FLUCTUATING AND TIME AVERAGED HEAT TRANSFER CHARACTERISTICS OF A STEADY JET AT LOW REYNOLDS NUMBERS.

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ABSTRACT. A study has been carried out to compare fluctuating and time averaged heat transfer measurements of a steady impinging air jet at low Reynolds numbers. The steady jet issued from a 5mm diameter orifice plate with air being supplied by a compressor via a plenum chamber, and being controlled by a mass flow controller. Tests were conducted for Reynolds numbers ranging from 1000 to 4000, and for non-dimensional surface to jet exit spacings (H/D) from 1 to 6. Initial studies have shown a distinct difference between trends in time-averaged and fluctuating Nusselt number profiles. Whereas the time-averaged profiles show a uniform/linear increase with Re from 1000 to 4000, the fluctuating Nu profiles seem to indicate a transition from a nearly laminar jet at Re=1000 to turbulent jets at Re = 3000, 4000.'

Keywords: Steady Jet, Fluctuating, Nusselt Number, Reynolds number, Impinging, Velocity, PIV

INTRODUCTION

Steady impinging jet heat transfer has received considerable attention over the last few decades particularly in many industrial and engineering applications (e.g. manufacturing, material processing and electronics cooling). This is largely due to the fact that an impinging air jet is known to yield relatively high local and area averaged convective heat transfer coefficients, O'Donovan et al. [1]. Hollworth and Durbin [2] investigated the impingement cooling of electronics and Babic et al. [3] used jet impingement for the cooling of a grinding process. In these, and in other cases research has been conducted with a specific application as the focus but there has also been many investigations into the fluid flow and heat transfer characteristics. These have led to the identification of several parameters which influence heat transfer on the impingement surface.

The flow and heat transfer characteristics of an impinging jet are affected by numerous parameters like confinement and impingement angle but the two parameters with greatest influence are the jet Reynolds number, and the surface to jet exit spacing (H/D). Impinging jets are commonly divided into three regions on the basis of the flow structure (i) the free jet region, (ii) the stagnation region and (iii) the wall-jet region, as illustrated in figure 1. Numerical modelling and theoretical studies have also formed part of the extensive research into jet impingement heat transfer; a theoretical study by Shadlesky [4] developed a solution for heat transfer at the stagnation point. More fundamental research has been conducted into the effects of turbulence on heat transfer. Jet turbulence intensity has been augmented by Gardon and Akfirat [5] 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 [6]. Although jet impingement heat transfer has been studied extensively by many researchers including O'Donovan

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et al [7], the amount of information available for low Reynolds number jet flows, in the laminar to turbulent transition range, is more limited. In particular the approach of examining both the fluctuating and time averaged heat transfer characteristics at low Reynolds numbers has yet to be explored fully.

The purpose of this study is to acquire an understanding of how steady impinging air jets behave at low H/D values and low Reynolds numbers, by comparing time averaged and fluctuating heat transfer data, with the use of Particle Image Velocimetry (PIV) to elucidate the heat transfer results.

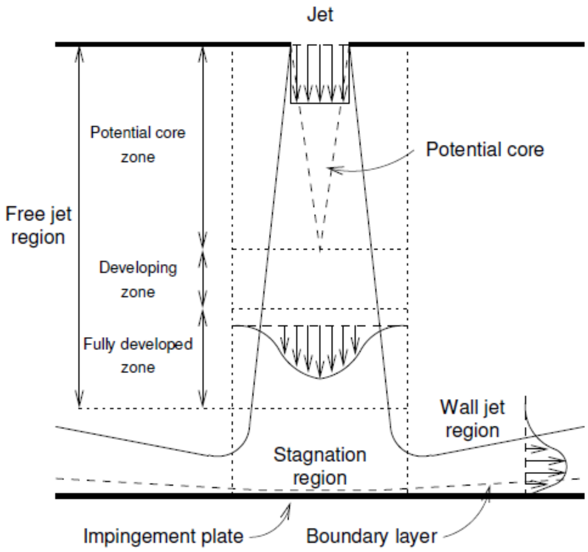


Figure 1: impinging jet flow regimes [8]

EXPERIMENTAL SETUP

There are three main elements that make up the experimental rig, these being the steady jet, heated, instrumented impingement surface and the particle image velocimetry (PIV) system. The impingement surface and the jet are mounted on independent carriages that travel on orthogonal tracks; the carriage for the impingement surface is moved using a computer controlled traverse, whereas the carriage for the jets is moved using a manually operated lead screw actuator. The instruments associated with the heated impingement surface are two single point heat flux sensors and two thermocouples. The rig design is presented in figure 2.

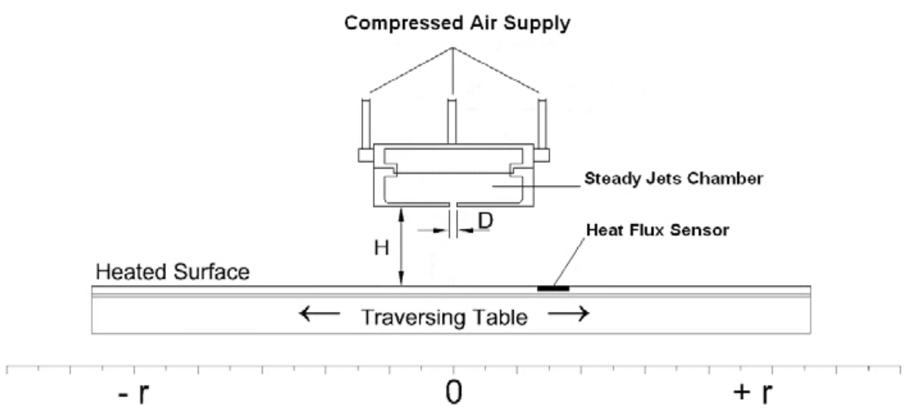


Figure 2: Schematic of the steady jet apparatus

The cavity body and orifice plate are constructed from a thermoplastic and were fabricated using a method of rapid prototyping called fused deposition modelling. The cavity has an approximate volume of 100cm^3 with an orifice diameter of 5mm. Care was taken to ensure a well defined outlet velocity profile with the use of a flow straightener. The steady jet is clamped onto a carriage, which allows for height adjustment from 1 to 6 diameters above the impingement surface. The jet is fixed at a normal angle of impingement (90°).

The steady jet was formed from a constant compressed air source that was maintained at flow rates corresponding to jet Reynolds numbers ranging from 1000 to 4000 by an MKS® Instruments 1579A Digital Mass Flow Controller, which has an accuracy of 1% of full scale and a repeatability of ± 0.2% of full scale (300 slm). The jet impinges onto a surface that consists of a 5mm thick flat copper plate measuring 425mm x 550mm. To the underside of the plate a silicon rubber heater mat is glued with a thin layer of adhesive. The mat is approximately 1.1mm thick. The underside of the plate and mat assembly is insulated from the surroundings. The entire assembly is such that it approximates a uniform wall temperature boundary condition. The system is typically operated at a surface temperature of 40°C.

Two sensors were utilised in this study (i) an RdF Micro-Foil® heat flux sensor, and (ii) a Senflex® hot film sensor. The first sensor measures the temperature differential across a known thermal barrier using a differential thermopile. The heat flux through the sensor is based on the following equation:

$$\ddot{q} = k_s \frac{\Delta T}{\delta} \tag{1}$$

where ΔT is the temperature difference across the thickness (δ) of the barrier and k_s is the thermal conductivity of the barrier (kapton). A single pole T-type thermocouple is also embedded in this sensor to measure the local temperature.

Both sensors are positioned centrally on the plate, and together with the jet and plate carriage arrangement, allow for heat transfer measurements beyond 20 diameters from the geometric centre of the jet. In this study, testing has only been concerned within the region of 1 to 6 diameters from the geometric centre of the jet.

The Senflex® hot film sensor operates in conjunction with a Constant Temperature Anemometer (CTA) to measure the fluctuating heat flux to the impinging jet. This sensor consists of a nickel sensor element that has been electron beam deposited onto a 0.051mm thick Upilex S polyimide film. The hot film element has a physical area of approximately 0.1mm x 1.44mm and is less than 0.2µm thick. The sensor has a typical cold resistance of between 6 and 8 Ohms. Two copper strips are also deposited on the film; these provide terminals for connection to the CTA. These strips have a resistance of approximately 0.002 Ω /mm. The hot film is maintained at a slight overheat (≈1°C) above that of the copper plate using a Dantec StreamLine CTA. The power required to maintain the film at this overheat is equal to the heat actively being dissipated from the film. The CTA essentially acts as a Wheatstone bridge where the hot film acts as one resistor in the bridge. The resistance of the film varies with temperature and therefore, this film temperature can be controlled by varying a decade resistance which forms one arm of the Wheatstone bridge. The square of the voltage required to maintain the film at a constant temperature is proportional to the heat transferred to the air as described in equation (2):

$$q_{dissipated} \propto \frac{V_{out}^{2}}{R} \tag{2}$$

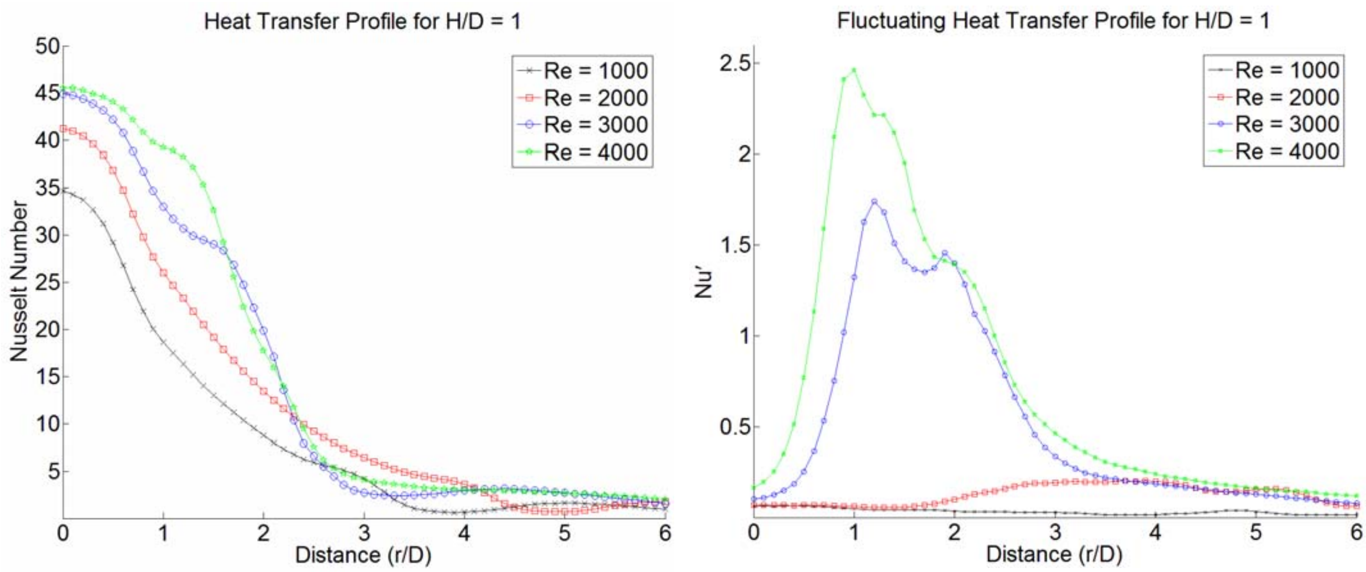
Time averaged velocity measurements have been performed using time resolved Particle image velocimetry (PIV). PIV has provided insight into both velocity and fluctuating nature of the jet flow at different Reynolds numbers.

The PIV system comprises of a Quantronix Darwin-Duo Nd:YLF twin cavity laser and a LaVision HighSpeedStar6 CCD camera.(1024x1024 pixels, 12 bit) with a 105mm lens. A glycol-water aerosol is used for seeding, with particle diameters between 0.2 and 0.3 µm. The velocity fields have been processed with LaVision's Davis 7 software using multi-pass cross-correlation with an interrogation window size decreasing from 64x64 to 16x16 pixels at 50% overlap.

RESULTS AND DISCUSSION

Results have been obtained for a jet that impinges perpendicularly to a flat plate. Four Reynolds numbers have been considered (1000, 2000, 3000, and 4000) and four surface to jet exit spacings (H/D = 1, 2, 4, and 6) were selected as this range provided a clear indication of the effects on impinging jets at low Reynolds numbers and low H/D values.

The results are presented as time averaged heat transfer distributions and fluctuating heat transfer distributions for each H/D from 1 to 6. A direct comparison between the time averaged and fluctuating heat transfer profiles of the jet will be made, with reference to corresponding PIV data where available.

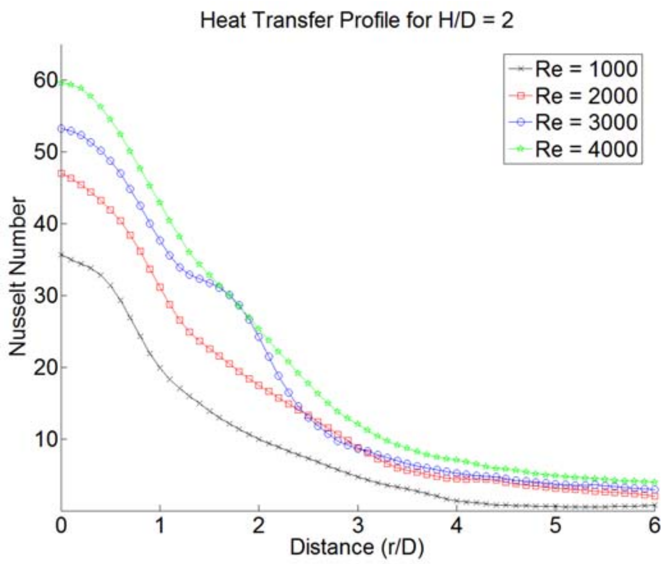


(a) Time averaged heat transfer distribution (b) Fluctuating heat transfer distribution

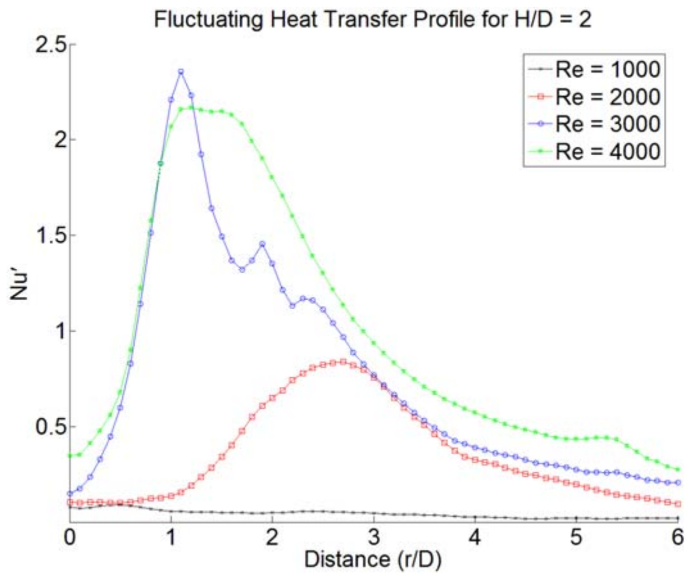
Figure 3: Time averaged and fluctuating heat transfer profiles H/D=1

Figures 3a and 3b illustrate the heat transfer profiles for the steady impinging jet at matching Reynolds numbers and jet exit to plate spacings (H/D). the presence of secondary peaks at a radial distance of approximately 1 < r/D < 2 for the time averaged heat transfer distribution there appears to be a transition to turbulence at Reynolds numbers 3000, 4000. This is clearly evident in the fluctuating heat transfer distribution (figure 3b) at Re = 3000 and 4000. There is quite a large fluctuation in heat transfer present between Re = 3000 and 4000 in figure 3b.

It is worth noting that at this H/D the orifice plate imposes a strong degree of confinement on the jet. As the height of the jet above the impingement surface is increased there is less likelihood of confinement occurring, therefore allowing the jet to fully propagate, and maximum heat transfer to take place.



(a) Time averaged heat transfer distribution



(b) Fluctuating heat transfer distribution

Figure 4: Time averaged and fluctuating heat transfer profiles H/D=2

Figure 4 shows the heat transfer results for H/D = 2. From figure 4b, at a Reynolds number of 3000 there is what appears to be a laminar core which results in a transition to turbulence at r/D of approximately 1.5. This is reflected in a local increase in time averaged heat transfer, as shown in figure 4a.

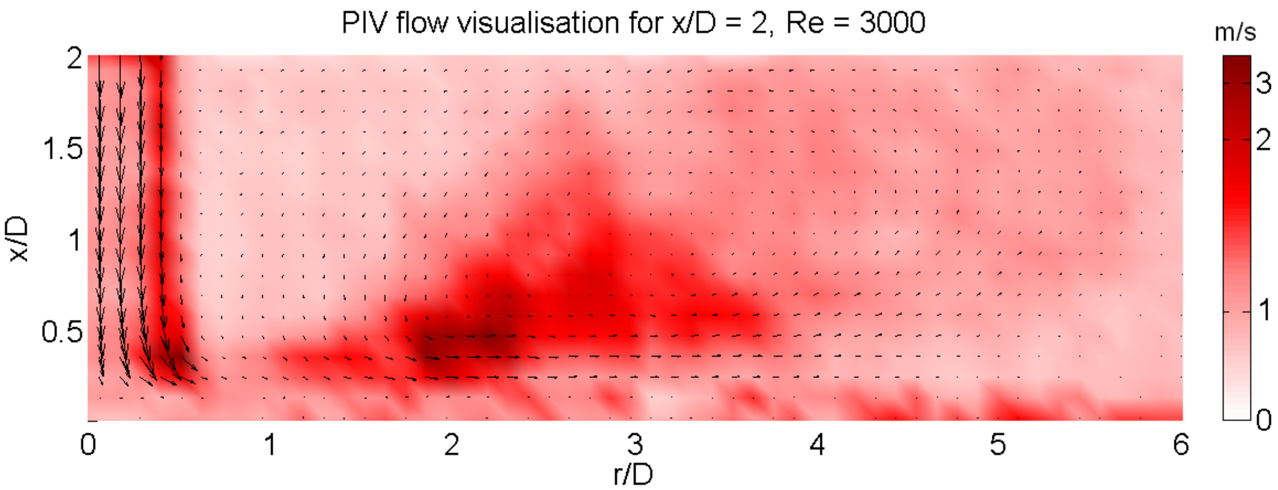
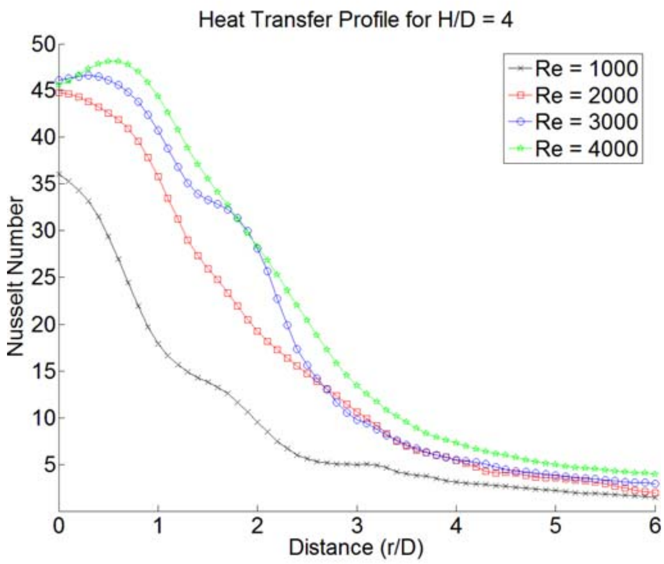


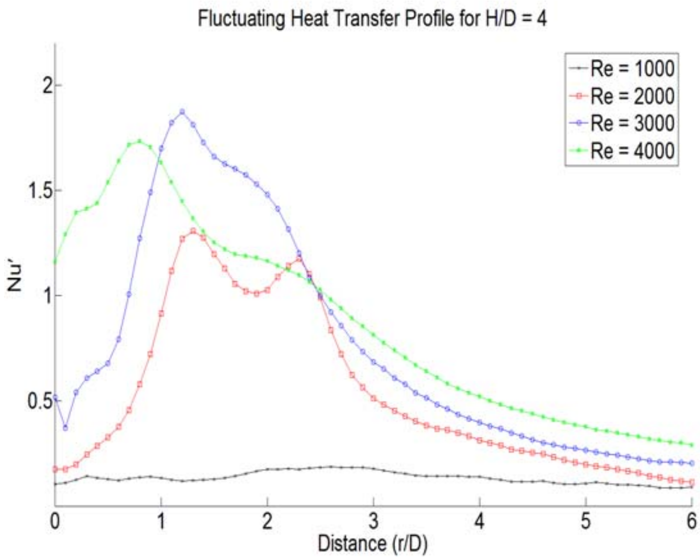
Figure 5: PIV fluctuating velocity plot x/D=2, Re=3000

The PIV fluctuating velocity plot is a combination of the mean velocity vectors and the root mean square velocity colour map. The PIV fluctuating velocity scale is defined using a custom colour map where deep red signifies large fluctuations and white represents zero fluctuation. This scale was defined by using the mean and root mean square velocity maps that were outputted from DaVis 7. The R.M.S. velocity scale is defined using the colour bar on each PIV image. The mean velocity scale is from 0 – 17 m/s.

Figure 5 shows the PIV time average velocity plot for x/D = 2 and Re = 3000. It can be seen that there are very large fluctuations in the shear layer before impingement and subsequently a transition to turbulence. Since the jet is laminar at the parameters shown, transition to turbulence after impact and this can be seen in the fluctuating heat transfer and PIV data at 0.5 < x/D < 4. The core is maintained right up until impingement; there are relatively low velocity fluctuations in the core, a coherent core which results in low fluctuating heat transfer. Where the jet impinges an area of high fluctuation are evident. Areas of recirculation can be clearly seen in the figure 5. The heat transfer fluctuations are caused by vortices in the shear layer impacting upon the plate, this can be seen at 0.5 < x/D < 2 for both the PIV data and fluctuating heat transfer.



(a) Time averaged heat transfer distribution



(b) Fluctuating heat transfer distribution

Figure 6: Time averaged and fluctuating heat transfer profiles

In figure 6 for H/D of 4, there is what appears to be a local minimum at a Reynolds number equal to 4000 on the time averaged heat transfer distribution which correlates with the fluctuating heat transfer distribution. Overall the fluctuations in heat transfer are higher in the stagnation zone, which could mean turbulence starts earlier in the jet due to the higher Reynolds number. There appears to be a loss of momentum in the jet at this H/D, which could be due to a diminished potential core.

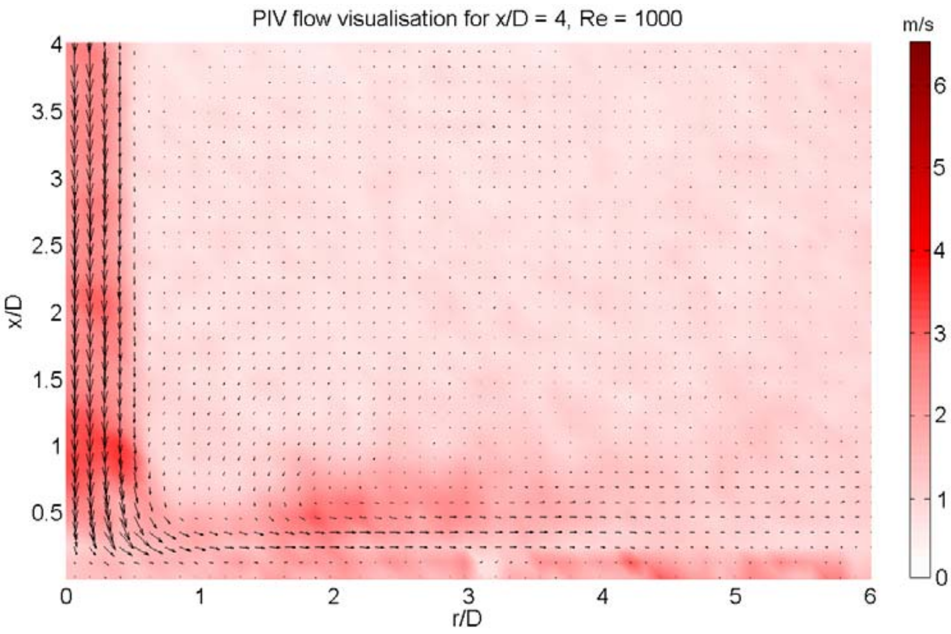


Figure 7: PIV fluctuating velocity plot x/D=4, Re=1000

Figure 7 shows very slight fluctuation in velocity in the core, this is due to a reduced potential core from a loss of momentum at H/D = 4 and low Reynolds number.

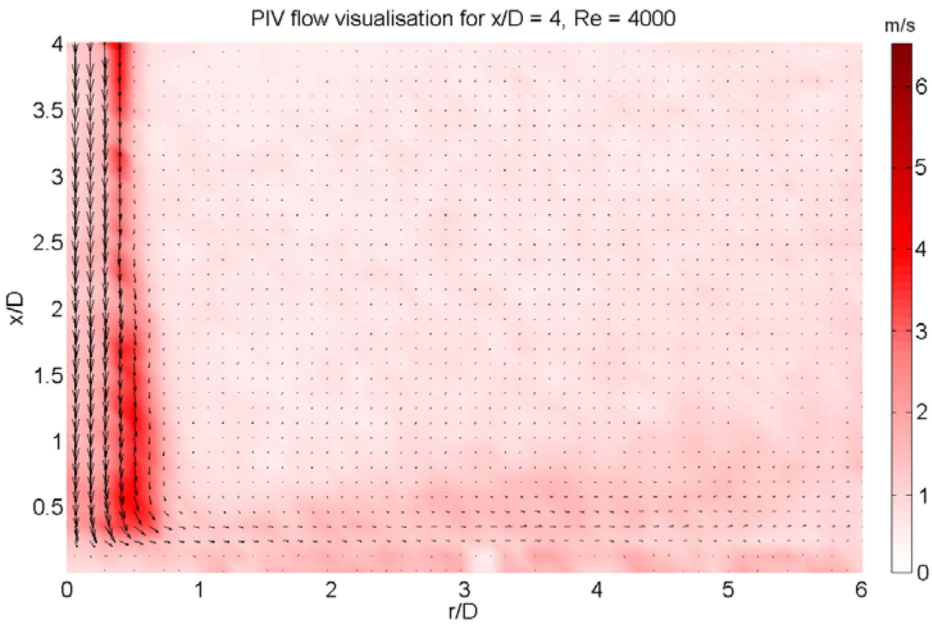
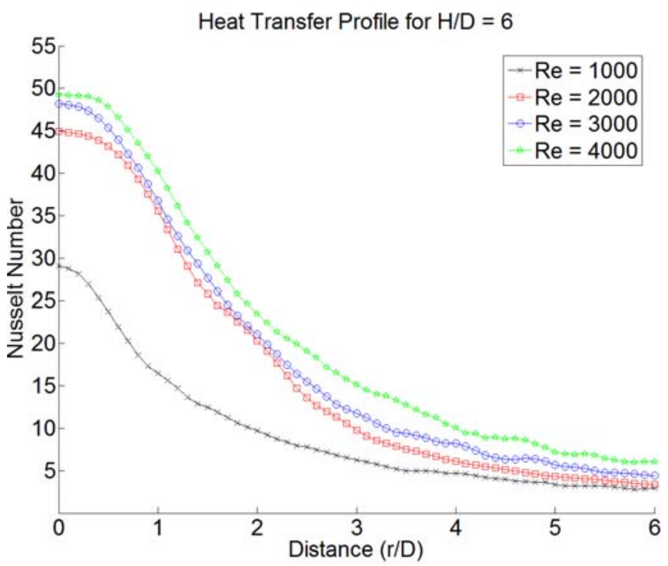
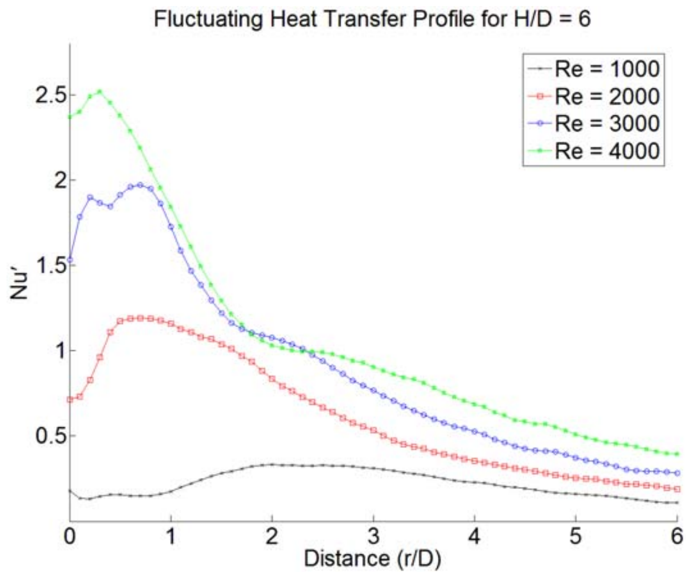


Figure 8: PIV fluctuating velocity plot x/D=4, Re=4000

The potential core is maintained at a greater height above the surface due to increased flow momentum, the lower fluctuating velocity in the core corresponds with the low local fluctuating heat transfer (figure 6b) observed at this jet exit to plate spacing and Reynolds number. A turbulent shear layer is observed at this height



(a) Time averaged heat transfer distribution



(b) Fluctuating heat transfer distribution

Figure 9: Time averaged and fluctuating heat transfer profiles

Comparison of figures 6a and 9a helps to illustrate the effects that jet exit to plate spacing have on the jet. A loss of momentum for Reynolds number of 1000 is apparent at H/D of 6, as the Nusselt numbers for this Reynolds number are lower than observed in figure 6a at the H/D = 4. The distributions in figure 9a show no evidence of secondary peaks and the Nusselt numbers reduce uniformly with increasing r/D, which indicates that the jet has been able to fully propagate; this is due to the higher jet to plate spacing allowing there to be a more fully developed flow. In figure 9b the Reynolds number of 2000 shows low levels of heat transfer fluctuation upon impingement. It is noted that Re = 1000 remains laminar throughout the H/D values tested (1, 2, 4 and 6)

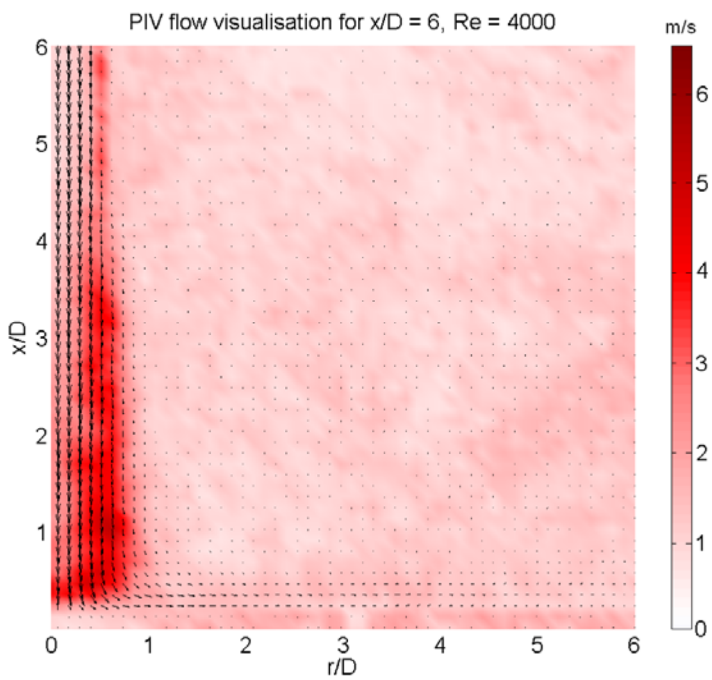


Figure 10: PIV fluctuating velocity plot H/D=6

Figure 10 shows a core that is almost diminished by the time it impinges on the surface and has greater fluctuating velocities. Less momentum in the potential core means that the core is less coherent and more turbulent before impingement, this is indicated by the intensity of the colours and this is also indicated in figure 9b for the fluctuating heat transfer profile. It is worth noting that there are lower fluctuations in the wall jet region for all PIV fluctuating velocity plots.

CONCLUSIONS

Results have been presented for time averaged heat transfer, fluctuating heat transfer and PIV fluctuating velocity plots relating to an axially symmetric impinging air jet. It has been shown that at low jet exit to plate spacing the time averaged heat transfer distribution in the radial direction shows evidence of secondary peaks. These peaks have been reported by several investigators including Farrelly et al [9] and have been attributed, in general, to an abrupt increase in turbulence in the wall jet boundary layer.

The fact that the peaks occur at approximately the same location for both Nu and Nu' distributions indicates a correlation between the heat transfer fluctuations and the mean heat transfer. Increasing Nusselt numbers for the time averaged heat transfer plots can be seen with increasing H/D at large r/D values, due to less confinement.

Peak Nusselt numbers recorded for the range of jet exit to plate spacings occur at an H/D = 2. An increase in H/D results in a loss of momentum and the onset of turbulent flow is reflected in the fluctuating heat transfer distributions at H/D = 4 and above. It is noted from the PIV data presented that there are lower fluctuations in the wall jet region.

NOMENCLATURE

Nu' – Fluctuating heat transfer

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