PIV measurements and convective heat transfer of an impinging air jet

by

T. S. O’Donovan\(^{(1)}\), D. B. Murray\(^{(2)}\) and A.A. Torrance\(^{(3)}\)

Department of Mechanical & Manufacturing Engineering,
Trinity College Dublin

\(^{(1)}\)E-Mail:odonovts@tcd.ie
\(^{(2)}\)E-Mail: dmurray@tcd.ie
\(^{(3)}\)E-Mail: atorrnce@tcd.ie

ABSTRACT

This paper reports on measurements of fluid flow and heat transfer for an impinging air jet with a wide range of test parameters; these include Reynolds Numbers (Re) from 10000 to 30000 and nozzle to impingement surface distance (H/D) from 0.5 to 8.

Both mean and fluctuating heat transfer profiles are presented for the range of variables considered. The heat transfer distribution exhibits secondary peaks at a radial position which is dependent on H/D. The PIV method has been used to measure the fluid flow for the free jet and for the jet in its impingement configuration. The local velocity and turbulence intensity profiles are in qualitative agreement with the heat transfer distributions.

The limitations of the PIV technique are addressed by using LDA measurements to achieve better spatial and temporal resolution. The heat transfer characteristics at the stagnation point are investigated in some detail and LDA data are used to relate the flow characteristic in the vicinity of the stagnation point to the heat transfer. It is well established that convective heat transfer at the stagnation point increases with increasing distance of the jet from the impingement surface. This is true up to a point just beyond the potential core of the jet. Both time averaged and fluctuating heat transfer data are reported and compared to the local flow velocity data. Again the local flow characteristics are in good agreement with the heat transfer. Differences between the two have primarily been attributed to the variation in air temperature in the stagnation region.

The use of both whole field, as shown in figure 1, and local flow measurement techniques has improved the understanding of the convective heat transfer mechanisms to the impinging jet.

\[\text{Velocity, } U/U_{exit}\]

\[\text{Turbulence Intensity, } \%\]

Fig. 1: Full Field Flow Measurements for Impinging Air Jet; Re=10000; H/D = 2
1. INTRODUCTION

An impinging air jet is known to yield relatively high local and area averaged convective heat transfer coefficients. Such a jet is of interest for the cooling of electronic components, gas turbine blades and manufacturing processes such as grinding. A grinding process, in common with other machining processes, produces large quantities of heat. This heat must be dissipated as high temperatures would have an adverse affect on the metallurgical composition of the workpiece. Grinding produces very high localised temperatures and as such the peak or stagnation point heat transfer is of particular interest in this investigation.

There are many parameters of an impinging jet that have an effect on the heat transfer. The two fundamental parameters include the jet Reynolds number and the nozzle to impact surface spacing. Several more parameters exist however. The angle of impingement has been investigated by Goldstein and Franchett [1], Yan and Saniei [2] and Beitelmal et al. [3]. It has been established that jet inclination has the effect of moving the stagnation point and therefore the location of the peak in the heat transfer distribution in the direction of the acute angle made between the jet and the surface. Collucci and Viskanta [4] and Garimella and Nenaydykh [5] have presented work conducted on the effects of nozzle geometry on the heat transfer. Arrays of impinging jets in various configurations have been investigated by Goldstein and Timmers [6]. More recently various methods of jet excitation have been studied. Hwang and Cho [7] and Liu and Sullivan [8] acoustically excited an impinging jet to show that the excitation frequency has the potential to either reduce or enhance the heat transfer. Jet excitation has also been achieved by controlling the shear flow by Hwang et al. [9]. The effects of crossflow, confinement and entrainment fluid temperature have also been the source of some investigation.

Numerical modelling and theoretical studies have also formed part of the extensive research into the area of jet impingement heat transfer. A theoretical study by Shadlesky [10] developed a solution for the heat transfer at the stagnation point.

More fundamental research has been conducted into the effects of turbulence on heat transfer. Jet exit turbulence intensity has been augmented by Gardon and Akfirat [11] by the use of turbulence grids at the jet exit. The flow along the plate from the geometric centre through the stagnation zone and the eventual formation of the wall jet are investigated by Lytle and Webb [12] and Hoogendoorn [13].

The current research is directed towards correlating the local fluid velocity and turbulence intensity in the wall jet to the mean and fluctuating heat transfer. The stagnation region has been investigated in further depth as previous studies have not quantitatively compared the flow to the heat transfer. It is anticipated that the measured turbulence intensity at the stagnation point will explain the variations in the heat transfer from the theoretical model of Shadlesky [23].
2. EXPERIMENTAL METHOD and PROCEDURES

Figure 2 (a) below depicts an overview of the experimental rig used in this investigation. The rig consists of a 550 mm × 475 mm × 5 mm horizontal copper plate that is heated from below by an electric silicone-rubber heating element. This approximates a uniform wall temperature boundary condition. A jet of air issues from a long (20 diameters) pipe with a 45° chamfer at its exit. The flow condition from such a nozzle is a hydro-dynamically fully developed turbulent jet with minimal entrainment. The airflow is metered using a volume flow meter to an accuracy of 1%. The meter consists of a Laminar Flow Element (L.F.E.), which is a restriction that is designed to force the air along a parallel path across which the differential pressure is measured and so the Poiseuille equation is valid. The flow meter is rated up to 400 l/min, which corresponds to a Reynolds number of 40,000 approximately for the nozzle diameter used (13.5 mm). However a Reynolds number up to 30,000 only is investigated here.

As illustrated in figure 2 (b), two flush mounted sensors are used for heat transfer measurements. An RdF micro-foil heat flux sensor is used for time-averaged data. This sensor consists of two thermopiles separated by a substrate of known conductivity and thickness. The difference in temperature across the substrate produces a voltage proportional to the heat flux. This sensor however has relatively poor spatial and temporal resolution. A Senflex hot film sensor is used in conjunction with a TSI Constant Temperature Anemometer (CTA) to acquire fluctuating heat transfer data. The sensor consists of copper leads on a thin (0.051 mm) Upilex S Polyimide film substrate. Nickel sensor elements are electron beam deposited onto the polyimide substrate to a thickness of < 0.2 µm. The CTA is essentially a Wheatstone bridge where the hot film forms one arm of the bridge. It is therefore possible to vary the temperature of the hot film by varying the resistance of another arm of the bridge. The output voltage is the voltage required to maintain the temperature of the bridge and is proportional to the power or heat dissipated from the bridge.

Two types of flow measurement have been carried out. Firstly, the flow velocity field has been measured and mapped using digital particle image velocimetry (DPIV). The DPIV system consists of a 15 mJ Nd:YAG double pulse laser and a double shutter PCO Sensicam. The resolution of the camera is 1280 x 1024 pixels and the minimum time between frames is 200 ns. Glycerine particles (5 µm diameter approx.) were used to seed the main flow and the entrained air of the jet. A plain and a cylindrical lens were used to illuminate a plane in the direction of the flow and through the centre of the jet. A two component fibre optic laser Doppler anemometer (LDA) system was used to determine instantaneous local velocities and turbulence intensities in flow regions identified from the PIV measurements as of particular interest. The LDA is a two component Dantec Argon Ion system (BSA 50) with beams of wavelengths 528 nm and 482 nm. It is a continuous laser that is used in burst mode and the system is capable of both forward and backscatter acquisition. Reconstruction of the irregularly sampled LDA signals was conducted using linear interpolation as the mean sample frequency was sufficiently high.
3. RESULTS AND DISCUSSION

Results have been obtained for a jet that impinges perpendicularly to a flat plate. Three Reynolds numbers have been considered, (10000, 20000, 30000). Nozzle to impingement surface spacings, normalised by the jet diameter, (H/D = 0.5; 1.0; 1.5; 2.0; 2.5; 3.0; 4.0; 5.0 6.0; 7.0; 8.0) were chosen as this range gives good resolution in the region within the potential core of the jet and also extends far enough beyond the potential core to examine the effects of a fully developed jet as the centreline velocity falls off and the turbulence intensity plateaus.

PIV was used to measure the flow field for the free jet. Absolute velocities normalised by the exit velocity and turbulence intensity for a jet of Re = 10000 are presented in figure 3 (a) below. Centreline velocity and turbulence based on the local absolute velocity has been extracted from the PIV data and this is presented in figure 3 (b) below.

The centreline velocity of this jet is constant and equal to the exit velocity up to 2.5 diameters from the nozzle. The turbulence intensity is also low, at approximately 5%, within this range. As the jet develops further however the centreline velocity decreases quite sharply. The shear layer penetrates the core of the jet and the turbulence intensity rises to approximately 25% at H/D = 7. A comparison is made by Gardon and Akfirat [11] between the free jet and the heat transfer of the same jet in an impingement configuration. There is reasonable correlation between the two however a more accurate comparison is that of the jet in its impingement configuration.

Several investigators including Gardon and Akfirat [11] and Hoogendoorn [13], have established that the heat transfer distribution from a surface to an impinging jet exhibits lateral peaks at low nozzle to surface spacings. These peaks are thought to be due to the development of the wall jet boundary layer. The heat transfer is a maximum at the stagnation point and decreases with increasing radial distance from the geometric centre. This is due to the thickening of the laminar boundary layer. As the wall jet escapes beyond the lip of the jet nozzle transition to turbulence occurs. The combined effects of sustained high local velocity and now high turbulence result in a peak in the heat transfer distribution. Beyond this peak, at greater radial distance from the stagnation point, the heat transfer decreases as the increasing turbulence intensity can no longer compensate for the decreasing fluid velocity and local temperature difference.

It is apparent from the heat transfer distributions in figure 4 below that the magnitudes of the lateral peaks are reliant on both Re and H/D whereas the radial locations of these peaks depend only on H/D. The peak radial position increases linearly with increasing H/D for the range considered.
To achieve a better understanding of the heat transfer mechanisms PIV measurements were conducted on the jet in several impingement configurations. It was intended that the heat transfer be correlated with the actual flow conditions for the impinging jet and not merely with the flow field of a free jet. Figure 5 below is one such comparison of the fluid flow field with time averaged and fluctuating heat transfer distributions.
The absolute velocity measurements in figure 5 above clearly depict the stagnation point and the absolute velocity variation along the plate. The flow accelerates from the stagnation point along the plate but soon after this the velocity decays. The rms velocity is a local minimum at the stagnation point; it is a maximum at the location where the shear layer impinges on the plate. The induced turbulence in the wall jet remains high at large radial positions. The mean and fluctuating Nusselt number distributions are in qualitative agreement with the flow data. However a peak in the fluctuating heat transfer occurs at a greater radial location than the location of maximum rms velocity. A limitation of the PIV has therefore been realised. Due to geometric constraints accurate PIV data were not obtained for the distance of 0.1 diameters above the impingement surface. It has therefore been difficult to relate the position of the peak in the heat transfer distribution to the location of the most favourable combination of high velocities and turbulence intensities.

It is clear however that the boundary layer that forms within the wall jet is controlled by the impinging free jet. The impinging jet ensures laminar flow in the stagnation region. Once the wall jet is free from the pressure gradient induced by the impinging jet, i.e. once it has escaped the lip of the nozzle, the transition to turbulent flow with boundary layer growth is aided by the existing high levels of turbulence from the free jet shear layer. Absolute velocities and rms velocities are presented in figure 6 below for the 11 nozzle to impingement surface spacings.

Fig. 6: Full Field Absolute and R.M.S. Velocities for H/D = 0.5 to 8.0; Re = 10,000

It is evident that the velocity gradient near the stagnation region is decreasing with increasing height of the nozzle above the plate. The jet has also spread more at large H/D. It appears that the stagnation point rms velocity does not change considerably until the impingement surface is beyond the potential core of the jet. Thereafter it increases and this is in accordance with the free jet flow measurements. The linear relationship between the radial location peak in the heat transfer distribution, shown in figure 4, and H/D appears to have the same slope as the line of maximum turbulence intensity in the free jet (figure 3). At larger H/D (> 3) the rms velocity is relatively uniform across the plate and the heat transfer distribution no longer exhibits lateral peaks. The heat transfer distribution simply decays from a peak at the stagnation point as the local velocity and temperature difference decrease towards zero.

Laser Doppler Anemometry was employed to investigate specific points of interest as it has better temporal and spatial resolution than PIV. One such point of interest is the stagnation point. In a theoretical correlation by Shadlesky [10], it was suggested that the heat transfer at the stagnation point is independent of nozzle to impingement surface spacing. This is contrary to experimental evidence presented by Lee et al. [14] and others because the assumption that the flow in the neighbourhood of the stagnation point is laminar regardless of the mainstream flow only holds true for H/D < 2. This
has been shown to be true in several experimental investigations including one by Liu and Sullivan [8]. With increasing H/D the flow in the stagnation region is less confined by the impinging jet and the flow at the geometric centre does not truly stagnate but has a zero velocity on a time averaged basis.

Figure 7 below depicts the variation in mean and fluctuating heat transfer parameters with H/D. The Nusselt number is a minimum where the fluctuating heat transfer is lowest and the flow stagnates. As the jet to plate spacing increases so too does the heat transfer. For all three Reynolds numbers the heat transfer is relatively constant at low H/D. This is well within the potential core as can be seen from free jet results presented in figure 3. In this case the fluctuating heat transfer parameter is also relatively low and constant.

![Fig. 7: Variation of Stagnation Point Nusselt Number and Nu Fluctuations with H/D](image)

It is expected that the potential core length increases with Reynolds number and this explains why the peak in the heat transfer occurs at H/D=6 for Re=10000 and at higher H/D for the Reynolds numbers of 20000 and 30000. After H/D=2.5 both the time averaged and fluctuating heat transfer rises sharply for Re=10000. Local velocity data from LDA measurements in the stagnation region are shown in figure 8 for the Reynolds number of 10000. The measurement volume of the two component LDA was 1.3 mm above the plate and as such the velocity of the jet normal to the impingement surface was greater than zero. The turbulence intensity is calculated by dividing the rms velocity in the x-direction (parallel to the impingement surface) by the absolute arrival velocity (normal to the impingement surface). Gardon and Akfirat [21] used the arrival velocity, measured from an equivalent position in the free jet to calculate the turbulence intensity at the stagnation point. Lytle and Webb however used the mean jet exit velocity in this calculation. It was decided in the present investigation that the local absolute fluid turbulence intensity was a more appropriate comparative factor for the fluctuating heat transfer.

There is a reasonable correlation between the fluctuating heat transfer and the local fluid turbulence intensity. For the range of nozzle to plate heights considered the turbulence intensity varies from 6 – 27 %. The normalised heat transfer fluctuations however vary from 12 – 25 %. The shapes of the curves are also well correlated as they are both relatively
constant from 0.5 to 2.5 diameters. Thereafter there is a sharp increase that continues in an almost linear fashion for the range of heights considered.

\[\text{Fig 8: X-Component R.M.S. Velocity 1.3 mm above the Stagnation Point; Re = 10000}\]

Differences between the fluctuating heat transfer and fluid flow parameters can be attributed in part to the distance between the LDA measurement volume and the impingement surface, and to the fact that the fluid flow is measured in only the x-direction. Another significant factor which would result in the two graphs being at variance to one another is the variation in local air temperature at the stagnation point.

At low H/D when the flow truly stagnates, instead of having zero velocity on a time averaged basis only, the mean heat transfer at the stagnation point is low. This is not just because the turbulence level is low but also because the air stagnates and consequently its temperature rises, reducing the local temperature difference. Even though the rms value is comparable to that of the flow, when it is normalised by the mean heat transfer at the stagnation point the ratio of the fluctuations is higher. The results show that this is only the case up until 2.5 diameters. This is as expected because beyond this height the shear layer has penetrated to the core of the jet and the stagnation point is now a time averaged stagnation point only, i.e. there is some fluid motion in the horizontal direction. No longer is the effect of conduction to stagnant air a factor which reduces heat transfer locally and the ratio of the rms fluid velocity to its mean correlates significantly better with the ratio of the rms Nu to the mean Nusselt number.

4. CONCLUSIONS

Time averaged and fluctuating heat transfer data have been presented for a range of impinging jet configurations. To further our understanding of the convective heat transfer mechanisms present in such configurations laser techniques were employed to measure the fluid flow characteristics. Initially PIV was used to obtain full field flow measurements of the air jet impinging on the heated flat surface. There was qualitative agreement between the heat transfer distributions and the flow along the surface. The PIV also revealed how the wall jet boundary layer formed.

Limitations of the PIV method for local velocity measurements were soon realised however and LDA was employed to look more closely at the correlation between flow and heat transfer at the stagnation point. The stagnation point heat transfer is virtually constant and independent of H/D for as long as the flow in the stagnation region can be considered laminar. This is only true however for low H/D < 2.5. Beyond this limit the average velocity at the geometric centre is zero however the instantaneous velocity is not. Referring to horizontal velocity components only here? The geometric centre can no longer be considered a true stagnation point for H/D > 2.5.

Fluctuating heat transfer correlates well with the turbulence intensity measured close to the geometric centre. Differences between the two have been attributed in the main to variations in the air temperature close to the plate. Further work in this area will include the correlation of heat transfer to local fluid velocity at various radial distances from the geometric centre.
REFERENCES


