PULSATING CIRCULAR AIR JET IMPINGEMENT HEAT TRANSFER

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ABSTRACT

The purpose of this study is to determine the effect of pulsating frequencies on the local and average heat transfer characteristics of a heated circular air jet. Pulsation of the air jet was produced by a rotating cylinder valve mechanism at frequencies between 10 to 80 Hz. Flow structures of the heated steady and pulse single circular axisymmetric air jet velocity are measured using a calibrated hot-wire anemometer and presented in non-dimensionalized form. Reynolds number was set at 16 000, 23 300 and 32 000. The heat flux of the heated air jet impinging on the plate was measured using a heat flux sensor at different radial positions. Results obtained show that the average pulsed jet Nusselt number was higher than the average steady jet Nusselt number for all values of frequencies due to the higher localised heat transfer of the pulsing jet. This was the result of enhanced turbulence intensity due to pulsing the jet. The stagnation point heat transfer of the pulsed jet obtained was lower for the same frequency range due to the small turbulent intensity at this position.

Keywords : *Flow Structures, Heat Flux, Pulsating Frequency, Pulse Jet, Turbulence Intensity*

1.0 INTRODUCTION

High convective heat transfer coefficient is a very important factor that leads to the many usage of impingement jets in industry for heating and cooling purposes. Applications of impinging air jets include the cooling of electronic equipment, aircraft engine nacelle and blade, drying of textiles, annealing of metals and tempering of glass. Extensive research has been conducted on steady impinging jet to understand their heat and mass transfer characteristics. Numerous studies and reviews on the subject of steady jet heat transfer have been published over the last several decades [1-3]. However, heat transfer in pulsating flows has been the subject of renewed interest in recent years since the present of flow pulsations has been found to increase the heat transfer coefficients.

Earlier findings regarding the effect of pulsation on heat transfer have been conflicting due to many different factors. Pulsing a flow is widely believed to increase the heat transfer rate but in some cases the literature shows that heat transfer decreases. Test carried out by Nevins and Ball [4] on heat transfer between a flat plate and a pulsating jet showed that no significant heat transfer enhancement was obtained by using a pulsed air jet. The test was conducted at $1200<Re<120\ 000$, $10^{-4}<St<10^{-2}$, and nozzle to plate spacing between 8 to 32 nozzle diameters. Nevins and Ball [4] did not document the extent of secondary flow structures in their experiments and since the experiment was studied at a very low Strouhal number this might affect their ability to demonstrate pulsed flow heat transfer enhancement. The application of pulse air jet was left dormant for many years due to this earlier finding and the difficulty of accurately controlling many pulse air jet parameters.

Recently, more researchers started to study experimentally and numerically the effect of flow pulsations on heat transfer enhancement. Sheriff and Zumbrunnen [5] investigated experimentally the effect of flow pulsation on cooling performance using arrays of jet. The present of coherent structures was reported but no significant enhancement with respect to the heat transfer characteristics was recorded.

Kataoka and Suguro [6] show that stagnation point heat transfer for axisymmetric submerged jets is enhanced by the impingement of large-scale structures such as vortex rings on the boundary layer which occurred in pulse flow. Further tests carried out by Sailor et al. [7] on the effect of duty cycle variation on heat transfer enhancement for an impinging air jet showed significant heat transfer enhancement. In this test, the effect of traditional variables such as jet to plate spacing, Reynolds number and pulse frequency were studied together with a new parameter, duty cycle representing the ratio of pulse cycle on time to total cycle time. Duty cycle was shown to have a significant effect on the heat transfer enhancement.

Mladin and Zumbrunnen [8] investigated theoretically the influence of pulse shape, frequency and amplitude on instantaneous and time-averaged convective heat transfer in a planar stagnation region using a detailed boundary layer model. They reported that there exists a threshold Strouhal number, $St>0.26$ below which no significant heat transfer enhancement was obtained. Results obtained by Zumbrunnen and Aziz [9] on the effect of flow intermittency on convective heat transfer to a planar water jet impinging on a constant heat flux surface reinforces this finding. This experiment carried out at $St>0.26$ found that local Nusselt number increases by up to 100%. However, Sailor et al. [7] used Strouhal number between 0.009 and 0.042 and still recorded significant enhancement in stagnation point heat transfer for pulse flow.

Azevedo et al. [10] investigated impingement heat transfer using a rotating cylinder valve for a range of pulse frequency. The results show that heat transfer degraded for all frequencies. In their experiments, velocity profile of the pulse jet shows the existence of a two-peak region for every flow cycle. This results in disturbance to the pulse flow and affects the flow structure and heat transfer. The dependence of pulse characteristics on convective heat transfer was discussed by Mladin and Zumbrunnen [8].

In order to study the heat transfer characteristics of an impinging pulse air jet, the characteristics of a non-impinging pulse air jet needs to be initially understood. Farrington and Claunch [11] carried out a test to determine the influence of flow pulsations on the flow structures of an unforced planar jet with Re=7200 and $0<St<0.324$. The results of the test were captured using infrared imaging and smoke-wire visualization. They concluded that for pulsating jets, the vortices were larger than the steady jet and occurred closer to the nozzle. These larger vortices resulted in an increased entrainment and led to a wider angle of the potential core. Jets with large amplitude of pulsations entrained surrounding fluid more rapidly and decayed more quickly than steady jets. An increase in turbulence intensity can be associated with the pulse decay.

In pulsed flows, the size and formation of coherent structures are influenced by the amplitude and frequency [8, 12]. Large coherent flow structures can evolve from shear layers formed between a free jet flow and a surrounding fluid. The formation and interaction of flow structures can be influenced by the mixing within the boundary layer and a marked increase in turbulence intensities has been noted with pulse flows. Recent findings [7] on the enhancement of heat transfer due to pulse air jets have encouraged new research in this subject. Comprehensive data showing the effect of pulse frequency on local and average heat transfer profile are still limited and there is need of further investigation.

The purpose of this study is to investigate steady and pulsating single circular jet heat transfer characteristics. The focus of the study is given on the effect of flow pulsation frequencies on the average and stagnation Nusselt number. Comparison between steady and pulsed jet heat transfer was discussed in details together with other published results. In this paper the stagnation point Nusselt number of a pulse jet means the time average value at the impingement point of the jet axis. The local Nusselt number of a pulse jet is the time average at a point on the impingement surface. The local Nusselt number is assumed to be radially symmetrical about the stagnation point. The average Nusselt number of a pulse jet is both a time average and an area average over the impingement surface. The total heat flux is proportional to the average Nusselt number.

2.0 EXPERIMENTAL APPARATUS AND PROCEDURE

A. Pulse Flow System

Figure 1 shows the schematic diagram of the experimental test set-up. The pressurised air used in the experiment is supplied through the boiler house. The air is continuously fed through a permanent piping to two air storage tanks. The highest pressure that could be achieved in the tanks was 5 bar. The air contained in the tank was piped to a common manifold to facilitate control. The supply pipe of the air storage tanks are controlled by a stop valve. A pressure regulator was placed in between the air heater and the stop valve to regulate the supply of air. A vortex flow meter is placed just downstream of the pressure regulator and is used to measure the mass flow of the air jet impinging on the surface of the plate. A Secomak model 15/2 air heater is used to heat the air jet. The maximum power of the heater is 18kW with an associated maximum air temperature of 300˚C. The air heater is wired to the 3-Phase Thyristor Controller. The controller is connected to a 3-phase power supply of 440 Volts, 50Hz.

The pulse air jet is generated using a rotating cylinder valve driven by an electric motor controlled by an electronic motor controller. The system consists of a rotating cylinder enclosed inside a block aluminium alloy body. A 20 mm diameter hole was bored in the rotating cylinder normal to its axis to allow air passage. The aluminium alloy body is fixed to the supporting shaft and has a 40.2 mm hole diameter bored through its centre. The rotating cylinder has a diameter of 40 mm and it was aligned inside the body such that a 0.1mm radial clearance is achieved between the stationary body and the cylindrical valve. This led to minimal leakage through the gap with the valve is in the closed position. Figure 2 shows a schematic diagram of the rotating valve pulse jet system.

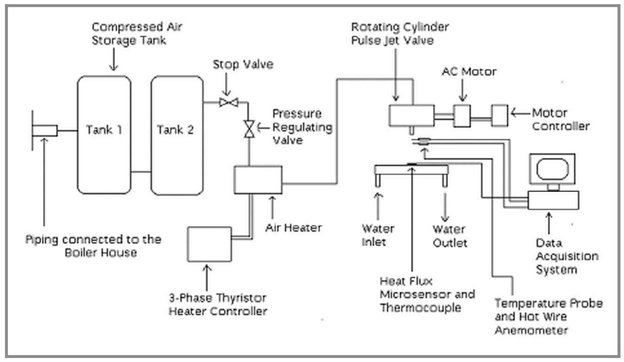

Figure 1: Schematic Diagram of the Experimental Test Set-up

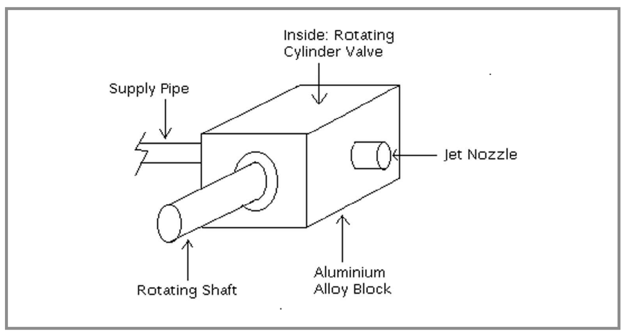

Figure 2: Schematic Diagram of the Rotating Cylinder Valve Pulse Jet System

The 15 mm diameter shafts at each end of the rotating valve were press-fits on sealed bearings to prevent air leakage through the shafts when the valve is closed. A 20 mm diameter hole was bored on each side of the aluminium block and the hole was aligned with the hole through the rotating cylinder. One end of the bored hole was connected to the compressed storage air tank and the other end was connected to the jet nozzle. In the tests, a jet nozzle of 20 mm diameter and 50 mm long was used.

The system can produce a maximum operating frequency of 80Hz. The tests were carried out for both continuous and pulse jet flow. The data acquisition system was used to record the jet exit velocity and the temperatures on the impingement plate surface. Heat transfer coefficients were calculated from the value of temperature drop between the jet exit air and the plate surface.

B. Pulse Flow Measurement

Time-averaged velocity of the centreline jet exit air close to the nozzle was measured with a calibrated hot wire anemometer. The time-averaged centreline velocity was used in calculation of the nominal Reynolds numbers. Due to the high pulsation frequency, the data were recorded at a sampling frequency of 1kHz. Figure 3 shows the velocity profile at pulsation frequency of 10 and 20 Hz for a jet nozzle diameter of 20 mm. The ON part of the pulsation duty cycle has velocity variation similar to a half-cycle of a sinusoid and the velocity at the nozzle exit is close to zero during the nominally OFF part of the duty cycle. This shows minimal air leakage in contrast with the work carried out by Azevedo et al. [10] where the leakage is quite significant. The stable flow structure created from the experiment is important in order to correctly measure the instantaneous heat transfer.

The jet exit velocity profile for all the test frequency is determined by plotting the graph of radial distance against the non-dimensional jet exit velocity. The corresponding Reynolds number in this test is based on time-averaged centreline velocity. From the jet exit velocity profile obtained, it is found that mass flow rate for different test frequencies are slightly different due to the difference in the local velocity measurement affected by the pulses. The average mass flow rate based on the local jet exit velocity difference was calculated by numerically integrating the local mass flow rate measurement over the nozzle area.

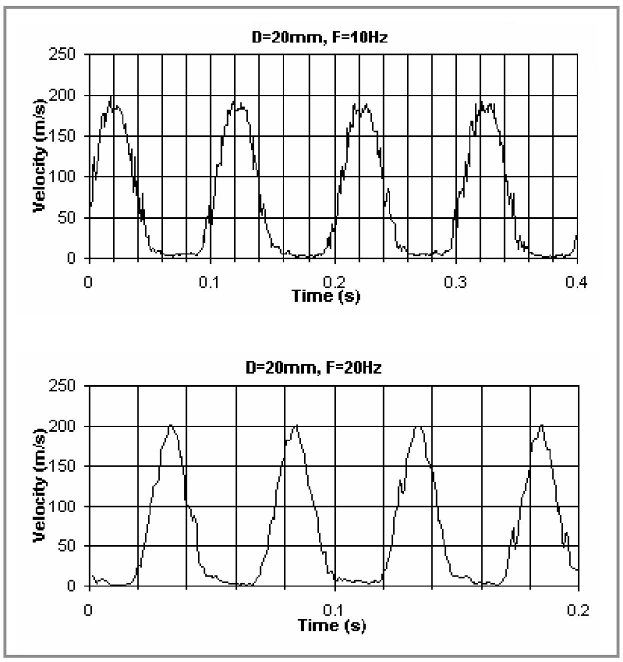

Figure 3: Velocity Profile for Pulsation frequencies of 10 and 20Hz

Figure 4 shows the non-dimensional jet exit velocity profile for steady and pulse flow for Reynolds Number, Re=16000, 23300 and 32000. The velocity profile shows that certain local velocities away from the stagnation point are higher than the velocity at those stagnation points. Since calculations of Reynolds number based on time-averaged centreline velocity do not necessarily mean the mass flow rate is kept constant, hence the Reynolds number is calculated based on the average mass flow rate measurement.

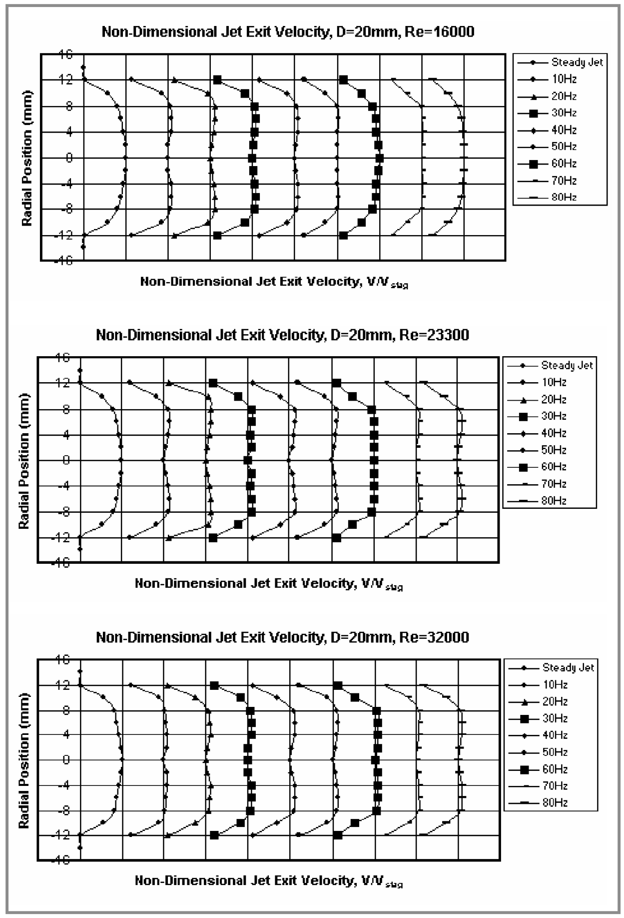

Figure 4: Jet exit velocity profile for pulsation frequencies of 10 to 80Hz for Re=16000, 23300 and 32000

The Reynolds Number based on average mass flow rate for each test frequencies are then determined using equation (1) [13] below,

$$Re = \frac{wD}{A_{hole}\mu} \qquad (1)$$

where *w* is the average mass flow rate calculated from the jet exit velocity profile, D is the nozzle diameter, *A* is the nozzle area and m is the dynamic viscosity of air at supply air temperature.

C. Heat Transfer Measurement

Heat transfer measurements were recorded from a heat flux microsensor bonded on the water-cooled aluminium block as shown in Figure 1. The impingement block was constructed from two 12 mm thick, 300 mm by 300 mm wide aluminium plates. Both plates were milled to a depth of 10mm so that only a 2 mm thick wall between the impingement area and the water passage remained. The plates were bonded to each other to create a water-tight aluminium block. Three 12 mm connecting nozzles were attached to the lower part on the rear of the block to allow the cooling water in and another three on the upper part to allow for the discharge of the water. Two K-type thermocouples were attached on the rear of the plate at a distance 120 mm apart to monitor the plate temperature. The plate was maintained at a temperature of 10^0C throughout each of the test.

The heat flux of the heated air jet impinging on the plate was measured using a heat flow and integral thermocouple sensor from RdF Corporation. Calibration information and related measurement uncertainties for the sensor was provided by the manufacturer. The sensor is bonded on the plate by a laminating adhesive and is located at the centre of the plate. The impingement plate surface was covered by a Kapton sheet having the same thermal conductivity as the heat flux sensor so that the presence of the sensor would not alter the temperature distribution. The sensor provided voltage outputs corresponding to heat flux and plate temperature.

The values of heat flux and plate temperature for the stagnation point and local measurements at different radial positions were monitored and recorded by the data acquisition system. Local heat transfer measurements were recorded at radial distances from 1 to 6 nozzle diameters. The instantaneous Nusselt number was calculated using equation (2) [14] below,

$$Nu = \frac{q''}{(T_j - T_w)} \frac{D}{k} \tag{2}$$

Where q" is the stagnation point heat flux measured by the sensor, D is the nozzle diameter, k is the thermal conductivity of the air jet evaluated at film temperature, T_j is the temperature of the hot air jet and T_w is the temperature of the plate at the stagnation point. The average Nusselt number based on the local temperature difference was calculated by numerically integrating the heat flux measurement over the impingement area.

Experimental Uncertainty

In each of the tests carried out, the measured centreline velocity immediately before and after the experiment fluctuated by less than 5%. The uncertainty for the Reynolds numbers is estimated at ±2%. The uncertainty in thermal conductivity of the impinging jet is believed to be less than 3% due to the small fluctuations in jet temperature. The uncertainty in the heat flux measurement is estimated at 5%.

3.0 RESULTS AND DISCUSSION

Table 1 shows the values of the parameters that were investigated. The values of both the local and average Nusselt numbers are considered to be functions of Reynolds number and radial distance from the stagnation point. Duty cycle and x/D were not varied in this experiment. To confirm the accuracy of the present work, the value of the steady jet Nusselt number versus Reynolds number was plotted and compared to the results of several previous researchers work as reported by ambunathan et al. [2].

Table 1: Parameters investigated in the experiments

Parameter	Values
Re	16 000, 23000, 32000
St	0.008<St<0.123
x/D	4
DC	33%
D	20mm
F	10-80Hz

Figure 5 shows that the measured Nusselt numbers for the steady jet are comparable with previous experiments. The steady Nusselt number values obtained by Sailor et al. [7] were higher than those documented by other research work. Their calculations of Nusselt numbers were based on the difference of temperature between the stagnation point and the local adiabatic wall temperature. The average air temperature at the stagnation point is lower than the air temperature at the jet nozzle because ambient cool air is entrained in the pulse jet. This explains why Sailor et al. [7] recorded a higher value of Nusselt number for the same test parameters.

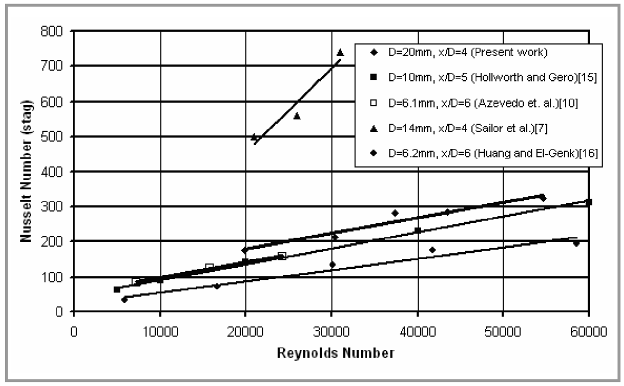

Figure 5: Comparison of steady flow stagnation point nusselt number versus Reynolds number from previous research

Figure 6 shows the variation of stagnation point and average Nusselt number with Reynolds number at nozzle to plate spacing, *x/D* equal to 4 for frequencies from 20 to 80 Hz. Stagnation Nusselt number increases with higher Reynolds

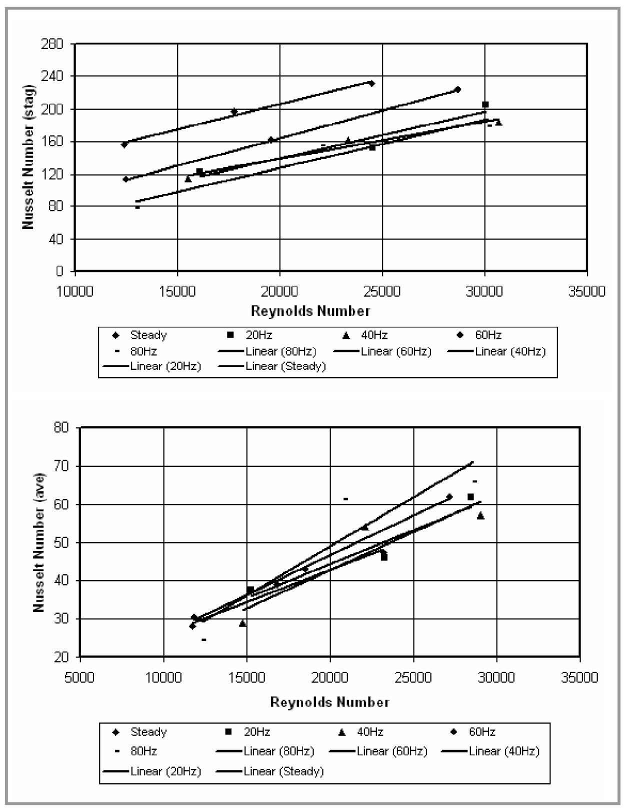

Figure 6: Variation of stagnation point and average Nusselt number with Reynolds number, Re at nozzle to plate spacing, x/D=4 for frequencies from 20 to 80 Hz

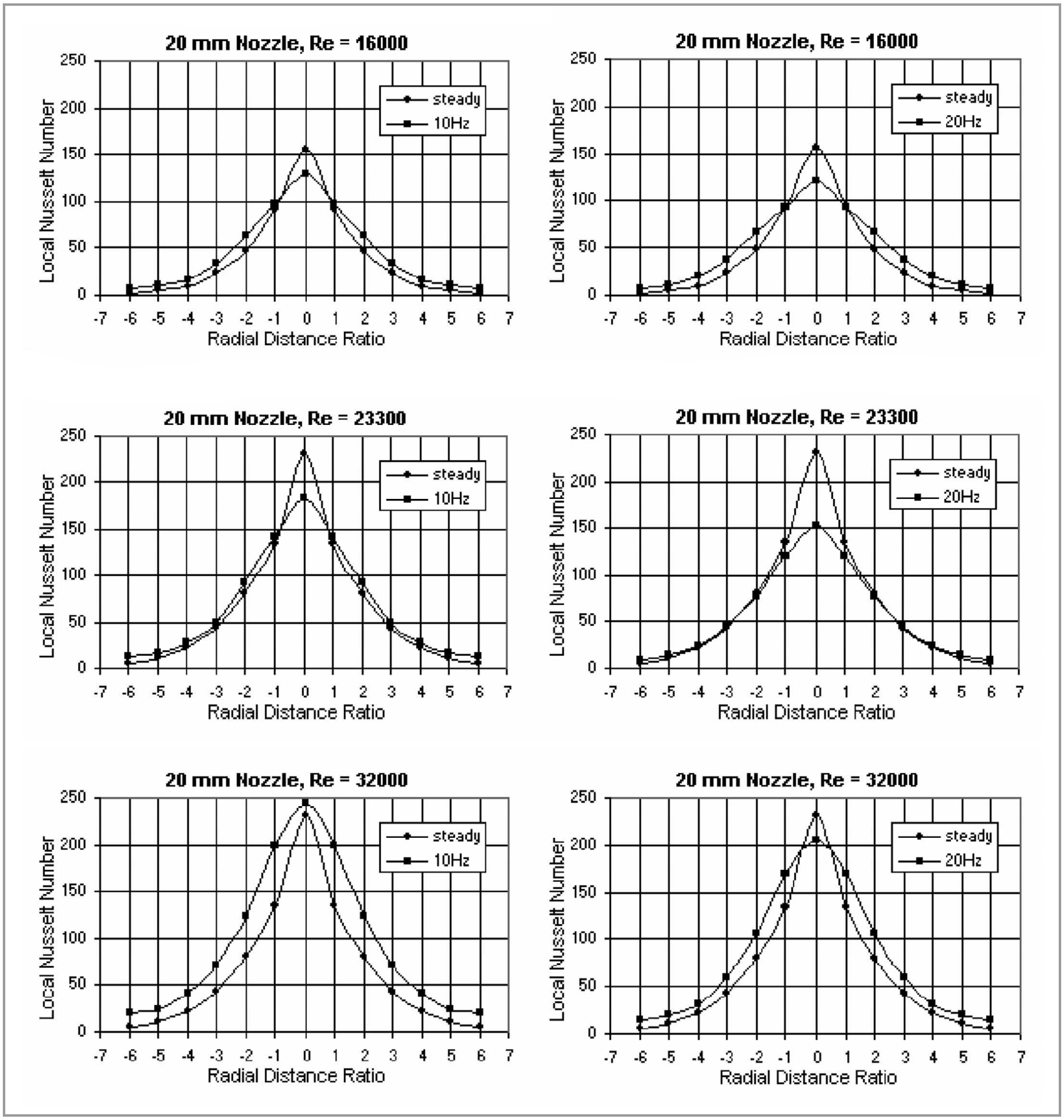

Figure 7: Variation of local nusselt numbers with radial distance at frequencies of 10 and 20Hz for Re=16000, 23300 and 32000

numbers as predicted. Figure 7 shows a graph of local Nusselt number against radial distance for frequencies of 10Hz and 20Hz for all the three Reynolds number under investigation. The graph shows that the local Nusselt numbers for pulse flow is higher than for steady flow at position from 1 diameter outwards. Higher turbulence intensity at these positions is believed to contribute to the increase in heat transfer. Figure 8 shows the dependence of stagnation Nusselt number on frequency for three Reynolds numbers at x/D=4. For stagnation point heat transfer, jet pulsation has the effect of decreasing the heat transfer for all the frequencies studied. The highest frequency degradations are between 20 to 50Hz and 80Hz. This generally is in agreement with the results obtained by Azevedo et al. [10]. In their experiments, where the Reynolds number used in the tests were less than 25 000, the heat transfer results at all frequencies shows significant degradation. The results show that higher mass flow rate can influence the heat transfer measurements.

Figure 9 shows a graph of stagnation and average Nusselt numbers against frequency for Reynolds number at 16000, 23300 and 32000. The flow structure changes with frequency in a complex manner so the Nusselt number versus frequency curve shows no clear trend. The average Nusselt numbers for the pulse jet are higher than the steady jet for all the frequencies tested by at least 30%. The maximum average heat transfer enhancement occurs at frequency of 70Hz for

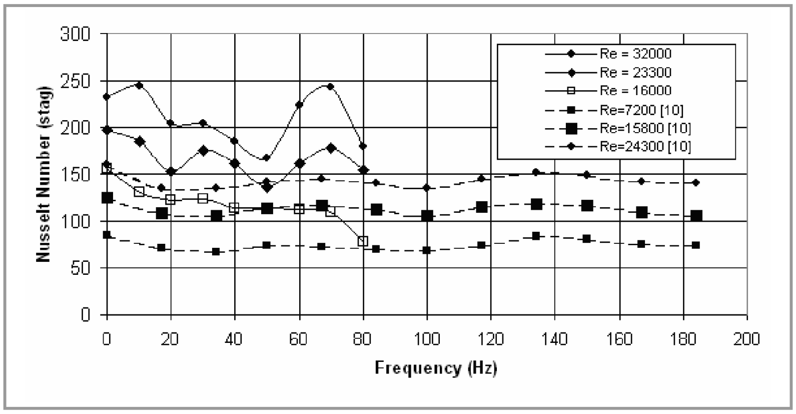

Figure 8: Variation of stagnation nusselt numbers with frequency for Reynolds number of 16000, 23300 and 32000. Comparison of current results with Azevedo et al. [10]

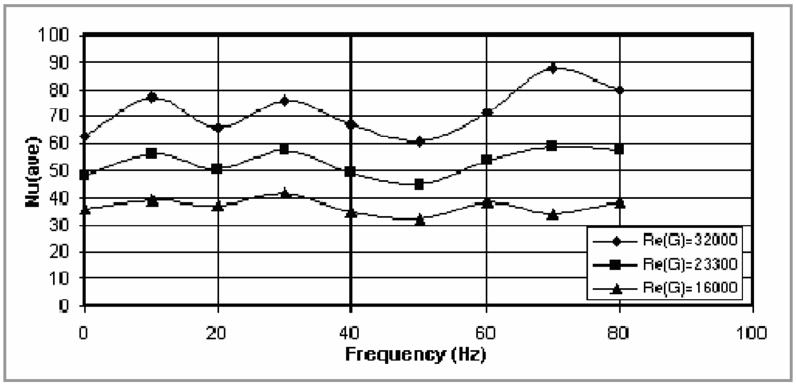

Figure 9: Variation of stagnation and average nusselt numbers with frequency for Reynolds number = 16000, 23300 and 32000

Reynolds number of 32000. The percentage of enhancement at this frequency is approximately 80%.

The present results show that the average heat transfer on the impingement area is enhanced quite significantly even though the stagnation point heat transfer decreases. These increases are shown to be available for a system that uses a single pulse jet impinging on an area with radius up to 6 times the nozzle diameter.

4.0 CONCLUSIONS

The results of the experiments show that there is significant enhancement in the local heat transfer of the pulse flow at positions one nozzle diameter or more away from the stagnation point for all the pulse frequencies. The stagnation point heat transfer does not show any enhancement for the three Reynolds numbers investigated. The average Nusselt number for the pulse jet is enhanced for all the frequencies investigated. The degree of enhancement is in the range 30-80% with the greatest benefit at frequency of 70Hz for Re=32000. Heat transfer in the pulse flow mode is complex and dependent on the flow structure of the jet. The significant enhancement of the heat transfer at local distances away from the stagnation point resulted in higher average Nusselt numbers for pulse flow compared to steady flow. Significant turbulence intensity caused by pulsating the jet resulted in the increase recorded. The degradation in heat transfer at the stagnation point is believed to be due to small turbulent intensities of the pulse flow at this position. As a conclusion, this system is suitable for applications requiring overall heat transfer enhancement on the impingement surface. ■

NOMENCLATURE

D	Diameter of nozzle (m)
DC	Duty cycle of flow (ratio of ON time to total cycle time)
F	Frequency of flow pulsations (Hz)
$\overline{h}$	Heat transfer coefficient (W/m^2K)
	Average heat transfer coefficient (W/m^2K)
k	Thermal conductivity of air (W/mK)
Nu	Nusselt number, hD/k
$\overline{Nu}$	Average Nusselt number
Pr	Prandtl number
q"	Heat flux per unit area (W/m^2)
Re	Reynolds Number, VD/n
Re(G)	Reynolds Number, $wD/A_{hole}\mu$
St	Strouhal number, FD/V
T_j	Air Jet Temperature (K)
T_w	Plate Temperature (K)
V	Velocity of jet exit air (m/s)
w	Average mass flow rate (kg/s)
x	Distance from nozzle to impingement surface (m)
x/D	Ratio of distance between nozzle and impingement surface to diameter
v	Kinematic viscosity of air at flow temperature (m^2/s)
μ	Dynamic viscosity

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PROFILES

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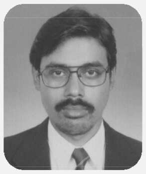

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