EFFECT OF VORTICES ON JET IMPINGEMENT HEAT TRANSFER

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Abstract

Convective heat transfer to an impinging air jet is known to yield high local and area averaged heat transfer coefficients. Such jets are of interest in the cooling of electronic components and of turbine blades and in manufacturing processes such as grinding. The current research is concerned with the measurement of heat transfer to impinging air jets over a wide range of test parameters. These include Reynolds numbers, Re , from 10000 to 30000, nozzle to impingement surface distance, H/D , from 0.5 to 8 and angle of impingement, α , from 30° to 90° (normal impingement). The current research reports both mean and fluctuating heat transfer distributions up to 6 diameters from the geometric centre of the jet. The heat transfer results are compared to local velocity data. Spectral and coherence information is also reported for simultaneous velocity and heat flux measurement.

At low nozzle to impingement surface spacings the heat transfer distributions exhibit secondary peaks at a radial location that varies with both Re and H/D . These peaks are shown to be due to an abrupt increase in turbulence in the wall jet boundary layer. At certain test configurations vortices that initiate in the shear layer impinge on the surface and move along the wall jet before being broken down into smaller scale turbulence. This vortical characteristic of an impinging air jet has also been shown to vary considerably with the angle of impingement; depending on the distance between the near nozzle edge and the impingement surface, vortices at different stages of development will impact with the target surface.

1. Introduction

Impinging jets have been used to transfer heat in diverse applications, which include the drying of paper, the cooling of turbine blades and the cooling of a grinding process. Hollworth and Durbin [1] investigated the impingement cooling of electronics. Roy et al. [2] investigated the jet impingement heat transfer on the inside of a vehicle windscreen and Babic et al. [3] used jet impingement for the cooling of a grinding process. The main variables for jet impingement heat transfer are the angle of impingement, the jet Reynolds number and the height of the nozzle above impingement surface. A comprehensive review of heat transfer to an impinging circular jet has been conducted by Jambunathan et al. [4]. Martin [5] also conducted a review of jet impingement heat transfer for a wider range of parameters that included jet arrays, confinement and submergence. Polat et al. [6] presented a review of numerical investigations that have been conducted in the area of convective heat transfer to impinging jets. Comprehensive studies of the mean fluid flow characteristics of both free and impinging jets have also been presented by Donaldson and Snedeker [7], Beltaos [8] and Martin [5].

In many investigations, including that of Gardon and Akfirat [9], the heat transfer to an impinging jet has been correlated with what is often termed the "arrival flow condition". This is the flow condition at an equivalent location in a free jet. It is well established that the shape of the heat transfer distribution changes significantly with nozzle to impingement surface spacing. 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 have been reported for a jet impinging at low nozzle to surface spacing ($H/D \leq 2$). According to Hoogendoorn [13] for a jet with low turbulence, two radial peaks occur in the heat transfer distributions. Goldstein and Timmers [14] compared heat transfer distributions of a large nozzle to plate spacing ($H/D = 6$) to that of a relatively small spacing ($H/D =$

2). It was shown that while the Nusselt number decays from a peak at the stagnation point for the large H/D , the Nusselt number is a local minimum at the stagnation point when $H/D = 2$. For the same jet Reynolds number the average heat transfer coefficient was found to be lower for the lower nozzle to surface spacing; this was attributed to the fact that the mixing induced in the shear layer of the jet had not penetrated to the potential core of the jet. The flow within the potential core has relatively low turbulence and consequently the heat transfer is lower in this case. Goldstein et al. [12] continued research in this area. Once again, at small spacings ($H/D \leq 5$), secondary maxima were evident at a radial location in the heat transfer distribution; at $H/D = 2$ these maxima were greater than the stagnation point Nusselt number. The secondary maxima occur at approximately r/D of 2 and were attributed to entrainment caused by vortex rings in the shear layer. Several investigators including Donaldson et al. [15] presented heat transfer data for a jet impinging at very large H/D . According to Mohanty and Tawfek [16] the heat transfer rate peaks at the stagnation point and decreases exponentially with increasing radial distance beyond $r/D = 0.5$ for $4 < H/D < 58$.

In more recent years research in this area has been concerned with the vortical nature of the impinging jet flow. In a jet flow vortices initiate in the shear layer due to Kelvin Helmholtz instabilities. Vortices, depending on their size and strength, affect the jet spread, the potential core length and the entrainment of ambient fluid. Fleischer et al. [17] employed a smoke wire technique to visualise the initiation and development of a vortex in an impinging jet flow. The vortex break-up location indicated a transition to turbulent flow that cannot sustain large scale flow structures. At large H/D the vortices break-up as they reach the end of the potential core, before impinging on the surface. This occurs following a vortex merging process where the size of the vortex increases but the strength decreases. Vortices merge because the vortex does not move fast enough to prevent being entrained by the fluid flow. Hui et al. [18] and Hussain [19] describe the break-up process of a vortex. It is apparent that the vortical nature of an impinging jet has a significant influence on the mean heat transfer distribution. Vortices also affect the magnitude and direction of velocity fluctuations close to the impingement surface. To date however, little has been reported on the effect of these fluctuations on the heat transfer characteristic. The current investigation reports on the fluctuating heat transfer to an obliquely impinging air jet. The effect of naturally occurring vortices on the temporal nature of the heat transfer is reported.

2. Experimental Setup

Figure 1: Schematic of Experimental Rig

The main elements of the experimental rig are a nozzle and an impingement surface as indicated in figure 1. This schematic also illustrates two directions which are commonly referred as the uphill and downhill direction of the flow. A heated impingement surface measuring 425mm x 550mm consists of three main layers. 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. It has a power rating of 15kW/m² and a voltage rating of 230V. The voltage is varied using a variable transformer that controls the heat supplied to the copper plate. A thick layer of insulation prevents heat loss from the heating element other than through the copper. The plate assembly is such that it approximates a uniform wall temperature boundary condition, operating typically at a surface temperature of 60°C.

The jet nozzle consists of a brass pipe of 13.5mm internal diameter. The pipe is 20 diameters long and a 45° chamfer is machined at the nozzle lip to create a sharp edge to minimise entrainment. The nozzle is clamped on a carriage in an arrangement that allows its height above the impingement surface and its angle of impingement to be varied. The height of the nozzle can be varied from 0.5 to 10 nozzle diameters above the impingement surface and the jet can be set at oblique angles ranging from 15°, in 15° increments, up to the normal angle of impingement (90°). Air is supplied to the jet nozzle by a compressor. 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. It is important that the jet exit temperature is maintained within 0.5°C of the ambient air temperature. To this end a heat exchanger is installed on the air line. The heat exchanger consists of a controlled temperature water bath in which a series of copper coils are placed. The air flows through the copper coils to increase the jet exit temperature to the required setting.

An RdF Micro-Foil® heat flux sensor is flush mounted on the heated surface to measure surface heat flux. 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:

$$q'' = k_s \frac{\Delta T}{\delta}$$

where k_s is the thermal conductivity of the barrier (kapton) and ΔT is the temperature difference across the thickness (δ) 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. The sensor consists of a nickel sensor element that is electron beam deposited onto a 0.051mm thick Upilex S polyimide film. The hot film element has a thickness of < 0.2µm and covers an area of approximately 0.1mm x 1.4mm. 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 0.002Ω/mm. A TSI Model 1053B 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 below:

$$q_{dissipated} \propto \frac{V_{out}^2}{R_{probe}}$$

Corrections due to the slight overheat of the sensor above the impingement surface were made to acquire accurate measurements. Both the heated plate and the nozzle are mounted on carriages that travel on orthogonal tracks, thus allowing the sensor to be placed in any location in a 2D plane.

Seeding particles are introduced to the jet flow by means of a venturi seeder installed in the compressed air line. This enabled the use of laser techniques to measure the flow velocity. Laser Doppler anemometry is employed to measure the velocity at locations within the impinging jet flow. The laser Doppler anemometry system is based on a Reliant 500mW Continuous Wave laser from Laser Physics. This is a two component system and therefore the laser is split into 2 pairs of beams that have wavelengths of 514.5nm (green) and 488nm (blue), to measure the velocity in orthogonal directions at the same point location. The four beams, each of diameter 1.35mm, are focused on a point 250mm from the laser head. The system works in backscatter mode and a Base Spectrum Analyser (BSA) acquires and processes the signal to compute the velocity.

A complete calibration and uncertainty analysis for this experimental set-up is presented by O'Donovan [20]. The uncertainty of the mean and root-mean-square Nusselt number is calculated to be $\pm 5.7\%$ and $\pm 30.1\%$ respectively. The jet exit velocity can be set to an accuracy of 4.7%.

3. Results & Discussion

3.1 Time-Averaged Heat Transfer Distributions

Both time averaged and temporal measurements of the heat transfer and fluid flow of an impinging air jet have been made. This section contains results in the form of local, time-averaged and root-mean-square Nusselt number distributions up to 6 diameters from the geometric centre of the impinging air jet, for a range of experimental configurations.

Figure 2: Obliquely Impinging Jet Mean Nusselt Number Distributions; $Re = 10000$

Figure 2 presents Nusselt number distributions for a jet Reynolds number of 10000, impinging at various oblique angles. A characteristic of the heat transfer distribution due to an obliquely impinging jet is that the peak heat transfer that occurs at the stagnation point is displaced in the

uphill direction from the geometric centre (as defined in Figure 1). As the angle made between the jet and the impingement surface decreases from normal impingement ($\alpha = 90^\circ$) the displacement of the stagnation point from the geometric centre is increased. Figure 2 (a) presents heat transfer distributions for a normalised jet to plate spacing of $H/D = 2$ and figure 2 (b) presents similar data for a larger spacing of $H/D = 6$. These two heights correspond to locations where the impingement surface is within the potential core of the jet and just beyond the core of the jet respectively. It is well documented that at low nozzle to impingement surface spacings, secondary peaks in the Nusselt number distribution occur at a radial location of approximately 1 to 3 diameters from the geometric centre for a normally impinging jet. Figure 2 (a) shows that these peaks in the Nusselt number distribution occur for oblique jet impingement also but this feature is not present in the data shown in Figure 2 (b) where $H/D = 6$.

It can be seen also from the results presented in figure 2 that secondary peaks may not occur in the uphill direction. For example, when the jet impinges at an oblique angle of 30° there is no secondary peak in the uphill direction but a broad secondary peak does occur in the downhill direction. As the angle of impingement increases towards normal impingement a narrow secondary peak appears in the uphill direction. The magnitude of this peak is greater than that in the downhill direction. Secondary peaks have been attributed to increased turbulence in the wall jet boundary layer and therefore the turbulence characteristic of the flow in both uphill and downhill directions is of interest.

Figure 3 presents root-mean-square Nusselt number distributions for the same range of parameters as in figure 2. The rms profiles are broadly similar to the mean heat transfer profiles, indicating that the eventual magnitude of the Nusselt number is influenced by the level of fluctuations in heat transfer and hence by the turbulence intensity close to the impingement surface. This is consistent with expectations. Also, it can be seen that the secondary peaks in the mean Nusselt number distributions at $H/D = 2$ correspond to sharp gradients in the corresponding Nu' distributions, suggesting abrupt increases in wall jet turbulence at these locations.

Figure 3: Obliquely Impinging Jet Root-Mean-Square Nusselt Number Distributions; $Re = 10000$

At $H/D = 2$, the greatest Nu' magnitudes correspond with the greatest mean Nu magnitudes; this is not the case for $H/D = 6$. While the radial positions of the peaks in the mean and fluctuating heat transfer distributions match for the latter case, the fluctuations are greatest for the most acute angle (30°) whereas the opposite is true for the mean Nusselt number. The conclusion that can be drawn from this analysis is that the controlling influence on heat transfer in this case is not the turbulence intensity but the turbulent flow structure. This is addressed in section 3.2.

The effect of Reynolds number on the heat transfer distributions is presented in figures 4 and 5 for a jet impinging at an oblique angle of 45° . Overall, the form of the heat transfer distribution does not change with Reynolds number. Increasing the Reynolds number has the effect of elongating the potential core of the jet. Effectively, this means that the impingement surface is at a different location relative to the potential core. For this reason the secondary peaks that occur in the downhill direction are more pronounced for larger Reynolds numbers. It has been shown that secondary peaks do not occur once the impingement surface is beyond the length of the potential core of the jet; for a Reynolds number of 10000 this was the case for $H/D = 6$ (Figure 2(b)). In figure 4 (b), Nusselt number distributions are presented for a jet impinging with $H/D = 6$. In this case, it can be seen that secondary peaks exist at $Re = 20000$ and become more pronounced at $Re = 30000$. This is again due to the elongation of the potential core at higher Reynolds numbers, with the impingement surface effectively being located within the potential core length of the jet, even at large H/D .

Figure 4: Obliquely Impinging Jet Mean Nusselt Number Distributions; $\alpha = 45^\circ$

Figure 5 presents root-mean-square Nusselt number distributions for the same range of test parameters ($\alpha = 45^\circ$; $Re = 10000$ to 30000 ; $H/D = 2, 6$). Again, peaks in the mean heat transfer distributions correlate well with the magnitude of the fluctuating component of the heat transfer. The magnitude of the heat transfer fluctuations generally scale with the jet Reynolds number, which is based on the jet exit velocity.

(a) $H/D = 2$

(a) $H/D = 6$

Figure 5: Obliquely Impinging Jet Root-Mean-Square Nusselt Number Distributions; $\alpha = 45^\circ$

The effect of H/D on the heat transfer distribution for a given angle of impingement (45°) and Reynolds number (10000) is presented in figure 6. Increasing the nozzle to impingement surface spacing has the effect of decreasing the magnitude of the peak Nusselt number that occurs at the stagnation point. As expected the increased spacing also diminishes the magnitude of the secondary peak, until there is no discernable change in slope in the downhill direction at $H/D = 8$. Figure 6 (b) presents the root-mean-square Nusselt number distributions for the same range of parameters. By comparing the rms and mean heat transfer distributions it appears that peaks and troughs in both profiles occur at the same location. The peaks in the rms heat transfer are more pronounced, however, indicating that the level of heat transfer fluctuation is not the only factor influencing the mean heat transfer. The same inference can be drawn from the fact that Nu' at the stagnation point is almost independent of jet to plate spacing whereas the peak in the mean Nusselt number distribution varies considerably with H/D .

(a) Mean

(b) Root-Mean-Square

Figure 6: Obliquely Impinging Jet Nusselt Number Distributions; $Re = 10000$; $\alpha = 45^\circ$

3.2 Temporal Analysis of Heat Transfer and Local Fluid Velocity

In order to understand the dominant heat transfer mechanisms at play, the local velocity was measured approximately 3mm from the impingement surface, using a 2 component laser Doppler

anemometer (LDA). Figure 7 presents mean and root-mean-square axial and radial velocity measurements together with the corresponding Nusselt number distributions for a jet Reynolds number of 10000 impinging at an oblique angle of 60° and for $H/D = 2$.

Figure 7: Velocity & Heat Transfer Distributions; $Re = 10000$; $H/D = 2$; $\alpha = 60^\circ$

It can be seen from figure 7 (a) that the mean axial velocity is a maximum near the stagnation point and that this velocity decreases sharply in the uphill direction and more gradually in the downhill direction. The radial velocity is zero at the stagnation point, as expected, and increases with increasing radial location. The magnitude of the velocity is greater in the downhill direction as this is the direction of the main flow. The radial velocity increases to a peak but must eventually decrease as the jet flow spreads. The magnitudes of the two rms velocity components are similar throughout the distribution. High turbulence intensity occurs at the stagnation point which contributes to a peak in the mean Nusselt number distribution at this location. A combination of high mean and root-mean-square velocities result in a secondary peak in the heat transfer distribution at $r/D = 2$.

Simultaneous surface heat transfer and local velocity measurements have been acquired at various locations in the impinging jet flow. In the case where the jet impinges at an angle of 60° , the measurements were acquired at $r/D = -1.3, -1.11$, (both uphill direction) and at $r/D = 0.37, 1.11$ (both downhill direction). These locations are indicated on the heat transfer and velocity distributions in figure 7. Spectral and coherence analysis was conducted on the individual velocity and heat transfer signals and the results are presented in figures 8 – 9 for the uphill locations and figures 10 – 11 for the downhill positions.

A small, local peak in the heat transfer and velocity spectra in figures 8 and 9 indicates that a vortex passes in the uphill direction at a Strouhal number of 0.6. This vortex is weak however as the relative magnitude of the peak in each spectrum is small. The coherence between the individual velocity signals and the heat flux is low also. Figures 10 and 11 present equivalent data for the downhill direction of the wall jet flow. It is apparent that the vortices that pass in this direction are stronger and have more of an influence on the heat flux. At a radial location of $0.37D$ from the geometric centre, peaks in all three spectra are evident at a Strouhal number of 0.6. At $r/D = 1.11$ however, this peak is only evident in the velocity normal to the surface and in the heat flux signal. This radial location highlights the effect that relatively small fluctuations in the axial direction can have on the heat flux. In all, the coherence between the axial velocity and the heat flux signal is greater than that between the radial velocity and the heat flux. This is especially noticeable in

figures 10 and 11, at locations where it was shown in figure 7 that the radial component has a greater mean velocity and similar rms velocity to the axial component. This is attributed to a stronger thermal gradient normal to the impingement surface (axial direction) rather than parallel to the surface (radial direction).

**Figure 8: $Re = 10000$; $H/D = 2$; $\alpha = 60^\circ$;
 $r/D = -1.30$**

**Figure 10: $Re = 10000$; $H/D = 2$; $\alpha = 60^\circ$;
 $r/D = 0.37$**

**Figure 9: $Re = 10000$; $H/D = 2$; $\alpha = 60^\circ$;
 $r/D = -1.11$**

**Figure 11: $Re = 10000$; $H/D = 2$; $\alpha = 60^\circ$;
 $r/D = 1.11$**

A similar analysis has been carried out for a jet impinging at an angle of 45° . Again, the distribution of the velocities and the heat transfer profiles are presented in figure 12.

(a) Velocity

(b) Nusselt Number

Figure 12: Velocity & Heat Transfer Profiles; $Re = 10000$; $H/D = 2$; $\alpha = 45^\circ$

Spectral and coherence analysis has been carried out for three locations in the downhill direction ($r/D = -0.81, 0.76, 1.41$) and is presented in figures 13 – 15. This focus on the downhill direction is primarily due to the low magnitude of the velocity in the uphill direction for the oblique angle of impingement of 45° .

The simultaneous data exhibit many of the characteristics discussed earlier for a jet impinging at 60° . However, at $r/D = -0.81$ each spectrum exhibits peaks at Strouhal numbers of 0.6 and 1.1. It is thought that vortices initiate at the higher frequency ($St = 1.1$) at the jet lip and undergo a merging process, resulting in larger vortices passing at a lower frequency ($St = 0.6$). Because of the proximity of the jet nozzle to the impingement surface at this radial location, it is thought that heat transfer is affected by vortices passing in both the uphill and downhill directions. The jet flow in the uphill direction has had insufficient time or space to undergo these merging processes completely and so two peaks remain in the velocity and heat transfer spectra. At the larger radial locations in the downhill direction, the distance from the nozzle lip where the vortices initiate to the location where the vortex impinges on the surface is sufficient for the vortex to complete the merging process.

**Figure 13: $Re = 10000$; $H/D = 2$; $\alpha = 45^\circ$;
 $r/D = -0.81$**

**Figure 14: $Re = 10000$; $H/D = 2$; $\alpha = 45^\circ$;
 $r/D = 0.76$**

**Figure 15: $Re = 10000$; $HD = 2$; $\alpha = 45^\circ$;
 $r/D = 1.41$**

This is evident from the fact that at locations in the downhill direction of the geometric centre, one dominant frequency peak appears in the spectra. This can be seen in figures 14 and 15. This peak occurs at a Strouhal number of 0.6 and therefore it is clear that the vortex merging process has been completed. At $r/D = 1.41$ the coherence between the axial velocity and the heat flux is particularly high, indicating the influence that velocity fluctuations normal to the surface have on the heat flux.

4. Conclusions

Results have been presented of the mean and rms Nusselt number distributions for a range of impinging air jet configurations. It has been shown that the magnitude of the fluctuations has an influence on the mean heat transfer from the impingement surface. Measurements of the fluid velocity in the wall jet close to the impingement surface have also been presented. Comparisons of the rms velocity and heat transfer distributions have shown that heat transfer fluctuations are caused by the turbulence in the wall jet boundary layer. Analysis of simultaneous heat transfer and velocity measurements has shown that velocity fluctuations normal to the impingement surface have a more significant influence on the heat transfer. Spectral analysis of the heat transfer and velocity signals has shown the effect of vortices on the heat transfer. At oblique angles of impingement the vortical characteristic of the flow and consequently heat transfer is different in the uphill and downhill directions. This is due to the proximity of the jet nozzle to the impingement surface. In the downhill direction the vortices have time and space to develop and pass as large scale low frequency vortices. In the uphill direction however, the vortices have not completed a merging process and pass intermittently as large scale low frequency vortices and as small scale high frequency vortices.

5. References

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