INTERACTION EFFECTS OF HYDRODYNAMICALLY FULLY DEVELOPED PRIMARY FLOW AND SECONDARY FLOW IN THE THERMAL ENTRANCE REGION OF ANNULAR DUCT

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ABSTRACT

Experiments have been conducted to study the local and average heat transfer by mixed convection for hydrodynamically fully developed, thermally developing and fully developed laminar upward air flow in an inclined annulus with adiabatic inner cast iron tube and uniform heated outer aluminum tube with an aspect ratio ($\Omega = 0.72$) and ($L/D_h \approx 40$) for both calming and test sections). A wide range of Reynolds number from 859 to 2024 has been covered, and heat flux has been varied from 159 W/m^2 to 812 W/m^2 (these values of heat flux and Reynolds number gave Richardson number range from 0.03 to 0.٣٨), with angles of annulus inclination $\phi = 0^o$ (horizontal position), $\phi = 60^o$ (inclined position), and $\phi = 90^o$ (vertical position). The hydrodynamically fully developed condition has been achieved by using aluminum annulus (calming section) has the same dimensions as test section and has connected with it by Teflon piece. The average Nusselt numbers have been correlated with the product of (Richardson number and Reynolds number) and compared with available literature and showed satisfactory agreement. The temperature and local Nusselt number profiles results have revealed that the secondary flows created by natural convection have a significant effect on the heat transfer process.

الخلاصة:

أجريت تجارب لدراسة أنتقال الحرارة الموقعي والمعدل بواسطة الحمل المختلط لتمام التشكيل الهيدروديناميكي ولجريان الهواء الطباقي خلال مرحلة التشكيل الحراري وتمام التشكيل الحراري لتجويف حلقي بأنبوب داخلي حديدي المعدن معزول وأنبوب خارجي المنيومي المعدن مسخن تسخين منتظم بنسبة باعية ٠,٧٢ ونسبة ($L/D_h \approx 40$) لكل من مقطعي الخمد والأختبار . أمتد رقم رينولدز من ٨٥٩ الى ٢٠٢٤ ويتغير الفيض الحراري من 159 W/m^2 الى 812W/m^2 (قيم الفيض الحراري ورقم رينولدز تعطي مدى لرقم ريجاردسن من ٠,٠٣ الى 38 .٠) بزوايا ميل (وضع افقي $\phi = 0^o$) ، ($\phi = 60^o$ وضع مائل)، ($\phi = 90^o$ وضع عمودي) . تم الحصول على حالة تمام التشكيل الهيدروديناميكي بوضع مخمد في مدخل مقطع الأختبار الحلقي ذو أبعاد مماثلة لأبعاد مقطع الأختبار ويرتبط معه بواسطة قطعة تفلون. تم استنباط معادلة تجريبية لمعدل أرقام نسلت مع حاصل ضرب كل من رقم ريجاردسن ورقم رينولدز و قورنت بعمل سابق و أعطت قبول مرضي. بينت نتائج توزيع درجة الحرارة و رقم نسلت الموقعي , بأن الجريان الثانوي المتولد من قبل الحمل الحر يمتلك تأثير مهم على عملية أنتقال الحرارة.

Key wards: Combined convection, thermally developing, and inclined annulus.

INTRODUCTION

Heat transfer takes place between a solid surface and a fluid whenever a temperature difference exists. Where mixing of the fluid particles occurs, the heat is transferred by convection. The latter may either be forced or natural depending on whether the fluid motion is imposed or whether it occurs because of a difference in density **[Holland and Moores 1970]**. Combined free and forced convection is known as mixed convection which is characterized by Richardson number (Ri) which represents the ratio of buoyancy force (Gr) and inertial force (Re^2).

The interaction of the natural and forced convection currents is very complex and difficult since it depends not only on all the parameters determining both forced and free convection relative to one another but also on a large number of interacting parameters including the relative direction of the natural and forced convection to each other (i.e. aiding or opposing flow), the geometry of the arrangement, the velocity profile at annulus entrance and the heating surface boundary conditions**[Bergles and Simonds 1971]**.There are many employments for heat transfer by combined convection in concentric annular tubes because of special importance in many industrial engineering applications for examples; double pipe heat exchangers designed for chemical process, food industrial, heating of process fluids, the cooling of electrical cables and nuclear fuel rods, and the collection of solar energy.**[Kotake and Hattori 1985]** studied numerically the mixed convection in a horizontal annulus by examining the similarity condition of fully developed laminar flows of fluid (Pr=1) over the range of($10^4 \leq Gr \leq 10^6$), with radius ratio (0.5). Results of the inner and outer tubes temperature along the annulus, the streamlines and isotherm, and the relation of Nu_m with Re_r , Gr were obtained. **[Nieckele and Patankar 1985]** presented a numerical study for fully developed region of the buoyancy affected flow with an axial laminar flow in a horizontal annular pipe with radius ratio of 0.2, 0.33, 0.5, and 0.66 and ($10^4 \leq Ra \leq 10^7$). The inner wall was heated isothermally while the outer wall was adiabatic. The axial velocity profiles, the temperature variation in the cross section along the annulus and the effect of the radius ratio on the circumferential variation of Nu_x were depicted. Steady-state, fully developed velocity and temperature fields in mixed convection through a horizontal annulus (radius ratio equals 0.8), with a prescribed constant heat flux on the inner cylinder and an adiabatic outer cylinder, were analyzed by (kaviany 1986), using finite difference approximation over the range ($10^5 \leq Ra \leq 10^9$) and (Pr=0.7, 7, 70). Results of inner surface temperature, development of axial velocity profiles, and the effect of the buoyancy on the radial temperature distribution along the annulus were calculated. Numerical calculations have been performed systematically by **[Ihsan and Akeel 2009]** to investigate the parametric influences on the heat and fluid flow patterns and heat transfer rate in the hydrodynamically and thermally fully developed region of inclined annulus of radius ratio fixed at 0.5 with uniformly heated inner cylinder and adiabatic outer cylinder. The ranges of governing parameters covered in the calculations are ($10^3 \leq Ra \leq 10^6$) and Pr=0.7&5. The results show that, for the two values of Prandtl numbers investigated, the transition from single-eddy pattern to the double-eddy pattern appears to occur between 10^5 and 10^6. Large temperature gradients and higher local Nusselt number attain at the bottom except at vertical position in which the angular variation of the Nusselt number remains constant because of symmetry about vertical axis. Only one available experimental work has the same thermal boundary conditions used in the present work. This work was presented by **[Gada**

2009] who studied simultaneously developing laminar mixed convection heat transfer in the entrance region of inclined concentric annuli with a radius ratio of o.555. The investigation covered Reynolds number range ($383 \leq Re \leq 1500$) and Rayligh number range from (1.005×10^5 to 1.52158×10^5) for horizontal position ($\phi=0^o$), vertical position ($\phi=90^o$), and inclined position with aiding and opposing flow ($\phi=\pm30^o,\pm60^o$) where the minus sign refers to opposing flow and the plus sign refers to aiding flow. Results show that the heat transfer process in the aiding flow is better than that in opposing flow and an empirical correlation has been deduced for each angle of inclination.

The present work is included using of entrance section (calming section) in which the upward flow is hydrodynamically fully developed at entrance of heat transfer cylindrical annulus with uniformly heated outer tube and adiabatic inner tube. From the experimental viewpoint, there are no available investigations have dealt experimentally to study the effect of laminar mixed convection to hydrodynamically fully developed, thermally developing and thermally fully developed for air flow in a concentric annulus on the heat transfer process. So , the present work is a step toward broadening the scope of experimental investigations and fulfilling the existing gap in the experimental data for laminar range so that more empirical correlation in a vertical, horizontal, and inclined circular concentric annulus can be developed for assisting flow since these correlations are limited .

EXPERIMENTAL APPARATUS

The scheme of experimental rig is shown in Fig.(1). Generally, it consists of a centrifugal fan (1) which has an air control valve (2) to regulate the air that push to the test section, orifice plate (2), manometer (3), flexible hose(4), settling chamber (5), calming section (8), and heating section (10). The test section (heating section) consists of 1.2m length concentric annulus with (21.9) mm outside diameter of inner tube made of cast iron and insulated from its inside by fiber glass, and heated outer tube made of aluminum with 52.3mm inside diameter and insulated from outside by 60 mm and 5.7mm as thickness for asbestos rope layer and fiber glass, respectively. To enable the calculation of heat loss through the lagging to carry out, eight thermocouples are inserted in the lagging as two thermocouples at four points along the heated section. Using the average measured temperature drop and thermal conductivity of lagging the heat losses through it can be calculated. The outer tube surface temperatures were measured by eighteen asbestos sheath thermocouples (type K) arranged along the outer tube. The calming section of concentric annulus with the same dimensions as test section and connected with it by a two Teflon connection pieces. The first piece has the same outer diameter of inner tube, and the other has the same inner diameter of outer tube. The air induced by the centrifugal fan, enters the orifice pipe section (14) (British standard unit) and then settling chamber through a flexible hose (4). The settling chamber was carefully designed to reduce the flow fluctuation and to get a uniform flow at the beginning of calming section by using flow straightener (6). A uniform velocity profile by a well designed Teflon bell mouth (7) which was fitted at the begging of outer tube of calming section (8). The inlet air temperature was measured by one thermocouple located in the settling chamber (5) while outlet bulk air temperature was measured by two thermocouples located in the test section exit. The local bulk air temperature was calculated by fitting straight line-interpolation between the measured inlet and outlet bulk air temperatures. The choice of linear distribution of the bulk air temperature is attributed to the following reason: for constant wall heat flux (q) boundary condition, the bulk temperature gradient is calculated from:

$$\frac{dt_b}{dx} = \frac{q.p}{m^{\cdot} c_p} = \frac{p}{m^{\cdot} c_p} h(t_s - t_b) \qquad \ldots(1)$$

Where p is the perimeter of outer tube = πd_2.

From eq.(1) the axial variation of t_b may be determined. The heat is transferred to the fluid and t_b increases with x. For constant heat flux (q) it follows that the right hand side of Eq.(1) is constant and independent of the distance (x), hence,

$$\frac{dt_b}{dx} = \frac{q.p}{m^{\cdot} c_p} \qquad \ldots(2)$$

By integrating and applying the boundary condition (at x=0 : t=t_i), it follows that:

$$t_b(x) = t_i + \frac{q.p}{\dot{m}.c_p} x \qquad \ldots(3)$$

Where t_i is the inlet bulk air temperature accordingly, the bulk temperature varies linearly with the distance (x) along the tube annulus. Moreover, from the following equation:

$$q = h(t_s - t_b) \qquad \ldots(٤)$$

The temperature difference (t_s - t_b) varies with the distance (x) **[Incropera and Dewitt 2003]**. The difference is initially small (due to the large value of the heat transfer coefficient at the tube entrance) but increases with increasing the distance (x) due to the decrease in heat transfer coefficient of the outer tube that occurs as the boundary layer develops **[Incropera and Dewitt 2003]**.

EXPERIMENTAL PROCEDURE

Voltage regulator (variac), accurate ammeter and digital voltmeter were used to control and measure the input power to the working outer tube of annulus. The flow becomes hydrodynamically fully developed, thermally developing and thermally fully developed at the entrance to the test section by using a calming section with the same ($L/D_h \approx 40$) as test section. The ratio is enough for the flow to reaches these conditions. The Reynolds number under consideration is ranged from 859 to 2024, actually this range has been selected after so many experimental attempts so as to ensure that the mixed convection regime has been covered and accordingly this range gives the thermally developing flow and the thermally fully developed flow conditions. Moreover, this range was selected since if any fluid enters the annular gap at a uniform temperature is less than the surface temperature, convection heat transfer will be occurred and a thermal boundary layer begins to develop. In addition, if the outer tube surface conditions is fixed by imposing either a uniform temperature is constant or a uniform wall heat flux q is constant a thermally fully developed condition is eventually reached. For both surface conditions, however, the amount by which fluid temperatures exceed the entrance temperature increases with increasing the distance (x). For laminar flow the thermal entry length my be expressed as **[Incropera and Dewitt 2003]**:

$$(X_{fd,t} / D_h)_{Laminar} \approx 0.05 \text{ Re Pr} \qquad \ldots(5)$$

UNCERTAINTY ANALYSIS

The accuracy of experimental results depends upon the accuracy of the individual measuring instrument and the manufacturing accuracy of the circular inner and outer tubes. Also, the accuracy of any instrument is limited by its minimum division (its sensitivity). In the present work, the uncertainties in heat transfer coefficient (Nusselt number), Reynolds number and Richardson number were estimated following the differential approximation method **[Holman 1984]**. For a typical experiment, the total uncertainty in measuring the heater input power, temperature difference (t_s – t_b), the heat transfer rate , the circular tube surface area and the air flow rate were ± 0.2%,± 0.33%, ± 1.8%, ± 1.5%, and ± 0.02%, respectively. These were combined to give a maximum error of 1.45 % in heat transfer coefficient (Nusselt number) and minimum error of ± 1.35% in Reynolds number and ± 1.41%, in Rayliegh number.

RESULTS

Generally, when Richardson number is kept constant, the effect of buoyancy forces in a horizontal annulus is larger than other annulus inclination angle. Therefore, it can be expected for the same conditions of flow rate and input heat flux, the distribution of surface temperature along the outer tube distance increases as the annulus inclination angle changes from horizontal to vertical position as shown in Fig.(2) for Ri=0.03.This behavior can be attributed to that in horizontal position, the direction of forced convection is perpendicular to the direction of secondary motion caused by natural convection, so a spiral vortex will be generated along the axial distance causes a reduction in the surface temperature. This vortex will be weak if the angle of inclination deviates from the horizontal position towards the vertical position in which it diminishes. If the behavior of temperature distribution is studied alone in this figure, it will be noticed that the temperature value increases with the axial distance because the free convection effects do not start at the annulus inlet but require a starting length before being established. Then, the temperature value begins to decrease down stream due to strong natural convection in this region and heat losses. Fig.(3) shows the effect of angle of inclination on the local Nusselt number with the inverse Greatz number $(Gz)^{-1}$ (dimensionless axial distance) for Ri=0.076. It is apparent from this figure that the values of Nu_L decrease as the angle of inclination moves from horizontal to vertical position. The general variation of Nu_L reveals that the Nu_L near the inlet of annulus heated region are very high values since at the onset of heating, the wall to fluid temperature difference is larger (i.e; the thickness of thermal boundary layer is zero) leading to no fluid thermal profile has developed (there is only hydrodynamically fully developed fluid profile which distorts continuously as the flow moves further downstream of the test section). As a result, there is no net buoyant force upstream. Then the value of Nu_L decreases continuously due to the thermal boundary layer develops, and then near the exit of annulus heated region, the Nu_L value slightly increases because the secondary flow resulting from the natural convection which accelerates the approach of the fluid to the wall temperature through enhancement of the convection process down stream. It is expect that the length to hydraulic diameter ratio of the test section would have some bearing on the heat transfer performance and it is enough in the present work ($L/D_h \approx 40$) to obtain thermally fully developed. Taking into account the stronger of secondary currents in the horizontal position which is coupled with primary flow to produce strong vortex, the heat transfer process in the horizontal position is batter that in other angles of inclination. The relation between average Nusselt number and Reynolds number with Rayleigh number as a parameter is shown in Fig.(4). The results of Nu_m obtained for Re between 859 to ٢٠٢٤ and for Ra varied from 7.3×10^4 to 1.9×10^5. As be shown, the mean Nusselt number increases as Re increases where Ra is kept constant, and as Ra increases where Re is kept constant. The same result is obtained in Fig.(5) which shows the relation between Nu_m and Ra with Re as a parameter. Fig.(6) shows the effect of angle of annulus inclination on the relation between Nu_m and Ra where Re equals 2024. It is noticed that the heat transfer process improves as the angle of inclination moves from vertical to horizontal position. The same result will be obtained if Ra is kept constant ($Ra=1.9 \times 10^5$) as shown in Fig.(7). The values of the average Nusselt number (Nu_m) for horizontal (ϕ=0°), inclined (ϕ=60°), and vertical positions (ϕ=90°) are plotted in Figs.(8-10); respectively in the from of log(Nu_m) against log(Ri.Re) for the range of Re from 895 to 2024 and Ri from 0.03 to 0.٣٨. The dashed line in this figure represents the empirical correlation that deduced by Gada. It was shown that the heat transfer equations for all the positions have the same following from:

$$Nu_m = c\,(Ri\,.\,Re)^m \qquad \ldots(6)$$

Where c and m are shown in table 1.

These figures show that the values of Nu_m in the present work are higher than in the work of Gada. This result leads to the important physical fact that the heat transfer process in the hydrodynamically fully developed region of annulus is better than that in the hydrodynamically developing region.

CONCLUSIONS

As a result from the experimental work conducted in the present work to study mixed convection heat transfer for thermally developing and hydrodynamically fully developed laminar air flow in an inclined annulus with uniformly heated outer tube, the following conclusions can be made:

1. The horizontal position gives a stronger spiral vortex along the axial distance of the annular gap causes a reduction in the surface temperature and increasing in the heat transfer coefficients.
2. The variations of surface temperature and the local Nusselt number were found to be strongly dependent on Richardson number and angle of inclination.
3. The heat transfer results for horizontal, inclined, and vertical position for mixed convection in laminar range were correlated as $Nu_m=c(Ri.Re)^m$
4. The mixed convection region has been bounded by the suitable selection of Richardson number (i.e., Reynolds number and Rayliegh number ranges) which is varied approximately from 0.03 to 0.38.
5. The present experimental results have been compared with the available literature and showed similar trend and important physical fact says the heat transfer process in the hydrodynamically fully developed region of duct (any geometry) is better than that in the developing region of this duct.

NOMENCLATURE

Cp Specific heat at constant pressure, (J/Kg.^{0}C)
D_h Hydraulic diameter, (m)
h Coefficient of heat transfer, (W/m^2. ^{0}C)
L Annulus length, (m)
q Convection heat flux, (W/ m^2)
r_1 Radius of inner tube (m)
r_2 Radius of outer tube (m)
$\dot{m}$ Mass flow rate (Kg/sec)
κ Thermal conductivity of air (W/m.°C)
t Outer tube surface temperature (^{0}C)
x Axial distance

Greek

ϕ Angle of inclination, (degree)
μ Dynamic viscosity, (Kg/m.s)
ν Kinematics viscosity, (m^2/s)
ρ Air density at any point, (kg/m^3)
β Thermal expansion, (1/K)
Ω Aspect ratio= r_1/r_2-r_1

Dimensionless Gropes:

Gr Grashof number $= \dfrac{g\,\beta\,q r_1^{\,4}}{k\,\nu^{\,2}}$

Nu Nusselt number $= \dfrac{hD_h}{k}$

Ra Rayligh number $= Gr.Pr$

Re Reynolds number $= \frac{u_i D_h}{\nu}$

Ri Richarson number $= \frac{Gr}{\mathrm{Re}^2}$

Nu Nusselt number $= \frac{qD_h}{k(t_s - t_b)}$

G_Z Graetz number $= Re.Pr.D_h / x$

Subscript:

L Local
b Bulk
f Film
i Inlet
s Surface
fd,t Thermally fully developed
m Average

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Table 1: Constants in Eq.(٦) for various angles of inclination

ϕ	Present work		Gada work	
	c	m	c	m
0^o	3.991	-0.469	3.749	-0.487
60^o	3.48	-0.495	3.172	-0.493
٩$0^o$	2.8681	-0.481	2.402	-0.438

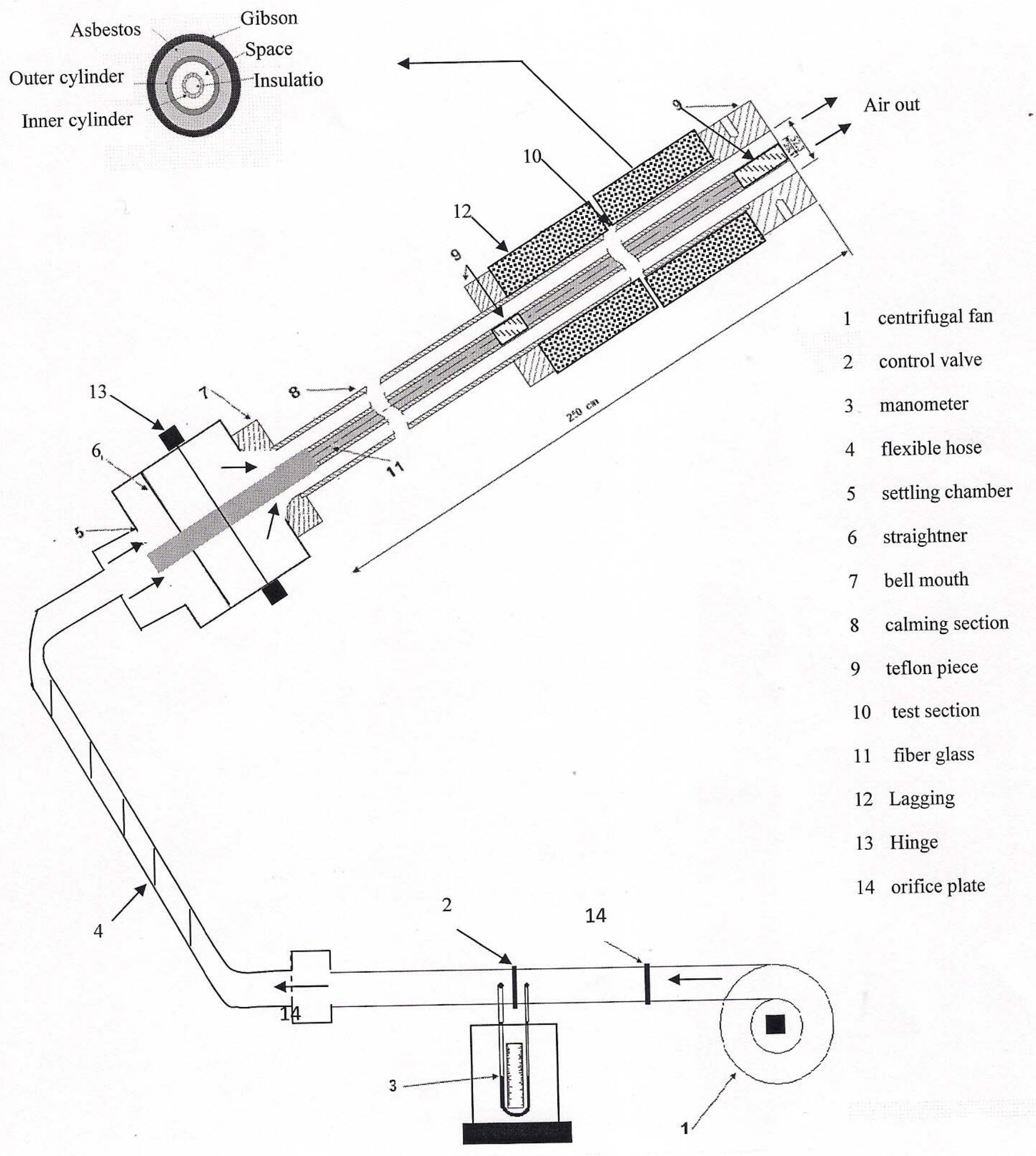

Fig .1: Diagram of experimental arrangement.

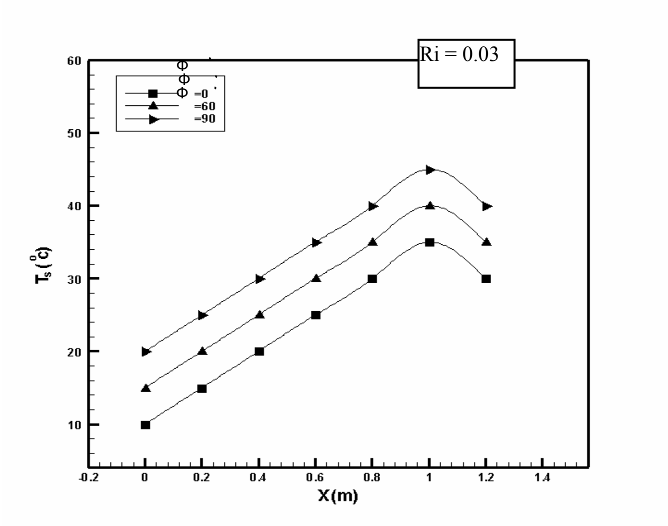

Fig.2: Variation of surface temperature along x-axis.

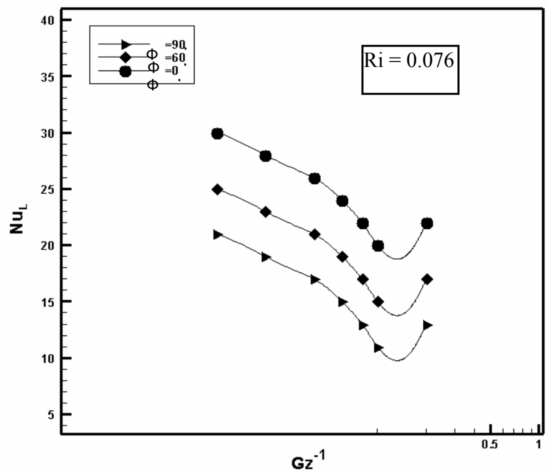

Fig.3: Local Nusselt number versus inverse Greatz number.

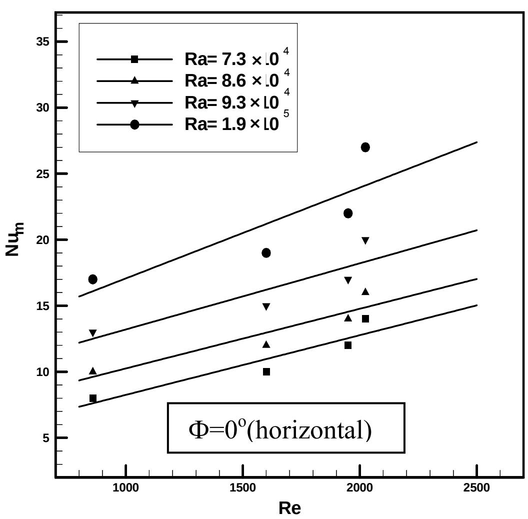

Fig.4: Mean Nusselt number versus Reynolds Number for various Rayliegh number.

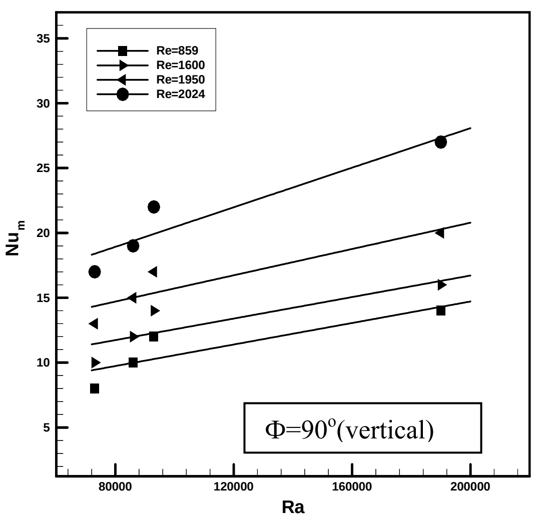

Fig.5: Mean Nusselt number versus Rayliegh number for various Reynolds number.

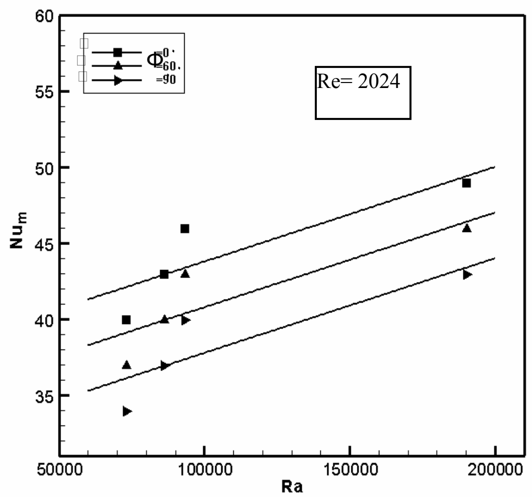

Fig.6: Mean Nusselt number versus Rayliegh numbe for various angles of inclination and Re=2024

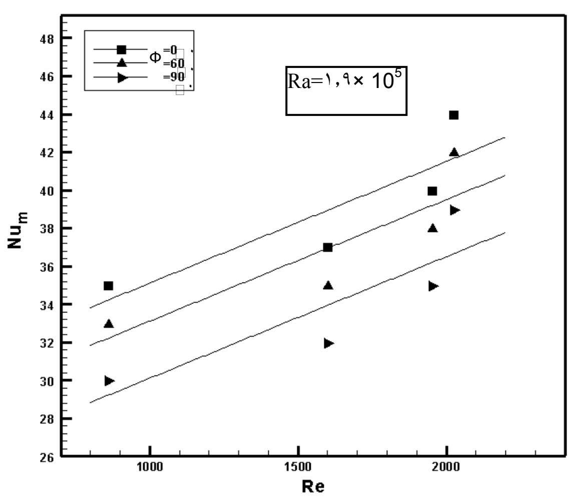

Fig.7: Mean Nusselt number versus Reynolds number for various angles of inclination and Ra=1.9×10^5

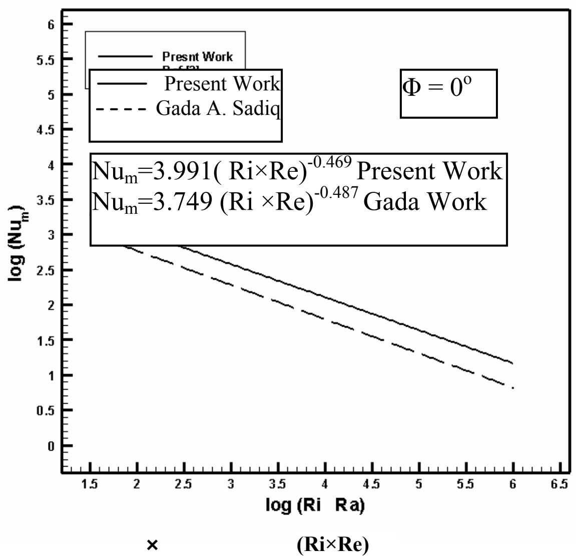

Fig.8: Log(Nu_m) versus Log(Ri.Re) for $\Phi = 0^o$

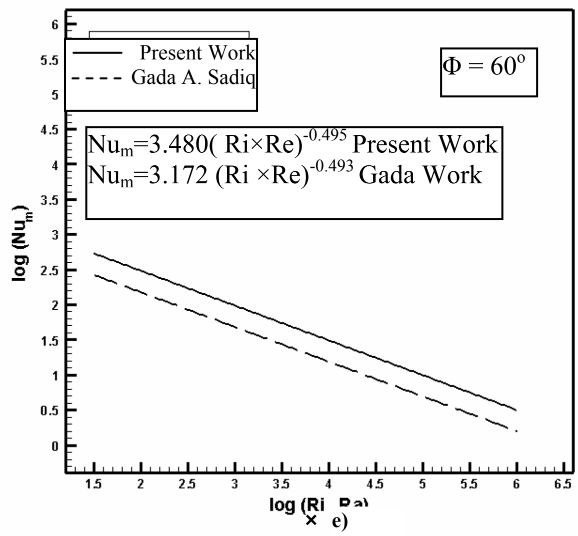

Fig.9: Log(Nu_m) versus Log(Ri.Re) for $\Phi = 60^o$

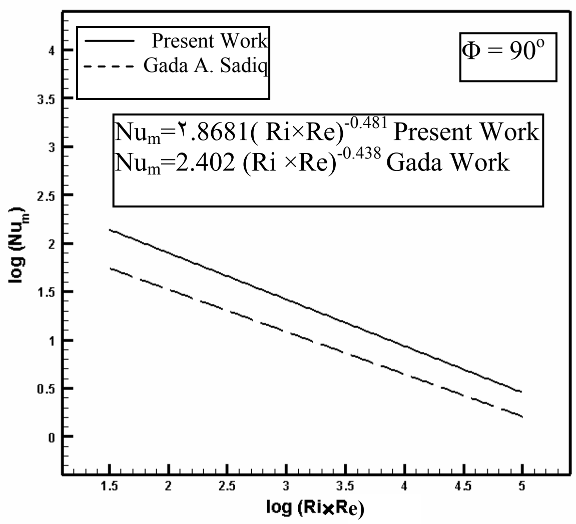

Fig.10: Log(Nu_m) versus Log(Ri.Re) for $\Phi = 90^o$