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Experimental study of heat transfer in pulsating turbulent flow in a pipe

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ABSTRACT

Heat transfer characteristics of pulsating turbulent air flow in a pipe heated at uniform heat flux were experimentally investigated. The experiments were performed over a range of $10^4 < Re < 4 \times 10^4$ and $6.6 \leqslant f \leqslant 68$ Hz. This situation finds applications in modern power generation facilities and industrial processes. With installing the oscillator downstream of the tested tube exit, results showed that *Nu* is strongly affected by both pulsation frequency and Reynolds number. Its local value either increases or decreases over the steady flow value. The variation is more pronounced in the entrance region than that in the downstream fully developed region. It is observed also that the relative mean *Nu* either increases or decreases, depending on the frequency range. Although the deviations are small, it seems to be obvious at higher values of Reynolds number. The obtained heat transfer results are classified according to turbulent bursting model and looked to be qualitatively consistent with previous investigations.

1. Introduction

There are many engineering practical situations where heat is being transferred under conditions of pulsating and reciprocating flows such as the operation of modern power producing facilities and industrial equipment used in metallurgy, aviation, chemical and food technology. Cavitations in hydraulic pipelines, pressure surges and flow of blood are also some of familiar instance of such flows. The performance of this equipment in thermal engineering applications is affected by the pulsating flow parameters (Al-Haddad and Al-Binally, 1989).

During the past few decades, numerous studies have been devoted to this pulsating flow and its associated heat transfer problems. A review of these studies with emphases on the onset of turbulence, velocity distribution and pipe flow as well as the heat transfer characteristics including axial heat transfer enhancement and convective heat transfer are presented in the following sections.

Pulsating flows can be produced by reciprocating pump or by steady flow pump together with some mechanical pulsating devices. It may normally be expected that the heat transfer to or from the flow would be changed since the pulsation would alter the thickness of the boundary layer and hence the thermal resistance.

Pulsating flow is assumed to be consisted of a steady Poiseuille flow and purely oscillatory (Zhao and Cheng, 1998). The amplitude of the oscillatory velocity is less than the time mean velocity and flow direction never reverse. Pulsating flow is one of the unsteady flows that are characterized by periodic fluctuations of the mass flow rate and pressure.

Most of investigators (Al-Haddad and Al-Binally, 1989; Zhao and Cheng, 1998; Hesham et al., 2005a; Habib et al., 1999; Gupa et al., 1982; Barid et al., 1996; Gbadebo et al., 1999; Zheng et al., 2004; Zohir et al., 2005; Erdal and Gainer, 1979; Habib et al., 2004, 2002) considered in their studies a small number of operating variables and confined it to relatively narrow range. As a result, some investigators reported little increase, no increase, and even decrease in the rate of heat transfer. These conflicting in results showed that the heat transfer characteristics in pulsating flow are still not clearly understood. Due to the complicated nature of unsteady turbulent flow, too much theoretical investigations are needed to find a solution for the problems of hydrodynamics and heat transfer of such a flow. Therefore the experimental investigation is still the most reliable way to deal with the pulsating flow.

Several researchers have presented experimental, analytical and numerical studies on the effect of pulsation on heat transfer characteristics. The characteristic of laminar pulsating flow inside tube under uniform wall heat flux have been experimentally investigated by Habib et al. (2002), $6.6 \leqslant f \leqslant 68$ Hz. It is reported that an increase and reduction in Nusselt number are observed, depending on the values of both the frequency and Reynolds number. Zheng et al. (2004) used self-oscillator in their investigations and concluded that the convective heat transfer rate is greatly affected by the configuration of the resonator.

An analytical study on laminar pulsating flow in a pipe by Faghri et al. (1979) reported that higher heat transfer rates are produced. They related that to the interaction between the velocity and temperature oscillation which introduces an extra term in the energy equation that reflects the effect of pulsations. On the

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Nomenclature

A_0	dimensionless pulsation amplitude; x_{max}/D
A	area of the orifice meter (m^2)
C_p	specific heat capacity of air ($J\ kg^{-1}\ K^{-1}$)
D	pipe diameter (m)
f	frequency (Hz)
h_m	mean heat transfer coefficient; ($W\ m^{-2}\ K^{-1}$)
h_x	local heat transfer coefficient; ($W\ m^{-2}\ K^{-1}$)
k	thermal conductivity of air ($W\ m^{-1}\ K^{-1}$)
L	length of the test section (m)
L_e	effective length of the test section (m)
m	mass flow rate ($kg\ s^{-1}$)
N	revolution of rotating valve spindle per minute (rpm)
Nu_m	mean Nusselt number; (h_m D/k)
Nu_{sx}	local Nusselt number without pulsation (h_{sx} D/k)
Nu_{px}	local Nusselt number with pulsation (h_{px} D/k)
Nu_{sm}	mean Nusselt number without pulsation (h_{sa} D/k)
Nu_{pm}	mean Nusselt number with pulsation (h_{pa} D/k)
Pr	Prandtl number
Δp	pressure drop across the orifice meter ($N\ m^{-2}$)
Q	input heat (W)
Q_{trans}	total net heat transfer (W)
Q_{loss}	total heat loss through insulation (W)
q_o	total heat transfer heat flux ($W\ m^{-2}$)
Re	Reynolds number
Re_w	kinetic Reynolds number, $\omega D^2/\nu$
T_{bi}	mean inlet air temperature (K)
T_{bx}	local mean temperature (K)
T_{sx}	local pipe surface temperature (K)
x	local distance along the test section (m)
x_{max}	amplitude of fluid displacement (m)
u_{max}	max cross sectional mean velocity of pulsating flow ($m\ s^{-1}$)
U^*	friction velocity ($m\ s^{-1}$)
U_m	Mean velocity of mean flow ($m\ s^{-1}$)
U_A	Amplitude velocity of pulsating component ($m\ s^{-1}$)
U	time average velocity, U_A = (1.63–31.81) ($m\ s^{-1}$)
$U(t)$	velocity component at time t ($m\ s^{-1}$)

Greek symbols

β	(orifice diameter/pipe diameter) ratio
Δ	velocity ratio, $u_{max}/u_s = (A_0 Re_w/2Re)$
ε	velocity ratio = u_{max}/u_∞
η	local relative enhancement ratio, (Nu_{px}/Nu_{sx})
η_m	average relative Enhancement ratio, (Nu_{pm}/Nu_{sm})
λ	Womersly number, $D/2\sqrt{\omega/\nu}$
μ	dynamic viscosity of air ($N\ s\ m^{-2}$)
ν	kinematic viscosity ($m^2\ s_{-1}$)
ω	angular pulsation frequency (s^{-1})
ω^*	dimensionless frequency ($\omega D/U^*$)

Subscripts

b	bursting
I	inner
m	mean
p	pulsating state conditions
s	steady sate conditions
x	local

other hand, Chang et al. (2004) reported that the pulsation has no effect on the time averaged Nusselt number.

An investigation to pulsating pipe flow with different amplitude was carried out by Guo and Sung (1997). In case of small amplitudes, both heat transfer enhancement and reduction were detected, depending on the pulsation frequency. However, with large amplitudes, the heat transfer rates are always enhanced.

Hemeada et al. (2002) analyzed heat transfer in laminar incompressible pulsating flow, the overall heat transfer coefficient increases with increasing the amplitude and decreases with increasing the frequency and Prandtl number.

The effect of many parameters on time average Nusselt number was numerically studied by Cho and Hyun (1990), Moschandreou and Zamir (1997), Lee et al. (1998), Chattopadhyay et al. (2006) and Hesham et al. (2005b). It is reported that the increase of Nusselt number depends on the value of the pulsation frequency and its amplitude. With amplitude less than unity, pulsation has no effect on time averaged Nusselt number (Chattopadhyay et al., 2006). In the thermally fully developed flow region, a reduction of the local Nusselt number was observed with pulsation of small amplitude. However, with large amplitude, an increase in the value of Nusselt number was noticed.

In summary, the time average Nusselt number of a laminar pulsating internal flow may be higher or lower than that of the steady flow one, depending on the frequency. The discrepancies of heat transfer rate from that of the steady flow is increased as the velocity ratio (Δ) is increased. For hydrodynamically and thermally fully developed laminar pulsating internal flow, the local heat transfer rate in the axial locations for $X/D < \pi Re/20\ \lambda^2$ can be obtained based on a quasi-steady flow (Zhao and Cheng, 1998).

Numerical investigations on turbulent pulsating flow have been carried out by several researchers (Xuefeng and Nengli, 2005, 2001, 2003). It is reported that there is an optimum Womersly number at which the rate of heat transfer is enhanced (Xuefeng and Nengli, 2005, 2001). In pulsating turbulent flow through an abrupt pipe expansion, Said et al. (2003) reported that the percentage enhancement in the rate of heat transfer of about 10 was observed for fluids having a Prandtl number less than unity.

Experimental investigations on pulsating turbulent pipe flow have been conducted by many authors (Al-Haddad and Al-Binally, 1989; Hesham et al., 2005a; Barid et al., 1996; Gbadebo et al., 1999; Zheng et al., 2004; Zohir et al., 2005; Erdal and Gainer, 1979; Habib et al., 2004, 2002; Faghri et al., 1979). The results of Hesham et al. (2005a, 1996, 1999, 2004,) showed an increase and reduction in the mean Nusselt number with respect to that of the steady flow.

Many parameters have an influence on heat transfer characteristics of pulsating turbulent flow. Among those, pulsation frequency, its amplitude, axial location, Reynolds number, Prandtl number and pulsator type and its location. In order to understand the phenomena of the effect of pulsation on the heat transfer coefficient and to resolve these problems of contradictory results, different models of turbulence for pulsating flows were considered. Two of these models are well known and mostly applied; the quasi-steady flow model (Shemer, 1985) and the bursting model (Habib et al., 1999, 2004; Gbadebo et al., 1999; Zohir et al., 2005).

In order to check whether the steady flow analysis is applicable to the prediction of heat transfer analysis in a pulsating turbulent flow, Park et al. (1982) carried out a series of measurements on heat transfer to a pulsating turbulent flow in a vertical pipe subjected to a uniform heat flux over a range of $1.9 \times 10^4 \leqslant Re \leqslant 9.5 \times 10^4$. *Their data are presented in the form of the ratio* $\eta_m(Nu_{pm}/Nu_{sm})$ as a function of the Womersly number, λ. It is reported that for the cases of high mean Reynolds number ($Re > 5.5 \times 10^4$), the experimental data approaches to the quasi-steady predictions. However, at low mean Reynolds number ($Re < 4 \times 10^4$), the data show an increasing discrepancy as λ is increased.

In the previous studies, few of operating parameters are considered and were confined to relatively narrow range. Thus, in order to have a complete understanding of introducing pulsation into a flow with heat transfer, it is necessary to consider various parameters and cover a wide range of these controlling parameters. Therefore, the present work aims to investigate experimentally the effect of pulsation frequency as well as the Reynolds number on the heat transfer characteristics of turbulent pulsating pipe flow over a range of the frequency (6.6–68 Hz) and Reynolds number (10,850–37,100). The experiments are carried out while the outer surface of the pipe is subjected to a uniform heat flux.

2. Experimental setup and instrumentation

2.1. Test rig

The experimental facility shown in Fig. 2 is designed and constructed to investigate the effect of pulsation on the convective heat transfer from a heated pipe over a range of $10^4 < Re < 4 \times 10^4$. It is an open loop in which air as a working fluid is pumped and passed the test section to the atmosphere after being heated. The rig basically consists of three part; the air supply unit with necessary adoption and measuring devices, the test section and the pulsating mechanism.

The air supply unit and its accessories consists of an air compressor, storage tanks, flow control valves, orifice meter, settling chamber and calming joints.

2.1.1. Air supply unit

Air is pumped by a compressor (1) into storage tanks (2), from which the air is discharged to the system as shown in Fig. 2. The airflow rate was controlled by a two flow control valves (3) and (4), placed upstream of a calibrated orifice meter (6) which measured the flow rate of the air using a U-tube manometer connected to a two static pressure taps, one 1.0 D upstream and the other 0.5 D down stream of that sharp edged orifice plate. Besides controlling the flow, the control valves also served as a major pressure drop in the system.

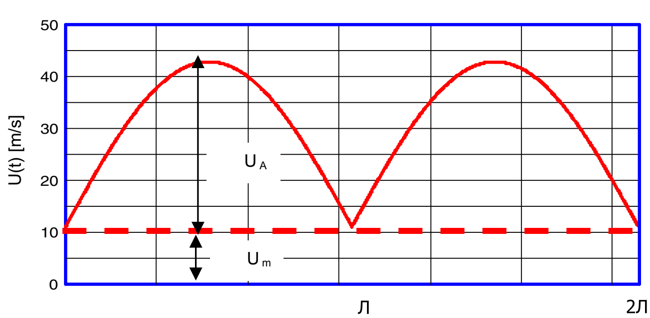

Fig. 1. Variations of inlet velocity.

The air flow passes through a settling chamber (7) to reduce the disturbance of the flow in the pipe. Afterward the air was passed over the test section (9) via the upstream clamping joint (8) and then exhausted to the atmosphere after being passed through the pulsating valve (14).

The flow measuring sections as well as rest of test loop were completely isolated from vibrations source by two flexible connections (5). The flow enters the test section at uniform temperature and fully developed turbulent after being stabilized and adjusted.

2.1.2. Test section

The test section (9) is shown schematically in Fig. 3. It is a copper tube of 15.5 mm inner diameter, 22 mm outer diameter and 1200 mm in length ($L/D_i = 77$). The test section is clamped from both sides by flexile joints.

The pipe wall temperatures are measured by 12 k-type thermocouples distributed along the tube surface from outside as shown in Fig. 2. Thermocouple junctions were fitted into 12 holes along the outer surface of the pipe, each of 2 mm in diameter and 2.5 mm in depth.

The thermocouple wires were then embedded inside the grooves milled in the outer pipe surface parallel to its axis by epoxy collected out of the test section and connected to a multi-channel temperature recorder via a multipoint switch.

The main heater of 17.3 m total length is a nickel chromium wire which has a resistivity of 15.5 Ω/m was divided into four equal lengths to heat up the test section tube. The tapes of the heaters were electrically insulated by fitting them inside very ductile Teflon pipes of 0.1 mm thickness and 2 mm in diameter and wrapped uniformly along the outside tube surface. These heater tapes were sandwiched by aluminum foils of 0.2 mm thickness for achieving a uniform heat distribution. Two auto-transformers were used to supply and control the power of the four electric heater sections, which are connected in parallel with the required voltage.

A layer of glass wool insulation of 35 mm thickness was applied over the heater, followed by a thin sheet of aluminum foil. Moreover, another 45 mm layer of glass wool was applied to assure perfect insulation of the tested pipe.

The distribution of thermocouples along the tested tube was concentrated at the beginning to measure the dramatic change of the temperature in the entrance region. The locations of these thermocouples relative to pipe diameter are shown in Fig. 2, and are as

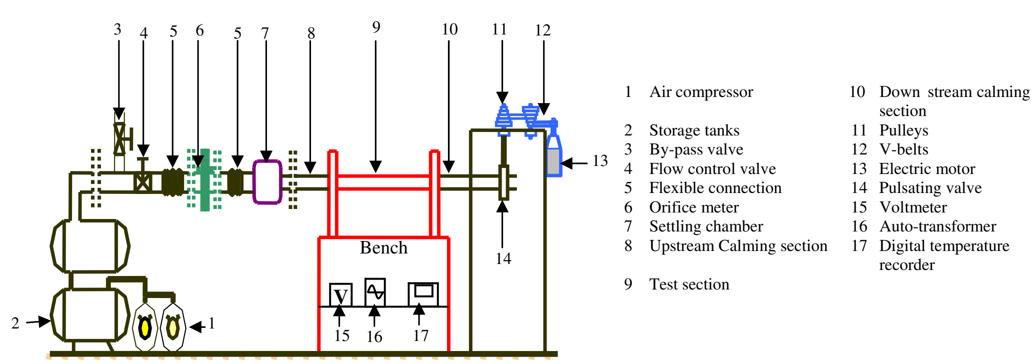

Fig. 2. Experimental equipment.

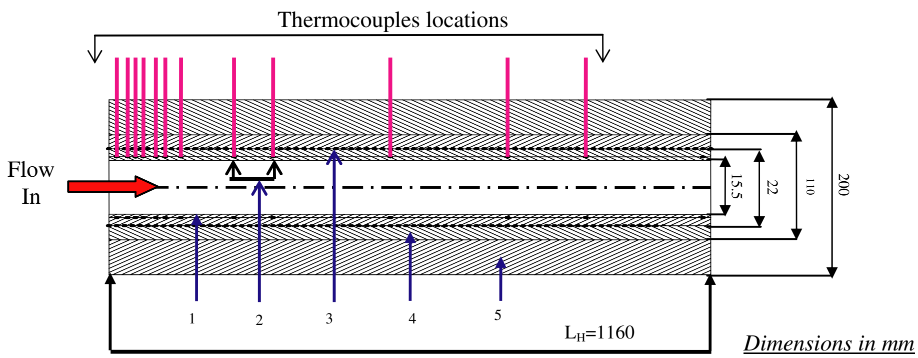

Fig. 3. Test section.

follows: X/D_i = 0.5, 1.0, 2.0, 3.0, 4.6, 6.3, 8.0, 15.0, 20.0, 35.0, 45.0, 60.0.

The bulk air temperatures at inlet and outlet of the test section were measured by two k-type thermocouples inserted in two holes drilled through the flanges at the entrance and exit of this test section.

2.1.3. Pulsating mechanism

The pulsating mechanism (11)–(14) shown in Fig. 1 is located down stream at the exit of the tested pipe. It is constructed of three main parts; an AC electric variable speed motor (13), a variable speed transmission mechanism 11, 12, and a rotating butterfly valve (14) of 15.5 mm inner diameter. The valve spindle was connected to the motor through two stepped pulleys and two V-belts. The output of the transmission mechanism, which is connected to the pulsator valve spindle through a sleeve, could be adjusted manually to rotate the butterfly valve with variable speed within the range of 200 up to 2050 rpm.

Pulsation in the air stream was generated by the butterfly valve. The pulsator valve could be adjusted to rotate to give different frequencies, which were measured by a digital tachometer. The valve has the same inner diameter of the test section pipe and was located at 300 mm downstream the end of the test section tube.

For each revolution of the butterfly valve, the flow is stopped and released twice, so the frequency of pulsation can be expressed as

$$f = \frac{(N \times 2)}{60} \tag{1}$$

Sinusoidal pulsating flow is assumed to be entering to the pipe as shown in Fig. 1, where the velocity only is oscillating. This assumption is considered when comparing the experimental results with numerical predictions reported by Elshafei et al. (2007), (illustrated in Figs. 22–25). Thus the velocity of flow is a periodic function of time and can be expressed as

$$U(t) = U_{\mathrm{m}} + \mathrm{abs}(U_{\mathrm{A}} \sin \omega t) \tag{2}$$

3. Procedure and calculations

3.1. Experimental procedure

In this investigation, an experimental program is conducted to study the heat transfer characteristics of steady and pulsating turbulent pipe air flow. Several parameters affect the performance of heat transfer of such a flow. Among all the amplitude and frequency of pulsation, Reynolds number and the location of pulsation mechanism relative to the test-section may have the great effect. Experiments in both cases, steady and pulsating flow were executed while the pipe wall is heated with a fixed uniform heat flux of 922 W/m^2 and the pulsation mechanism was located downstream of the test section. The mass flow rate of air was adjusted and held unvaried while varying the pulsation frequency from 0.0 up to 68 Hz. The investigation covered different values of Reynolds numbers in the range of $10^4 < Re < 4 \times 10^4$.

3.2. Data collection

The pressure drop across the orifice plate (Δp), is measured and used to calculate the maximum mass flow rate of air as White (2003).

$$\dot{m} = C_{\mathrm{d}} \rho_{\mathrm{air}} A \left| \frac{2\Delta p}{\rho_{\mathrm{air}}(1-\beta^4)} \right|^{\frac{1}{2}} \tag{3}$$

The discharge coefficient; C_{d} is obtained form calibration of the orifice meter. To measure the heat input to the test section through the heating tapes segments, the digital multi-meter that measures input currents, voltages and power simultaneously was used. The voltage coming from the main voltage stabilizer was adjusted through the two auto-transformers to achieve uniform wall heat flux.

The results are parameterized by the factors having more influence on heat transfer characteristics of flow under investigation. Reynolds number and dimensional frequency of turbulent flow are playing important roles. The maximum Reynolds number of air flowing in a pipe of diameter D_i is given by

$$Re = \frac{4\dot{m}}{\pi D_i \mu} \tag{4}$$

The dimensional frequency of turbulent pipe flow is defined as

$$\omega^* = \omega D / U^* \tag{5}$$

where ω is the angular pulsation frequency, given by

$$\omega = 2\pi f, \tag{6}$$

and U^* is the friction velocity described as Habib et al. (2004)

$$U^* = 0.199 U_{\mathrm{m}} / Re^{0.125} \tag{7}$$

Axial heat loss by conduction from both ends of the tested tube was eliminated by a two Teflon washers located between the test section flanges. The heat loss in radial direction through insulation was checked and found to be about 2% of heat input that can be ignored. So

$$Q_{\mathrm{trans}} = (Q - Q_{\mathrm{loss}}) = \dot{m} c_{\mathrm{pm}} (T_{\mathrm{bo}} - T_{\mathrm{bi}}) \tag{8}$$

The local bulk mean temperature of the air at the end of each segment was determined using energy balance (Incropera and Dewitt, 1996) as follows:

$$q_o = \frac{\dot{m}C_{pm}(T_{bx} - T_{bi})}{\pi D L_x}, \tag{9}$$

$$T_{bx} = \frac{\dot{q}_\circ \pi D L_x}{\dot{m}C_{pm}} + T_{bi} \tag{10}$$

The local mean bulk temperatures at the corresponding points along the test section are determined by linear interpolation of the mean temperature at the end of the heater segments, where the pipe wall temperature is measured. The local and mean heat transfer coefficients are determined, respectively by

$$h_x = \frac{\dot{q}_\circ}{T_{sx} - T_{bx}}, \tag{11}$$

$$h_m = \frac{1}{L}\int_0^L h_x \mathrm{d}x \tag{12}$$

Since the values of the local heat transfer coefficient are discrete, the above integration is carried out numerically using trapezoidal rule (Chapra and Canale, 1998). The local and mean values of Nusselt numbers are calculated, respectively as

$$Nu_x = \frac{h_x D}{k_a}, \tag{13}$$

$$Nu_m = \frac{h_m D}{k_a} \tag{14}$$

The value of k_a is detected at the mean bulk temperature of the flowing air.

The frequency of pulsation; f is determined by employing a digital tachometer. A photo electric probe emits light on to the flange that connected by the butterfly valve, which is taped at one side with a reflector. As the flange rotates, the reflection is counted by the probe. The number of reflections per minute is a measure of the speed of the motor, hence the frequency of pulsation.

Uncertainty analyses of various measured parameters were carried out based on method of Kline and McClintock (1953). The uncertainty levels in the mean Nusselt number is ±5.2%.

4. Results and discussion

The present investigation includes 82 runs for different values of Reynolds number and frequency. Each pulsation test was preceded by steady flow one at the same Reynolds number. For the sake of validating the present experimental data, it has been compared with that reported in literature (Dittus and Boellter, 1930) for the steady flow, which is given by

$$Nu_m = 0.023Re^{0.8}\ Pr^{0.4} \tag{15}$$

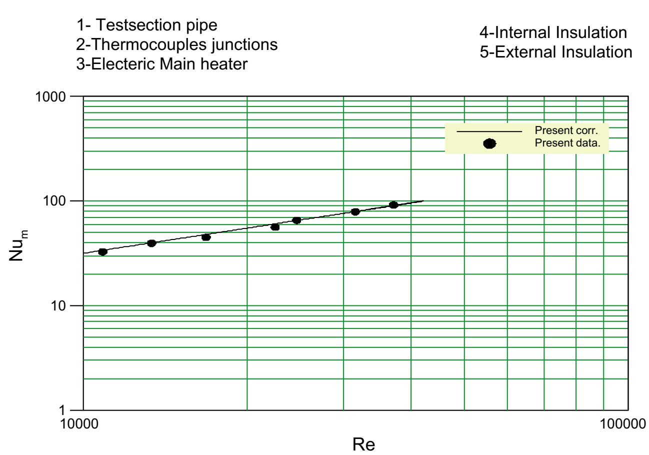

Fig. 4. Comparison of measured fully developed Nusselt numbers and correlated data of Eq. (13) (steady flow), refer to Dittus and Boelter correlation.

It is shown from Fig. 4 that the present data for the steady flow are in good agreement with the published data.

4.1. Pulsating flow

The effect of pulsation on the heat transfer characteristics are presented in terms of both relative local and relative mean Nusselt number defined as the ratio of the value of local and mean Nusselt number for pulsated flow to the corresponding ones for steady flow at the same Reynolds numbers. The mean Nusselt number is calculated by integrating the local Nusselt number along the pipe as explained earlier. The thermally fully developed flow was attached at $X/D \geqslant 15$, because the value of the local Nusselt number is normally constant.

Results showed that the relative local value of Nu either increased or decreased with increasing both the value of Reynolds number and pulsation frequency. This can be declared from the following sections.

For pulsation frequency of 6.7 Hz, it can be seen from Fig. 5 that little enhancement in the ratio of local Nusselt numbers was accompanied with Reynolds number of 16,800, 22,500 and 37,100 in the entrance region; with about 8% for Re equals to 37,100 and the percentage of enhancement smoothly decreased with X/D. However, for the rest of the tested flow Reynolds number, a reduction in the local Nu with X/D was observed (with maximum reduction of about 11% at the tested pipe end for Re of about 13,350.

For f = 13.3 Hz, the enhancement was raised up to 14% for the highest value of Reynolds number of about 37,100 at X/D equal 0.5 and 15, the maximum reduction in the local Nu of 11%, at Reynolds number of 13,350 as can be seen from Fig. 6.

For f = 20.7 Hz, the % age of enhancement is about 8% at inlet for Reynolds number of 37,100, while suffering reduction for the other values of Reynolds number as indicated in Fig. 7.

Another set of experiments have been carried out at higher frequencies to assess such ambiguous phenomena. For f equals to 28.3 and 39.3 Hz, Nu increased only for Reynolds numbers of 22,500 and 37,100, respectively. As can be seen from Figs. 8 and 9, the enhancement at Reynolds number of 37,100 was of about 19% and 14% at the inlet of the tested pipe, respectively.

For $42.5 \leqslant f \leqslant 49.2$ Hz, over a range of $10{,}850 \leqslant Re \leqslant 37{,}100$, an enhancement and reduction in the value of local Nu are observed as shown in Figs. 10–12. The maximum enhancement was of about 7% for the operating conditions of f = 42.5 Hz, Re = 22,500 and the maximum reduction of about 17% for the case when f = 42.5 Hz and Re = 13,350. The enhancement is always noticed at the entrance of the pipe.

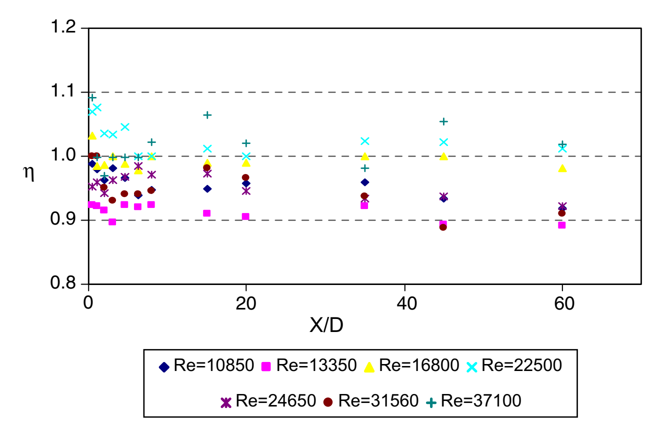

Fig. 5. Relative local Nusselt number of pulsated flow for different Re (f = 6.7 Hz).

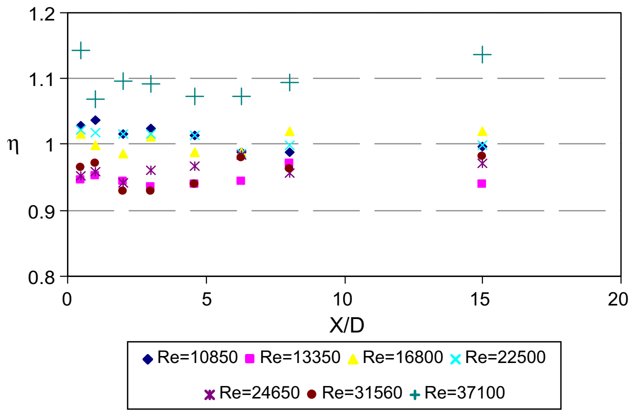

Fig. 6. Relative local Nusselt number of pulsated flow for different *Re* (f = 13.3 Hz).

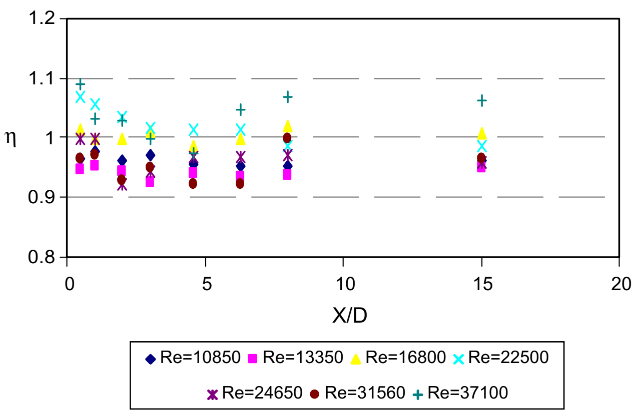

Fig. 7. Relative local Nusselt number of pulsated flow for different *Re* (f = 20.7 Hz).

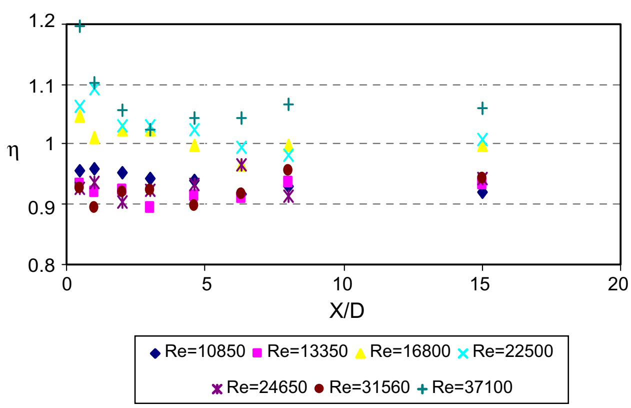

Fig. 8. Relative local Nusselt number of pulsated flow for different *Re* (f = 28.3 Hz).

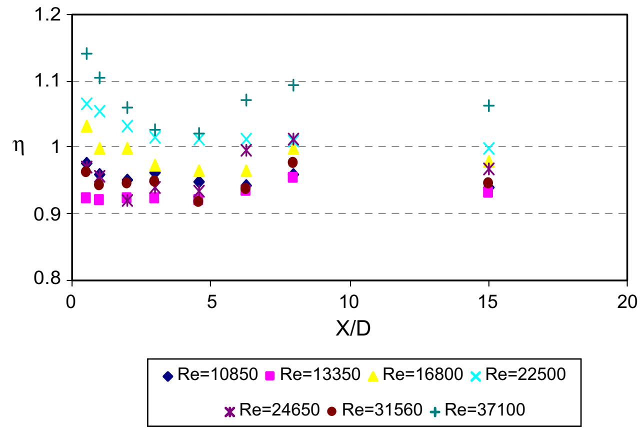

Fig. 9. Relative local Nusselt number of pulsated flow for different *Re* (f = 39.3 Hz).

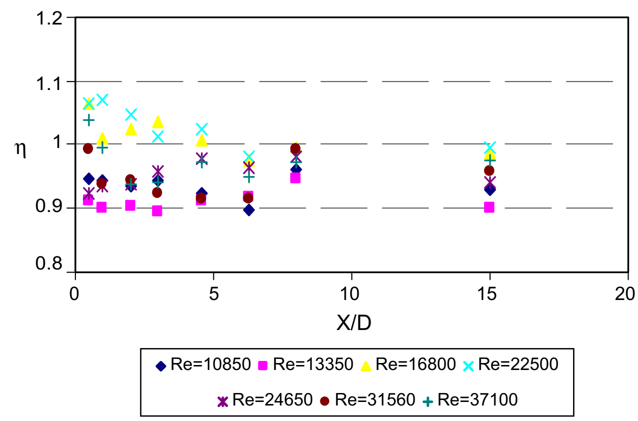

Fig. 10. Relative local Nusselt number of pulsated flow for different *Re* (f = 42.5 Hz).

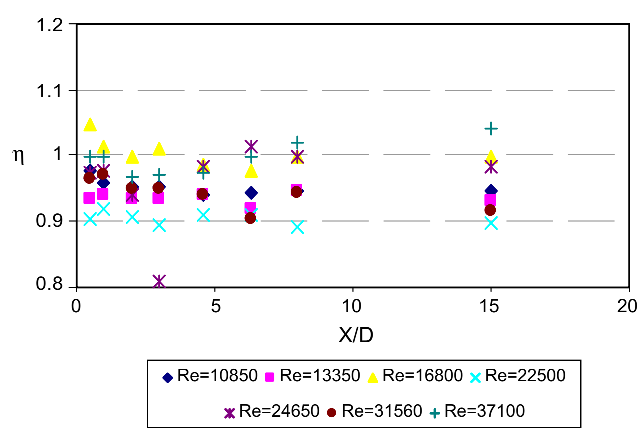

Fig. 11. Relative local Nusselt number of pulsated flow for different *Re* (f = 45.7 Hz).

The same trend can be observed in Figs. 13–15 for frequency $53.3 \leqslant f \leqslant 68$ Hz respectively.

The always happening of enhancement in Nusselt number for certain operating conditions (frequency Reynolds number combination) close to the pipe entrance may be argued to the agitation of the thermal boundary layer whose growth has just begun by those imposed matched frequency with Reynolds number.

The effect of pulsation frequency on the ratio of the mean Nusselt number for pulsated turbulent flow to that for steady flow; η_m at various Reynolds number is shown in Figs. 16 and 17. Over the tested range of $6.7 \leqslant f \leqslant 68$ Hz and for $10{,}850 \leqslant Re \leqslant 22{,}500$, it can be seen from Fig. 16 that for Reynolds number equals to 10,850 and 13,350, the ratio η_m is slightly decreases with increasing pulsation frequency. At any pulsation frequency it can be noticed that the value of η_m is firstly less than unity at Reynolds number of 10,850 after which it slightly decreases for Reynolds number of 13,500, followed by a noticeable increase at Reynolds number of 16,800 and 22,500. Also the same trend for η_m as a function of pulsation frequency can be noted in Fig. 17 for a range of 24,650 $\leqslant Re \leqslant$ 37,100.

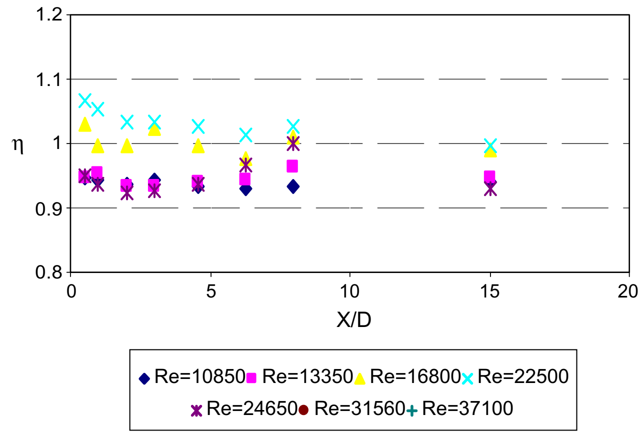

Fig. 12. Relative local Nusselt number of pulsated flow for different *Re* (f = 49.2 Hz).

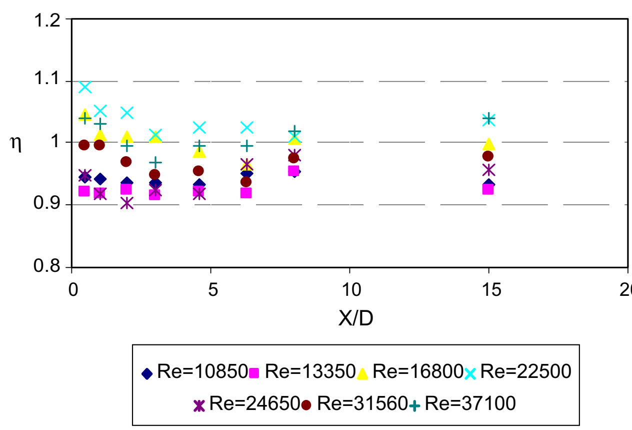

Fig. 13. Relative local Nusselt number of pulsated flow for different *Re* (f = 53.3 Hz).

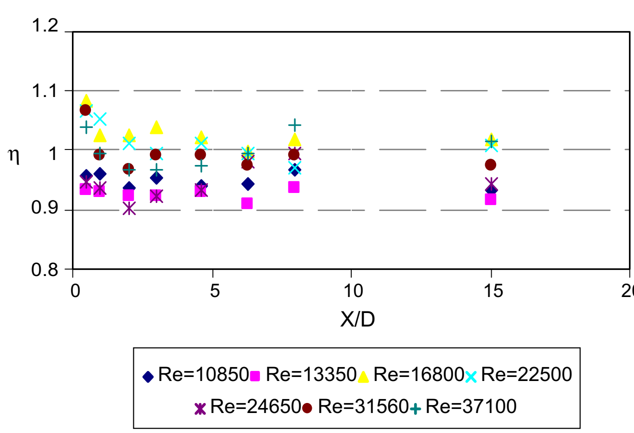

Fig. 14. Relative local Nusselt number of pulsated flow for different *Re* (f = 61.7 Hz).

An interesting observation in Fig. 17 is that at any value of pulsation frequency, the value of η_m is less than unity at *Re* equals to 24,650 and 31,560 after which it suddenly increases to more than unity in the range of $0 \leqslant f \leqslant 40$ Hz.

The variation of η_m versus Reynolds number over a pulsation frequency range of $6.7 \leqslant f \leqslant 68$ is shown in Figs. 18 and 19 from which the previous obtained results can be clearly noticed having the same trend. These results agree enough with that reported by Zhao and Cheng (1998), Gbadebo et al. (1999) and Faghri et al. (1979).

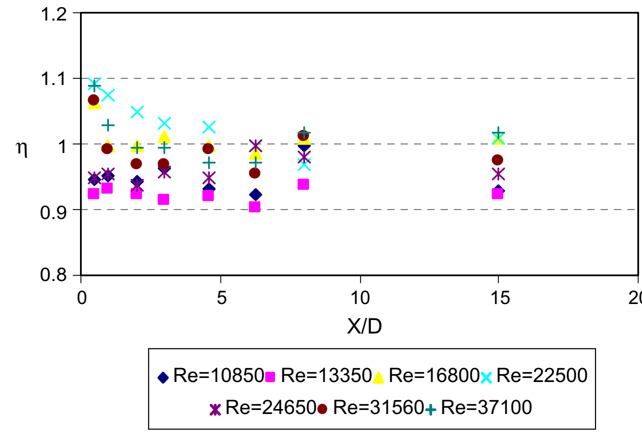

Fig. 15. Relative local Nusselt number of pulsated flow for different *Re* (f = 68 Hz).

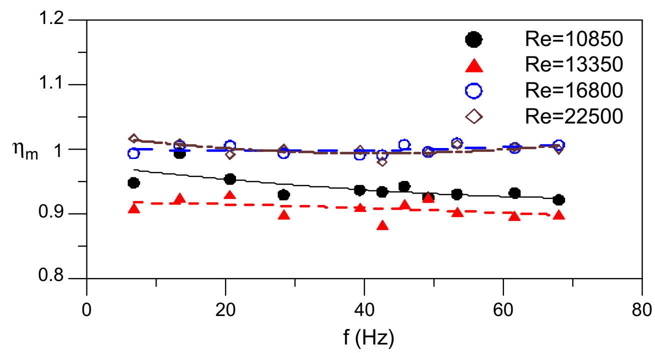

Fig. 16. Relative mean Nusselt number as a function of pulsation frequency.

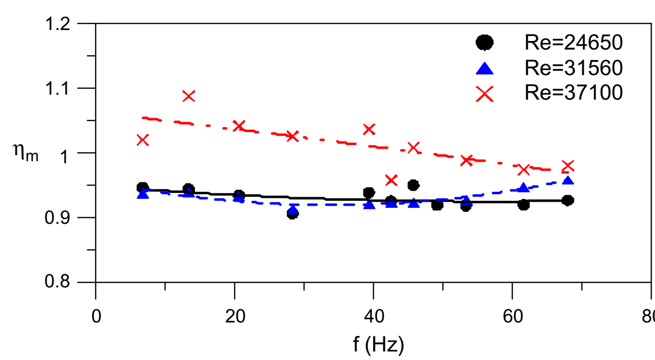

Fig. 17. Relative mean Nusselt number as a function of pulsation frequency.

The heat transfer results are also presented in the form relative mean *Nu* as a function of turbulent dimensionless frequency; ω^* that combine the effect of both Reynolds number and pulsation frequency. In the frequency range of 6.7 up to 68 Hz and Reynolds number of $10{,}850 \leqslant Re \leqslant 24{,}650$ ($0.28 < \omega^* < 8.4$), the experimental data are shown in Figs. 20–22. It is generally seen for all tested Reynolds numbers that the value of η_m slightly decreases with increasing ω^*. In the mean time, at a fixed value of ω^*, η_m fluctuates periodically as Reynolds number changed.

The noticed enhancement and reduction of heat transfer rate accompanied with pulsating turbulent flow need to be discussed and analyzed.

The heat transfer characteristics of such a flow are influenced by several parameters. These are: Reynolds number, the imposed

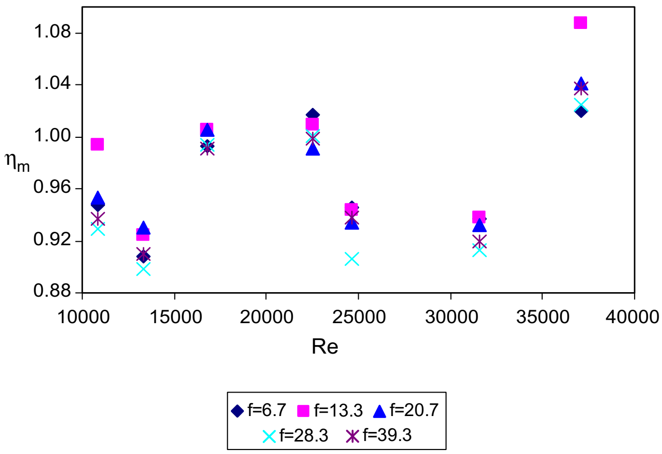

Fig. 18. Relative mean Nusselt number as a function of flow Reynolds number.

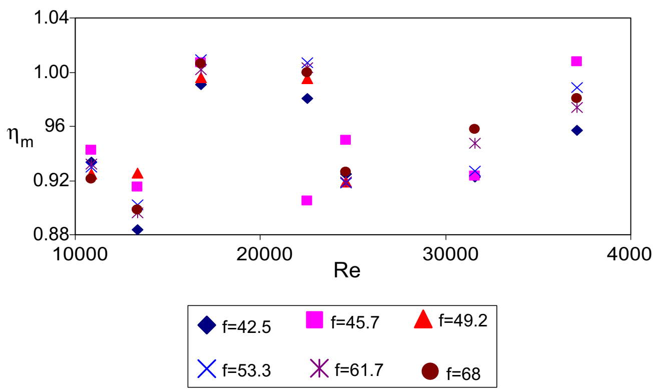

Fig. 19. Relative mean Nusselt number as a function of flow Reynolds number.

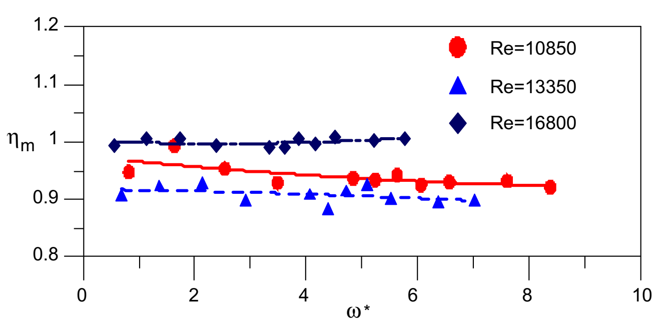

Fig. 20. Relative mean Nusselt numbers as a function of dimensionless turbulent frequency.

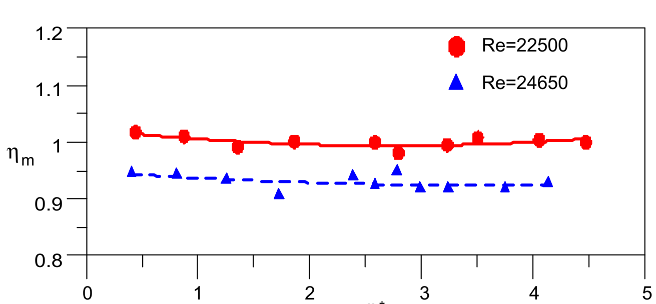

Fig. 21. Relative mean Nusselt numbers as a function of dimensionless turbulent frequency.

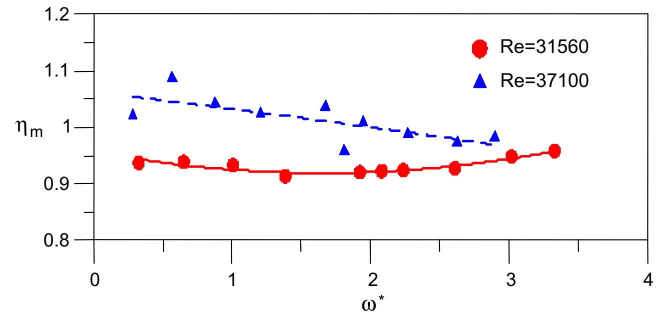

Fig. 22. Relative mean Nusselt numbers as a function of dimensionless turbulent frequency.

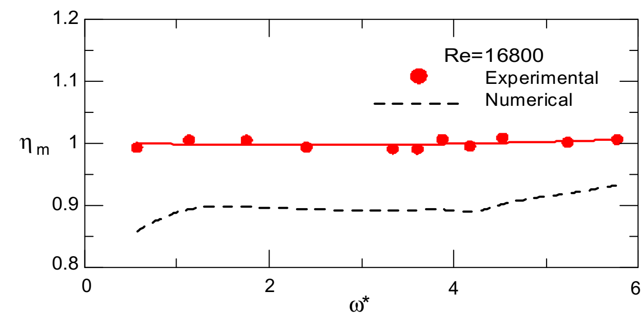

Fig. 23. Comparison between experimental and numerical values at $Re = 16{,}800$.

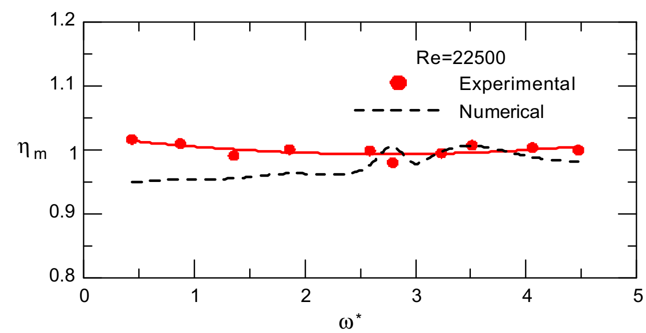

Fig. 24. Comparison between experimental and numerical values at $Re = 22{,}500$.

frequency and its amplitude, Prandtl number, length to diameter ratio, location of pulsator, pipe diameter and type of pulsator.

A comparison between the present experimental results with the numerical ones (Elshafei et al., 2007), over a ranges of $16.8 \times 10^3 \leqslant Re \leqslant 37.1 \times 10^3$ and $6.67 \leqslant f \leqslant 68$ Hz are described in Figs. 23–25. The data are presented in the form of η_m as a function of bursting frequency ω^*. It can be observed that for all values of Reynolds number, the numerical predictions of η_m as a function of ω^* have the same trend as those for the experimental data. The discrepancies between the experimental data and the computed ones at low bursting turbulent frequency are of about 10%. As ω^* increases, the computed results closes to the experimental ones, specially, at higher value of Reynolds number.

The pulsation of this turbulent flow is related to bursting phenomenon that occurs in the turbulent boundary layer. These turbulent bursts are a series of quasi-cyclic or periodic activities near the wall. Therefore, affecting the heat transfer processes.

The bursting model (Hemeada et al., 2002) is widely used in the analysis of heat transfer characteristics of such pulsating turbulent

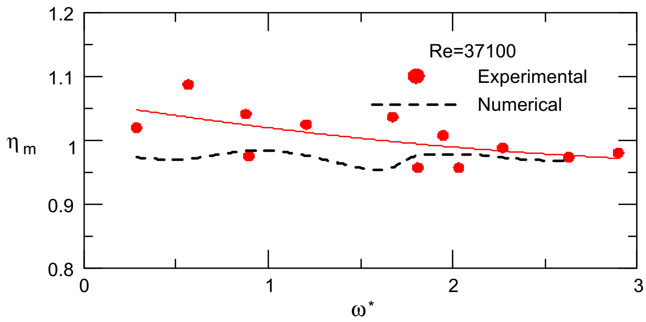

Fig. 25. Comparison between experimental and numerical values at $Re = 37{,}100$.

flow. Liao and Wang (1985) and Genin et al. (1992) investigated the hydrodynamics and heat transfer with pulsating turbulent fluid flow in pipes and reported that, when the imposed frequency is close to the bursting frequency, resulting in a renewable viscous sub-layer, and hence causes the occurrence of certain resonance interaction. This interaction may change strongly the heat transfer characteristics of turbulent flow.

The heat transfer results of the present study and that obtained by other investigators (Habib et al., 1999, 2004; Gbadebo et al., 1999; Liao and Wang, 1985; Havemann and Rao, 1954; Artineelli et al., 1943) are discussed and analyzed in view of that turbulent bursting model presented in Fig. 26. These collected heat transfer data are classified and presented in the form of dimensionless frequency; ω^* versus Re for each investigation. As shown in Fig. 22, there are three lines bounding the level of bursting frequency; the upper limit, the mean and the lower limit of this bursting frequency.

Ordinates of ω^* as function of Re for different regions are given to define pulsating flow regimes. In quasi-steady condition; represented by region (A), the rate of heat transfer will be reduced due to pulsation of the turbulent flow (Barid et al., 1996; Liao and Wang, 1985). Regions (B) and (C) are categorized as preferred regions in which the imposed pulsation frequency becomes close to the bursting frequency, leading to a resonance interaction of both frequencies. This interaction agitates the boundary layer and therefore the rate of heat transfer is expected to increase with the imposed pulsation, as reported by Liao and Wang (1985), Genin et al. (1992) and Mamayev et al. (1976).

Few experiments have been carried out in region D (Gbadebo et al., 1999; Ramaprian and Tu, 1983) who reported that the heat transfer coefficient is increased by about 27% due to pulsation. The heat transfer rate is expected to increase in this region due to the strong interaction between the imposed frequency and the turbulent bursting processes at the wall as reported by Ramaprian and Tu (1983).

Most of the data belong to Habib et al. (1999, 2004) and Havemann and Rao (1954) lay in regions B and C, where the heat transfer coefficient increases with pulsating frequency above a certain value. This conclusion is supported by Liao and Wang (1985) who reported that the rate of heat transfer might be increased with higher imposed frequency than that of bursting for the corresponding steady flow.

The bulk of the present data fall in one of the suggested preferred regime of bursting frequency, especially, in the intermediate frequency region represented by regime (B) shown in Fig. 26. In this regime, the mean bursting frequency is subdued to the pulsation frequency leading to the occurring of resonance that is dependent only on the pulsation frequency. Accordingly, the heat transfer process is expected to be affected resulting in either reduction or enhancement in the rate of heat transfer.

5. Conclusion

The effects of frequency and Reynolds number on heat transfer characteristics of pulsating turbulent pipe flow are experimentally investigated. The pipe was subjected to a uniform heat flux and the pulsator was placed down stream of the tested pipe. From the obtained results, the following may be concluded:

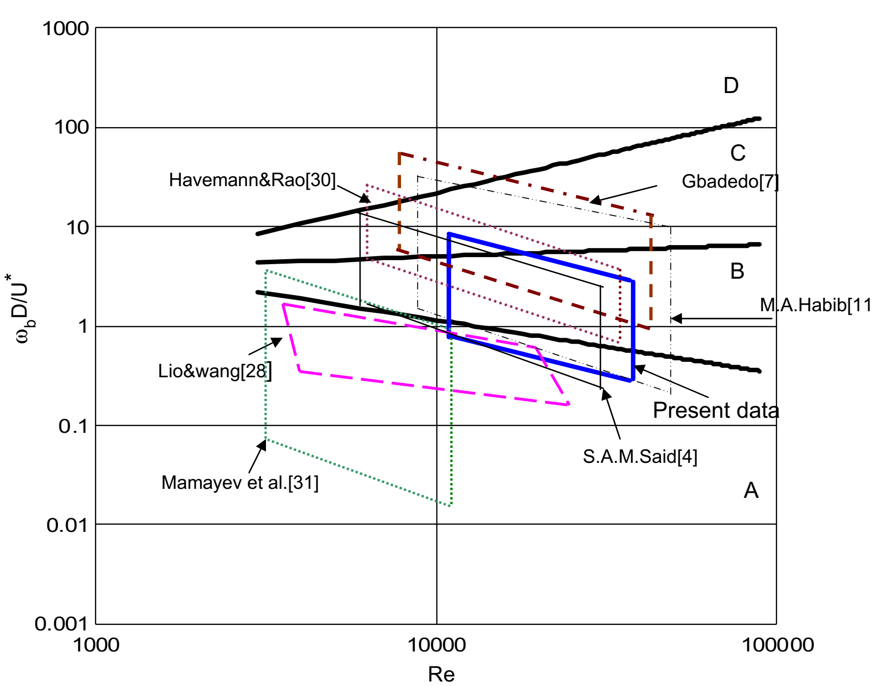

Fig. 26. Classification of data of heat transfer with turbulent pulsating flow.

- Observations of the local Nusselt number revealed that the heat transfer coefficient may be increased or decreased, depending on the value of frequency and Reynolds number.
- Higher values of the local heat transfer coefficient occurred in the entrance of the tested tube.
- The maximum enhancement in η_m of about 9% is observed at $Re = 37{,}100$, $f = 13.3$ Hz and ω^* of about 0.5 where the interaction between the bursting and pulsation frequencies is expected.
- The maximum reduction in η_m of about 12% was detected for $Re = 13{,}350$, $f = 42.5$ Hz and ω^* of about 4.4.

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