THE EFFECT OF LONGITUDINAL VIBRATION ON LAMINAR FORCED CONVECTION HEAT TRANSFER IN A HORIZONTAL TUBE

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ABSTRACT

The effect of longitudinal vibration on the laminar forced convection heat transfer in a horizontal tube was studied in the present work. The flow investigated is an internal, laminar, incompressible, developing, steady and oscillatory flow. The vibration vector was parallel to the fluid flow direction. The boundary layer was studied in the entrance region for Prandtl numbers greater than one and the hydrodynamic boundary layer grows faster than thermal boundary layer. The governing equations which used were momentum equation in the axial direction, continuity and energy equation. The finite difference technique was introduced to transform the partial differential equations into algebraic equations and the later equations were solved using the upwind scheme. The vibrational Reynolds number used as indication to the vibration and vibrational velocity which is used as boundary condition. The effect of vibration on laminar forced convection heat transfer is to increase the local and average Nusselt number when the vibrational Reynolds number increases; therefore, heat transfer coefficient increases too. Correlations for local Nusselt number with vibration concluded for constant wall temperature and constant heat flux.

الخلاصة:

يهدف البحث لدراسة تأثير الاهتزاز الطولي على انتقال الحرارة بالحمل القسري للجريان المستقر داخل أنبوب أفقي. الدراسة كانت للجريان الداخلي المستقر, اللاانضغاطي المتطور والاهتزازي.متجه الاهتزاز موازي لاتجاه جريان المائع. جرى البحث على الطبقة المتاخمة في منطقة الدخول للأنبوب حيث رقم برانتل(Prandtl no.) كان أكبر من واحد لذلك فأن الطبقة المتاخمة الهيدروليكية قد نمت قبل الطبقة المتاخمة الحرارية. حدد النموذج الرياضي بمعادلة الاستمرارية والزخم والطاقة. حولت المعادلات التفاضلية إلى معادلات جبرية باستخدام طريقة الفروق المحددة وتم حلها باستخدام طريقة (Upwind scheme). رقم رنولد الاهتزازي (Re_v) استخدم لإظهار الاهتزاز كظرف حالة. تأثير الاهتزاز على انتقال الحرارة بالحمل القسري للجريان المستقر بزيادة رقم رينولد الاهتزازي كان بزيادة رقم نسلت (Nusselt no.) وبهذا ازداد

معامل انتقال الحرارة. تم التوصل إلى علاقات لرقم نسلت الموقعي بوجود الاهتزاز لدرجة حرارة ثابتة ولحالة الفيض الحراري الثابت.

KEYWORD

Forced Convection Heat Transfer, Oscillatory Flow, Flow in the Entrance Region, Tube Vibration, Vibration Reynolds Number.

INTRODUCTION

It is well known that the heat transfer rate is greater in a turbulent flow than in laminar flow. Vibration of the heat transfer surface gives rise to turbulence in the case of laminar boundary layers and increases the heat transfer rate [1]. Studies in this field can be broadly classified into two types- one in which the heat transfer surface is vibrated experimentally by two different methods. In the first method, the surface is held stationary and acoustic vibrations are established in the fluid medium surrounding the surface. In the second method, an oscillatory motion is impressed upon the surface itself. Theoretically the flow medium is subjected to pulsation, vibrations or agitation [2].

LITERATURE REVIEW

Tessin and Jacob (1953), [3] While studying the effects of starting length on heat transfer from a cylinder to a parallel gas stream, they found that the heat transfer coefficients were not appreciably altered when the cylinder was subjected to a vibrational velocity of a bout (2%) of the flow velocity.

Anantanarayanan and Ramachandran (1958), [4] observed an increase of (130%) in the heat transfer from a vibrating horizontal nichrome wire, (0.018 in) in diameter to an air stream flowing parallel to the wire. The wire was vibrated in a vertical plane with varying values of vibrational velocity. The ratio of the vibrational velocity to the flow velocity used in their study varied from (0 to 30%).

Van der Hegge Zijnen (1958), [5] reported that the heat transfer from a tungsten wire (0.0005 cm) in diameter to a normal air stream decreased when the wire was subjected to vibrations in the direction of the air stream. A maximum decrease of (4.3%) in the heat transfer coefficient was observed when the ratio of the root-mean square velocity of vibration of the flow velocity was as high as (45%).

Kenji Takahashi and Kazuo Endoh, (1990) [1] investigated experimentally the effect of vibration on forced convection heat transfer from a vibrating sphere, a cylinder and a square-section tube to water. The obtained heat transfer data with the vibration effect is well correlated in terms of the energy dissipation calculated from the fluid drag acting on the vibrating bodies.

Sreenivasan and Ramachandran (1961), [2] investigated the effect of vibration on heat transfer from a horizontal copper cylinder (0.344) in diameter and (6 in) long. The cylinder was placed normal to an air stream and was sinusoidally vibrated in a direction perpendicular to the air stream. The flow velocity varied from (19 ft/s) to (92 ft/s), the double amplitude of vibration from (0.75 cm) to (3.2 cm), and the frequency of vibration from (200) to (2800 cycles/min). A transient technique was used to determine the heat transfer coefficients.

Lemlich (1961), [6] and [7] published two interesting studies of the effect of vibration on the forced convection heat transfer in a double pipe heat exchanger. In one study a clarinet mouthpiece placed on the center tube was used to obtain air vibration of about 600 Hz. He concluded that the average heat transfer was increased by (35%). In a similar study a specially designed pulsator was used to obtain vibrations of about 1.5 Hz. In this study the overall heat exchanger efficiency was increased by (80%).

Deaver, Penny and Jefferson, (1962), [8] studied the effect of low frequency and relatively large amplitude vibration on the average heat transfer from a small wire to water. They also concluded that above a critical value (af<0.36 fps), increases in either the frequency or the amplitude increase the convective coefficient of heat transfer.

MATHAMATICAL MODEL

The geometry of the tube and coordinates system are shown in figure (1) which represent two dimensional cylindrical coordinates (r, z), with a tube diameter of a (1.7cm). The tube length introduced as thermal entrance length. To solve the flow equations (conservation of momentum, mass and energy equations) for water flows in a horizontal tube, some assumptions should be taken to fit the case, which is under study. These assumptions are: -

1- Steady state flow.

2- Two dimensional axisymmetric flow.

3- Incompressible flow.

4- Laminar flow.

5- No inlet swirl.

6- No heat generation.

7- Negligible body force, dissipation function and buoyancy effect.

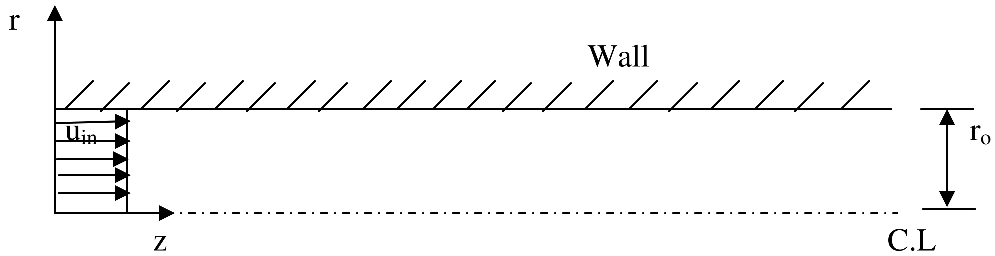


Fig (1). Problem configuration and coordinates system.

The governing equations used in this problem were given as:

Momentum equation in axial direction:-

$$\rho v\frac{\partial u}{\partial r}+\rho u\frac{\partial u}{\partial z}=-\frac{dp}{dz}+\mu\left[\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial u}{\partial r})+\frac{\partial^2 u}{\partial z^2}\right] \tag{1}$$

Continuity equation

$$\frac{1}{r}\frac{\partial}{\partial r}(rv)+\frac{\partial u}{\partial z}=0 \tag{2}$$

The energy equation

$$\rho v\frac{\partial T}{\partial r}+\rho u\frac{\partial T}{\partial z}=\frac{\mu}{\Pr}\left[\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial T}{\partial r})+\frac{\partial^2 T}{\partial z^2}\right] \quad (3)$$

Shear Stress

From Newton's law of viscosity for simple shear flow:

$$\tau_w=-\mu\frac{\partial u}{\partial r}\bigg|_{r=R} \quad (4)$$

Local Friction Coefficient

The local friction coefficient is defined by:

$$C_f=\frac{\tau_w}{\frac{1}{2}\rho u_{in}^2} \quad (5)$$

The average friction coefficient computed as follow:

$$\overline{Cf}=\frac{1}{L}\int_0^L Cfdz \quad (6)$$

Bulk temperature

The bulk (mixed-mean) temperature for the circular tube is: -

$$T_b=\frac{\int_0^R uTrdr}{\int_0^R urdr} \quad (7)$$

The friction coefficient and bulk temperature may be calculated by applying Simpson's rule.

Local Nusselt Number

The local Nusselt number is given by:

$Nu_z=(hd_h/k)$

Where d_h=2 r_o

And so the local Nusselt numbers are given in the following formula:

- For constant wall temperature:

$$h(T_w-T_b)=k[\partial T/\partial r]_{r=r_o}$$

$$Nu_z=[-[\partial T/\partial r]]_{r=r_o}\times(d_h)/(T_b-T_w) \quad (8)$$

• For constant heat flux:

$h(T_w - T_b) = q$ So:

$$Nu_z = \left[q(d_h)/(T_b - T_w)\right] \tag{9}$$

• The mean Nusselt number is computed as

$$\overline{Nu} = \frac{1}{L}\int_0^L Nu_z dZ \tag{10}$$

Entrance Length

For an internal flow we must be concerned with the existence of entrance and fully developed region. For laminar flow in a circular tube (Re≤2300), the hydrodynamic entry length may be obtained from an expression:-

$$(Z_{fd,h} / D) \approx 0.05\,\mathrm{Re} \tag{11}$$

The thermal entry length may be expressed as: -

$$(Z_{fd,t} / D) \approx 0.05\,\mathrm{Re}\,\mathrm{Pr} \tag{12}$$

. **Boundary Conditions**

Inlet Section Boundary Conditions

In inlet section the axial velocity supposed to be uniform and its value calculated from Reynolds number as follows:-

$$u_{in} = \frac{\mathrm{Re} \times \mu}{\rho \times d_h}$$

The radial velocity in inlet section was equal to zero and the inlet temperature was ($T_{in} = 20^\circ C$)

$u = u_{in}$, $v=0$ and $T=T_{in}$

Exit Section Boundary Conditions

In the exit section the flow became fully developed, the axial and radial velocity gradient was zero and the temperature gradient was constant. So the boundary conditions were as follows:

$(\partial u/\partial z) = 0$, $v = 0$ and $(\partial T/\partial z) = \text{Constant}$

Boundary Conditions at the Wall

The boundary conditions must be divided into two types, the forced convection heat transfer without vibration and the forced convection heat transfer with vibration.

Wall Boundary conditions without Vibration

At the wall, boundary conditions were taken from boundary layer theory that the axial and vertical velocity at the walls in the viscous flow must be equal zero, also the tube wall temperature was taken according to the case of constant wall temperature and of constant heat flux as following:-

$u = u_w = 0, v = 0$, ($T = T_w$) (Constant wall temperature)

And ($\partial T/\partial r = q/k$) (Constant heat flux)

Wall Boundary conditions with Vibration

The steady oscillatory flow was used in order to show the effect of vibration on the forced convection heat transfer and the effect of the vibration on the flow. Oscillatory flow [9] used vibrational velocity as boundary condition as follows:

$u = u_w = 2\pi af$, $v = 0$, $T = T_w$ (Constant wall temperature)

And ($\partial T/\partial r = q/k$) (Constant heat flux).

Boundary Conditions in centerline

The axial velocity gradient, radial velocity and temperature gradient occurred in the centerline was equal to zero so:-

$(\partial u/\partial r) = 0$, $v = 0$ and $(\partial T/\partial r) = 0$

SOLUTION METHODS

Because of the complexity of the analytical solution of the set of the algebraic equations, they may be solved by finite difference technique. In this study the governing equations are expressions of conservation of mass, momentum and energy equations. The Gauss-Siedal method was used to solve the continuity and momentum equations and Tri-diagonal method was used to solve the energy equation. A digital computer used to solve the finite difference equations using FORTRAN 90 program and Tec plot program used to plot the results.

RESULTS AND CONCLUSIONS

Reynolds number can be classified into two types; Reynolds number without vibration (Re) and Reynolds number with vibration (Re_v) which is called vibrational Reynolds number. For each value of Reynolds number (Re) three vibrational Reynolds number (Re_v) were taken in order to show the effect of vibration on the laminar forced convection heat transfer. For Reynolds number (Re=1000), three vibrational Reynolds number were taken as (Re_v =750, 1000 and 1250), for (Re=1500), (Re_v =1250, 1500 and 1750), and for (Re=2000), Re_v (=1750, 2000 and 2250) was taken. The inlet velocity calculated from the flow Reynolds number and vibrational velocity calculated from vibrational Reynolds number. The properties of water were taken as follows ($\rho = 1003 kg/m^3$, $\mu = 8.55 \times 10^{-4}$ Pa.s, Pr=5.83 and k=0.613 $W/m.^oC$).

Results without vibration

Temperature Variation for Constant Heat Flux:

The temperature variation in the axial and radial directions, for constant heat flux (q=10000 W/m^2) and different Reynolds number (Re =1000) are shown in figure (2). The figure shows that the temperature increase in the axial and radial directions and a small increase in temperature contours can be observed with the increases of flow Reynolds number.

Temperature Variation for Constant Wall Temperature

The temperature variation in the axial and radial directions for constant wall temperature ($T_w = 100^o C$) and different Reynolds number (Re =1000) are shown in figure (3). The figure shows that the temperature increase in the axial and radial directions and the change of flow Reynolds number causes a small change in temperature contours.

Local Nusselt Number (Nu_z)

The behavior of local Nusselt number with axial directions, for constant heat flux and for constant wall temperature is shown in figures (4a, b and c) respectively. The local Nusselt number decreases with the axial directions in the entrance region. While in the fully developed region the local Nusselt number become constant with the axial directions. These figures show the effect of Reynolds number (Re=1000, 1500 and 2000) on the local Nusselt number. The value of the local Nusselt number for constant heat flux is (4.364) and for constant wall temperature is (3.66) at the fully developed region (Z^+=1) because the length is introduce as thermal entrance length.

Velocity Profile

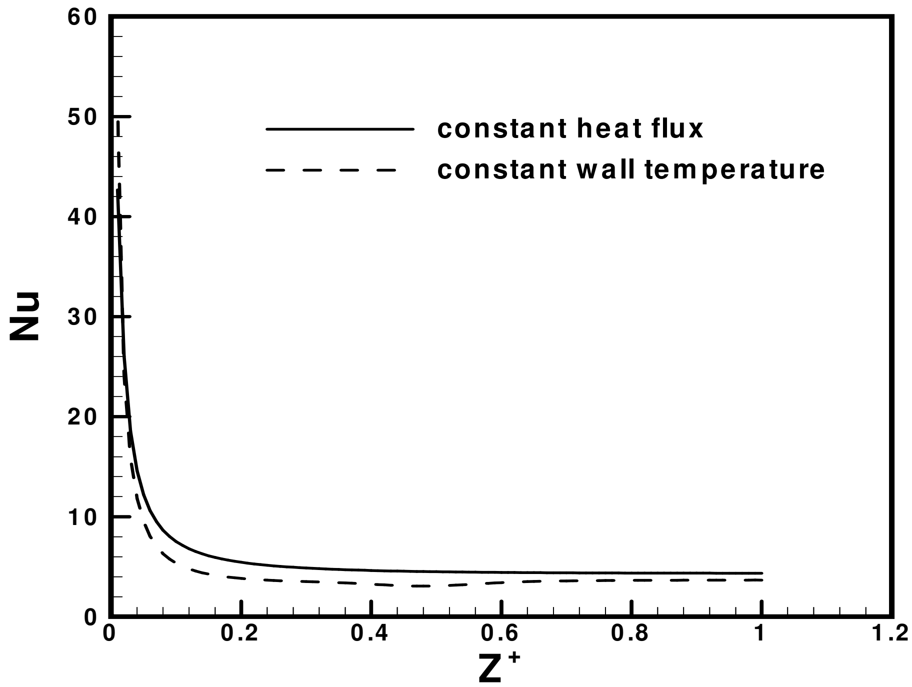
Development of velocity profile along the axial directions, for different Reynolds number (Re=1000, 1500 and 2000) are shown in figures (5a, b and c) respectively. The velocity profile developed from the entrance region until ($Z^+ \approx 0.55$), after this axial distance the velocity profile becomes constant and the flow from this axial distance and on is said to be fully developed. The maximum velocity occurs at dimensionless radial distance (R^*) was equal to zero (centerline).

Local Friction Coefficient:

The variations of the local friction coefficient along the axial directions is shown in figure (6) for different Reynolds number (Re=1000, 1500 and 2000), in which the velocity increase with the increasing of flow Reynolds number, while the local friction coefficient was decreased. The flow became fully developed at the dimensionless axial distance ($Z^+ \approx 0.55$), therefore from this distance and on the local friction coefficient became constant. From these figures it will be observed that the local friction coefficient decrease with the increasing of Reynolds number. The local friction coefficient values decreased until ($Z^+ \approx 0.1$) but it was increased after this axial distance because of numerical solution accuracy and the assumptions which were taken in this study.

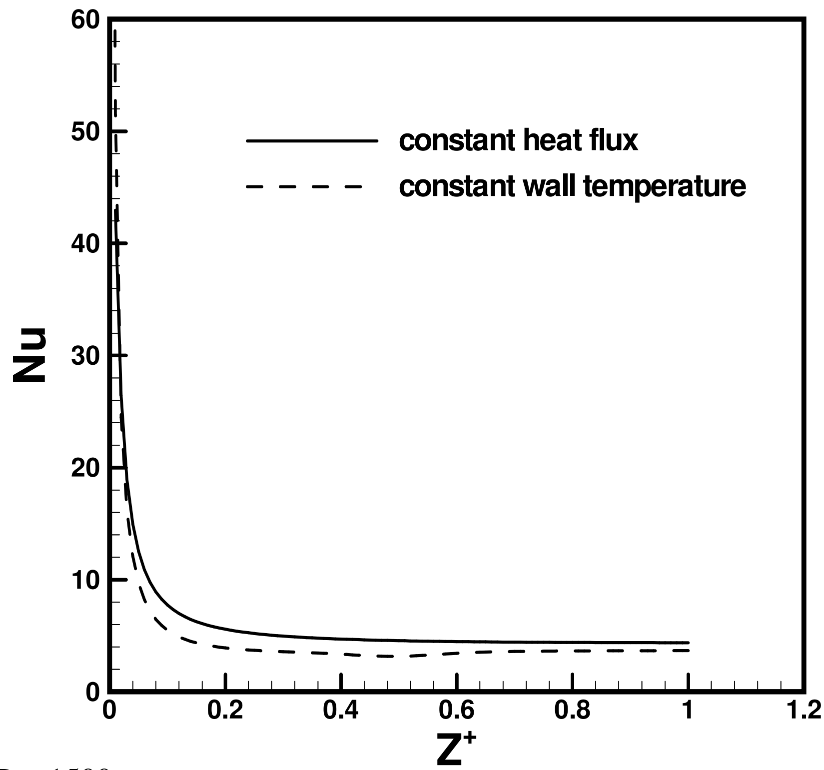
a) Re=1000, q=10000 W/m^2.

Fig (2): Contour of temperature for constant heat flux.

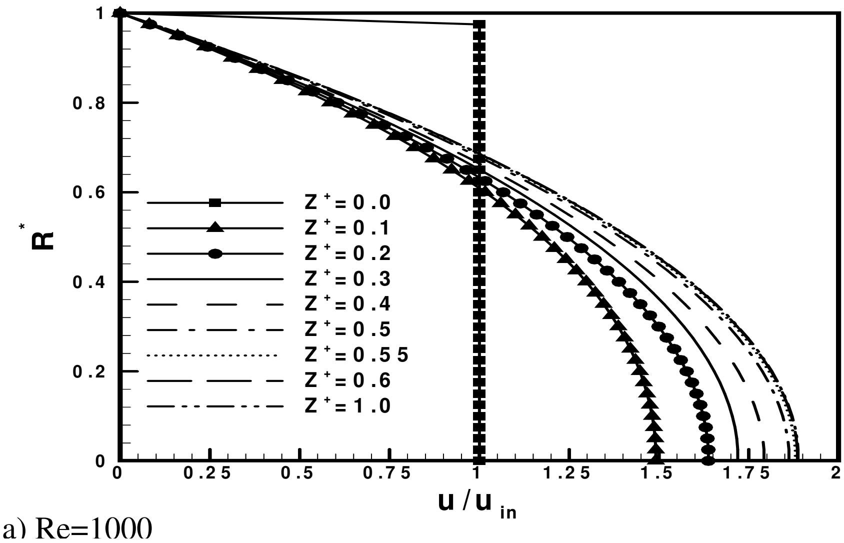


a) Re=1

Fig (3): Contour of temperature for constant wall temperature (T_w=100 ^oC).

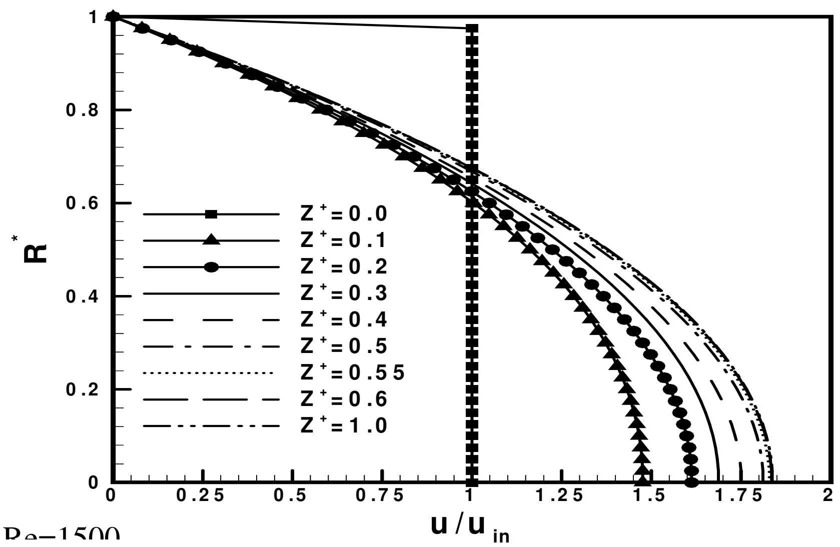


b) Re=1500



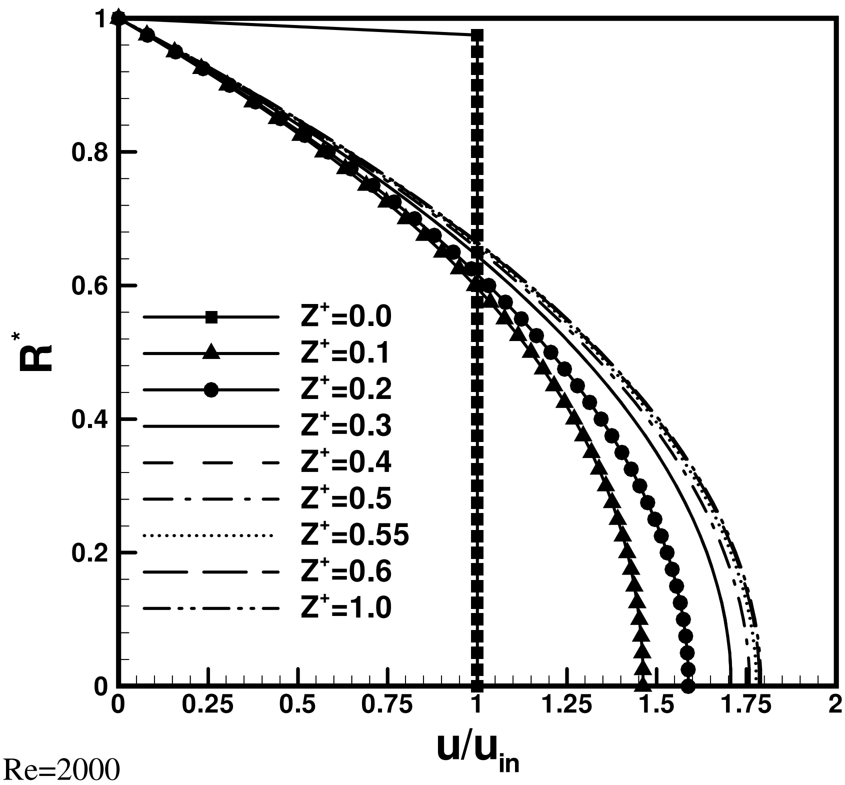
a) Re=1000

Fig (4). axial distribution for local Nusselt number without vibration.



b) Re=1500

Fig (5).developing profile of axial velocity without vibration.



c) Re=2000

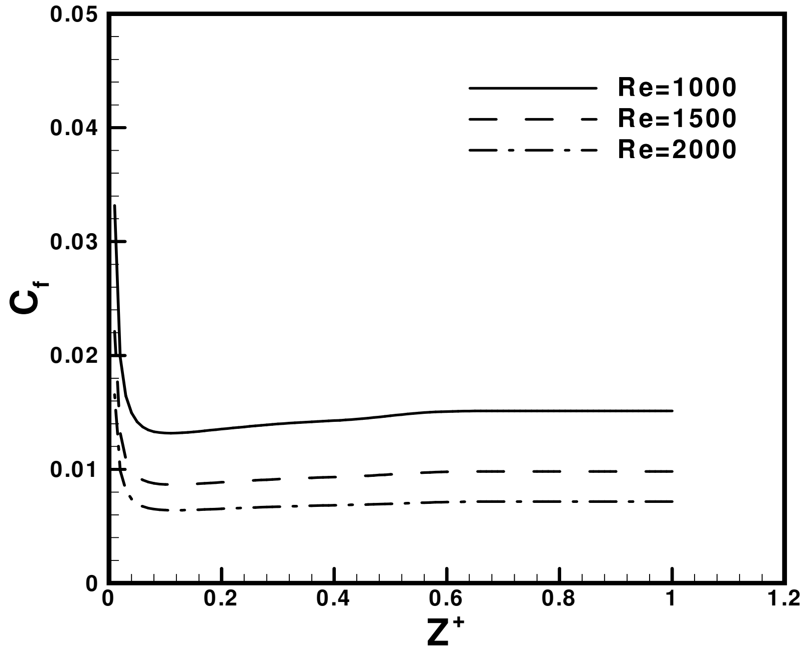


Fig (6).: axial distribution of local friction coefficient without vibration.

Results with vibration:

The parameters effects on the problem are frequency (f), amplitude (a) and the product of them which is represents by vibrational Reynolds number (Re_v). The results as follows:

Temperature Variation for Constant Heat Flux and Constant Wall Temperature: -

The effect of vibration on the temperature for constant heat flux and constant wall temperature are as follows: the case of Reynolds number (Re=1000) and vibrational Reynolds number (Re_v= 1000) are shown in figures (7) and (8). For this case the temperature increased along the axial distance (Z^+), and the temperature increased along the radial distance (R^*). The other cases had the same behavior. Generally when vibrational Reynolds number increased the lines of temperature contour were shift with small distance to the left.

Local Nusselt Number (Nu_z) and Average Nusselt Number ($\overline{Nu}$):

The effect of vibration on local and average Nusselt numbers are shown in figures (9a and b), (10) and (11) where Reynolds number (Re=1000) and vibrational Reynolds number (Re_v=1000), for constant heat flux and constant wall temperature respectively. A vibrational Reynolds number (Re_v=0) represents the behavior of local Nusselt number without vibration. Generally local and average Nusselt numbers increases when a vibrational Reynolds number increases because of the modifications of the flow in laminar boundary layer; earlier transition from laminar to turbulent boundary layer flow. For all cases fully developed occurs at the axial distance (Z^+=1) because the length is introduce as thermal entrance length.

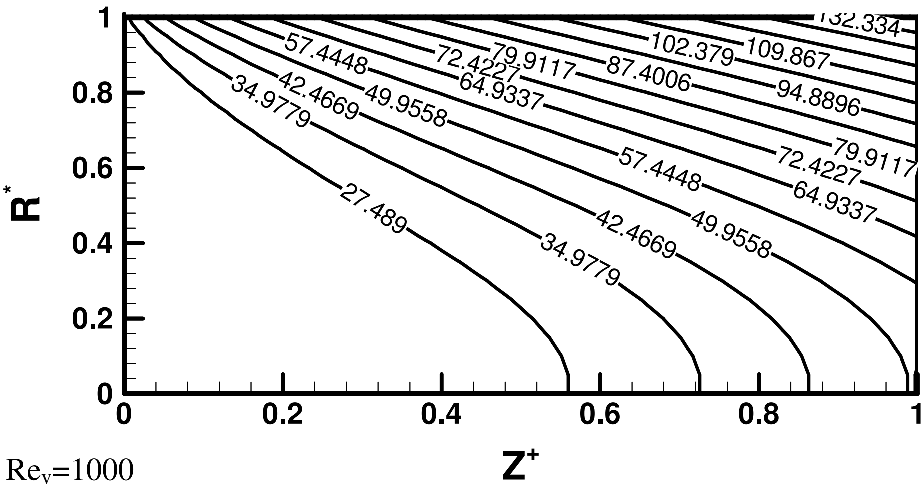
Velocity Profile:

For the case of Reynolds number (Re=1000) and vibrational Reynolds number (Re_v=750, 1000 and 1250), the velocity profile is shown in figures (12 a) at (Z^+=0.4) and (12 b) at (Z^+=1.0) respectively. For the first figure the curve of (Re>Re_v), the velocity profile from the axial distance ($Z^+ \approx 0.025$) was constant to the exit section, therefore, the flow was fully developed from

($Z^+ \approx 0.025$) to the exit section. In this figures the velocity profile was for turbulent flow because of the vibration influence, in the second curve of ($Re=Re_v$), the velocity profile is uniform from inlet section to exit section of the circular tube (flat velocity) and for the third curve of ($Re<Re_v$), the flow from axial distance ($Z^+ \approx 0.025$) to the exit section so the velocity profile is constant and the flow is fully developed.

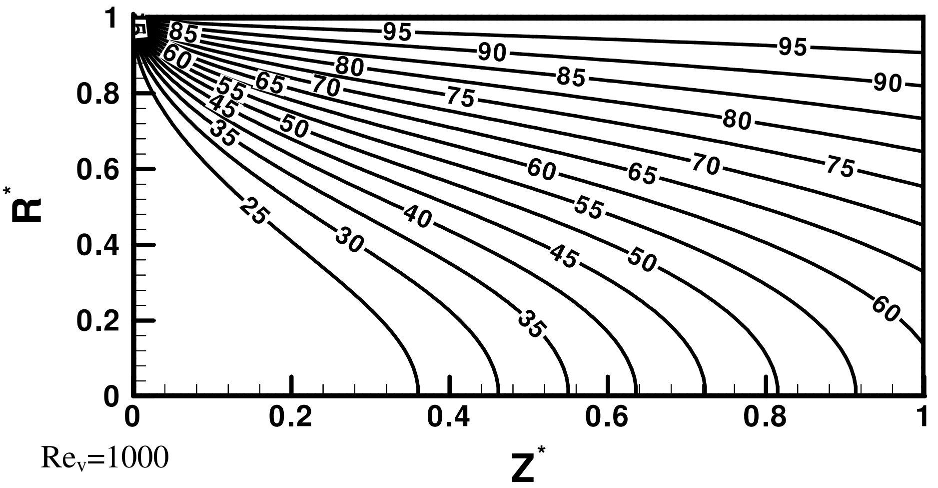
Local Friction Coefficient (C_f) and Average Friction Coefficient ($\overline{C_f}$): -

The local and average friction coefficient are affected by vibration and this is illustrated in figures (13) and (14) respectively for Reynolds number (Re=1000) and vibrational Reynolds number (Re_v=750, 1000 and 1250). A vibrational Reynolds number (Re_v=0) represents the behavior of local friction coefficient without vibration. Generally the local and average friction coefficients decreases for a vibrational Reynolds number increases, since the axial velocity gradient decreases in case of the vibration influence. For ($Re>Re_v$) local friction coefficient decreases at entrance region until axial distance ($Z^+ \approx 0.025$), then from this distance to the exit section the local friction coefficient behavior became constant (fully developed flow). For ($Re=Re_v$), local friction coefficient values were zero because the axial velocity gradient is zero whereas the flow is uniform from the inlet section to the exit section. For ($Re<Re_v$), local friction coefficient values were negative because the values of the axial velocity gradient is negative. The local friction coefficient values decreases until ($Z^+ \approx 0.1$) but its increases after this axial distance because of numerical solution accuracy and the assumptions which taken in this study.



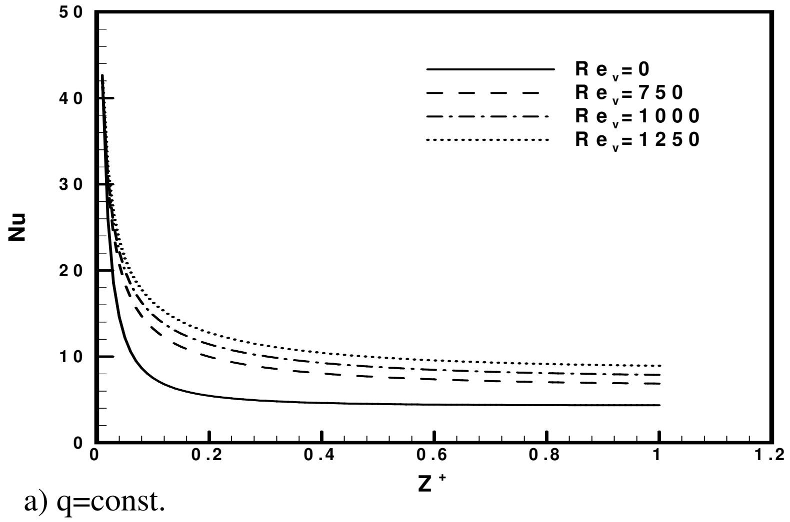
Re_v=1000

Fig (7). Contour of temperature for constant heat flux.

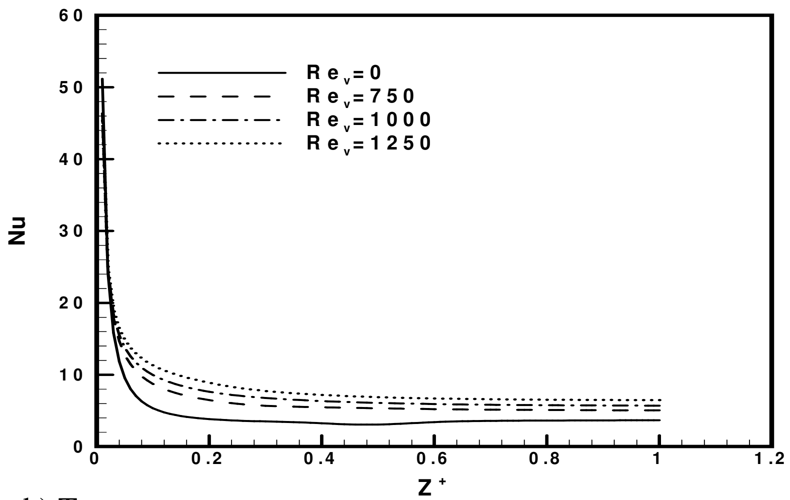


Re_v=1000

Figure (8): Contour of temperature for constant wall temperature.



a) q=const.



b) T_w=const.

Figure (9): The effect of vibration on developing Nu No.

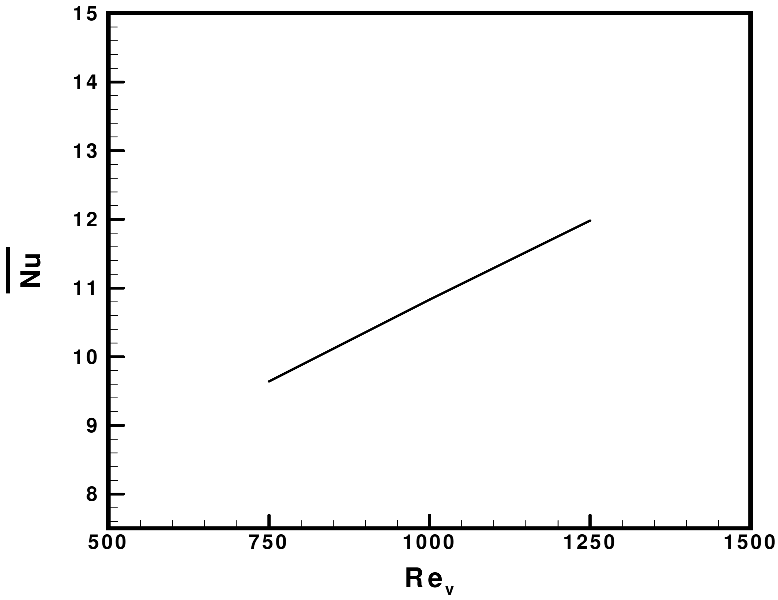


Figure (10): effect of vibration on $\overline{Nu}$ for (q=10000 W/m^2) at Re=1000.

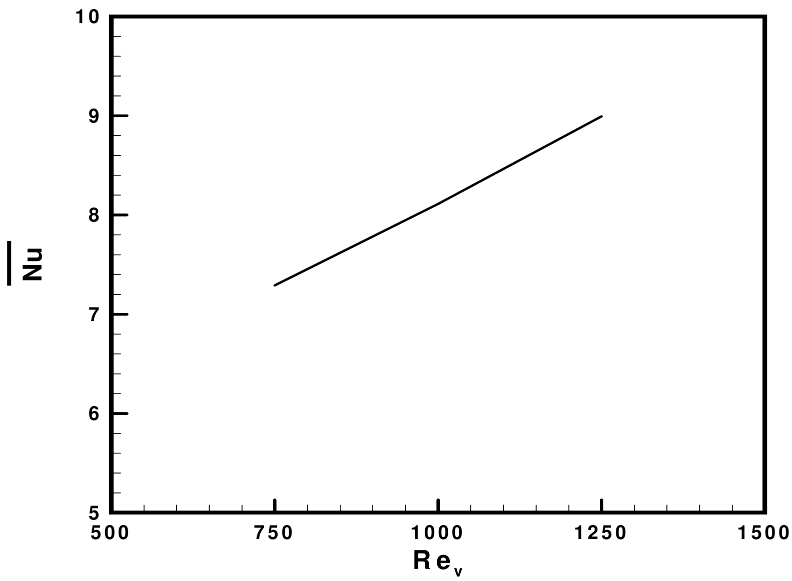
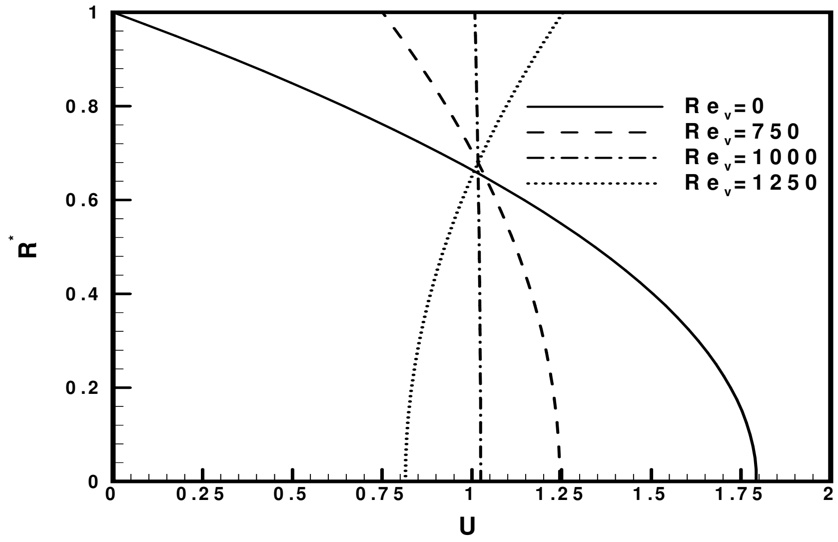
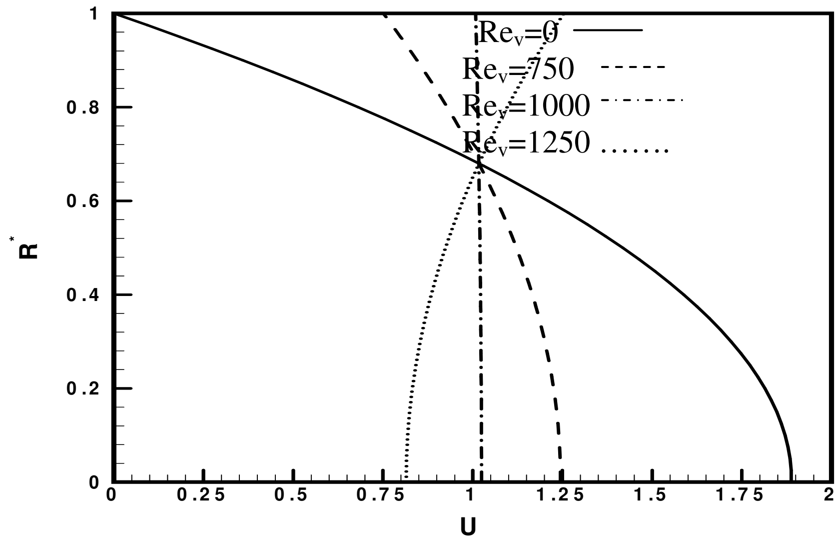


Figure (11): effect of Re_v on $\overline{Nu}$ at Re=1000 for (T_w=100 C^o).



a) Z^+=0.4



b) Z^+=1.0

Figure (12): developing profile of axial velocity with vibration.

CONCLUSIONS WITH VIBRATION:

1- The vibration influence is to convert the flow from laminar to turbulent flow.
2- The growth of hydrodynamic boundary layer with vibration is much faster than that without vibration.
3- Local Nusselt number with vibration is greater than that without vibration and that means increases in the heat transfer coefficient.
4- New correlation predicated for local Nusselt number due to vibration effect as follows:
Vibrational local Nusselt number for constant wall temperature as follows: -

$$Nu_v = \left[Nu\% + 3.657 + \frac{0.0677(\mathrm{Re}\,\mathrm{Pr}\frac{D}{L})^{1.33}}{1 + 0.1\mathrm{Pr}(\mathrm{Re}\frac{D}{L})^{0.3}} \right]$$

Where

Percentage of increasing in $Nu = Nu\% = \left[\frac{Nu_v - Nu}{Nu} \right]$

Vibrational local Nusselt number for constant heat flux is: -

$$Nu_v = \left[Nu\% + 4.364 + \frac{0.086(\mathrm{Re}\,\mathrm{Pr}\frac{D}{L})^{1.33}}{1 + 0.1\mathrm{Pr}(\mathrm{Re}\frac{D}{L})^{0.83}} \right]$$

This correlation is Valid for Pr=5.83 or if RePrD/L<33.

REFRENCES

[1] Kenji Takahashi and Kazuo Endoh. (1990), "Anew correlation method for the effect of vibration on forced convection heat transfer" Journal of chemical engineering of Japan. Vol.23, No.1, pp.45-50.
[2] K.Sreenivasan and A.Ramachandran, (1961) "Effect of vibration on heat transfer from a horizontal cylinder to a normal air stream" Int.heat mass transfer, Vol.3, pp.60-67.
[3] W. Tessin and M.Jakob, (1953), "Influence of unheated starting section on heat transfer from a cylinder to gas streams parallel to the axis" Trans.Amer.Soc.Mech.Engrs, Vol.75, pp.473-481.
[4] R.Anantanarayanan and A.Ramachandran, (1958), "Effect of vibration on heat transfer from a wire to air in parallel flow" Trans.Amer.Soc.Mech.Engrs, Vol.80, pp.1426-1432.
[5] B.G.Van Der Hegge Zijnen, (1958), "Heat transfer from horizontal cylinders to a turbulent air flow" Appl.Sci.Res.A7, pp.205-223.
[6] R., Lemlich, (1961), "A musical heat exchanger" Journal of heat transfer, Trans.ASME.Vol.83, pp.385-386.
[7] R., Lemlich, (1961), "Vibration and pulsation boost heat transfer" Chemical engineering, Vol.68, pp.171-176.
[8] F.K., Deaver, W.R., Penny, and T.B., Jefferson, (1962), "Heat transfer from an oscillating horizontal wire to water" Journal of heat transfer, Trans.ASME.Vol.84, series C, pp.133-148.
[9] Takahiko Tanahashi, Tsuneyo Ando, Hideki Kawashima and Teruyuki Kawashima, (1985), "Flow in the entrance region of a porous pipe" Bulletin of JSME, Vol.28, No.237, pp.420-427.

NOMENCLATURE

a	= amplitude	[m]
a_o	= projected area	[m^2]
Cf	= local friction coefficient	—
Cd	= drag coefficient	—
d_h	= hydraulic diameter ($2r_o$)	[m]
f	= frequency of vibration	[Hz]
Nu	= local Nusselt number	—
Pr	= Prandtl number	—
q	= heat flux	[W/m^2]
r	=tube radius	[m]
r_o	= outer radius	[m]
R^*	= dimensionless radius (r/r_o)	—
Re	= Flow Reynolds number	—
T	= temperature	[^oC]
u	= axial velocity	[m/s]
U	= dimensionless axial velocity (u/u_{in})	—
v	= radial velocity	[m/s]
Z^+	=axial distance (z/L)	—

GREEK SYMBOLS

ρ	= density	[kg/m^3]
μ	= viscosity	[Pa.s]
k	= thermal conductivity	[$W/m.^oC$]
ω	=angular frequency of vibration	[rad/s]
τ_w	= wall shear stress	[N/m^2]

SUPERSCRIPT

($\overline{}$)	= average quantity

SUBSCRIPT

v	= vibration
w	= wall
in	= inlet
b	= bulk