EFFECT OF ACOUSTIC EXCITATION ON THE HEAT TRANSFER TO AN IMPINGING AIR JET

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ABSTRACT
Impinging air jets are known as a method of achieving particularly high heat transfer coefficients and are employed in many applications including the cooling of electronics, manufacturing processes such as grinding, etc. The current investigation is concerned with acoustically exciting an impinging air jet to enhance its overall cooling capacity. Distributions of the heat transfer to an axially impinging air jet for a range of Reynolds numbers ($Re$) from 10000 to 30000, non-dimensional nozzle to impingement surface heights ($H/D$) from 0.5 to 2 and excitation frequencies ($f$) that range from 0.5 to 1 times the natural frequency of the jet are presented. For this low range of nozzle to impingement surface spacings it has been shown that the heat transfer distribution exhibits a peak at the stagnation point and secondary peaks at a radial location that is both excitation frequency and Reynolds number dependent. Distributions of the fluctuating component of the heat transfer coefficient are also presented for the range of parameters tested. These have been used, along with spectral analysis of the heat flux signal, to discern whether local variations in heat transfer are due to changes in the local vortex flow or to changes in the mean flow structure of the impinging jet.

INTRODUCTION
Impinging air jets are employed in a wide range of applications for enhanced cooling as they are known as a means of achieving particularly high heat transfer coefficients. Jets have been used to transfer heat in diverse applications, which include the cooling of turbine blades, electronic components and manufacturing processes such as grinding. Many fundamental investigations of the impinging jet fluid flow and heat transfer have identified several parameters which influence heat transfer on the impingement surface. Thus, the main variables for jet impingement heat transfer are the angle of impingement, the jet Reynolds number, $Re$, $H/D$, and $f$. The current investigation is concerned with acoustically exciting an impinging air jet to enhance its overall cooling capacity. Distributions of the heat transfer to an axially impinging air jet for a range of Reynolds numbers ($Re$) from 10000 to 30000, non-dimensional nozzle to impingement surface heights ($H/D$) from 0.5 to 2 and excitation frequencies ($f$) that range from 0.5 to 1 times the natural frequency of the jet are presented. For this low range of nozzle to impingement surface spacings it has been shown that the heat transfer distribution exhibits a peak at the stagnation point and secondary peaks at a radial location that is both excitation frequency and Reynolds number dependent. Distributions of the fluctuating component of the heat transfer coefficient are also presented for the range of parameters tested. These have been used, along with spectral analysis of the heat flux signal, to discern whether local variations in heat transfer are due to changes in the local vortex flow or to changes in the mean flow structure of the impinging jet.

NOMENCLATURE

- $D$: jet diameter, $m$
- $f$: frequency, $Hz$
- $f_e$: excitation frequency, $Hz$
- $h$: convective heat transfer coefficient, $W/m^2K$
- $H$: nozzle to plate spacing, $m$
- $k_a$: thermal conductivity of air, $W/mK$
- $k_s$: thermal conductivity of sensor barrier, $W/mK$
- $Nu$: Nusselt number, $hD/k$
- $Nu'$: root-mean-square Nusselt number
- $\dot{q}$: heat flux, $W/m^2$
- $r$: distance along plate from geometric centre, $m$
- $R$: sensor resistance, $\Omega$
- $Re$: Reynolds number, $\rho U_j D/\mu$
- $St$: Strouhal number
- $T$: temperature, $°C$
- $U_j$: jet exit velocity, $m/s$
- $V_{out}$: output voltage, $V$
- $\Delta T$: temperature difference, $°C$
- $\delta$: sensor thickness, $m$
- $\mu$: viscosity, $kg/ms$
- $\rho$: density, $kg/m^3$
number and the height of the nozzle above the impingement surface. Several other parameters such as the free stream jet turbulence intensity, confinement and submergence of the jet flow have also been shown to influence the heat transfer. In recent years, vortices which occur naturally in an impinging jet flow have also been shown to influence the surface heat transfer. By controlling the formation of vortices within the impinging jet flow the current investigation has quantified the influence vortices have on surface heat transfer.

Comprehensive studies of the mean fluid flow characteristics of both a free and an axially symmetric impinging air jet have been presented by Donaldson and Snedeker [1] and Beltaos [2]. In many investigations, including that by Gardon and Akfirat [3], the heat transfer to an impinging jet has been correlated with what is often termed the “arrival” flow condition. Comprehensive reviews of the heat transfer to impinging jets have been presented by Martin [4], Jambunathan et al. [5] and Polat et al. [6]. The heat transfer distribution to an impinging jet varies significantly in shape and magnitude with the various test parameters. The shape of the radial heat transfer distribution is affected primarily by the height of the nozzle above the impingement surface. Several investigators, including Donaldson and Snedeker [7], Goldstein and Behbahani [8] and others presented heat transfer data for a jet impinging at large $H/D$. According to Mohanty and Tawfek [9] the heat transfer rate peaks at the stagnation point and decreases exponentially with increasing radial distance beyond $r/D = 0.5$ for a relatively large range of nozzle to impingement surface spacings ($4 < H/D < 58$). In studies by Baughn and Shimizu [10], Huang and El-Genk [11], Goldstein et al. [12] and others, secondary peaks in the heat transfer distribution to an impinging air jet have been reported. In some cases two radial peaks are present in the heat transfer distributions. Both Hoogendoorn [13] and Lytle and Webb [14] have proposed that at low $H/D$, the wall jet boundary layer thickness decreases with distance from the stagnation point as the flow escapes through the minimum gap between the nozzle lip and the impingement surface. In the case of a low turbulence jet this thinning results in a local maximum in the heat transfer distribution. With further increase in distance from the stagnation point the laminar boundary layer thickness increases before transition to a fully turbulent flow. Effectively the thickening of the laminar boundary layer decreases the rate of heat transfer and upon transition to a fully turbulent wall jet, the heat transfer distribution increases to a secondary peak.

In a jet flow, vortices initiate in the shear layer due to Kelvin-Helmholtz instabilities. As the vortices move downstream of the jet nozzle each vortex can be wrapped and develop into a three dimensional structure due to secondary instabilities. These secondary instabilities can lead to the “cut and connect” process as described by Hui et al. [15] and Hussain [16] in which the toroidal vortices break down into smaller scale motions, generating high turbulence. Vortices, depending on their size and strength, affect the jet spread, the potential core length and the entrainment of ambient fluid. In certain cases jet vortices can pair, forming larger but weaker vortices. In general, vortices pass in the shear layer of the jet at the same frequency as that at which they roll up but in the vortex pairing case the passing frequency halves as the vortices pair off. Turbulent jets have a fundamental frequency at which the pairing process stabilises and this is determined by the turbulence level of the jet. With distance from the jet nozzle the vortices break down into random small scale turbulence. It is clear that vortices influence the arrival velocity of the impinging jet flow and therefore influence the shape and magnitude of the heat transfer distribution.

In recent times control of the jet vortex flow has attracted much research interest as the latest parameter identified as important for impinging jet heat transfer. Hui et al. [15] and Gao et al. [17] installed mechanical tabs at the nozzle exit to instigate streamwise vortical structures. These have the effect of increasing the secondary instabilities in the jet and therefore hasten the “cut and connect” process that breaks the vortices down into small scale turbulence. Hwang et al. [18] investigated the effect of acoustic excitation on a coaxial jet flow and explored the resulting effect on heat transfer. Hwang and Cho [19] continued this research for a wider range of test parameters. Liu and Sullivan [20] excited an impinging air jet acoustically and reported on the resulting flow and heat transfer distributions. It was found that, depending on the frequency of excitation, the area averaged heat transfer could be enhanced or reduced at low nozzle to impingement surface spacings. While the research to date has shown possible enhancement of the mean heat transfer at various excitation frequencies, much of this has been attributed to changes in the arrival velocities. O’Donovan and Murray [21], [22] have shown that for $H/D < 2$ the arrival flow velocity and turbulence intensity does not change significantly. There is however a significant variation in the size of the secondary peaks in this range. It was shown that the magnitude of these peaks is influenced by the vortices within the jet flow. Strong small-scale vortices instigate large velocity fluctuations normal to the surface in the wall jet resulting in enhanced heat transfer. Artificial jet excitation can control the development of vortices in the jet flow and therefore has the potential to enhance heat transfer from the surface.

The current investigation presents surface heat transfer data for a jet impinging at heights of $H/D \leq 2$ and jet Reynolds numbers of 10000 to 30000. A reduced range of data are also presented for a similar impinging jet that is acoustically excited at frequencies ranging from 0.5 to 1 times its natural frequency. Both mean and fluctuating heat transfer distributions from a heated surface to the impinging air jet are compared for the range of parameters. Spectral analysis of surface heat flux signals are presented to show the influence of both naturally occurring and acoustically controlled vortices on the local and area averaged heat transfer.
EXPERIMENTAL SETUP

The main elements of the experimental rig are a nozzle and an impingement surface. Both are mounted on independent carriages that travel on orthogonal tracks. The flat impingement surface is instrumented with two single point heat flux sensors and the ability of the carriages to move in this way enables the jet to be positioned relative to the sensors at any location in a two dimensional plane. The rig design is presented in figure 1.

As illustrated in figure 1 the jet issues from a contoured nozzle at one end of a cylindrical chamber. Air is supplied by a compressor to the chamber via four separate inlets located near the top of the chamber. An Alicat Scientific Inc. Precision Gas Flow Meter is installed on the compressed air line to monitor both the air volume flow rate and temperature. An 8Ω Visaton acoustic speaker is positioned directly opposite the jet nozzle and is driven by a sine wave produced by a TTI TG210 function generator with an acoustic amplifier.

The impingement surface is a flat plate, measuring 425mm × 550mm, that consists of two main layers mounted on a carriage. The top surface is a 5mm thick copper plate. A silicon rubber heater mat, approximately 1.1mm thick, is fixed to the underside of the copper plate with a thin layer of adhesive. The plate assembly is such that it approximates a uniform wall temperature boundary condition, operating typically at a surface temperature of 60°C. Grooves are machined in the impingement surface to allow the flush mounting of the heat flow sensors. These are positioned in a central location and, together with the nozzle and plate carriage arrangement, allow for heat transfer measurements beyond 20 diameters from the geometric centre of the jet. For the present study, testing has only been concerned with a region extending to 6 diameters from the geometric centre.

An RdF Micro-Foil® Heat Flux Sensor is flush mounted on the heated surface. This sensor contains a differential thermopile that measures the temperature above and below a known thermal barrier. The heat flux through the sensor is therefore defined by equation 1.

\[ \dot{q} = k_s \frac{\Delta T}{\delta} \]  

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

A Senflex® hot film sensor operates in conjunction with a Constant Temperature Anemometer to measure the fluctuating heat flux to the impinging jet, as it has higher temporal resolution than the Micro-Foil® sensor and can accurately acquire data in excess of 8kHz. This equates to a Nyquist frequency of 4096Hz and for the maximum jet exit velocity investigated, the Strouhal number is calculated to be approximately 5. This upper value
was not required however as the maximum Strouhal number associated with coherent structures within the impinging jet flow is less than 2. The sensor consists of a nickel sensor element that is electron beam deposited onto a 0.051\,mm thick Upilex S polyimide film. The hot film element has a thickness of < 0.2\,\mu m and covers an area of approximately 0.1\,mm \times 1.4\,mm. The typical cold resistance of the sensor is between 6 and 8 Ohms. Copper leads are also deposited on the film to provide terminals for connection to the CTA. The leads have a resistance of approximately 0.002\,\Omega/mm. A Dantec StreamLine Constant Temperature Anemometer is used to control the temperature of the hot film. It maintains the temperature of the film at a slight overheat (≈ 5°C) above the heated surface. The power required to maintain this temperature is equal to the heat dissipated from the film. A CTA is essentially a Wheatstone bridge where the probe, or hot film in this case, forms one arm of the bridge. The resistance of the film varies with temperature and therefore, by varying a decade resistance that forms another arm of the bridge, the temperature of the film is controlled. The voltage required to maintain the temperature of the film constant is proportional to the heat transfer to the air jet as described in equation 2. Corrections due to the slight overheat of the sensor above the impingement surface temperature were made to acquire accurate measurements.

\[ q_{dissipated} \approx \frac{V^2_{out}}{R} \quad (2) \]

Both the RdF Micro-Foil® heat flux sensor and the Senflex® hot film sensor were calibrated in situ against a reference stagnation point heat transfer sensor presented by Shadlesky [23]. The calibration of the sensors compared favourably with that provided by the manufacturer’s specification. Heat transfer results are presented as distributions of the time averaged mean Nusselt number (\(Nu\)) and fluctuating Nusselt number (\(Nu'\)). The magnitude of the fluctuations of the Nusselt number (\(\nu'\)) is the root-mean-square (rms) of the Nusselt number signal. The mean and fluctuating Nusselt numbers have calculated uncertainties of 5.7% and 30.0% respectively and are based on the local heat transfer coefficient that is defined by equation 3.

\[ h(r) = \frac{\dot{q}(r)}{(T_{surf}(r) - T_j)} \quad (3) \]

where \(\dot{q}\) is the heat flux signal (corrected for losses) from the surface, \(T_j\) is the jet exit temperature and \(T_{surf}\) is the local surface temperature. 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 \(Nu'\) is, in general, less than 30%. A complete calibration and uncertainty analysis for this experimental set-up is presented by O’Donovan [24].

RESULTS AND DISCUSSION

Time averaged and fluctuating heat transfer distributions for a steady jet impinging at \(H/D \leq 2\) are presented in figures 2 and 3 respectively for \(Re = 10000, 20000\) and \(30000\). For the entire range of parameters presented it is apparent that the maximum heat transfer coefficient occurs at the stagnation point. With increasing distance from the stagnation point the magnitude of the Nusselt number decreases before increasing again to a secondary peak. This peak occurs between \(r/D = 1\) and 2 and its location is dependent on both \(H/D\) and \(Re\). It is also apparent that the relative magnitude of the secondary peak is dependent on \(H/D\). For the entire range of parameters studied the heat transfer at the stagnation region (0 < \(r/D < 0.5\)) does not vary with \(H/D\); beyond a radial location of two diameters the distributions coalesce once more. The variation of the magnitude of this secondary peak is the subject of the current investigation.

It has been shown by O’Donovan and Murray [22] that the magnitude of the secondary peak is dependent on the passing frequency of vortices in the wall jet. The vortices which break down in the wall jet, just beyond the stagnation region, contribute to the turbulence in the wall jet which enhances the heat transfer to a secondary peak. With increasing radial distance the combined effects of increased temperature and reduced velocity in the wall jet result in the steady decrease of heat transfer.

Variations in the magnitude of the Nusselt number fluctuations with radial location are presented in figure 3. The general trend of these distributions is similar to the time averaged heat transfer distributions. This indicates that the local heat transfer magnitude is highly dependent on the fluctuations of velocity and temperature of the flow locally. The overall magnitude of the fluctuations increases with increasing Reynolds number. The fluctuations are smaller in the stagnation zone for low \(H/D\) and increase with increasing \(H/D\). Conversely, the magnitude of the secondary peak reduces as the height of the nozzle above the impingement surface increases. As the wall jet becomes fully developed however, the distributions of the heat transfer fluctuations coalesce at radial locations greater than \(r/D = 2\).

Liu and Sullivan [20] have shown that the vortices within the jet flow can be influenced by acoustic excitation and O’Donovan and Murray [22] have attributed the magnitude of the secondary peaks in heat transfer to the magnitude and passing frequency of the vortices. The current investigation used acoustic excitation to influence the magnitude of the local and area averaged heat transfer.

Figure 4 compares time averaged heat transfer distributions for a steady jet (\(Re = 10000; H/D = 1.0\)) excited at a range of
Figure 2. Time Averaged Nusselt Number Distributions

Figure 3. Fluctuating Nusselt Number Distributions
frequencies less than the natural frequency of the jet. It can be seen that the only effect of acoustic excitation is to change the magnitude of the secondary peak. Acoustically exciting the jet at frequencies less than the jet’s natural frequency has the effect of promoting vortex merging and therefore reduces the vortex passing frequency in the wall jet. Results presented in figure 4 have shown that exciting the jet at lower frequencies than the natural frequency reduces the magnitude of the secondary peak in the heat transfer distribution. As the excitation frequency is increased incrementally towards the natural frequency, then the magnitude of the peak is incrementally restored. The distribution of the fluctuating Nusselt number for the same range of parameters is presented in figure 5. It is clear that the same trends for an unexcited jet occur in the case of the jet excited at a range of excitation frequencies, however the fluctuating heat transfer distribution for an excited air jet exhibits two secondary peaks. As the frequency increases the magnitude of the fluctuations decrease in the stagnation zone. The opposite is true at a radial location of \( r/D = 1.5 \). This indicates that exciting the jet at frequencies lower than the natural frequency has the effect of making the arrival flow condition similar to that of a jet impinging at a larger \( H/D \). The resulting heat transfer characteristic for a jet excited at a frequency less than its natural frequency is equivalent to a jet impinging at a larger \( H/D \). As the excitation frequency nears the naturally occurring frequency, both the mean and fluctuating distributions of the Nusselt number become similar to that of an unexcited jet.

O’Donovan and Murray [22] acquired surface heat flux and local velocity measurements simultaneously and have shown that regular fluctuations in the wall jet velocity are attributed to vortices in the flow. Cross-correlating the heat flux and velocity signals have shown that peaks in the heat flux spectra are indicative of the frequency at which vortices pass in the wall jet. Figure 6 presents spectral analysis of the heat flux signal for the unexcited jet at a radial location of \( r/D = 1.3 \) for the full range of \( H/D \). At greater radial distances the peak disappears as the vortices which pass in the wall jet are broken down into small scale turbulence. Sharp peaks in the heat flux spectra at frequencies greater than 1200 Hz can be ignored as these have been attributed to electrical noise. The natural frequency of the jet is a broadband, however at \( H/D = 0.5, 1.0, 1.5 \) and \( 2.0 \) the broadband centre frequencies are approximated as 720, 630, 510 and 420 Hz respectively. The natural frequency of the jet varies with distance from the jet exit as naturally occurring vortices within the jet flow undergo a series of merging processes. This has the effect of reducing the passing frequency of vortices in the wall jet as \( H/D \) is increased.

Spectral analysis of the heat flux signal is presented in figure 7 at a radial location of \( r/D = 1.3 \) for each of the excitation frequencies. These data are used to explain differences in the mean and fluctuating heat transfer distributions presented in figures 4 and 5. Again, peaks in the heat flux spectra at frequencies greater than 1200 Hz are attributed to electrical noise. The spectrum of the heat flux signal for an unexcited impinging jet is presented in figure 7 (a). The broadband peak in this heat flux spectrum has a centre frequency of approximately 630 Hz. As stated previously, this is the natural frequency at which vortices pass in the unexcited jet flow at \( H/D = 1 \). Figures 7 (b), (c) and (d) show similar data for an impinging jet that is acoustically excited at frequencies of 315, 472 and 630 Hz respectively. When the jet is excited at a frequency of 315 Hz (half the natural frequency) it can be seen from figure 7 (b) that the natural frequency has been replaced by the excitation frequency. The driving frequency has, in this case, successfully paired the natural frequency to its sub-
harmonic. This would have also occurred in an unexcited jet but at a much greater $H/D$. It is for this reason that exciting the jet at a subharmonic of its natural frequency will reduce the magnitude of the secondary peak, as these peaks are small or may not exist at larger values of $H/D$.

As the excitation frequency of the jet is increased to $472\, \text{Hz}$ (figure 7 (c)) the spectrum shows that this driving frequency does not have the same influence on naturally occurring frequency. Two distinct peaks now occur at the excitation frequency ($472\, \text{Hz}$) and the natural frequency ($630\, \text{Hz}$). This indicates that the excitation frequency has not been as successful in encouraging vortices to merge in the same manner as $f_e = 315\, \text{Hz}$. As this excitation frequency was partially successful however, a reduction in the magnitude of the secondary peak is achieved at this excitation frequency as shown previously in figure 4. Finally, figure 7 (d) presents the spectrum of the heat flux signal at $r/D = 1.3$ for an impinging jet that is acoustically excited at its natural frequency. It is apparent that this frequency has the effect
of maintaining the natural frequency at which the vortices pass in the wall jet. It is therefore understandable that the heat transfer distributions for an unexcited jet and one excited at a frequency equal to the natural frequency are similar, as indicated in figure 4.

CONCLUSIONS

Time averaged and fluctuating Nusselt number distributions have been presented for a jet impinging on a heated flat surface. This has been done for a range of Reynolds numbers and jet to surface spacings. It has been shown that secondary peaks in the heat transfer distributions occur for the range of $H/D \leq 2$ presented. These peaks have been attributed, in the past, to an abrupt increase in turbulence in the wall jet. The current research presents data in support of this assertion and investigated the influence of vortices within the jet flow on the magnitude of the secondary peaks. It was shown that for a steady jet the magnitude of the peaks decreased with increasing $H/D$. It was also shown, by measuring the spectrum of the surface heat flux signal, that vortices within the impinging jet flow merge and therefore pass at lower frequencies with increasing $H/D$. The magnitude of the secondary peaks is linked to the frequency of the vortices in the impinging jet flow. Excitation of the impinging jet flow has proved to be a successful method of controlling the frequency of the vortices. Heat transfer distributions and spectral analysis of heat flux signals for a impinging jet that was acoustically excited have shown that reducing the frequency of vortices in the jet flow decreases the magnitude of the secondary peaks. Future work in this area will include flow velocity measurement which will provide a greater understanding of the effect vortices have on heat transfer. This will be done with a view to using acoustic excitation to enhance the overall heat transfer.

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