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Effect of variable duty cycle flow pulsations on heat transfer enhancement for an impinging air jet

David J. Sailor *, Daniel J. Rohli, Qianli Fu

Department of Mechanical Engineering, Tulane University, 400 Lindy Boggs Center, New Orleans, LA 70118, USA

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Abstract

A series of experiments has been conducted in which a pulsed air jet is impinged upon a heated surface for the purpose of enhancing heat transfer relative to the corresponding steady air jet. Traditional variables such as jet to plate spacing, Reynolds number, and pulse frequency have been investigated. One additional flow variable – the duty cycle – representing the ratio of pulse cycle on-time to total cycle time is introduced and shown to be significant in determining the level of heat transfer enhancement. Specifically, heat transfer enhancement exceeding 50% is shown for a variety of operating conditions. In each case, the duty cycle producing the best heat transfer is shown to depend upon each of the other flow parameters. Recommendations are made for further experimentation into optimizing the duty cycle parameter for any particular application. © 1999 Elsevier Science Inc. All rights reserved.

Keywords: Pulsating flow; Heat transfer enhancement; Air jet; Impingement

Notation

d	inner diameter of jet nozzle
$\mathrm{d}T$	temperature difference $T_{surf} - T_{jet}$
DC	duty cycle of flow field (ratio of on time to total cycle time)
F	frequency of flow field pulsations (Hz)
h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity of air at film temperature (W/m K)
Nu	Nusselt number, hd/k
q''	heat flux per unit area (W/m^2)
Re	Reynolds number, Vd/ν
St	Strouhal number, fd/V
T_{aw}	adiabatic wall temperature (K)
T_{jet}	jet temperature (K)
T_{surf}	centerline temperature of heat transfer surface (K)
U	uncertainty
V	centerline jet velocity (m/s)
z	axial distance between jet exit and plate surface
Greek	
ν	kinematic viscosity of air at flow temperature (m^2/s)
σ	standard deviation of sample points

* Corresponding author. Tel.: +1-504-865-5775; fax: +1-504-865-5345.

E-mail address: sailor@mailhost.tcs.tulane.edu (D.J. Sailor)

1. Introduction

Many industrial processes involve mass or heat transfer to or from a material being manufactured or processed. Jet impingement has been used extensively in such processes, and the subject has been studied in depth over several decades (e.g. Martin, 1977; Downs and James, 1987). Examples of industrial applications of impingement heat transfer are annealing of metals, tempering of glass, drying of textiles, and cooling of electronics. If these processes can be made more efficient, there is a potential for great savings in processing time and process energy consumption. For the most part, previous studies have focused on optimizing the transport processes associated with steady impinging jets. Parameters that have received attention include the spacing between the jet outlet and the impingement surface, the angle of impingement, the magnitude of imposed swirl velocities, and turbulence intensity.

Nevins and Ball (1961) were among the first to investigate the potential benefits of pulsed jet impingement. They studied spatially and temporally averaged heat transfer for impingement of a pulsating circular air jet with $1200 < \mathrm{Re} < 120{,}000$, $10^{-4} < \mathrm{St} < 10^{-2}$, and nozzle-to-plate spacing from 8 to 32 nozzle diameters. This work concluded that no significant heat transfer enhancement was obtained by introducing flow pulsations. This finding was left unchallenged for about 25 years. In the late 1980s and early 1990s several research teams revisited the issue of heat transfer enhancement via flow pulsations. Kataoka and Suguro (1987) documented the beneficial role of secondary flow structures on surface heat transfer. Specifically, they showed that stagnation point heat transfer for axisymmetric submerged jets is enhanced by the impingement

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of large-scale structures on the boundary layer. Such large-scale structures have been well-documented for pulsed jet flows (e.g., Eibeck et al. 1993). In addition to the presence of large scale structures (e.g., periodic formation of vortices which impinge upon the heat transfer surface), pulsed flows have the potential to result in thinner time- averaged boundary layers. This is especially true for intermittent (on-off) flows for which the boundary layer momentarily vanishes each cycle and regenerates from a zero thickness. Zumbrunnen (1992) and Mladin and Zumbrunnen (1996) used nonlinear dynamics models of momentum and energy transport within the boundary layer to estimate pulsation frequencies above which heat transfer enhancement might be expected due to the time-averaged boundary-layer thickness effect. The threshold identified was St > 0.26. Zumbrunnen and Aziz (1993) presented experimental results for high-frequency (St > 0.26) intermittent water jets. They found that flow pulsations resulted in increases in local Nusselt numbers by up to 100% for Re in the range of 3100–21,000. Eibeck et al. (1993) used experiments and computations to identify enhanced secondary flow structures resulting from flow pulsations. Although they investigated Strouhal numbers below Zumbrunnen's threshold they found pulsed impingement enhancement well in excess of 100%. This enhancement is believed to be largely a result of the effects of secondary flow structures (vortex rings) impinging upon the heat transfer surface. In retrospect it can be shown that the pioneering work by Nevins and Ball was also well below the threshold Strouhal number. Unlike Eibeck, they did not document whether and to what extent secondary flow structures were present in their experiments. Thus, their inability to demonstrate pulsed flow heat transfer enhancement is not surprising.

These studies of pulsed jet flows are encouraging, and motivate the need for additional work. Most previous pulsed impingement studies have significant limitations in terms of applicability in industry. Also, they fail to study in much detail the role played by the actual velocity profile in time. Few previous studies have focused on the periodic waveform. Notable examples are the theoretical study by Mladin and Zumbrunnen (1996) and the experimental study by Sheriff and Zumbrunnen (1994). In Sheriff's experimental study, however, the focus was on simple sinusoidal and square-pulse waveforms with an exploration into the effect of pulse frequency and magnitude. While sinusoidal and square wave velocity–time profiles are easy to generate and study, it is reasonable to hypothesize that another functional form for this flow profile may yield better heat transfer enhancement results.

In this paper we introduce a profile parameter termed the duty cycle. This parameter is defined as the ratio of flow on time to total cycle time for one period of the flow pulsation. This parameter is particularly useful in describing flows where the valve control profiles are rectangular rather than square in nominal shape. As discussed in this paper, we have found the duty cycle parameter to be crucial for the purpose of optimizing heat transfer enhancement associated with pulsed impingement.

2. Experimental methods

2.1. Apparatus

Fig. 1 is a schematic representation of the apparatus used in our experiments. The major components are an air compressor, a refrigerated drying unit, a pulse valve mechanism, thermal anemometry equipment, an enclosed heat transfer measuring cell including a brass heat transfer surface, a subsurface heater, and a high speed computerized data acquisition system. The cooling air jet used in the experiments was generated with a 25 hp air compressor and aftercooler connected to two surge tanks with a collective capacity of approximately 1.9 m^3. The compressed air was piped through the refrigerated drying unit to remove moisture/contaminants, and its temperature was measured with a 36-gage T-type thermocouple before entering the pulse drive mechanism.

After emerging from the sharp-edged exit of the tailpipe ($d = 14.00 \pm 0.05$ mm, $L = 51.00 \pm 0.05$ mm) of the pulse valve mechanism, the average velocity of the pulsed jet was measured with a calibrated hot wire anemometer. As noted by Bremhorst and Listjono (1987) wind tunnel calibration of hot wire probes to be used at nozzle exits can introduce error due to static pressure effects. To minimize calibration-related errors in our experiments we made jet exit measurements at $z/d < 1$. According to Bremhorst and Listjono (1987) the resulting velocity calibration errors should be less than 2%. The time-averaged centerline velocity was used in calculation of the nominal Reynolds numbers for each experiment. The heat transfer measuring cell consisted of a wooden enclosure measuring approximately 380 mm on a side and 250 mm high. At the top of the cell, flush with the enclosure surface, was a 178 mm diameter, 6.4 mm thick circular brass disk, which served as the heat transfer surface. A 0.2 mm thick heat flux sensor with a 40 junction thermopile and 36-gage T-type thermocouple were attached at the center point of the surface of the disk and connected to the data acquisition system. A 32 mm layer of magnesium oxide powder was packed between the lower surface of the brass disk and the 152 mm diameter silicon rubber heater. The purpose of the powder was to evenly distribute the

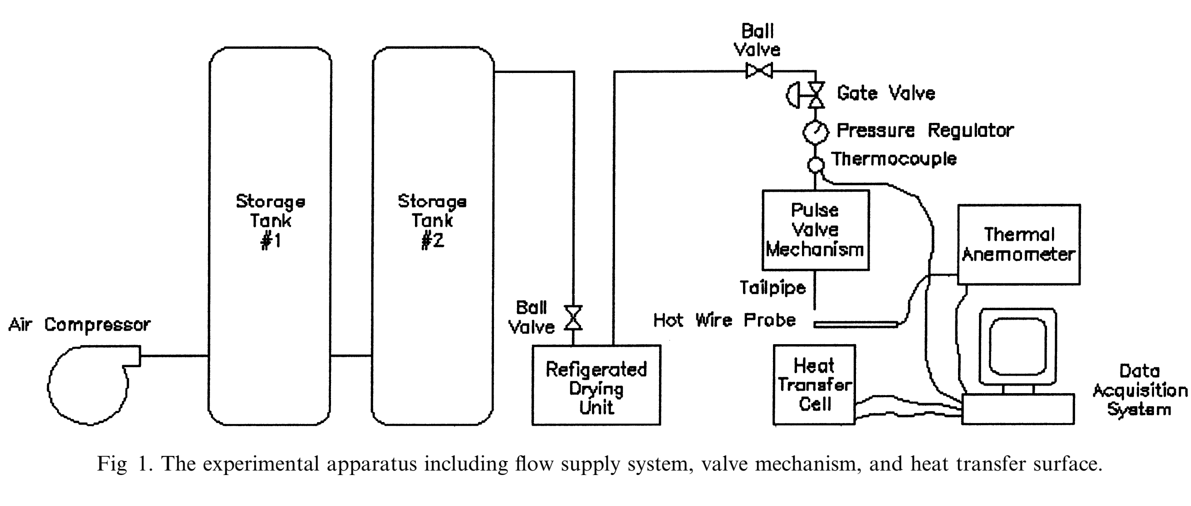

Fig 1. The experimental apparatus including flow supply system, valve mechanism, and heat transfer surface.

heat from the 1.55 W/cm^2 heater. The heater was powered with a direct current power supply and regulated by a commercial temperature controller to maintain the heater's temperature between preset limits. Insulating material was placed below the heater and on all sides of the heat transfer cell to maximize heat available to the brass heat transfer surface.

2.2. Pulse valve mechanism

Earlier impingement heat transfer experiments conducted on site employed a high speed solenoid valve controlled by a repeat cycle timer circuit (Sailor and Patil 1996). In those experiments, the duty cycle was adjusted by varying the electronic control signal fed to the solenoid. Due to the physical limitations of the timer circuit, the highest frequency achieved was approximately 10 Hz. In the second generation of this experiment the timer circuit was replaced by a solid state computer controlled relay, limited only by the solenoid charging time. With this arrangement the practical frequency limit for experiments was increased to 25 or 30 Hz. Because of the desire to incorporate the duty cycle parameter into higher frequency pulsed impingement studies and the limitations associated with using a computer controlled solenoid valve in industrial applications, a mechanically driven valve was designed and constructed. The new device was capable of generating profiles with duty cycles ranging from approximately 0.2–0.6 at frequencies in excess of 100 Hz.

The mechanical valve illustrated in Fig. 2 consists of a hollow shell and concentric rotating insert. The shell contains two threaded ports placed 180° apart to accommodate inlet and outlet ports. A passage is machined into each cylindrical insert to provide for a flow path. Twice during each rotation of the cylindrical insert this passage is aligned with the inlet and outlet, and air flows freely from the inlet to outlet. When the passage is not aligned with the inlet–outlet ports there is a small leakage flow. The size of the air passage dictates the duty cycle of the velocity profile created by the valve. Hence, different valve inserts are required to generate different valve duty cycles. Full details of this pulsed valve mechanism are disclosed in Sailor and Rohli (1998).

The mechanical valve was mounted above the heat transfer surface and driven by a 0.1 hp AC electric motor via a geared belt drive system. The speed of the motor and driven valve was controlled with an analog speed controller and measured with an optical tachometer. Since each rotation of the valve results in two pulse cycles, the frequency of the flow profile is therefore calculated by dividing the tachometer's rpm output by 30.

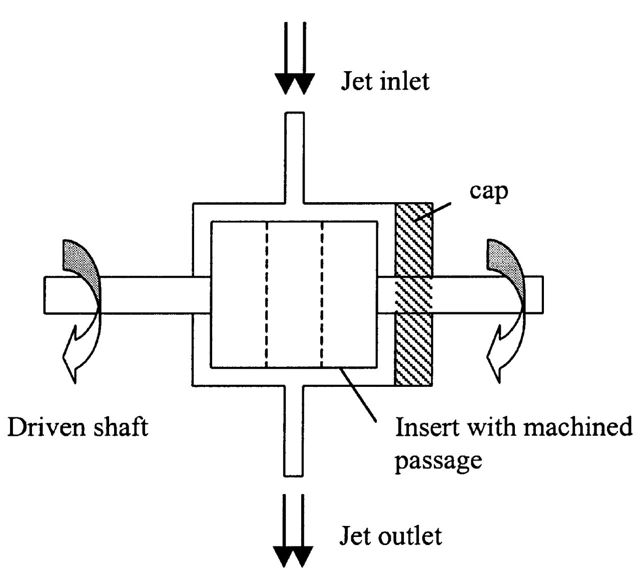

Fig 2. Schematic of pulsed valve mechanism.

2.3. Flow field measurements

The goal of this study was to use steady jet impingement results as a standard of comparison for evaluating the performance of various pulsed impingement conditions. The comparisons between pulsed and steady cases were based on equivalent bulk flow rates. This corresponds to the equivalence of bulk Re. In the pulsed flow cases the calculation of bulk Re involved averaging of multiple flow speed measurements in each of 50 or more integral cycles. There are two reasonable alternatives to this comparison based on constant Re. As suggested by Bremhorst and Hollis (1990) constant momentum flux might be a useful standard. This is particularly relevant for pulsed cases where the momentum flux develops over the first 10 jet diameters due to the existence of a streamwise pressure gradient. Another alternative particularly relevant in an industrial application would be to consider the case of a constant upstream pressure driving the flow through the jet valve and tailpipe.

Measurements of the radial profiles for the exit velocities of the time-averaged pulsed and steady flows indicate that these exit profiles are surprisingly similar, especially for the lower frequency experiments. As expected, using the traditional definition for turbulence intensities the pulsed jet profiles had much higher turbulence intensities (70–110%) than the steady jet cases (10–15%). These values are consistent with those presented by other researchers for similar jet studies (Bremhorst and Harch, 1979; Wygnanski and Fiedler, 1969; Bremhorst and Hollis, 1990). Fig. 3 compares representative steady and pulsed exit profiles, both non-dimensionalized by their centerline velocity. These profiles are fairly well characterized by the centerline velocity. That is, for a given centerline velocity value, the mass flow rates (and hence bulk Re) for the steady and pulsed cases have been found to be within 5% of each other. The centerline velocities were therefore measured and used to calculate nominal Re for all experimental results presented below.

The general experimental procedure for both the steady and pulsed cases was as follows. For any particular experiment, the heat transfer surface was heated, and initially isolated from the flow using a plexiglas barrier. The flow was generated using the compressor and valve mechanism. A hot wire probe was inserted in the exit stream of the jet at approximately 0.5 diameters from the exit plane. A pressure regulator was adjusted until the desired time-averaged centerline Re was achieved. The hot wire probe was then removed. The plexiglas barrier was removed allowing the flow to impinge upon the heat transfer surface. Measurements of surface temperature and surface heat flux were recorded by the data acquisition system. Due to the response times of the thermocouple and heat flux sensor, the calculated heat transfer coefficient erroneously appears to be a function of time during the initial few seconds of the experiment. Generally after 5–10 s, the temperature and heat flux transients are small enough that the sensor response times become inconsequential. Thus, after an initial delay of 10 s into the experiment, calculations from all subsequent measurements reveal an approximately constant heat transfer coefficient. Due to the high frequency of the pulsed experiments, it was deemed necessary to sample at a frequency of 1000 per s, and average the heat transfer coefficient over the sample period of 1 s. The result was that a minimum of 20 points in each of 50 or more full cycles were sampled and averaged for all experiments. Once the data were stored, the hot wire probe was

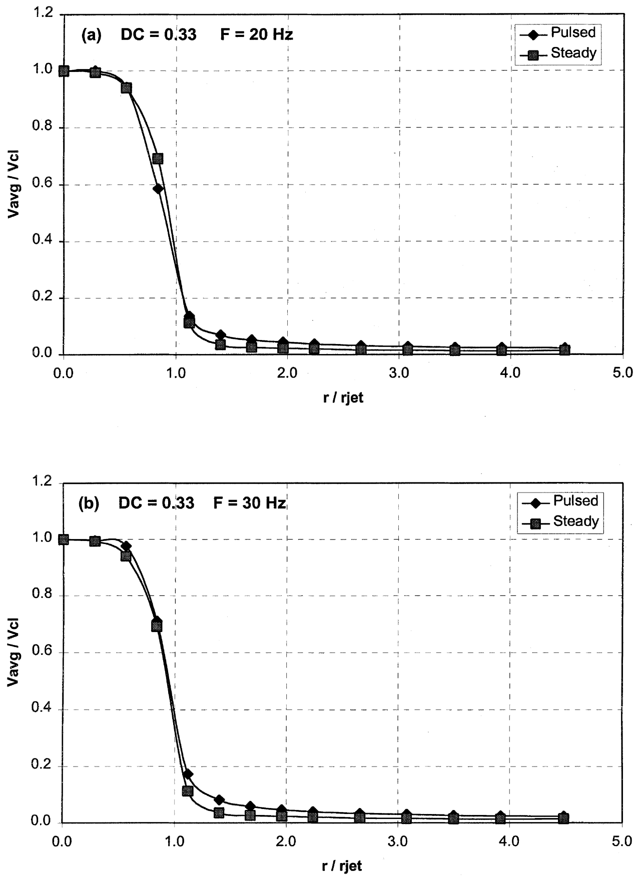

Fig 3. Representative steady and pulsed jet radial velocity profiles, measured at $z/d = 0.5$ and non-dimensionalized by their centerline velocity.

reinserted to verify that the flow velocity had remained essentially constant. In all cases, the centerline velocity immediately before and after the experiment changed by less than 5%. Due to the initial 10 s delay imposed upon the heat transfer measurements in each experiment, the latter velocity reading is more representative of experiment conditions and was used to calculate the Reynolds number for each experimental datum point presented here.

As heat transfer enhancement is the focus of this study, detailed velocity profiles and a discussion of the fluid dynamics of pulsed flows are not presented. The reader should refer to Eibeck et al. (1993) or Mladin and Zumbrunnen (1996) for such a discussion. Nevertheless, it is useful to illustrate the duty cycle concept through a representative flow profile. Fig. 4 shows the cyclic opening/shutting of the valve (solid line) associated with the cyclic rotation (approx. 20 Hz) of a valve insert with a duty cycle of 0.33. The actual velocity profile generated by this valve is shown by symbols.

2.4. Heat transfer measurements

In order to calculate the stagnation point heat transfer associated with each experiment we measured instantaneous values of the jet temperature T_{jet}, the centerline surface temperature T_{surf}, and the centerline surface heat flux q at each sampling instant. The instantaneous Nusselt number was then calculated from:

$$\mathrm{Nu} = \frac{q''}{(T_{\text{surf}} - T_{\text{aw}})} \frac{d}{k}, \tag{1}$$

where q'' is the stagnation point heat flux into the plate, d is the jet diameter, k is the thermal conductivity of the air jet evaluated at the film temperature, T_{surf} is the measured stagnation point temperature, and T_{aw} is the local adiabatic wall temperature.

The mean and standard deviation (σ) of instantaneous Nu were calculated for each experiment and any values outside of the 2σ confidence region were discarded as outliers. The average Nu was then calculated as the average of the remaining sample points.

The heat flux sensor and surface-mounted thermocouple both had response times on the order of 1 s. The introduction of a step change in the surface boundary condition at the onset of the experiment results in large temporal gradients of surface temperature and surface heat flux. Because of the sensor response times measurements near the beginning of an experiment yield erroneous heat transfer results. The temporal temperature and heat flux gradients decrease in time, however, resulting in a nominal delay time after the response times of the sensors become inconsequential. This minimum delay time was typically on the order of 10 s and was individually verified for each experimental case by ensuring that longer delay times produced equivalent heat transfer results to within the uncertainty of the measurements. In all experiments the measured temperature difference was at least 8°C with an uncertainty of roughly 6%. The uncertainty in the heat flux measurement is estimated at ±5%. Given the small fluctuations in jet temperature (and humidity ratio) during the experiment the uncertainty in thermal conductivity of the impinging fluid is believed to be less than 3%. Finally, the uncertainty in the jet diameter is approximately 0.35%. Assuming that errors are independent

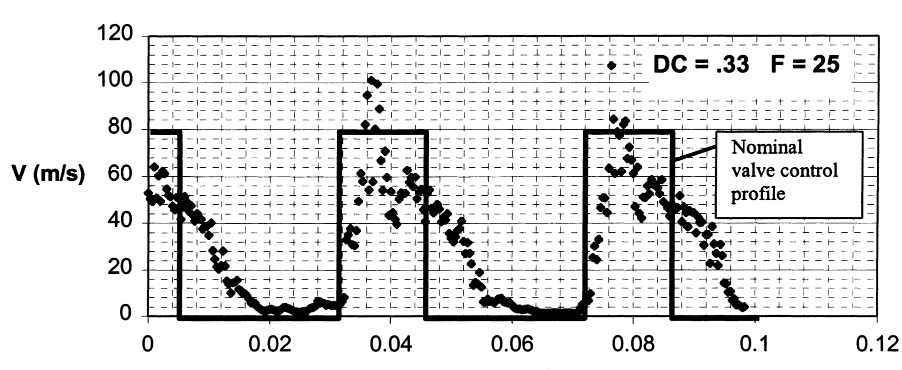

Fig 4. Representative centerline velocity at $z/d = 0.5$ as a function of cycle time for a 0.33 duty cycle case operating at approximately 25 Hz.

it is reasonable to estimate the overall uncertainty of Nusselt number calculations using a standard Euclidean norm:

$$U_y = \left(\sum_{1}^{n} \left(\frac{\partial f}{\partial x_i} U_{x_i} \right)^2 \right)^{0.5}, \quad \text{where } y = f(x_1, x_2, \ldots, x_n). \tag{2}$$

Applying this expression to Eq. (1) it can be shown that the uncertainty in calculated Nusselt numbers for these experiments is given by:

$$U_{\mathrm{Nu}} = \left(\left(\frac{d^* U_q}{k(\mathrm{d}T)} \right)^2 + \left(\frac{q^* U_{\mathrm{d}}}{k(\mathrm{d}T)} \right)^2 + \left(\frac{-qd^* U_k}{k^2(\mathrm{d}T)} \right)^2 + \left(\frac{-qd^* U_{\mathrm{d}T}}{k(\mathrm{d}T)^2} \right)^2 \right)^{0.5}, \tag{3}$$

where $\mathrm{d}T$ is the temperature difference and U is the uncertainty of the variable corresponding to the subscript. Based on this expression and the individual parameter uncertainty estimates, the uncertainty in the calculated Nusselt number is less than 8% for all experiments presented in this paper.

3. Results

Experiments were conducted in which the parameters of Reynolds number, Strouhal number (pulsation frequency), flow duty cycle, and jet-to-plate spacing were varied. The values of each parameter that were investigated are listed in Table 1.

Fig. 5 demonstrates the heat transfer for the smallest jet-to-plate spacing tested ($z/d = 4$). At this z/d location virtually all pulsed jet experiments demonstrated enhancement relative to the steady jet results. The steady jet results show an increase in Nusselt number from around 500 for the low Re case to nearly 700 for the high Re case. In each of these experiments the DC = 0.33 case resulted in the highest heat transfer enhancement, followed by the DC = 0.25 and DC = 0.50 cases. The optimum frequency for these experiments appears to be in the range of 25–35 Hz. The maximum heat transfer enhancement occurs for the highest flow speed, which resulted in pulsed flow enhancement over the steady jet of up to 65%.

When the jet-to-plate spacing was increased to $z/d = 6$, the middle duty cycle DC = 0.33 no longer exhibited the best heat transfer performance. As shown in Fig. 6, the DC = 0.25 experiments performed best at this axial spacing. Again the peak enhancement occurred for frequencies in the 25–35 Hz range for the lowest two flow speeds, and at a slightly higher frequency (30–40 Hz) for the highest flow rate case. As expected from published steady jet experiments (see literature review in Martin, 1977), the steady jet heat transfer was slightly higher at $z/d = 6$ than it was at $z/d = 4$ with Nusselt number values ranging from 500 to 800. The actual heat transfer enhancement factors for this spacing were less than those for the smaller spacing, with a peak enhancement factor of approximately 60% for the highest flow rate case.

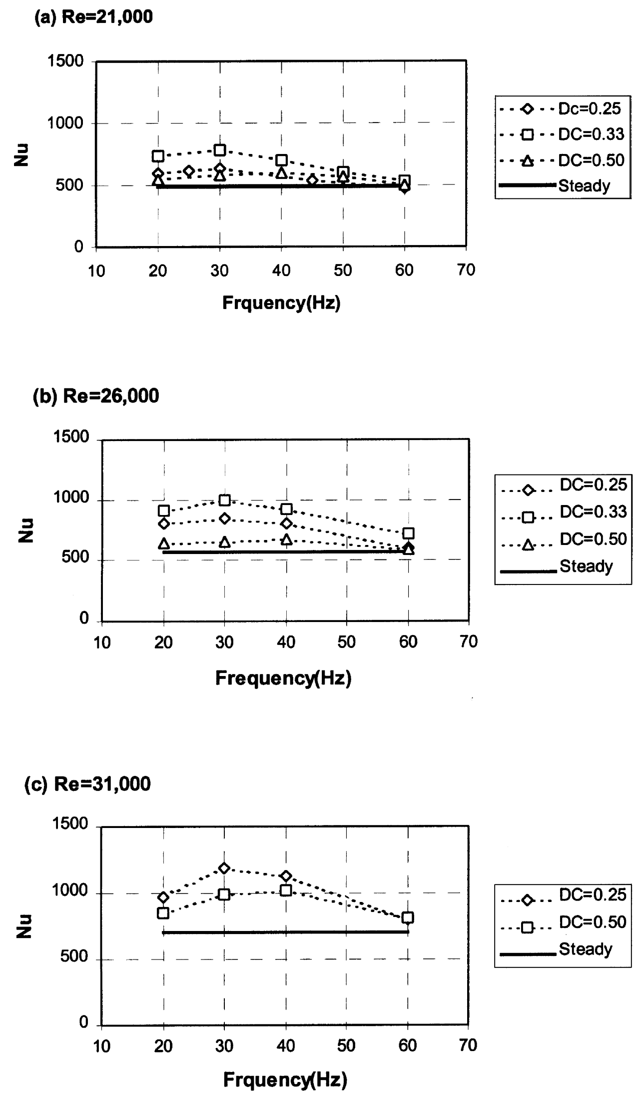

Fig 5. Pulsed and steady jet heat transfer results $z/d = 4$.

At the largest jet-to-plate spacing ($z/d = 8$) the steady jet heat transfer was less than for the other two spacings. The heat transfer enhancement for the pulsed cases (relative to the corresponding steady case) was also much less, with no statistically significant differences at the two lower Re settings. As shown in Fig. 7 the lowest duty cycle case gave modest enhancement (up to 30%) at the highest Re level.

4. Discussion

The results presented above reveal several important issues associated with pulsed impingement heat transfer. First, while

Table 1.
Parameter values investigated in this study and their associated uncertainties

Parameter investigated	Values of parameter used in experiments
Re	21,000, 26,000, and 31,000 (±4%)
z/d	4.0, 6.0, 8.0 (±0.1)
St (f)	0.009<St<0.042. Steady jets, and several frequency values between 20 and 60 Hz (±2 Hz)
DC	0.25, 0.33, 0.50 (±0.01)

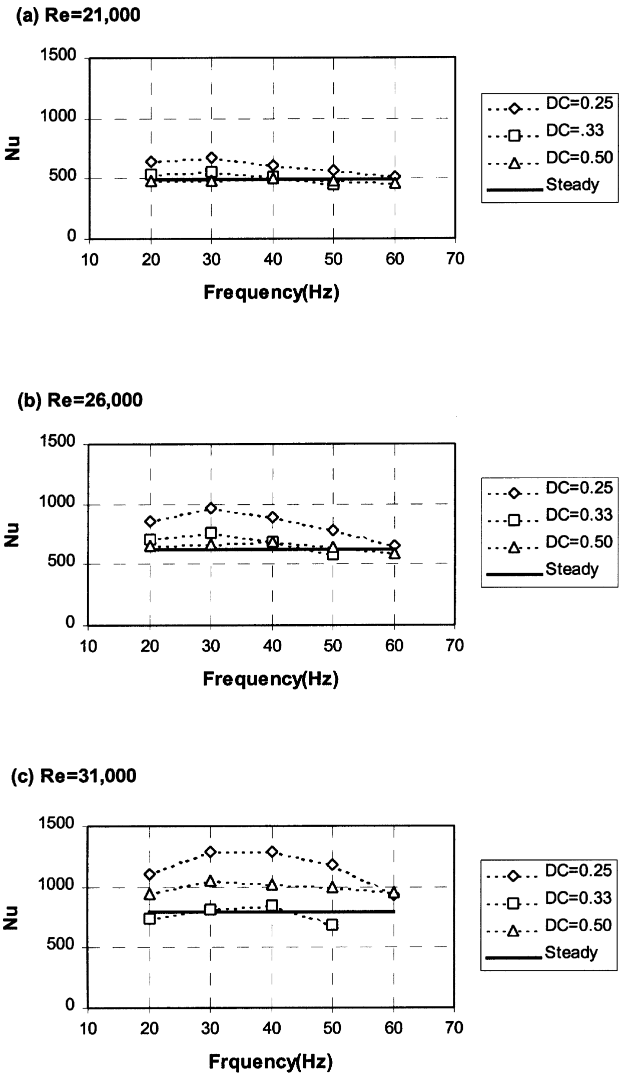

Fig 6. Pulsed and steady jet heat transfer results $z/d = 6$.

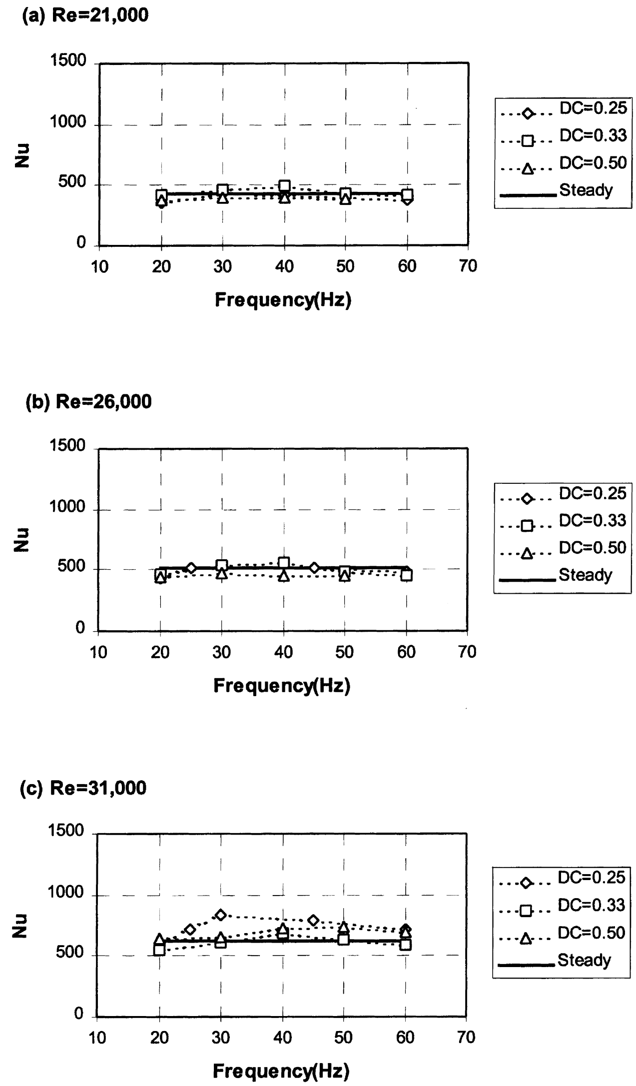

Fig 7. Pulsed and steady jet heat transfer results $z/d = 8$

DC = 0.50 corresponds to the common square-wave velocity profile used in pulsed flow studies, the lower duty cycle cases consistently produced better heat transfer results. Also, while some arguments (e.g., with respect to time-averaged boundary-layer thickness) would suggest that higher frequencies should produce more enhancement, these results show that intermediate frequencies generally performed best. Also, the highest enhancement rates were obtained for the highest flow rates investigated, suggesting that even higher enhancement may be possible for flow rates not investigated within the scope of this study.

Earlier it was noted that Nevins and Ball (1961) found no significant heat transfer in their early studies with pulsed impingement. This was explained at least partially by Zumbrunnen and Aziz (1993) since the Nevins and Ball experiments studied very low St number flows. While the present study is also below the threshold St noted by Zumbrunnen, we have found significant heat transfer enhancement. There may be several factors that contribute to this. One of the likely explanations is simply that Zumbrunnen's threshold Strouhal number argument focuses on enhancement due solely to the intermittent growth of the boundary layer and the effect that flow pulsations have on the time-averaged boundary layer thickness. As noted in Eibeck et al. (1993) secondary flow structures may also play an important role in pulsed impingement enhancement.

These findings are significant, but there are remaining questions. First, it would be instructive to know the spatial distribution of heat transfer, and be able to evaluate spatial averages of heat transfer enhancement. Obviously there are many important applications in the drying industry (e.g., Page and Kiel, 1990) where it would be useful to study the effects of heating the jet instead of the impingement surface. In such cases, the enhanced entrainment associated with the pulsed jet may have significant adverse effects. Specifically, entrainment of cool ambient fluid into a hot jet would lower the bulk flow temperature, hence reducing the driving temperature difference for convective heat exchange. Finally, while the technology exploited in this study is not yet ready for implementation in industrial processes, the mechanical valve developed for this study is quite promising as a means for implementing pulsed flows in industrial processes. Specifically, a valve or array of valves could be installed with optimum duty cycle and pulse frequencies for the application of interest. Such optimization studies are the subject of ongoing work.

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