



The Effect of Restriction Shape On Laminar Natural Convection Heat Transfer In A Vertical Circular Tube

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Abstract

Natural convection heat transfer is experimentally investigated for laminar air flow in a vertical circular tube by using the boundary condition of constant wall heat flux in the ranges of (Ra_L) from $(1.1 \cdot 10^9)$ to $(4.7 \cdot 10^9)$. The experimental set-up was designed for determining the effect of different types of restrictions placed at entry of heated tube in bottom position, on the surface temperature distribution and on the local and average heat transfer coefficients. The apparatus was made with an electrically heated cylinder of a length (900mm) and diameter (30mm). The entry restrictions were included a circular tube of same diameter as the heated cylinder but with lengths of (60cm, 120cm), sharp-edge and bell-mouth. The surface temperature along the cylinder surface for same heat flux would be higher values for circular restriction with length of (120cm) and would be smaller values for bell-mouth restriction. The results show that the local Nusselt number (Nu_x) and average Nusselt number (\overline{Nu}_L) are higher for bell-mouth restriction and smaller values for (120cm) restriction. For all entry shape restrictions, the results show that the Nusselt number values increases as the heat flux increases. From the present work an empirical correlations were obtained in a form of $(\text{Log } \overline{Nu}_L)$ versus $(\text{Log } Ra_L)$ for each case investigated and obtained a general correlation for all cases which reveals the effect of restriction existence on the natural convection heat transfer process in a vertical circular tube.

Keyword: Restriction Shape, Laminar Natural Convection, Heat Transfer, Vertical Circular Tube.

Introduction

Thermal convection heat transfer is the process by which heat transfer takes place between a solid surface and the fluid surrounding it. If the motion of fluid is due to solely to the action of buoyancy forces arising from the density variations in the fluid owing to the temperature difference between the fluid and the contacting surface, this case called free convection. If the motion of the fluid is caused by forces independent of the temperature differences in the fluid, arising from externally imposed pressure

differences, this case called as forced convection.

The available work on natural convection from the inside surfaces of vertical tubes open at both ends with restriction at entry is very limited. It is only recently that this case has attracted attention due to the use of this tube in practical fields including the storage of cryogenic fluids, petroleum storage vessels on hot days, the thermal response of building to a change in environment temperature,

and the storage of hot fluids for solar power plants and nuclear engineering. In all these cases, the heat loss from or to the tube is an important problem. Extensive analytical and experimental work has been conducted with fluids in a tube subject to different boundary conditions, but the specific problem that will be considered here is that of constant wall heat flux.

However, most of the available investigations are theoretical and deal with the vertical tube in special cases only. To the author knowledge no work is available on the case, which studied in the present work. The present work has been carried out in an attempt to fill a part of the existing gap. It provides experimental data on uniformly heated constant wall heat flux in vertical tube.

Dyer, J.R. (1968) conducted theoretical and experimental works for natural convection heat transfer from the inside surface of a vertical cylinder in the isothermal and constant wall heat flux conditions. He assumed that the incompressible flow is laminar, that the fluid approaching the entrance of the cylinder has a flat velocity profile, that the flow was symmetrical about the vertical axis of the cylinder and the properties of the fluid vary insignificantly with temperature except the density. The results obtained were plotted in the form of (LogNu_R) against $(\text{LogGr}_R \cdot \text{Pr} \cdot R/L)$ for the isothermal cylinder and in the form of (LogNu_R) against $(\text{LogGr}_R \cdot \text{Pr} \cdot \text{Nu}_R \cdot R/L)$ for the constant heat flux cylinder. He also made some experiments on an isothermal cylinder of (25mm) inside diameter and (1220mm) length in the range of $(\text{Gr}_R \cdot \text{Pr} \cdot R/L)$ from (8 to 80). The results obtained were compared with the theoretical results.

Dyer, J.R. (1970) investigated theoretically the development of

laminar natural convective air flow in heated vertical ducts that are restricted at the bottom with (Ra) ranges from (6 to 84). The governing equations (continuity, momentum and energy) were solved numerically by finite difference method. From this work, relationship between (Nu) versus (Ra) and axial temperature distribution were presented.

Kageyama, M. and Izumi, R. (1971) solved numerically by finite difference method natural heat convective fields in the entrance region of a heated vertical circular tube, in which the temperature and velocity profiles were developing simultaneously and the characteristics of heat transfer were obtained. The analysis presented was based on the assumption that the inlet velocity profile is uniform. Analytical and numerical verifications are given for uniform wall temperature and for both infinite and semi-infinite pipe lengths.

Davis, L.P. and Perona, J.J (1974) used a finite difference method to solve the incompressible thermal boundary layer equations for air flow by natural convection in a vertical tube having conditions of constant wall temperature and constant wall heat flux. Air enters the bottom of the tube with uniform velocity and temperature flows upward through the tube due to natural convection. The flow is assumed to be both stable and laminar. The velocity and temperature profiles were obtained for various stages of the flow development, a graphical correlation was found between dimensionless tube length and two dimensionless quantities representative of the volumetric flow rate and the rate of heat dissipation. Comparison was made with available theoretical investigations and showed excellent agreement.

Meric, R.A. (1976) analyzed the development of free convection in

finite vertical tubes by an analytical method, which is based upon a slug-flow linearization of the governing boundary-layer type equations. The resulting equations are solved by means of Laplace transformations to give simple closed form expression for the flow variables. The variation of inlet velocity with (Gr), velocity and temperature profiles for (Pr=0.7) were depicted, also the fluid pressure distribution and heat flux versus tube length were presented. The results were also compared with available numerical results and showed good agreement.

Barrow, R.D. (1978) investigated theoretically and experimentally upward free convection for air (Pr=0.7) in an internally heated vertical duct open at both ends. The experimental data for the wall temperature distribution and local heat transfer coefficient were compared with the results of numerical analysis of an idealized and laminar flow heat transfer model. Correlations for the average Nusselt number, maximum wall temperature and flow rate were presented with the Rayleigh number as the independent variable.

Dyer, J.R. (1983) presented a theoretical and experimental study of natural convective for a laminar air flow in heated vertical ducts. The ducts were open – ended, circular in cross-section and their internal surfaces dissipate heat uniformly. Temperature and velocity fields and the relationship between (Nu) and (Ra) numbers were obtained by solving the governing equations by a step- by- step numerical technique. Two (Ra) numbers were introduced, one expressed in terms of the uniform heat flux and the other in terms of the mean wall temperature. The influence of (Pr) number on the relationship between (Nu) and (Ra) was discussed. Also, experiments were conducted for (Ra) between (1 to

13000). Three ducts were used having same length of (1.22m) but of different internal diameter of (19.1, 25.4, and 46.7mm). From the experiments (Nu) number was determined with air as the working fluid. Comparison between experimental and theoretical studies was carried out and showed good agreement.

Kokugan, T. and Kinoshita, T. (1985) performed experimental work in a heated vertical open ends tube consisting of heated section at constant wall temperature. Correlations between (Gr) and (Re) were derived by setting up a mechanical energy balance in the tube. The following equation was obtained semi-empirically in the experimental conditions:

$$Gr_o = 6.3 Re_o^2 + 3.2(L_H + L_o)/(D.Re_o) \dots(1)$$

Where: L_H =heated length; L_o = entrance length and subscript (o) denoted to at room temperature. The results were compared with available numerical results obtained by (Davis et al, 1971).

Hess, C.F. and Miller, C.W. (1989) carried out experiments using a Laser Doppler Velocimeter (LDV) to measure the axial velocity of a fluid contained in a cylinder subject to constant heat flux on the side walls. The modified Rayleigh numbers ranged between $(4.5 \cdot 10^9)$ to $(6.4 \cdot 10^{10})$, which corresponds to the upper limit of the laminar regime. The flow field inside the boundary layer was divided into three regions along the axis of the cylinder and a parabolic distribution was used to fit the data within each region. Variation of axial velocity with radius for different height in the bottom and top parts of the cylinder and inside the boundary layer region and with radius for different time were presented. Also, the variation of radial position of maximum velocity and radial position of zero velocity with Rayleigh number

was depicted. Excellent agreement was found with an available numerical solution.

AL-Arabi, M. et al (1991) investigated experimentally natural convection heat transfer from the inside surfaces of vertical tube to air in the ranges of $(Gr_{L,Pr})$ from $(1.44 \cdot 10^7)$ to $(8.85 \cdot 10^8)$ and (L/D) ranged from (10 to 31.4). The results obtained were correlated by dimensionless groups as follows:

$$Nu_{mL} = \frac{1.11}{[1 + 0.05(t_{ms} - t_i)]} (Gr_{mL} Pr)^{0.25} \dots\dots (2)$$

In this study, the effect of (L/D) on (Nu_{mL}) was insignificant and the entrance length was practically constant. The results were compared with the available theoretical data of (Dyer, 1975) and showed a good agreement.

Wu, Yissu (1995) examined numerically and experimentally the problem of laminar natural convection in vertical tubes with one end open to a large reservoir, designated open thermosyphons, and to predict flow behavior and the heat transfer rates. In the numerical study, a semi-implicit, time-marching, finite -volume solution procedure was adopted to solve the three governing equations--mass, momentum, and energy--sequentially. Experimental work involved the use of a Mach-Zehnder interferometer to examine the temperature field for a modified rectangular open thermosyphons through the interpretation of fringe patterns. These experimental fringe patterns were used for the qualitative comparison with those obtained from the numerical analyses. Nusselt numbers were determined from the interferometer results and compared with numerical results. Heat transfer rates through the tube wall were found to be strong functions of the tube radius, and

approached an asymptotic limit as the tube radius was increased. Both experimental and numerical results exhibited an oscillatory nature for large height-to-width (aspect ratio) open cavities. Comparisons between experimental and numerically-generated fringe patterns indicated good agreement.

The purpose of the present investigation is to determine experimentally the effect of entry restriction lengths and shapes placed in bottom position of a uniformly heated vertical circular tube on the laminar air flow due to natural convection heat transfer process and to obtain general correlation for this problem.

EXPERIMENTAL APPARATUS

The experimental apparatus which was designed to have a heated section preceded with entry restrictions having different shapes and lengths, as well as, different Grashof number is shown schematically in **Fig. (1a)**, to investigate natural convection heat transfer in a vertical circular tube. The apparatus consists essentially of a cylindrical heated section open at both ends, mounted vertically on a wooden board and the lower end of the restriction shape is protected from outside air currents by shields.

The heated cylinder (1) provided with changeable entry restrictions of four different shapes, particulars of which are: cylindrical restrictions with lengths of (60cm, 120cm), sharp-edge restriction and bell-mouth restriction. The air was withdrawn from atmosphere flows due to buoyancy effect through the entry restriction into the heated section and then the heated air was exhausted to the atmosphere. The heated cylinder (1) is made of copper with (30mm) inside diameter, and (900mm) length. The teflon connection pieces: represents a part of the test section

inlet and another teflon piece represents the heated section exit. The entry restrictions are connected with the heated cylinder by Teflon connection piece bored with the same inside diameter of the (heated cylinder and entry restriction). The Teflon was chosen because of its low thermal conductivity in order to reduce the test section ends losses.

The cylinder is heated electrically by using an electrical heater as shown in **Fig. (1b)**. It consists of nickel-chrome wire (3) electrically isolated by ceramic beads, wounded uniformly along the cylinder as a coil in order to give uniform heat flux. An asbestos rope was used as spacer to secure the winding pitch. The outside of the test section was then thermally insulated by asbestos (4) and fiberglass (5) layers, having thicknesses of (15) mm and (16) mm respectively. Twenty - (0.2mm)-asbestos sheath copper-constantan (type T) thermocouples (2), to measure the cylinder surface temperatures were fixed along the cylinder. The thermocouples (2) were fixed by drilling twenty holes (2) and along the cylinder wall. The measuring junctions were secured permanently in the holes by sufficient amount of high temperature application defcon adhesive. All thermocouples were used with leads and calibrated using the melting points of ice made from distilled water as reference point and the boiling points of several pure chemical substances.

The inlet bulk air temperature was measured by one thermocouple placed in the beginning of the entry restriction, while the outlet bulk air temperature was measured by three thermocouples located in the heated section exit. The local bulk air temperature was calculated by fitting straight line - interpolation between the measured inlet and outlet bulk air temperatures.

To perform heat loss calculation through the heated section lagging, six thermocouples are inserted in the lagging (6) as two thermocouples at three stations along the heated section as shown in **Fig. (1b)**. By using the average measured temperatures and thermal conductivity of the lagging, the heat loss through lagging can be determined. Also, to evaluate the heat losses from the ends of the heated section, two thermocouples were fixed in each teflon piece. By knowing the distance between these thermocouples and the thermal conductivity of the teflon, the end losses could be calculated.

Voltage regulator (variac), accurate ammeter and digital voltmeter were used to control and measure the input power to the working cylinder. The apparatus was allowed to turn on for at least (3 hours) before the steady state conditions were achieved. The readings of all thermocouples were recorded every half an hour by a digital electronic thermometer until the reading became constant, then the final reading was recorded. The input power to the heater could be changed to cover another run in shorter period of time and to obtain steady state conditions for next heat flux.

Data Reduction Method

The following simplified steps were used to analyze the natural convection heat transfer process for air flow in a vertical circular tube when its surface was subjected to a constant wall heat flux boundary condition.

The total input power supplied to cylinder can be calculated as:

$$Q_t = V * I \quad \dots\dots\dots (3)$$

The convection heat transferred from the cylinder surface: -

$$Q_{conv.} = Q_t - Q_{cond.} \quad \dots\dots\dots (4)$$

Where $Q_{cond.}$ = is the total conduction heat losses (lagging and ends losses),

and its calculated from $\left(Q_{\text{cond.}} = \frac{\Delta T}{R_{\text{th}}}\right)$,

where R_{th} is the thermal resistance of the insulations.

The convection heat flux can be represented by: -

$$q_{\text{conv.}} = \frac{Q_{\text{conv.}}}{A_s} \quad \dots\dots (5)$$

Where $A_s = p * D * L$

The convection heat flux, was used to calculate the local and average heat transfer coefficient as follows:

$$h_x = \frac{q_{\text{conv.}}}{T_{\text{sx}} - T_{\text{bx}}} \quad \dots\dots (6)$$

Where: T_{sx} = local surface temperature.

T_{bx} = local bulk air temperature.

All the air properties were evaluated at the mean film temperature (Louis Burmeister, 1993).

$$T_{\text{fx}} = \frac{T_{\text{sx}} + T_{\text{bx}}}{2} \quad \dots\dots (7)$$

Where: T_{fx} = local mean film air temperature.

The local Nusselt number (Nu_x) can be determined as: -

$$Nu_x = \frac{h_x \cdot L}{k} \quad \dots\dots (8)$$

The average values of Nusselt number (\overline{Nu}_L) can be calculated based on the calculated average surface temperature and average bulk air temperature as follows:

$$\overline{T}_s = \frac{1}{L} \int_{x=0}^{x=L} T_{\text{sx}} dx \quad \dots\dots (9)$$

$$\overline{T}_a = \frac{1}{L} \int_{x=0}^{x=L} T_{\text{bx}} dx \quad \dots\dots (10)$$

$$\overline{T}_f = \frac{\overline{T}_s + \overline{T}_a}{2} \quad \dots\dots (11)$$

The average values of the other parameters can be calculated as follows:

$$\overline{Nu}_L = \frac{q_{\text{conv.}} \cdot L}{k(\overline{T}_s - \overline{T}_a)} \quad \dots\dots (12)$$

$$\overline{Gr}_L = \frac{g \beta L^3 (\overline{T}_s - \overline{T}_a)}{\nu^2} \quad \dots\dots (13)$$

$$\overline{Ra}_L = \overline{Gr}_L * Pr \quad \dots\dots (14)$$

Where: $\beta = \frac{1}{(273 + \overline{T}_f)}$, All the air

physical properties (ρ , μ , ν and κ) were evaluated at the average mean film temperature (\overline{T}_f), but all the physical properties in all papers which listed in REFERENCES were taken at a mean film temperature which based on ambient temperature at tube entrance and given by $[T_{\text{mf}} = (T_{\text{ms}} + T_i) / 2]$.

RESULTS AND DISCUSSION

A total of (28) test runs were conducted to cover (four) entry restrictions with different lengths and shapes of two circular tube with length [60cm ($L/D=20$) and 120cm ($L/D=40$), sharp-edge and bell-mouth]. The range of used heat fluxes was (249 W/m^2 to 1000 W/m^2).

Surface Temperature

Generally, many variables such as heat flux, and the flow restriction situation may affect the variation of the surface temperature along the cylinder. The temperature variation for selected runs is plotted in Figs. (3 - 6).

The variation of the surface temperature (T_s), with tube length is shown in Fig. (2) for different heat fluxes. The (T_s -x) curves for all restriction shapes have the same general shape. The value of (T_s) gradually increases with length until a limit beyond which it begins to decrease. This phenomena, which can be explained if Fig. (2) is considered. At the entrance to the tube (point a) the thickness of the boundary layer is zero.

Then it gradually increases until, at (point b), the boundary layer fills the tube. From (point a) to (point b) the heat transfer gradually decreases and (T_s) gradually increases. Beyond (point b) one would expect a straight-line (T_s -x) relation (bc) the case being that of constant wall heat flux. However, as the air is heated along the tube, its physical properties gradually change with the increased temperature. The thermal conductivity increases causing less resistance to the flow of heat and the viscosity increases causing radial flow of the hotter layers of air nearer to the surface to the tube center. A gradual increase of the local heat transfer beyond (point b) must then result. For constant wall heat flux this can only take place if the local difference between the bulk air temperature (as shown by straight-line ac'') and the surface temperature decreases resulting in the shape of the (T_s -x) curve (abc') shown.

Fig. (3) shows the variation of the surface temperature along the cylinder for different heat fluxes, for restriction with length of 60cm ($L/D=20$). This figure reveals that the surface temperature increases at cylinder entrance to reach a maximum value after which the surface temperature decreases. The location of maximum temperature seems to move toward the cylinder entrance as the heat flux increases. This can be attributed to the developing of the thermal boundary layer faster due to buoyancy effect as the heat flux increases, and as explained previously.

Fig. (4) is similar to **Fig. (3)** but pertains to restriction with length of 120cm ($L/D=40$). The curves in the two figures show same trend, but the surface temperature values in **Fig. (4)** are higher than values in **Fig. (3)** due to the length of restriction. **Figs. (5&6)** are similar in trends to **Figs. (3&4)** but pertains to restrictions of

sharp-edge and bell-mouth respectively. **Figs. (7&8)** show the effect of variation of restriction shape on the cylinder surface temperature for low heat flux (249 W/m^2) in **Fig. (7)** and for high heat flux (996 W/m^2) in **Fig. (8)**. It is obvious from these figures that the increasing of restriction length causes an increase of the surface temperature, compared to the case as the heat flux kept constant. It is necessary to mention that the friction between the inside surface of the restriction length and the air flowing through it caused the temperature at entrance to the heated tube to be higher than the ambient temperature. Also, in these figures the lower values of the cylinder surface temperature takes place in bell-mouth restriction because the turbulence is smaller in the sharp-edge restriction.

Local Nusselt Number (Nu_x)

For natural convection from a uniformly heated surface of length (L) exposed directly to the atmosphere, the mean heat transfer coefficient for the whole length is calculated from:

$$h_m = \frac{1}{L} \int_{x=0}^{x=L} h_x dx$$

Where: $h_x = \frac{q}{\Delta T} = \frac{q}{T_{sx} - T_{bx}}$

(ΔT) in above equation is taken as the difference between that local surface temperature (T_{sx}) and the air temperature far away the effect of the surface. All previous investigators written in introduction section calculated the heat transfer coefficient based on the temperature difference between the surface temperature and the fluid temperature at the entrance (T_i) [i.e. (ΔT)= ($T_{sx}-T_{bi}$)]. In the present work, the heat transfer surface is not exposed to the atmosphere (because the flow is confined). Heat is transferred from the hot surface of the cylinder to the air flowing in it.

Therefore, $(\Delta T)_x$ cannot be taken equal to $(T_s - T_i)$. It should be taken as $(T_{sx} - T_{bx})$ where (T_{bx}) is the local bulk air temperature in the cylinder.

The variation of the local Nusselt number (Nu_x) with the dimensionless axial distance (X/D) , is plotted for selected runs in **Figs. (9 - 14)**.

Figs. (9-12) show the effect of the heat flux variation on the (Nu_x) distribution for the four restriction shapes used in the present work respectively. It is clear from these figures that at the higher heat flux, the results of (Nu_x) were slightly higher than the results of lower heat flux. This may be attributed to the secondary flow effect that increases as the heat flux increases leading to higher heat transfer coefficient. Therefore, as the heat flux increases, the fluid near the wall becomes hotter and lighter than the bulk fluid in the core. As a consequence, two upward currents flow along the sides walls, and by continuity, the fluid near the cylinder center flows downstream.

Figs. (13&14) show the effect of variation of restriction shape on the (Nu_x) distribution with (X/D) , for low heat flux (249 w/m^2) in **Fig. (13)** and for high heat flux (996 w/m^2) in **Fig. (14)**. For constant heat flux, the (Nu_x) values give higher results in bell-mouth restriction than the lower values in $(L/D=40)$ restriction. This situation reveals that in bell-mouth the velocity of flow is uniform and the intensity of turbulence is smaller but in sharp-edge restriction there is extra turbulence at the tube entrance as a result of which the effect of viscosity becomes negligible, causes the heat transfer results to be lower than in bell-mouth restriction. But in the restrictions with lengths of $(L/D=20 \text{ and } L/D=40)$ the velocity profile in this condition will be fully developed at tube entrance that may be becomes as a resistance on the

flow and as (L/D) is higher the resistance becomes higher and the surface temperature will be higher, this causes that the (Nu_x) values will be lower than the other restriction shapes.

Average Nusselt Number (\overline{Nu}_L)

The variation of the average Nusselt number (\overline{Nu}_L) with the dimensionless axial distance (X/D) is depicted for selected runs in **Figs. (15 - 18)**. **Figs. (15&16)** show the effect of the heat flux variation on the (\overline{Nu}_L) for restrictions with lengths of $(L/D=20 \text{ and } L/D=40)$ respectively and the effect of variation of restriction shapes on the (\overline{Nu}_L) for low heat flux (249 W/m^2) and high heat flux (996 W/m^2) in **Figs. (17&18)** respectively. The (\overline{Nu}_L) variations for all restriction shapes are similar trend mentioned for (Nu_x) .

Average Heat Transfer Correlation

The general correlation obtained from dimensional analysis (**Ede, A.J., 1967**) for heat transfer by natural convection is:

$$\overline{Nu} = f_1 (\overline{Ra})^n \quad \dots\dots\dots (15)$$

In the case of heat transfer from the inside surface of vertical cylinders one expects that there is an effect of both length and diameter. For similarity with flat surface (which is a cylinder of infinite diameter) the characteristic linear dimension in $(Nu \text{ and } Gr)$ may be taken as the tube length (L) . Then equation (15) becomes:

$$\overline{Nu}_L = f_2 (\overline{Ra})^n \quad \dots\dots\dots (16)$$

Therefore, the following correlations were obtained from the present work for each restriction shape and obtained a general correlation for all restriction shapes as follows, and it shown in **Figs. (19 - 23)**:

$$\overline{Nu}_L = 1.176 (\overline{Ra}_L)^{0.23} \quad \text{For restriction with (L/D=40) (17)}$$

$$\overline{Nu}_L = 1.206 (\overline{Ra}_L)^{0.23} \quad \text{For restriction with (L/D=20) (18)}$$

$$\overline{Nu}_L = 1.372 (\overline{Ra}_L)^{0.23} \quad \text{For sharp-edge restriction (19)}$$

$$\overline{Nu}_L = 1.462 (\overline{Ra}_L)^{0.23} \quad \text{For bell-mouth restriction (20)}$$

$$\overline{Nu}_L = 1.248 (\overline{Ra}_L)^{0.23} \quad \text{For all restriction shapes (21)}$$

As a comparison between the present study and with the normal case, which included vertical cylinder open at both ends without any type of restriction in the laminar range, the correlation obtained for the normal case was (**Louis Berimester, 1993**):

$$\overline{Nu}_L = 0.59 (\overline{Ra}_L)^{0.25} \quad \text{..... (22)}$$

From the comparison, it is clear that the restriction shape and length has a tremendous effect on the heat transfer results.

CONCLUSIONS

As a result of the experimental work conducted in the present investigation to study natural convection heat transfer from the inside surface of uniformly heated vertical circular tube with different restriction shapes placed inside in bottom sections, the following conclusions can be made: the surface temperature increases as the heat flux increases. For the same heat flux the surface temperature for restriction of (L/D=40) is higher than that for other restriction shapes. The variation of (Nu_x) with (X/D), for all restriction shapes has the same trend and this variation is summarized as follows: for all restriction shapes, the (Nu_x) increases with the increase of heat flux, for the same heat flux the (Nu_x) for bell-mouth restriction is higher than for other restriction shapes. Correlations in the form of ($\overline{Log \overline{Nu}_L}$) against

($\overline{Log Ra}_L$) (using cylinder length as the characteristic linear dimension) represent the results for each restriction shape [eqs. (17-20)]. Also, a general correlation for all restriction shapes, which connects all the results [eq. (21)]. From the comparison, the restriction shape and length has a tremendous effect on the heat transfer results.

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NOMNCLATURE			
A_s = Cylinder surface area, (m^2).		Dimensionless Group	
C_p = Specific heat at constant pressure, ($J/kg \cdot ^\circ C$).		Gr = Grashof number = $g b L^3 (T_s - T_a) / \nu^2$	
D = Cylinder diameter, (m).		Nu = Nusselt number = $h.L / k$	
g = Gravitational acceleration, (m/s^2).		Pr = Prandtl number = $m \cdot C_p / k$	
h = Heat transfer coefficient, ($W/m^2 \cdot ^\circ C$).		Ra = Rayleigh number = $Gr.Pr$	
I = Heater current, (ampere).		X/D = Axial distance	
k = Thermal conductivity, ($W/m \cdot ^\circ C$).			
L = Cylinder length, (m).		Subscript	
$Q_{cond.}$ = Conduction heat loss, (W).		a	air
$q_{conv.}$ = Convection heat flux, (W/m^2).		b	bulk
$Q_{conv.}$ = Convection heat loss, (W).		f	film
Q_t = Total heat input, (W).		i	inlet
R = Cylinder radius, (m).		m	mean
T = Air temperature, ($^\circ C$).		s	surface
V = Heater voltage, (volt).		L	based on tube length
x = Axial distance, (m).		t	total
Greek		w	wall
		x	local
b	Thermal expansion coefficient, ($1/K$).		
m	Dynamic viscosity, ($kg/m.s$).	Superscript	
ν	Kinematic viscosity, (m^2/s).		
ρ	Air density, (kg/m^3).	—	average

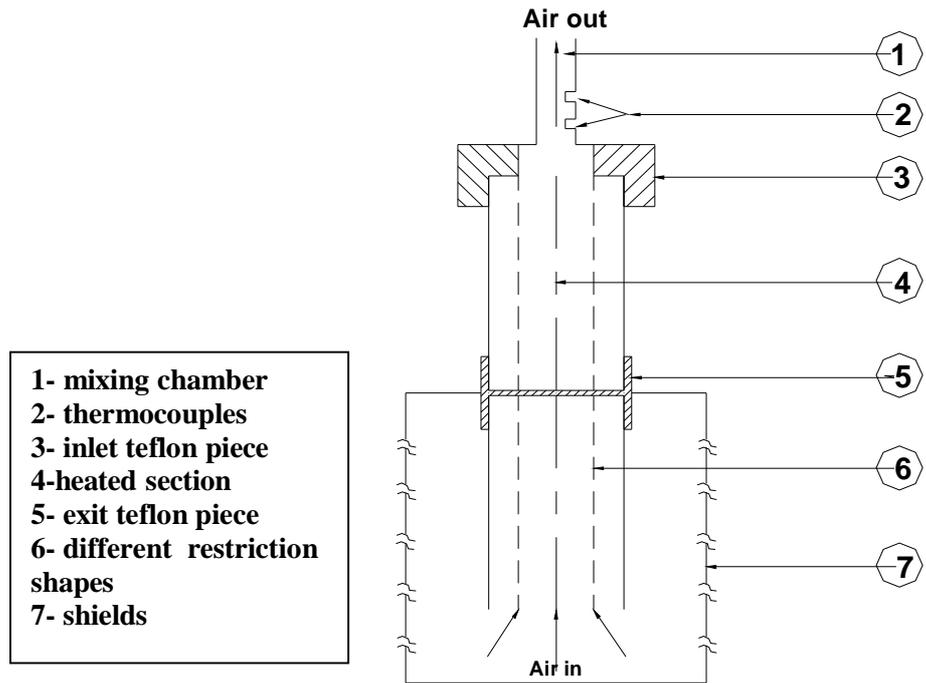


Fig. (1a) Layout of Experimental Arrangement

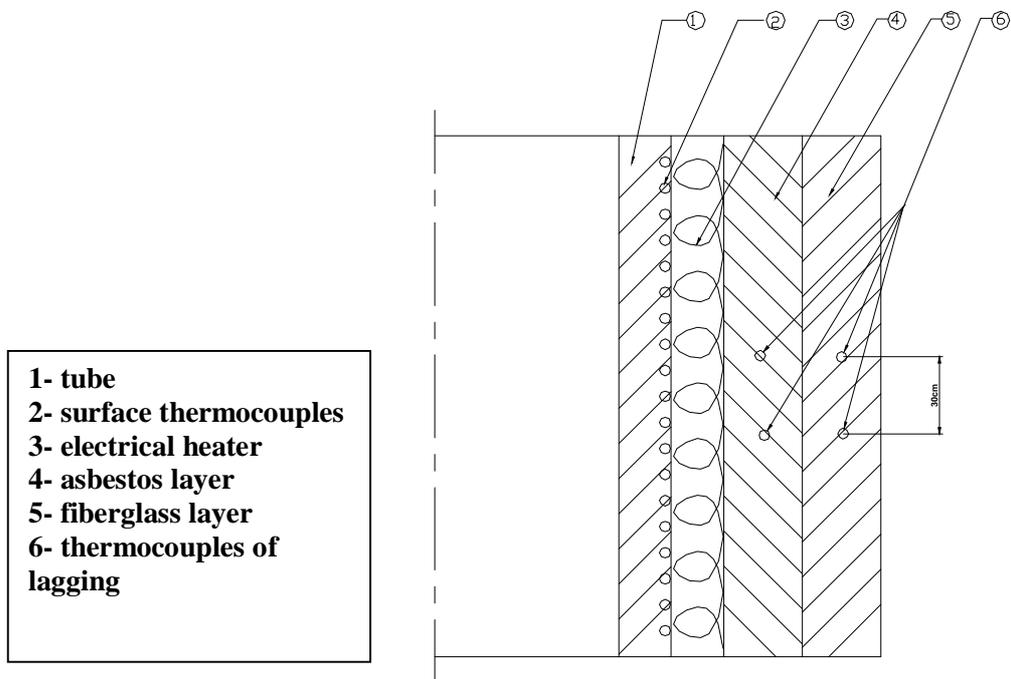


Fig. (1b) General Arrangement of Experimental Apparatus

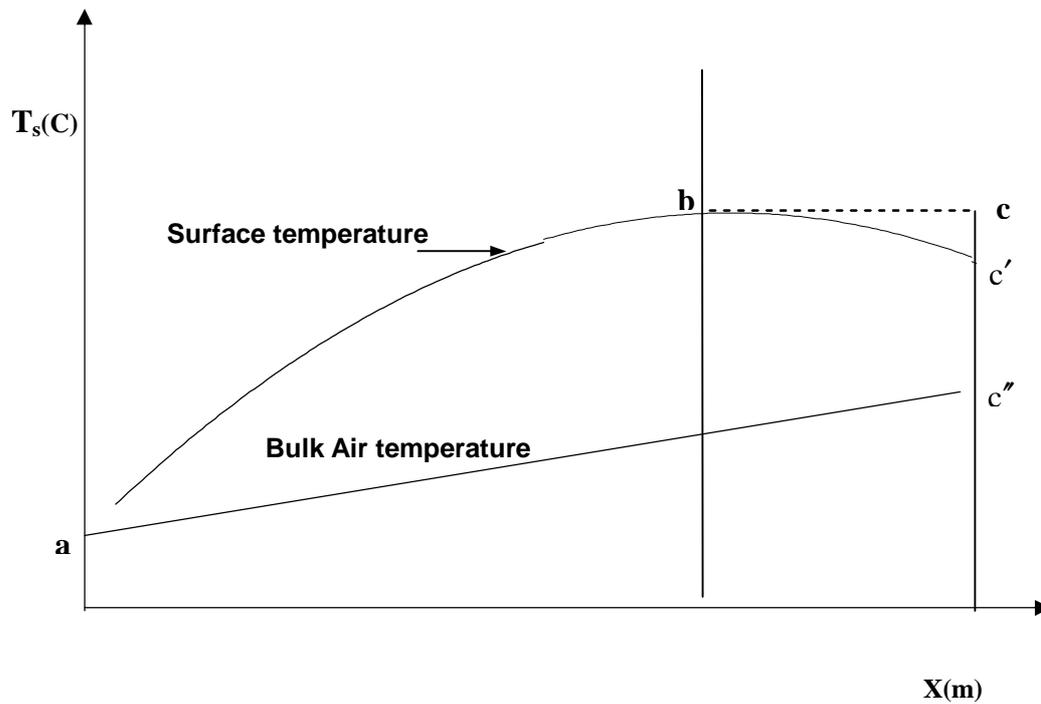


Fig. (2) Variation of surface temperature versus axial distance along the tube

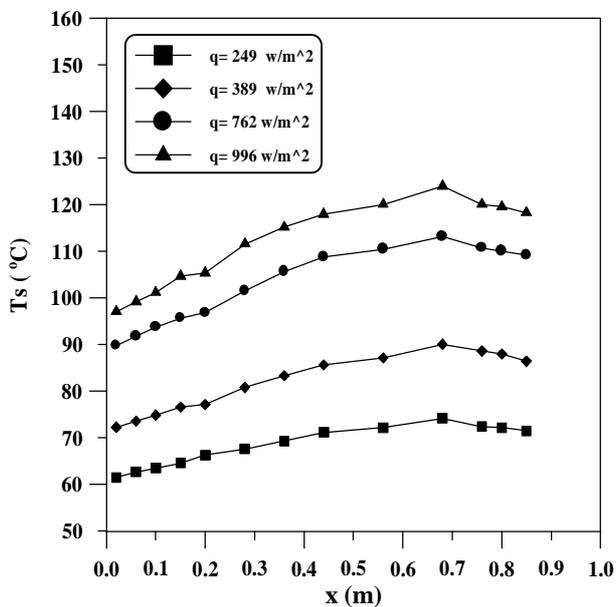


Fig. (3) Variation of surface temperature versus axial distance for restriction of (L/D= 20)

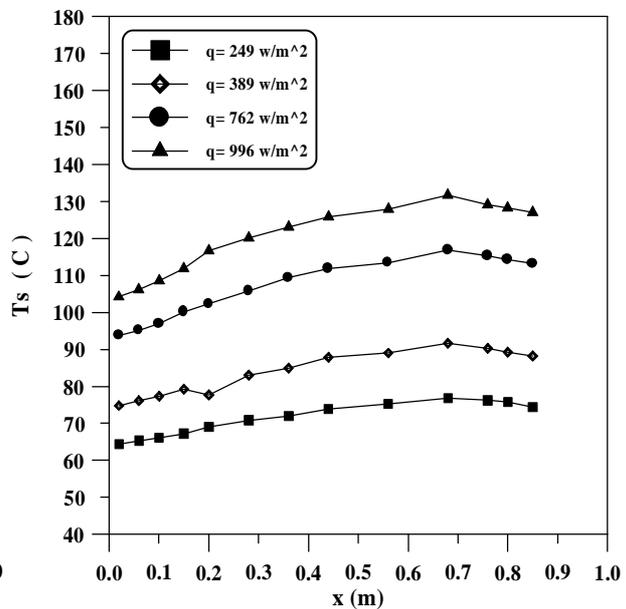


Fig. (4) Variation of surface temperature versus axial distance for restriction of (L/D= 40)

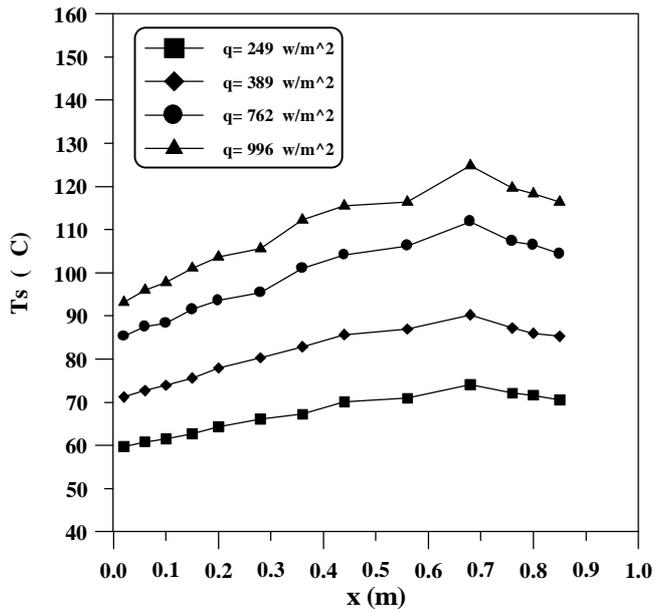


Fig.(5) Variation of surface temperature versus axial distance for sharp-edge restriction

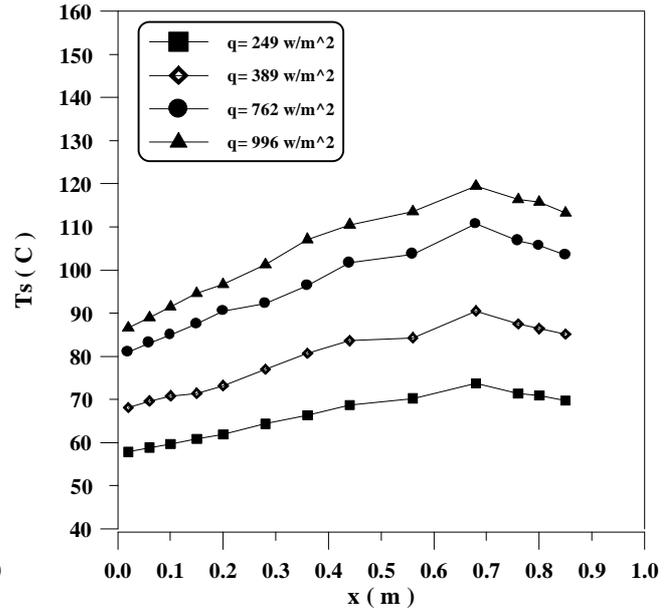


Fig. (6) Variation of surface temperature versus axial distance for bell-mouth restriction

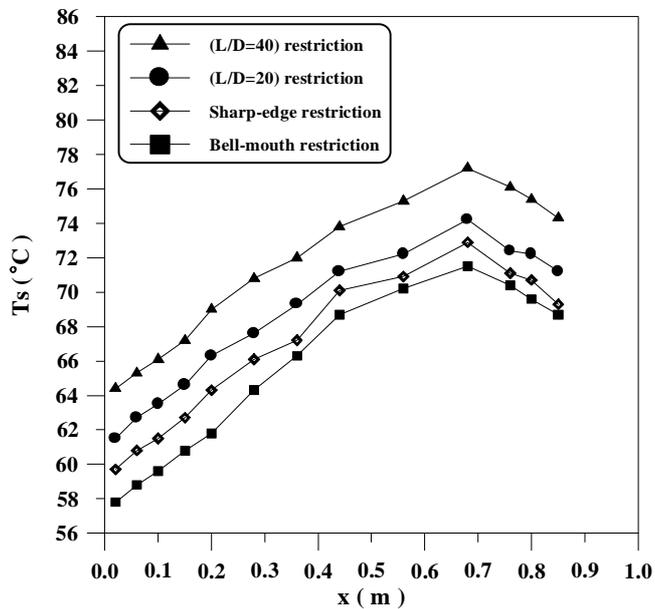


Fig. (7) Variation of surface temperature versus axial distance for different restriction shapes for $q = 249 \text{ w/m}^2$

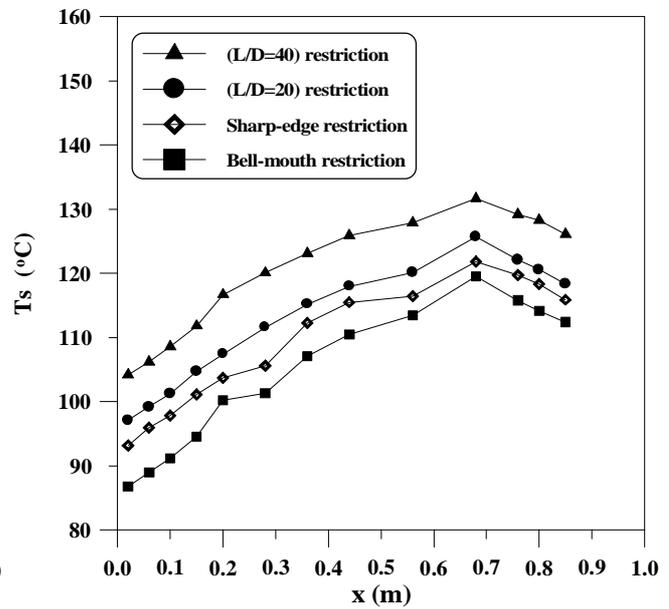


Fig. (8) Variation of surface temperature versus axial distance for different restriction shapes for $q = 966 \text{ w/m}^2$

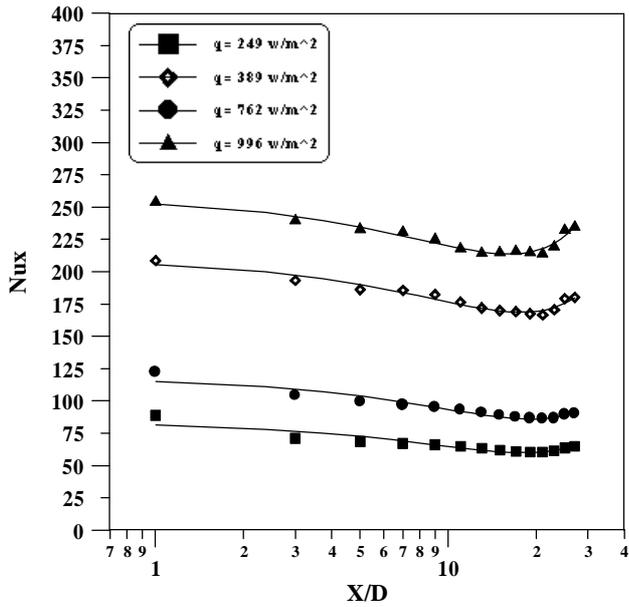


Fig. (9) Variation of local Nusselt number versus axial distance for restriction of (L/D=20)

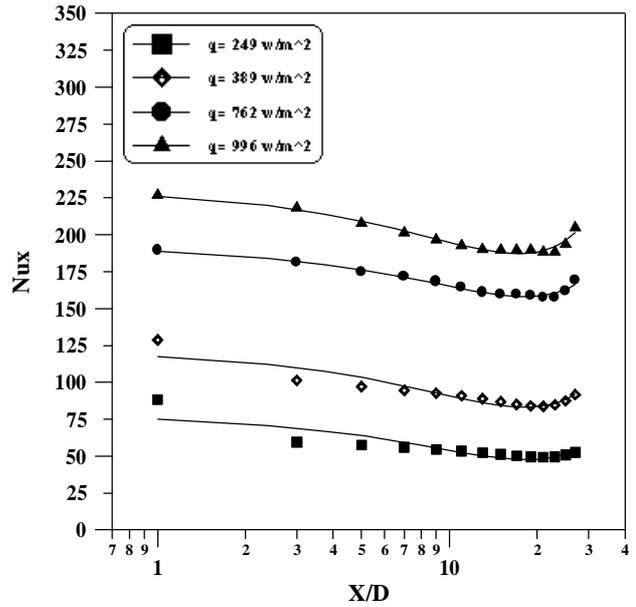


Fig. (10) Variation of local Nusselt number versus axial distance for restriction of (L/D=40)

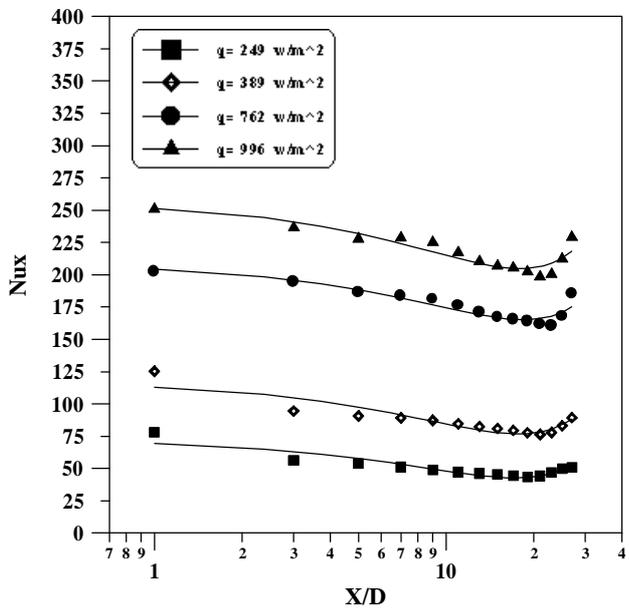


Fig. (11) Variation of local Nusselt number versus axial distance for sharp-edge restriction

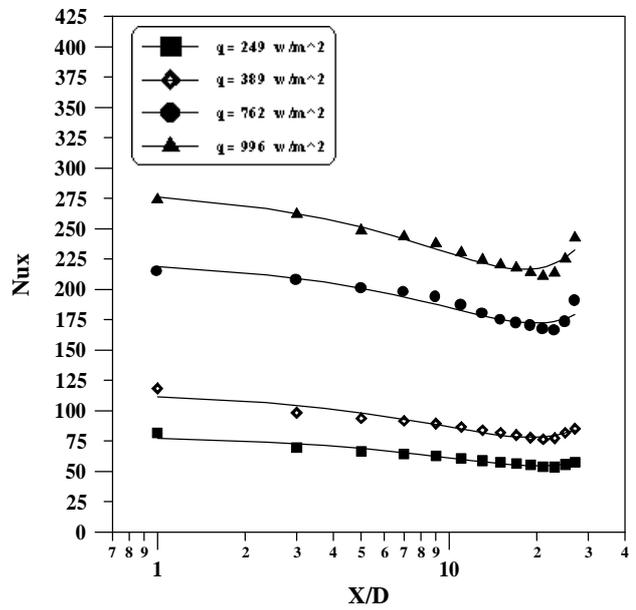


Fig. (12) Variation of local Nusselt number versus axial distance for bell-mouth restriction

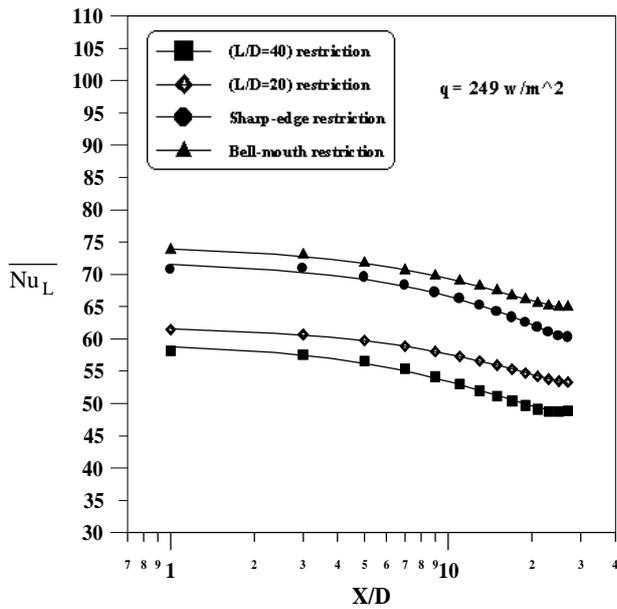


Fig. (17) Variation of average Nusselt number versus axial distance for different restriction shapes

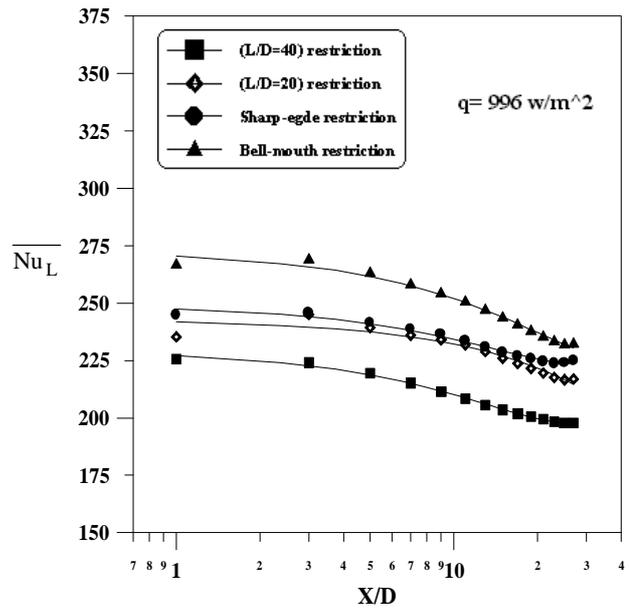


Fig. (18) Variation of average Nusselt number versus axial distance for different restriction shapes

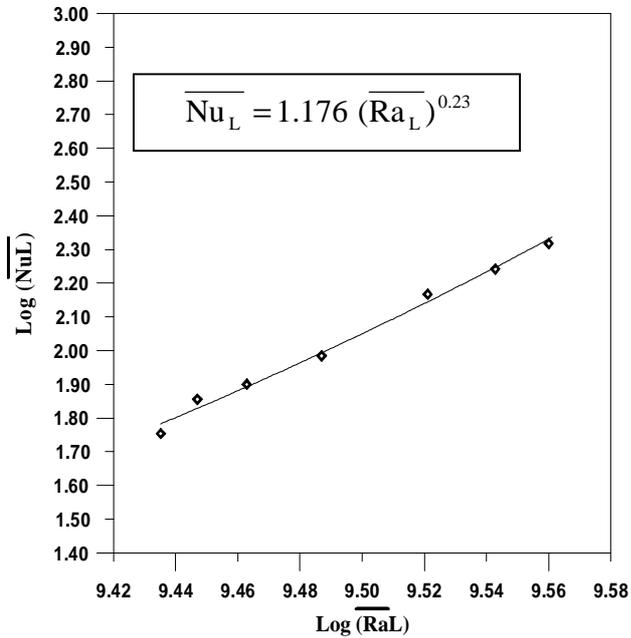


Fig. (19) A correlation between average Nusselt number versus average Rayleigh number for (L/D=40) restriction

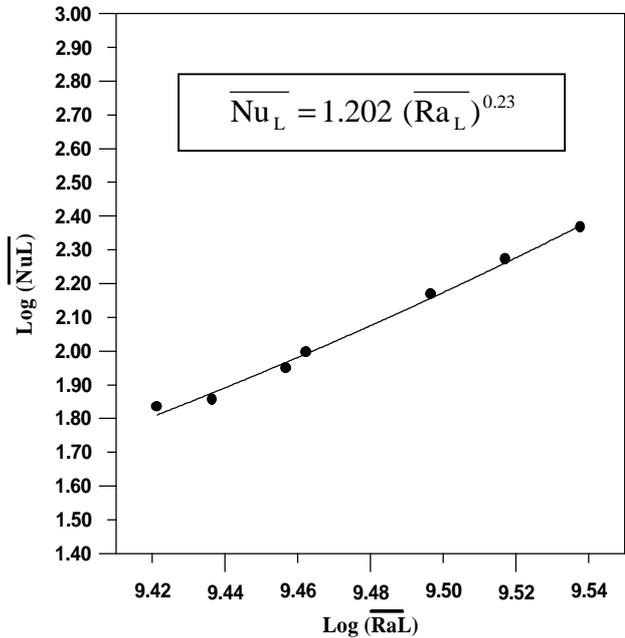


Fig. (20) A correlation between average Nusselt number versus average Rayleigh number for (L/D=20) restriction

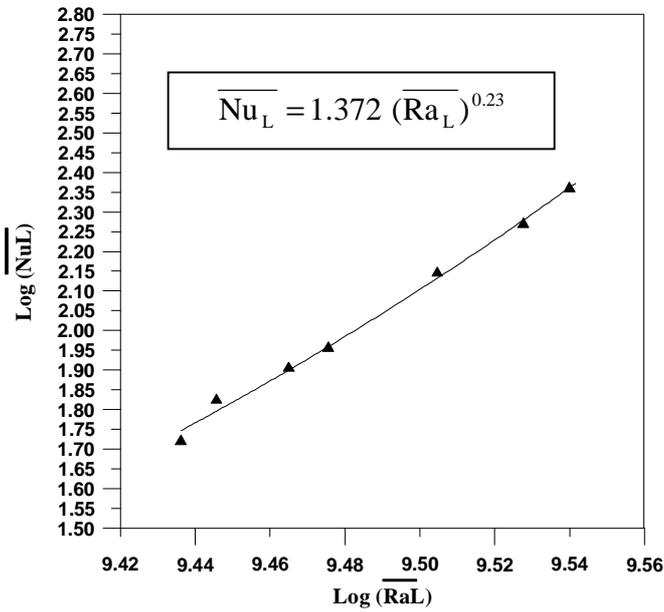


Fig. (21) A correlation between average Nusselt number versus average Rayleigh number for sharp-edge restriction

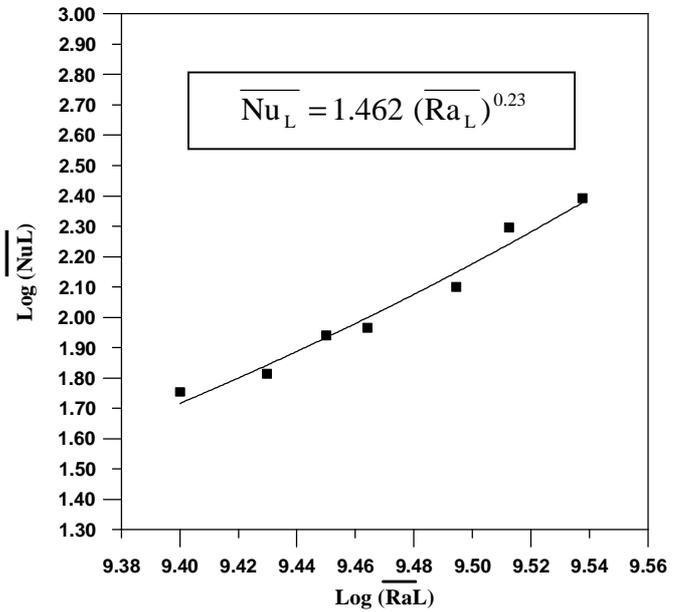


Fig. (22) A correlation between average Nusselt number versus average Rayleigh number for bell-mouth restriction

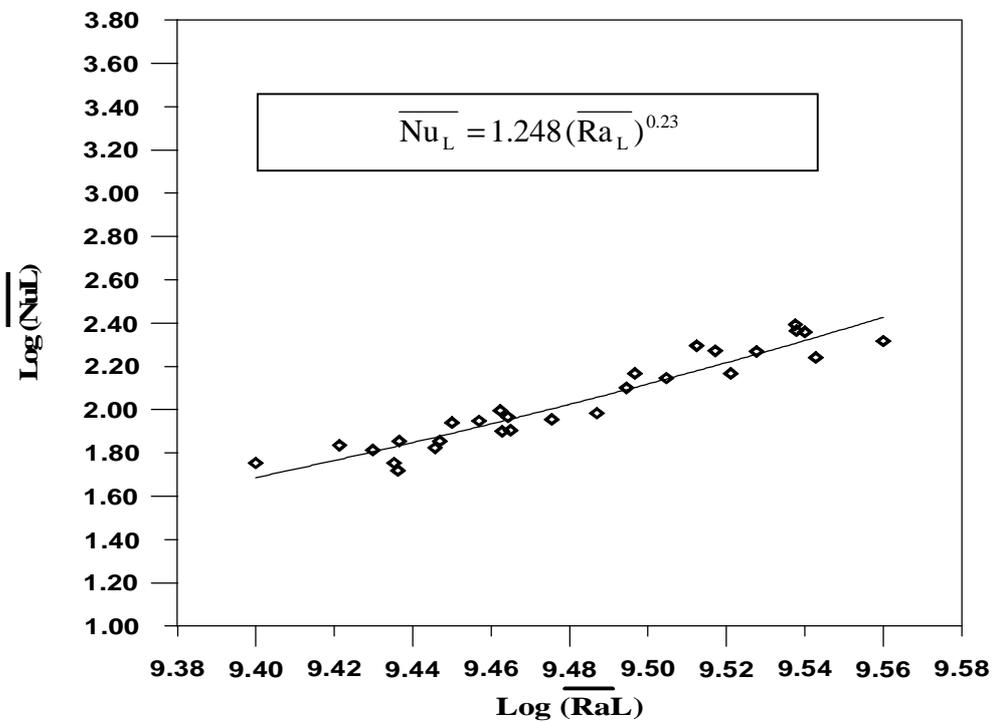


Fig. (23) A correlation between average Nusselt number versus average Rayleigh number for all restriction shapes

تأثير شكل المقيد على انتقال الحرارة الطباقى بالحمل الطبيعي في أنبوب دائري شاقولي

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الخلاصة:

ُبحث عملياً انتقال الحرارة بالحمل الطبيعي لجريان الهواء الطباقى في داخل أنبوب دائري شاقولي باستخدام شرط ثبوت الفيض الحراري ولمدى تغير رقم رايلي (Ra_L) من ($1.1 \cdot 10^9$) إلى ($4.7 \cdot 10^9$). صمم الجهاز العملي لإيجاد تأثير الأنواع المختلفة من المقيد (restriction) الموضوع في مدخل الأنبوب وفي الموقع السفلي لأنبوب التسخين، على درجة الحرارة على طول سطح الأنبوب المسخن وكذلك على معامل انتقال الحرارة الموقعي و المعدل. الجهاز العملي المستخدم يتكون من اسطوانة مسخنة حرارياً بطول (900mm) وبقطر داخلي (30mm). مقيدات الدخول تتضمن أنبوب دائري أسطواني له نفس القطر الداخلي لأنبوب التسخين ولكن بأطوال (120cm, 60cm), وكذلك مقيد بشكل حافة حادة (sharp-edge) بالإضافة إلى مقيد بشكل فم الجرس (bell-mouth). لقد وجد من خلال النتائج العملية أن درجة الحرارة على طول سطح الاسطوانة تكون أعلى ما يمكن للمقيد الأسطواني الذي طوله (120cm) وتكون اقل ما يمكن في حالة المقيد الذي بشكل فم الجرس (bell-mouth) عند نفس الفيض الحراري. أظهرت النتائج أن قيم رقم نسلت الموقعي (Nu_x) والمعدل (\overline{Nu}_L) تكون أعلى ما يمكن في حالة المقيد الذي بشكل فم الجرس (bell-mouth) وتكون اقل ما يمكن في حالة المقيد الأسطواني الذي طوله (120cm), وأيضاً لكل الحالات المستخدمة، بينت النتائج زيادة رقم نسلت كلما زاد الفيض الحراري. لقد تم الحصول على معادلات تجريبية (empirical correlations) بشكل (\overline{Nu}_L Log) ضد (Ra_L Log) لكل حالة من الحالات المدروسة في البحث وكذلك تم الحصول على معادلة عامة والتي تظهر تأثير وجود المقيد على عملية انتقال الحرارة بالحمل الطبيعي في داخل أنبوب دائري شاقولي.