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Numerical study of heat transfer in pulsating turbulent air flow

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A numerical investigation of heat transfer characteristics of pulsating turbulent flow in a circular tube is carried out. The flow is thermally and hydrodynamically fully developed and the tube wall is subjected to a uniform heat flux. The flow inlet to the pipe consists of fixed component and pulsating component that varies sinusoidally with time. The flow and temperature fields are computed numerically using computational fluid dynamics (CFD) Fluent code. Prediction of heat transfer characteristics is performed over a range of $10^4 \leq Re \leq 4 \times 10^4$ and $0 \leq f \leq 70$ Hz are observed. Results showed little reduction in the mean time-averaged Nusselt number with respect to that of steady flow. However, in the fully developed established region, the local Nusselt number either increases or decreases over the steady flow-values depending on the frequency parameter. These noticed deviations are rather small in magnitude for the computed parameter ranges. The characteristics of heat transfer are qualitatively consistent with the available experimental and numerical predictions.

Key words: Nusselt number, heat transfer, computational fluid dynamics (CFD), pulsating flow in pipe.

INTRODUCTION

The study of pulsating flow in pipe has been a subject of interest among many researches. The problem is important in biological applications in relation to blood flow and in industrial applications related to heat exchange efficiency, cavitation in hydraulic pipe lines, pressure surges and reciprocating machines. In general, pulsating flow field is assumed to consist of a steady poiseuille flow part and purely oscillating part. Many experimental, analytical and numerical works studied the heat transfer in pulsating laminar flow but little in turbulent flow. One of the key issues concerning pulsating convection heat transfer in tubes is whether a superposed flow pulsation effects on heat transfers in the original steady flow. The answer to this question in the previous studies can be classified into four different opinions: (1) flow pulsation enhances heat transfer (Zheng et al., 2004; Xuefeng and Nengli, 2005; Faghri et al., 1979); (2) it deteriorates heat transfer (Mostafa, 2005a, b; Hemeada et al., 2002; Guo and Sung, 1997); (3) it has no effect on heat transfer (Chattopadhyay et al.,

2006); and (4) it either enhances or deteriorates heat transfer, depending on flow parameters (Al-Haddad and Al-Binally, 1989; Martineelli et al., 1943; Cho and Hyun 1990; Gbadebo et al., 1999; Gupa et al., 1982; Moschandreou and Zamir, 1997; Said et al., 2003; Scotti and Piomelli, 2001; Zhao and Cheng, 1998; Zohir et al., 2005).

Numerical study presented by (Xuefeng and Nengli, 2005) showed that in a pulsating turbulent flow there is an optimum Womersley number at which heat transfer is maximally enhanced. Experimental studies by (Al-Haddad and Al-Binally, 1989; Habib et al., 1999; Barid et al., 1996; Habbib et al., 2002; Liao and Wang, 1985; Martineelli et al., 1943) showed that increment and reductions in mean Nusselt number depends on pulsation frequency, turbulent bursting frequency, amplitude, axial location, Reynolds number and Prandtl number. In order to understand these phenomena and to resolve the contradictory results, different models of turbulence for pulsating flows were considered. Two of these models are well known, the quasi-steady flow model (Shemer, 1985) and the bursting model discussed by (Liao and Wang, 1985; Genin et al., 1992; Havemann and Rao, 1954; Martineelli et al., 1943).

Due to its complexity, studies on heat transfer in

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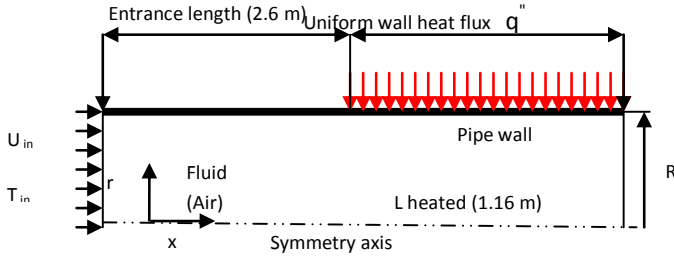


Figure 1. Schematic diagram for the studied problem.

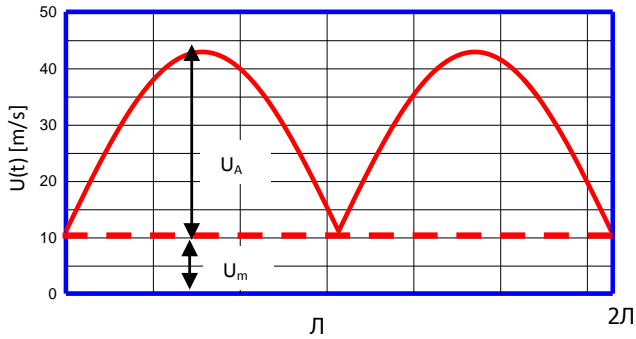


Figure 2. Variations of inlet velocity.

work aims to numerically and experimentally investigate the characteristics of heat transfer in pulsating pipe flow and how it is affected by pulsation frequency as well as the value of Reynolds number.

In the present study, a pipe subjected to a constant heat flux from outside is considered. The governing equations are solved with the aid of CFD code (Fluent 6-1, 2003). The obtained numerical results can be served as a useful source material against which the outcome of future analytical and experimental investigations can be checked.

MATHEMATICAL FORMULATIONS

Pipe geometry

In the present study, the numerical solution of the pulsating turbulent flow through an externally heated pipe is analyzed. Air is selected as a working fluid. The radius and length of the pipe are R and L respectively. The thickness of the pipe is neglected and the thermal boundary condition on the pipe wall is assumed to be a uniform heat flux. The two-dimensional axisymmetric model of the considered pipe is shown in Figure 1.

The pipe geometry at the inlet is long enough to assure a hydrodynamically fully developed flow at the heated section and simulated to the experiential facility of

Huseyin et al. (2005). Since the pipe cross-section is circular, it is assumed that the flow is axisymmetric. In cylindrical polar coordinates, this means that the flow variables depend only on the axial coordinate x and radial coordinate r . The pipe wall section is modeled as: unheated section of 2.6 m length and a heated one of 1.16 m long. Sinusoidal pulsating flow is assumed to be entering to the pipe of fixed period (the velocity is only oscillates) as can be seen in Figure 2.

The governing conservation equations

The governing equations for transient turbulent incompressible flow and heat transfer in the flow region ($0 \leq x \leq L$ and $0 \leq r \leq R$) can be written as follows (Huseyin et al., 2005):

Continuity:

$$\frac{\partial u}{\partial x} + \frac{v}{r} + \frac{\partial v}{\partial r} = 0 \quad (1)$$

Momentum in x-direction:

$$\begin{aligned} & \rho \left[\frac{\partial u}{\partial t} + \frac{1}{r} \frac{\partial}{\partial x} (ruu) + \frac{1}{r} \frac{\partial}{\partial r} (rvu) \right] \\ & = -\frac{\partial p}{\partial x} + \frac{1}{r} \frac{\partial}{\partial x} \left\{ r\mu \left[2 \frac{\partial u}{\partial x} - \frac{2}{3} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial r} + \frac{v}{r} \right) \right] \right\} \\ & \quad + \frac{1}{r} \frac{\partial}{\partial r} \left[r\mu \left(\frac{\partial u}{\partial r} + \frac{\partial v}{\partial x} \right) \right]. \end{aligned} \quad (2a)$$

Momentum in r-direction:

$$\begin{aligned} & \rho \left[\frac{\partial v}{\partial t} + \frac{1}{r} \frac{\partial}{\partial x} (ruv) + \frac{1}{r} \frac{\partial}{\partial r} (rvv) \right] \\ & = -\frac{\partial p}{\partial r} + \frac{1}{r} \frac{\partial}{\partial r} \left[r\mu \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial r} \right) \right] \\ & \quad + \frac{1}{r} \frac{\partial}{\partial r} \left\{ r\mu \left[2 \frac{\partial v}{\partial r} - \frac{2}{3} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial r} + \frac{v}{r} \right) \right] \right\} \\ & \quad - 2\mu \frac{v}{r^2} + \frac{2}{3} \frac{\mu}{r} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial r} + \frac{v}{r} \right). \end{aligned} \quad (2b)$$

Energy:

$$\begin{aligned} & \rho C_p \left(\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial r} \right) \\ & = \lambda_{eff} \left[\frac{\partial^2 T}{\partial x^2} + \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) \right] + \mu \phi \end{aligned} \quad (3a)$$

Where ϕ is viscous dissipation term, which is

$$\varphi = 2 \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial r} \right)^2 + \left(\frac{v}{r} \right)^2 \right] + \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial r} \right)^2, \quad (3b)$$

and λ_{eff} is the effective thermal conductivity, and for standard k - ε model, is given by

$$\lambda_{\text{eff}} = \lambda + \lambda_t \quad (3c)$$

$$\lambda_t = C_p \mu_t / Pr_t \quad (3d)$$

Turbulence model

Among the available turbulent models, those based on Eddy viscosity are widely applied to the turbulent kinetic energy – turbulent energy dissipation rate (k - ε). Two additional equations for the standard k - ε turbulence model: the turbulence kinetic energy, k , and the dissipation rate, ε , are determined using the following transport equations, respectively.

k – Equation:

$$\rho \frac{Dk}{Dt} = \frac{\partial}{\partial x} \left(\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \right) + \frac{1}{r} \frac{\partial}{\partial r} \left(r \left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial r} \right) + G - \rho \varepsilon \quad (4a)$$

Where

$$G = \mu_t \left(2 \left(\frac{\partial u}{\partial x} \right)^2 + 2 \left(\frac{\partial v}{\partial r} \right)^2 + 2 \left(\frac{v}{r} \right)^2 + \left(\frac{\partial u}{\partial r} + \frac{\partial v}{\partial x} \right)^2 \right) \quad (4b)$$

ε - equation:

$$\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x} \left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x} \right) + \frac{1}{r} \frac{\partial}{\partial r} \left(r \left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial r} \right) + \frac{\varepsilon}{k} (C_{1\varepsilon} G - C_{2\varepsilon} \rho \varepsilon) \quad (5)$$

The turbulent viscosity, μ_t is computed by combining K - ε as follows:

$$\mu_t = \rho C_\mu (k^2 / \varepsilon) \quad (6)$$

Where C_μ , $C_{1\varepsilon}$, and $C_{2\varepsilon}$ are the model constant; σ_k and σ_ε are turbulent Prandtl numbers for k and ε , respectively. These constants have the following default values (Lauder and Spalding, 2006): $\sigma_k = 1.0$, $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $\sigma_\varepsilon = 1.3$, and $C_\mu = 0.09$, respectively.

Heat transfer coefficient and Nusselt number

Due to the unique characteristics of the pulsating flow,

several heat transfer coefficients have to be carefully defined.

The instantaneous local heat transfer coefficient is given by

$$h(x, t) = \frac{q}{T_w - T_{av}} \quad (7a)$$

and the local heat transfer coefficient, defined as the average of heat transfer coefficient over a cycle, is formulated as

$$h(x) = \frac{1}{2\pi} \int_0^{2\pi} h d(\omega t) \quad (7b)$$

The instantaneous heat transfer coefficient, defined as the average of heat transfer coefficient over the whole length of pipe at a given moment, is expressed as

$$h(t) = \frac{1}{L} \int_0^L h dx. \quad (7c)$$

Subsequently, several relevant Nusselt numbers are rationally introduced. The instantaneous local Nusselt number is

$$Nu(x, t) = \frac{hD}{K} \quad (7d)$$

The overall Nusselt number; Nu is given by

$$Nu = \frac{1}{2\pi L} \int_0^L \int_0^{2\pi} Nu_{x,t} d(\omega t) dx. \quad (7e)$$

To ensure sufficient temporal resolution of the numerical solution, five to seven cycles are required to obtain a steady periodic solution with an acceptable accuracy.

Boundary conditions

The solution domain of the considered 2D, axisymmetric pipe flow is geometrically quite simple, which is a rectangle on the x - r plane, enclosed by the inlet, outlet, symmetry and wall boundaries. On walls no-slip condition are assumed for the momentum equations. The inlet velocity values have been derived from given Reynolds numbers. Inaccuracies due to an uncertainty in the shape of the inlet velocity profiles are not expected to play an important role, since all calculations were carried out at places sufficiently downstream from the inlet, so that

quite fully-developed conditions should be expected at the calculating section. For pulsating cases, the inlet velocity is varied with time. The outlet boundary condition is called "outflow", which implies zero-gradient condition at the outlet.

At the pipe inlet, ($x=0$):

For steady flow condition, $U(t) = U_{in}$, and for $\frac{\partial \phi}{\partial x} = 0$ sinusoidal pulsating flow, $U(t) = U_m + U_A \sin(\omega t)$

At the pipe exit plane, ($x = L$):

At the pipe axis, ($r = 0$): $\frac{\partial \phi}{\partial r} = 0$

At the pipe wall, ($r = R$): (u, v, k, ϵ) = 0, a uniform heat flux is imposed, $q'' = q_0$ for heated section and $q'' = 0$ for the adiabatic wall.

Values of k and ϵ are not known at the inlet, but, if they are not given by experimental data, some reasonable assumptions can be made. The kinetic energy of turbulence according to a certain value of the square of the average inlet velocity is expressed by Launder and Spalding (2006) as

$$k = 0.03 U_{in}^2$$

In addition, dissipation is calculated according to the equation

$$\epsilon = C_\mu \frac{k^{\frac{3}{2}}}{0.03R}$$

Simulation values

In the analysis, the following simulation values are taken as:

$U_{in} = (12.7 \text{ to } 42.8) \text{ m/s}$, $U_m = 11.07 \text{ m/s}$, $U_A = (1.63 \text{ to } 31.81) \text{ m/s}$, $\omega = (41.9 \text{ to } 427.25) \text{ rad/s}$, and $q = 922 \text{ W/m}^2$.

COMPUTATIONAL PROCEDURE

Calculation tools

Even the difficult general differential equations now yield to the approximating technique known as numerical analysis, whereby the derivatives are simulated by algebraic relations between a finite numbers of grid points in the flow field which are then solved on a digital computer. A suitable CFD computer code is used to solve the governing equations numerically along with the boundary and the initial conditions. The CFD program is a process by which the fluid flow characteristics can be predicted through arbitrary geometries giving such information as: flow speed, pressure, residence time,

flow patterns, etc. The Fluent 6.1 (Fluent, 2003) program is chosen as the CFD computer code for this work because of the ease with which the analysis model can be created, and because the software allows users to modify the code for special analysis conditions through the use of user subroutines. The Fluent computer code uses a finite-volume procedure to solve the governing equations of fluid flow in primitive variables such as velocity components in axial and radial directions and pressure. A variety of turbulence models is offered by the Fluent computer code. The standard $k-\epsilon$ model was used as a turbulence model in this study. This model is a semi-empirical one, based on model transport equations for the turbulent kinetic energy (k) and its dissipation rate (ϵ). The model transport equation for k is derived from the exact equation while the model transport equation for ϵ is obtained using physical reasoning and bears little resemblance to its mathematically exact counterpart. In the derivation of the $k-\epsilon$ model, it was assumed that the flow is fully turbulent, and the effects of molecular viscosity are negligible. The standard $k-\epsilon$ model is, therefore, valid only for fully turbulent flows. A detailed description of turbulence models and its applications to turbulence can be found in "Fluent (2003) (Launder and Spalding, 2006). In the case of the standard $k-\epsilon$ models, two additional transport equations, (for the turbulent kinetic energy and the turbulence dissipation rate) are solved, and turbulent viscosity, μ_t , is computed as a function of k and ϵ . The solution method for this study is axisymmetric. In order to define the pulsating inlet velocities in all cases, a UDF (User-Defined Function) file was introduced to the prepared fluent case file. By using the results of the calculations performed with the fluent code. This program, written in FORTRAN 77 language, calculates numerically the local and average Nusselt number.

Grid size

Grid-independent tests were carried out to ensure grid independence of the calculated results; consequently, the grid size and the grid orientation giving grid independent results were selected, and thus a total cell number of 45000 cells (300×150) were adopted. The grid points are clustered in the radial direction so that the finer spacing is formed near the wall and at the inlet. Figure 3 shows the grid.

RESULTS AND DISCUSSION

Steady flow

To validate the numerical calculations, the computational results for the fully developed steady turbulent pipe flow are compared with the experimental data observed in (Elshafei et al., 2006). For two different values of Reynolds number, the numerical results are in consistent with the experimental results; within about $\pm 5\%$ as can be noticed in Figure 4. The flow was observed to be thermally fully developed for $X/D \geq 20$.

Pulsating flow

In the present work, investigations are performed within the ranges of $10^4 \leq Re \leq 4 \times 10^4$, and $0 \leq f \leq 70 \text{ Hz}$.

The effect of pulsating frequency on the local time average Nusselt number at fixed Reynolds number of 16.8×10^3 is shown in Figure 5. It can be seen that, as

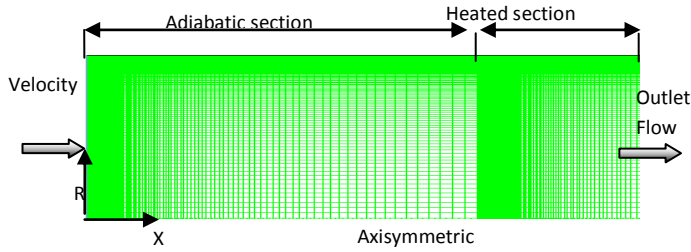


Figure 3. Computational grid for pipe.

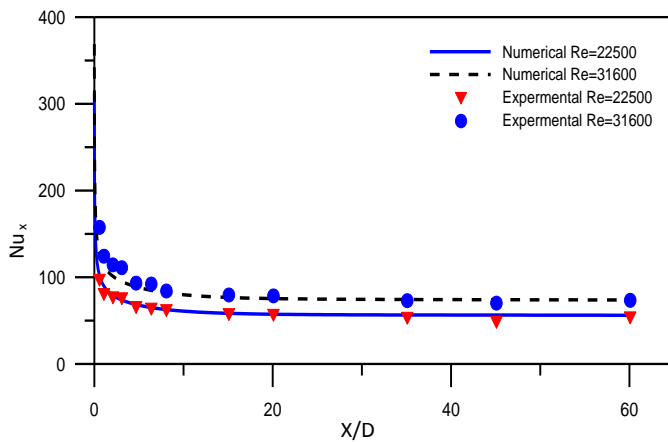


Figure 4. Variation of local Nusselt number at different Reynolds number.

the axial location moves downstream, the time average local Nusselt number for both pulsating and steady flows decays rapidly from a very large value near the entrance and the axial decrease becomes less step in the bulk of the pipe soon after leaving this entrance region.

The variation of the time averaged local Nusselt number for pulsating turbulent air flow exhibits a similar trend to that of the steady air flow case, but generally with lower values than that of the steady flow for $f < 61.13$ Hz. The maximum reduction of about 6% in the local Nusselt number was predicted in the thermal entrance region. For $f \geq 61.13$ Hz, little enhancement in the local time average Nusselt number of about 8% for $X/D \geq 50$ was detected.

Figure 6 illustrates the variation of the ratio of the local Nusselt number value for pulsating flow to that of the steady flow value; η with the axial distance at $Re = 22.5 \times 10^3$. For $f \leq 39.3$ Hz, it can be observed from Figure 6a that the ratio η starts rising rapidly in the entrance zone of the thermally developed region ($X/D < 20$) and becomes nearly fixed below unity where its value in the bulk of the pipe soon after leaving the entrance length ($X/D \geq 20$) implies that the effect of pulsation on the local heat transfer diminishes.

For $f \geq 42.5$ Hz, the value of η close to the end of the pipe enhanced by about 4% as shown in Figure 6b.

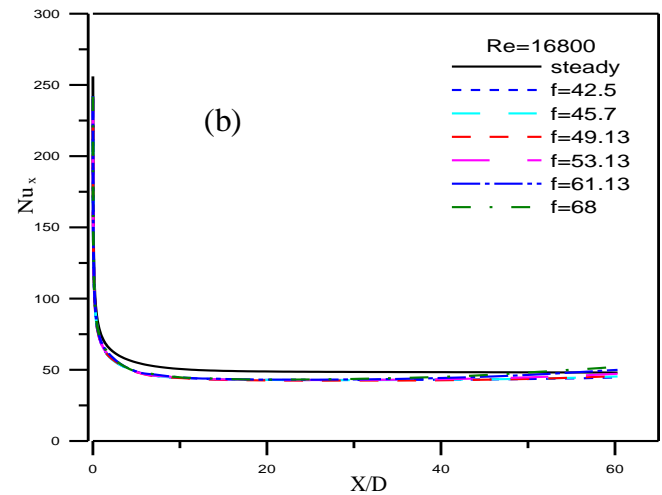
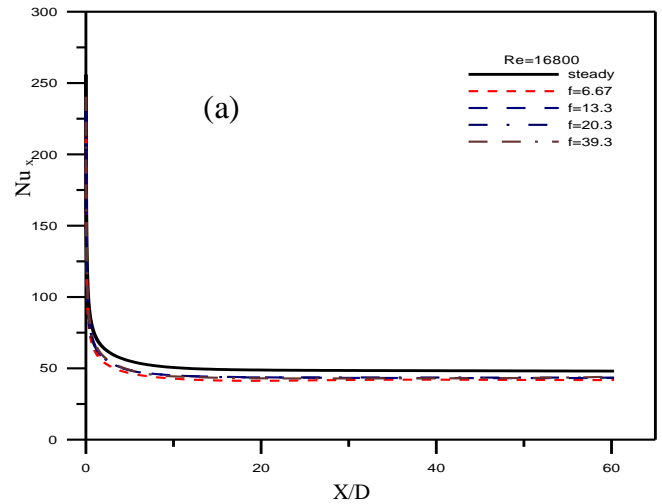


Figure 5. Distribution of Nu_x at different frequency, $Re=16800$.

The impacts of other relevant parameters on heat transfer are illustrated in Figure 7a and b. The detected values of η due to the variation of pulsation frequency is at higher value of Reynolds number ($Re = 31.6 \times 10^3$). At lower frequency ($f \leq 39.3$ Hz), a small change in heat transfer ratio is visible near the entrance region. At $f=13.3$ Hz, a rapid increases in η can be observed up to $X/D = 20$, after which it seems to be unchanged and generally, the value of η is less than unity. The same trends can be seen for $f = 28.3$ Hz and $f = 39.3$ Hz. However, in the downstream region the value of η reaches about unity.

The effect of changing pulsation frequency on the ratio η seems to be negligible specially, for higher values of pulsation frequency as can be noticed in Figure 7b.

At high values of Reynolds number; $Re \geq 37 \times 10^3$, certain qualitative trends can be shown in Figure 8a and b, which are similar to those described in the previous figures. For all studied values of pulsation frequency, the values of the ratio η are always less than unity.

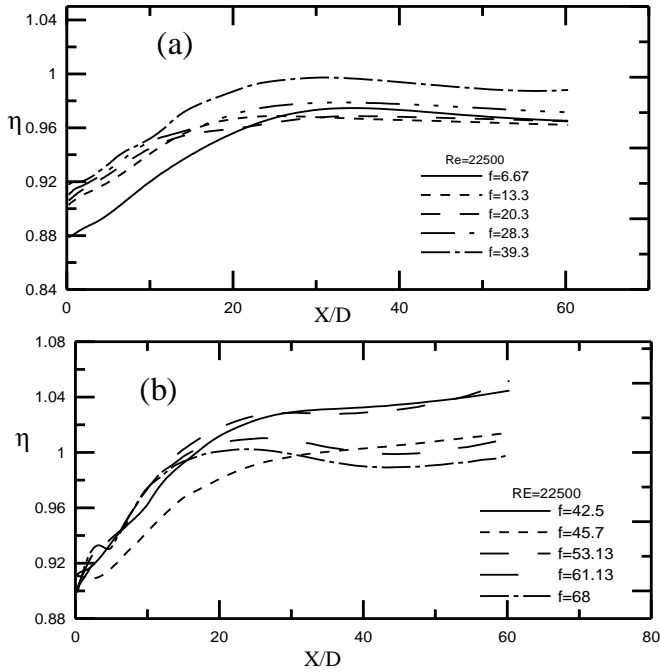


Figure 6. Distribution of η at different frequency, $Re=22500$.

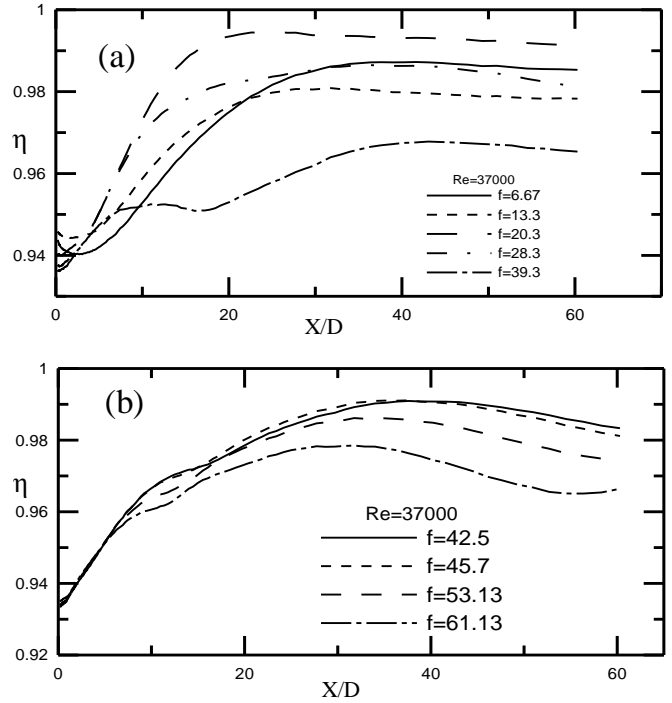


Figure 8. Distribution of η at different frequency, $Re=37000$.

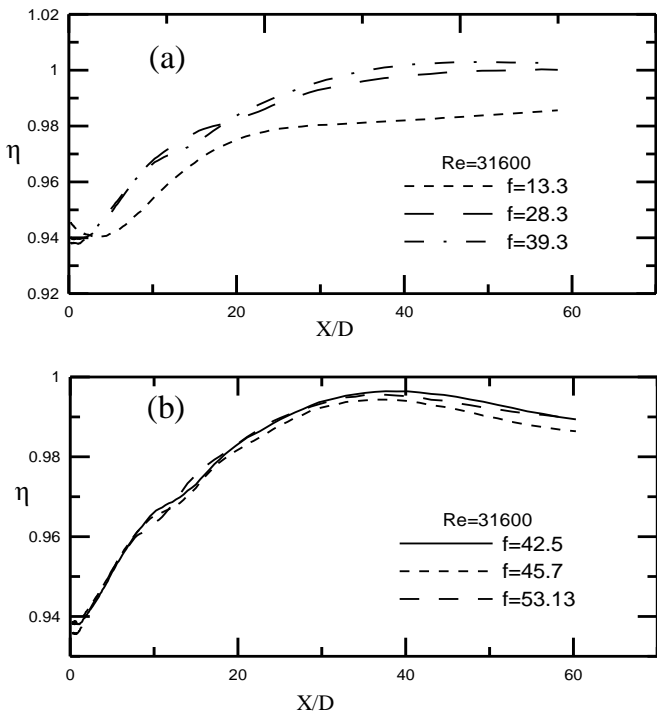


Figure 7. Distribution of η_x at different frequency, $Re=31600$.

This global behavior of pulsating flow is consistent with the analytical and numerical results of (Mostafa, 2005b;

Chattopadhyay et al., 2006; Lee et al., 1998; Shemer, 1985; Kim et al., 1993; Jie-Chang et al., 2004; Elshafei et al., 2006). The overall patterns of η curves are qualitatively similar as discussed earlier.

Number and pulsation frequency

The third set of results pertains to the effect of the Reynolds number on the relative mean time- averaged Nusselt; η_m in case of pulsating flow in pipes. The calculated data for η_m versus Re are described in Figure 9a and b. In the range of pulsating frequency; $f \leq 39.3$ Hz, it is found that Reynolds number strongly affects the heat transfer ratio; η_m when Reynolds number is relatively low; $Re < 25 \times 10^3$ as can be noticed in Figure 9a. The heat transfer ratio sharply increases with Reynolds number up to nearly unity. However, when Reynolds number exceeds this value, the effect of increasing Reynolds number diminishes and the value of η_m remains nearly constant. It is also noticed that the variation of the imposed pulsating frequency has a relatively small effect on η_m . These computed results are in consistent with that reported in (Xuefeng and Nengli, 2005).

For higher pulsation frequency; $f \geq 42.5$ Hz, the value of the ratio η_m increases also with the increase of Reynolds number. As η_m reaches to its peak value at $Re \approx 22.5 \times 10^3$, it then smoothly decreases with increasing Reynolds number up to 33×10^3 where η_m value is held nearly constant for higher values of Reynolds number as

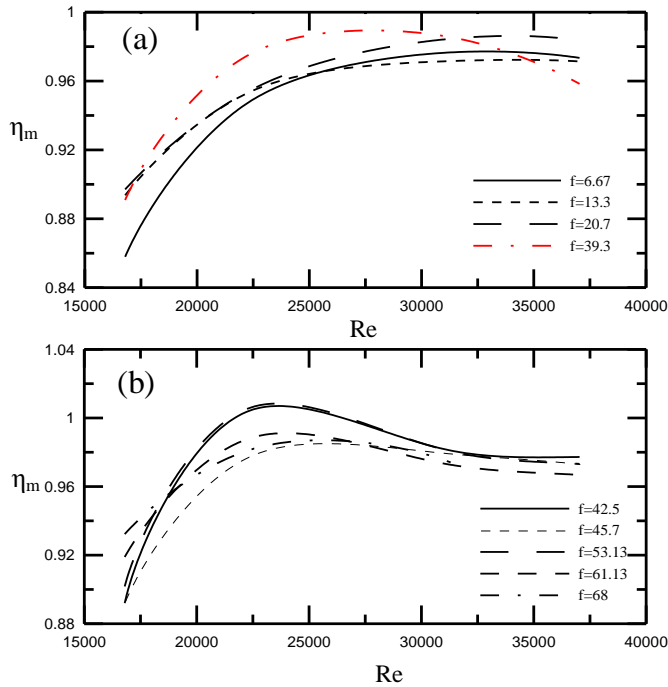


Figure 9. Heat transfer ratio for different Reynolds.

can be noticed in Figure 9b.

The behavior of Nusselt number corresponding to the variation in both Reynolds number and pulsation frequency can also be explained in view of the bursting phenomena (Liao and Wang, 1985; Genin et al., 1992). The bursting model defines certain regions on frequency-Reynolds number plane, where, the bursting frequency can be dependent or independent of the pulsation frequency.

The present results showed that the heat transfer coefficient may be increased or decreased with the change of both Reynolds number and frequency. The mean bursting frequency may be 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. These results are consistent with those reported by Habbib et al. (2004), and Liao and Wang (1985).

A comparison between the present numerical results with the experimental ones [32] over a ranges of $16 \times 10^3 < Re < 38 \times 10^3$ and $6.67 \leq f \leq 68$ Hz are described in Figure 10. 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^*

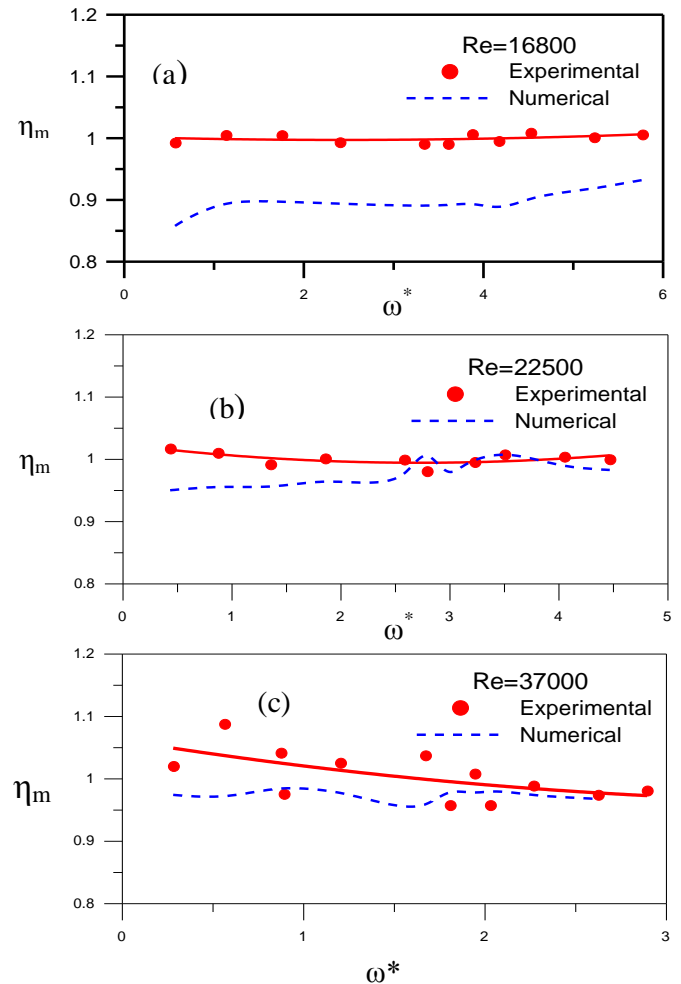


Figure 10. Comparison of the experimental data with numerical data at different (ω^*).

increases, the computed results closes to the experimental ones, specially, at higher value of Reynolds number.

In summary, the present elaborated numerical results for heat transfer characteristics of turbulent pulsating flow are useful. It can serve as a source of materials against which further analytical data may be checked for consistency.

Conclusion

Simulation was performed to describe heat transfer in pulsating turbulent air flow in a pipe using CFD code. The simulation results on the fully developed pulsating turbulent flow and heat transfer are compared with the available experimental data. The findings suggest that the heat transfer augmentation in the established flow regions takes place only for a certain band of

frequencies. For extreme low and high frequencies, heat transfer is actually reduced below that of steady flow. The following may be concluded from this study:

- i) At all values of pulsation frequency, the variations in the local time-averaged Nusselt number exhibit similar trends to the steady flows.
- ii) The local time-averaged Nusselt number either increases or decreases than steady flow values depending on the frequency and Reynolds number.
- iii) Frequency has little effect on the values of mean time-averaged Nusselt number especially at higher values of pulsation frequency.
- iv) The presence of any pulsation frequency has a notable effect on the heat transfer coefficient in comparison with that of steady flow, even at a very small value of pulsation frequency. However, the change of the value of pulsation frequency has relatively small effect on the heat transfer rates.
- v) No increase in the mean time averaged-Nusselt number for frequency $f \leq 39.34$ Hz. For $f \geq 42.5$ Hz, little increase in time averaged-Nusselt number is detected at $Re \approx 22.5 \times 10^3$.

Nomenclature

| | |
|---------------------|--|
| C_p : | Specific heat at constant pressure (J/Kg K) |
| C_1, C_2, C_μ : | Empirical constant of k - ϵ model. |
| D : | Diameter of pipe (m) |
| G : | The production of turbulent kinetic energy (Kg/m ³ s ³) |
| $h(x,t)$: | Instantaneous local heat transfer coefficient (W/m ² K) |
| $h(t)$: | Instantaneous heat transfer coefficient which is the average of heat transfer coefficient over the whole length of pipe at given moment (W/m ² K) |
| $h(x)$: | Local heat transfer coefficient which is the average of heat transfer coefficient over a cycle (W/m ² k). |
| p : | Pressure (Pa) |
| K : | Thermal conductivity (W/m K) |
| k : | Turbulent kinetic energy (m ² /s ²) |
| L : | Length of pipe (m) |
| Nu : | Overall Nusselt number |
| Nu_t : | Instantaneous Nusselt number |
| Nu_x : | Local Nusselt number |
| Pr : | Prandtl number (u/α) |
| q : | Heat flux per unit area (W/m ²) |
| r : | Radial coordinate (m) |
| R : | Radius of pipe (m) |
| Re : | Reynolds number (UD/u) |
| T_{av} : | Average bulk temperature along cross-sectional area of the pipe (°C) |
| T_{in} : | Inlet temperature of the pipe (°C) |
| U_{in} : | Average of mean velocity at $x=0$ (m/s) |
| | $U_{in} = U_m + U_A$ |

| | |
|---------|--|
| U_m : | Mean velocity of pulsating component, (m/s) |
| U_A : | Amplitude velocity of pulsating component, (m/s) |
| UDF : | User defined function |
| u : | Velocity component in the axial direction (m/s) |
| v : | Velocity component in the radial direction (m/s) |
| x : | Axial distance(m) |
| t : | Time (s). |

Greek symbols

| | |
|---------------------|--|
| α : | Thermal diffusivity of fluid $K/\rho C_p$ (m ² /s) |
| ϵ : | Turbulent energy dissipation rate (W/kg) |
| ϕ : | Viscous dissipation |
| μ : | Dynamic viscosity (kg/m s) |
| μ_t : | Eddy (or turbulent) viscosity (kg/m s) |
| ν : | Kinematics viscosity (m ² /s) |
| ρ : | Density (Kg/m ³) |
| σ_k : | Turbulent Prandtl number for k |
| σ_T : | Turbulent Prandtl number for T |
| σ_ϵ : | Turbulent Prandtl number for ϵ |
| ω : | Angular frequency (1/s) |
| ω^* : | Dimensionless frequency; ($\omega D/U^*$), U^* is the fraction velocity described as $U^* = 0.199 U_m / Re^{-0.125}$ |
| η : | Relative local Nusselt number = (Nu_{px}/Nu_{sx}) . |

Subscripts

| | |
|-------|--------------|
| s : | Steady state |
| p : | Pulsation |
| m : | Mean. |

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