Time and frequency domain investigation of the heat transfer to a synthetic air jet

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Abstract. Heat transfer to a synthetic air jet is investigated experimentally. The influence of peaks in heat transfer outwith the stagnation region of the jet are of particular interest. Heat transfer to the jet is reported for experimental parameters, jet exit to impingement surface spacings, $H/D = 1$, Reynolds number of 3000, non-dimensional Stroke length, $L_0/D$ of 14, and an excitation frequency of 70 Hz. Peaks in heat transfer outwith the stagnation region of the jet are investigated in both the time and frequency domain and a connection between the driving frequency of the jet and changes in the rate of heat transfer is outlined. It is shown that two types of changes in the rate of heat transfer outwith the stagnation region are present in synthetic jet impingement heat transfer; those associated with the jet excitation frequency and therefore attributed to interactions between the two jet flow regimes and those associated with the breakdown of coherent structures in the jet flow.

1. Introduction
Jet impingent heat transfer has, for many years, been used in industry for applications such as the cooling of turbine blades [1], manufacturing processes such as grinding [2] and the thermal management of electronics [3]. This is due to the high localised and area averaged heat transfer coefficients jet impingement can achieve. Synthetic air jets are a promising new technology which has been shown by both Smith and Swift [4] and Pavlova and Amitay [5] to be capable of providing more than twice the cooling of continuous air jet impingement for a similar range of Reynolds numbers.

A synthetic jet is a time averaged fluid motion which is formed by the periodic oscillation of a diaphragm bounded to a cavity containing an orifice. The periodicity and amplitude of the oscillatory flow dictate whether or not a jet is formed. Synthetic jets are characterised using two dimensionless parameters, Reynolds number and stroke length. The velocity scale used in these parameters is suggested by Smith and Glezer [6] to be the downstream directed velocity on the expulsion stroke of the jet cycle. Persoons and O’Donovan [7] have shown that by measuring the pressure in the jet cavity, the exit velocity of the jet can be calculated using conservation of momentum at the jet orifice. Synthetic air jets are compact and require no external plumbing which makes them an attractive technology for space restricted applications such as electronics cooling.

Numerous studies investigating steady jet impingement heat transfer have reported secondary peaks in the heat transfer distribution [8-12]. These secondary peaks in the heat transfer distribution are commonly observed at low $H/D (\leq 2)$ outwith the centre of the stagnation region at $r/D = 2$. Goldstein et al [11] noted secondary peaks only at the lowest spacing tested, $H/D = 2$ and attributed them to entrainment caused by vortex interactions in the shear layer. Gardon and Akfirat [9] who observed secondary and tertiary peaks in heat transfer attributed them to two separate effects, diffusion of mixing induced turbulence in the boundary layer and transition to a turbulent boundary layer. Hoogendoorn [12] report the occurrence of secondary peaks in the heat transfer for...
spacings up to $H/D = 8$ over the range of Reynolds numbers the secondary peaks where attributed to “very high” turbulence in the boundary layer.

This area was investigated more recently by O’Donovan and Murray [13, 14] who showed there to be little change in the arrival velocity or turbulence intensity of the jet in the range of $0.5 < H/D < 2$ but significant change in the magnitude of the secondary peaks in this range. O’Donovan and Murray [13, 14] clearly ascribed this to the interactions of vortices with the wall jet flow. Impinging vortices have been shown to enhance velocity fluctuations normal to impingement and resulted in an increase in heat transfer locally.

Smith and Swift [4] have shown that steady and synthetic jets have many similarities in the far field having near identical self-similarity velocity profiles. Therefore much of the work done investigating the far field behaviour of steady jets is applicable to synthetic jets. Conversely in the near field steady and synthetic jets perform dramatically differently. Steady jets in the near field are characterised by a potential core with a turbulent mixing layer in which exist vortices generated by Helmholtz instabilities. Synthetic jets in contrast have no potential core and the near field consists entirely of vortices generated by the periodicity of the oscillatory flow.

One of the factors impeding the implementation of synthetic jets as a cooling technology is there relatively poor performance at low jet to impingement surface spacings where heated air is recirculated into the jet flow increasing the jet temperature and decreasing its cooling efficacy. Another factor is the effect of confinement. Youn et al [15] investigated the effects of confinement of a mesoscale steady jet for a wide range of confining plate lengths and nozzle to impingement surface spacings. Overall, confinement is shown to reduce the heat transfer as the heated outflow is confined between the jet and the impingement surface. Youn et al [15] show that at low $H/D$, even relatively minor confinement can reduce the stagnation point heat transfer by more than 50%. In general, at low $H/D$ however, further increasing the confinement does not lead to substantial further reductions in the surface heat transfer, but it does however, affect the height at which the effect of the increased plate length will be overcome, for instance a jet confined to $L_{c}/D = 12$ overcomes the effect of confinement and cools at the same level as an unconfined jet at $H/D = 4$ while a jet confined to $L_{c}/D = 24$ does not overcome the effect of confinement until $H/D = 10$. Herein lies one significant difference between steady and synthetic air jet impingement heat transfer. Chaudhari et al [16] investigated the effect of increasing the confining plate length of a synthetic jet Heat transfer to the more confined impinging jet was less than 50% of the heat transfer to the less confined jet for the range of nozzle to impingement surface spacings tested. Therefore, it is clear that heat transfer to confined impinging synthetic jets is significantly different from steady jets, which only display the effects of increased confinement at low $H/D$.

Flow interactions between the exiting jet and the incoming jet flow form part of the reason for this difference and how understanding the nature of these interactions would be a powerful design tool in order to gain an understanding of synthetic jet behaviour at low $H/D$ the heat transfer is investigated both the time domain and frequency domain the focus of the current study is to investigate the connection between the magnitude of fluctuations in heat transfer in the time domain (Nu’) and the magnitude of the corresponding driving frequency in the frequency domain and the influence of both on the magnitude of the heat transfer coefficient.

2. Experimental setup

The experimental setup (figure 1) consists of two main components: a synthetic air jet, and a heated impingement surface. The synthetic jet is mounted directly above the heated surface with the jet flow directly perpendicular to it. As indicated in figure 1, the synthetic air jet consists of a cavity with an oscillating diaphragm or actuator on one end and the jet orifice opposite it. The cavity diameter is 76mm and has a depth of 30 mm and the circular orifice is 5mm in diameter and 10mm in length. The whole assembly is fabricated from a 90mm by 90mm by 35mm aluminium block. All dimensions are accurate to within 0.1mm. The synthetic jet actuator is a Visaton® FR8 8Ω acoustic speaker which is driven by national instruments 9263 analogue output module providing a sine wave which is amplified by a Kemo® MO 034 40W power amplifier. A GRAS 40PL microphone and a T-type thermocouple are embedded in the cavity of the jet to allow for cavity pressure and temperature measurements; these are recorded using National instruments DAQ 9233 IEPE Dynamic Signal Acquisition module and a 9211 thermocouple module and logged using LabVIEW. The jet assembly is attached to a Manfrotto 454 micro positioning plate, to allow for fine adjustment of the distance between the synthetic air jet and the impingement surface.

The impingement surface consists of a thin resistance heater foil that has a surface area of 150mm by 100mm and approximates a uniform wall flux thermal boundary condition. Three different thicknesses of Constantan® Resistance Alloy (Cu55/Ni45) foil were used these were 10, 25 and 50μm and were fabricated by Goodfellow Cambridge Limited. The undersides of the foils were lightly coated with a mat black paint to increase the surface
emissivity value to approximately 0.98; this was necessary to ensure accurate surface temperature measurements with an infrared thermal imaging camera. The foils are clamped between two copper bus bars. A direct current voltage is applied across the foil via the bus bars by a Farnel AP 20 - 80 regulated power supply.

Figure 1: schematic diagram of the experimental set up

A CEDIP Titanium 560 Indium Antimonide detector 3 – 5 µm waveband high speed thermal imaging camera is mounted directly beneath the heated surface. The camera is used to measure the surface temperature of the foil with high spatial and temporal resolution. The frame rate used in the current study is 400Hz which yields an image of 640 x 64 pixels. The field measured by the camera at this resolution is 105 x 10.4mm which corresponds to an image resolution of 30.7 pixels per jet diameter. The camera has a noise equivalent temperature difference (NETD) of <18mK at 30°C and a range of -20 to 3000°C with an absolute temperature accuracy of ±1%. The camera is also attached to a Manfrotto 454 micro positioning plate. For all tests the maximum operating temperature of the foils was less than 80°C. Each dataset is 20 seconds in duration which result in 8000 individual images.

Heat transfer data are presented in the form of the time-average and time-varying Nusselt number which uses the diameter of the jet to normalise the surface convective heat transfer coefficient. This can be calculated as follows:

\[ h = \frac{q_{\text{gen}}'}{(T_{\text{surf}} - T_{\text{gen}}')} \]  \hspace{1cm} (1)

\[ q_{\text{gen}}' = \frac{VI}{A_{\text{surf}}} \]  \hspace{1cm} (2)

\[ Nu = \frac{hD}{k} \]  \hspace{1cm} (3)

Equation 2 is based on the assumption that heat generated within the foil is conducted in 1 dimension through the thickness of the foil before convecting to the impinging jet. Patil and Narayanan [17], who also investigated the application of the thin foil infrared thermography technique for heat transfer measurements in convective flows, found that “lateral conduction heat flux rate contributed a significant portion to the net heat entering the control volume”. Therefore, in order to accurately assess the local heat transfer taking place from the foil to the impinging jet, lateral conduction must be considered. A correction for lateral conduction similar to the one used by Golobic et al [18] was used in this study to calculate the mean and time-varying surface heat transfer coefficient. Figure 2 shows the uncorrected heat transfer distributions for three foil thicknesses. Figure 3 shows the same data once the correction technique has been applied. Once lateral conduction has been taken into accounted all three heat transfer distributions converge, confirming the validity of the correction technique.

A synthetic jet is defined primarily by two dimensionless parameters, the Reynolds number and stroke length. The stroke length, \( L_0 \) is defined by Smith and Glezer[4]as the integral of the jet exit velocity, \( u_0 \) with time for the expulsion stroke:

\[ L_0 = \int_0^T u_0(t) \, dt \]  \hspace{1cm} (4)

Where \( r \) is the period of the cycle. The velocity used to calculate the jet Reynolds number, \( Re \) is based on this stroke length:
Hence the synthetic jet Reynolds number, $Re$, may be calculated as in equation 6:

$$Re = \frac{U_0 D}{v}$$

In this investigation the jet outlet velocity is not measured directly but rather the time varying relative pressure of the cavity is measured and the jet exit velocity calculated via the method proposed by Persoons and O’Donovan [7]. By using the conservation of momentum in the orifice the time varying jet exit velocity can be calculated as follows:

$$m \frac{dU_0}{dt} + F_d(U_0) = pA$$

Where $F_d(U_0)$ represents a damping force, and $m = \rho AL’$ is the mass of gas in the orifice. The effective length $L$ is the sum of the geometric length $L$ and end corrections $L’ = 2\beta D$, $\beta = 0.425$ [19] for a sharp-edged circular orifice.

The absolute uncertainty in the Nusselt number is calculated to be less than 3% for a fixed Reynolds number of 3000 and stroke length of 15.

3. Results and discussion

Distributions of time averaged and local Nusselt number for an impinging synthetic air jet are presented in this section. Profiles of the magnitude of the Nusselt number fluctuations in the time domain as well as magnitude of the fluctuations in the frequency domain are also presented as they give some insight into the level of turbulence in the flow next to the wall. The jet used in this investigation was generated using a constant excitation frequency of 70Hz; the Reynolds number is set at 3000 which has a corresponding dimensionless stroke length, $L_o/D$, of 14. The empirically found relationship between the three parameters is given by equation 8

$$f_c = 0.32 Re L_o/D^{0.944}$$

Time averaged heat transfer distribution are presented in figure 4 as can be seen heat transfer to the jet is a maximum in the stagnation region and decreases with increasing radial distance. Also included in figure 4 are the natural convective and radiative heat loss from the heated impingement surface, this value was measured whilst the jet was not in use. The magnitude of the free convective and radiative losses is something of an over estimation of the value to be expected during jet operation as this value includes natural convection from the top of the heated impingement surface and is calculated from a uniform temperature distribution both of which do not occur during jet operation. Nu’, the root-mean-square of the time varying Nusselt number signal is presented in Figure 5 for the same range of test parameters as figure 4. Nu’ is based on the time varying temperature difference between the heated surface and the synthetic air jet over the 20 second data capture period at a frequency of 70 Hz. By examining both the Nu and Nu’ profiles and comparing the relative position peaks in each profile it is possible to discern a link between changes in the fluctuations in heat transfer and changes in the mean heat transfer. As can be seen in figure 5 the Nu’ profile has a peak at centre of impingement and the fluctuations decrease in magnitude until $r/D=1$ at which point fluctuations begin to rise, this rise in fluctuations
is accompanied by a change in the slope of the heat transfer distributions visible in figure 4 at $r/D=1$. $\text{Nu}'$ rises to a maximum at $r/D=2$ and then falls until $r/D=2.5$ at which point an increase is observed which is accompanied by a sharp change in heat transfer visible at $r/D=2.5$ in figure 5. From figures 4 and 5 it can be seen that there exists a connection between the magnitude of the heat transfer and the magnitude of the fluctuations in heat transfer. The presence of secondary peaks in steady jet impingement heat transfer have been shown by O’Donovan and Murray [13] [14] to be caused by the breakdown of vortices in the jet flow.

In steady jets vortices are generated due to Helmholtz instabilities in the jet flow with a periodicity dictated by the frequency of these instabilities. In synthetic jets impingement heat transfer the periodicity of the vortex generated at each expulsion stroke of the jet is set by the jet excitation frequency. If the peaks in $\text{Nu}$ and $\text{Nu}'$ observed in figures 4 and 5 are attributable to breakdown of vortices then a frequency domain analysis of the time varying temperature signal at the jet excitation frequency will give some insight into the nature of the peaks in the heat transfer and fluctuations in heat transfer.

To investigate these peaks in the frequency domain, a fast Fourier transform is performed on the time varying temperature signal at each radial location from 0 to 9 diameters. Figure 6 shows the magnitude of the portion of time varying signal which is detectable at the synthetic jet actuation frequency with increasing distance from the centre of impingement. While these trends (figures 4 – 6) have the potential to shed light on the effect of vortices passing along the impingement surface, they should be interpreted with caution as the relative uncertainty is high and results presented are at greater precision than possible, according to the IR Camera specification. Figure 7 shows the phase of the excitation frequency across the surface of the foil with respect the phase at the centre of impingement. From figure 6 it is clear that the amplitude of the oscillations is in general greater around the centre of impingement ($r/D<2$). Three distinct peaks are discernable in the profile at $r/D=0.5$, 2 and 4.5. The peak at $r/D=0.5$ is attributed to vortices which are located at the edge of the jet impinging on the foil. The second peak at $r/D=2$ in figure 6 corresponds to the peak in $\text{Nu}'$ observable at $r/D=2$ in figure 6 and the change in then slope of the heat transfer at the same radial location in figure 4.
After the peak in magnitude of frequency domain fluctuations at the driving frequency \( r/D = 2 \) in figure 6 the magnitude falls sharply to a minimum at \( r/D = 3.2 \) and then begins to rise again to a local maxima at \( r/D = 4.5 \). Comparing this with the magnitude of the fluctuations in heat transfer seen in figure 5 it can be seen that after the peak at \( r/D = 2 \) the magnitude of the fluctuations begin to fall to a local minima at \( r/D = 2.5 \) and then rises sharply to a maximum at \( r/D = 3.5 \) this peak in Nu’ has a corresponding peak in the heat transfer visible at \( r/D = 3.5 \) in figure 4, after this peak the magnitude of Nu’ falls sharply.

From this comparison it can be seen that the peak in the time domain fluctuations and frequency domain fluctuations at \( r/D = 2 \) are clearly associated with the change in heat transfer at this point observable in figure 5 and can be attributed to interactions between the issuing jet and the fluid been drawn into the cavity on the suction stroke of the jet. This is supported by the fact that the frequency domain fluctuations at \( r/D = 2 \) are out of phase with the main jet excitation as is shown in figure 7. Conversely The sharp change in magnitude of the fluctuations in heat transfer at \( r/D = 2.5 \) in figure 5 and the corresponding change in the heat transfer at the same radial location in figure 4 clearly do not have any corresponding frequency domain associations and are attributed the breakdown of coherent structures in the jet flow and are similar to the secondary peaks observed in steady jet impingement heat transfer by O’Donovan and Murray [13] [14]

To further understand the convective heat transfer to a synthetic air jet the time varying temperature was examined. As previously mentioned surface temperature data were captured at 400Hz the duration of the test was 20 seconds as the excitation frequency of the jet was 70Hz this allowed the data to be phase locked to convert the 8000 images to 40 averages each of 200 images at a distinct point in the jet cycle. Figure 8 the surface temperature distribution with the black x’s indicating the locations where the time-resolved heat transfer signals are examined.

Figure 9 shows the phase averaged normalised temperature timeline at the points corresponding to those shown in figure 8. From the stagnation point it is clear that the temperature signal fluctuates sinusoidally at all locations up to \( r/D = 2.5 \). After this point the signal becomes incoherent, this corresponds with the sharp change in the magnitude of both Nu and Nu’ seen beyond \( r/D = 2.5 \) in figure 4 and respectively. The signal becomes more coherent again \( r/D = 4.5 \) which corresponds with the point at which Nu’ has reach the level observed at \( r/D = 2.5 \) and at which the driving frequency fluctuations have risen again at \( r/D = 4.5 \), observable in figure 6 the signal retains its sinusoidal nature until a radial location of 7 jet diameters and then losses coherence the decrease in signal coherence and increases the fluctuation in heat transfer between \( r/D = 2.5 \) and \( r/D = 4.5 \) give further indication that the secondary peaks in heat transfer at shown to be due to turbulence caused by the breakdown of coherent structures on the impingent surface by O’Donovan and Murray [13] [14] in their work on steady jets has relevance when applied to synthetic jets.
Figure 9: Normalised Phase Averaged Heat Transfer Signals; $f_e=70\text{Hz}$, $Re=3000$, $L_0/D=14$, $H/D=1$, $r/D$ 0 to 8.5

Figure 10: Normalised Phase Averaged Heat Transfer Signals; $f_e=70\text{Hz}$, $Re=3000$, $L_0/D=14$, $H/D=1$, $r/D$ 0 to 2

Figure 10 shows the phase averaged normalised temperature timeline at the points from $r/D$ 0 to 2 as can be seen the signal up until $r/D=1.5$ are in phase with each other but at $r/D=2$ the signal begins to move out of phase with the main excitation frequency which gives further indication the peak seen at this point in figure 6 is not associated with the main jet but is attributable to interaction between the two flow regimes.

4. Conclusions
Heat transfer to a synthetic air jets has been investigated experimentally. Heat transfer to the jet is reported for a jet exit to impingement surface spacings $H/D =1$, at a Reynolds number, $Re=3000$ with a non-dimensional Stroke length, $L_0/D=14$. 
Peaks in heat transfer outwith the stagnation region of the jet were investigated and a connection between the driving frequency of the jet and changes in the rate of heat transfer was outlined. It was shown that two types of changes in the rate of heat transfer are present in synthetic jet impingement heat transfer; those associated with the jet excitation frequency and therefore attributed to interactions between the two jet flow regimes and those associated with the breakdown of coherent structures in the jet flow.

5. Nomenclature

| Symbol | Description | Unit |
|--------|-------------|------|
| D      | jet diameter, m |      |
| p      | pressure, Pa   |      |
| f_e    | excitation frequency, Hz |      |
| H      | height of nozzle above plate, m |      |
| k      | thermal conductivity, W/mK |      |
| L_o    | stroke length, m |      |
| L_op   | orifice plate length |      |
| Nu     | Nusselt number |      |
| Nu'    | root-mean-square Nusselt number |      |
| q      | rate of heat transfer, W |      |
| r      | distance from centre of impingement, m |      |
| A      | area, m^2 |      |
| Re     | Reynolds number, ρU/D/μ |      |
| T      | temperature, °C |      |
| U_0    | downstream directed jet velocity, m/s |      |
| V      | voltage, V |      |
| ρ      | density, kg/m^3 |      |
| v      | kinematic viscosity, m^2/s |      |
| ν      | cycle period, s |      |

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