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Heat Transfer by Mixed Convection in the Opposing Thermally Developing Flow in a Vertical and Inclined Annulus

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Abstract

Combined convection heat transfer in an inclined ($\psi=60^o$) and vertical ($\psi=90^o$) annulus has been experimentally studied for opposing thermally developing and thermally fully developed laminar air flows with adiabatic inner tube and uniformly heated outer tube ($r_1/r_2=0.41$) for Reynolds number range from 450 to 1000 and heat flux is varied from 150 W/m^2 to 780 W/m^2. The hydrodynamically developed condition has been achieved by using entrance section annular pipe (calming section) having the same dimensions as test section ($L/D_h \approx 40$). The mixed convection regime has been bounded by the convenient selection of Re and heat flux ranges, so that the obtained Richardson number varied approximately from 0.05 to 0.97. The average heat transfer results have been correlated with an empirical correlation by dimensionless groups as $\log Nu_m$ against $\log Ra/Re$ and compared with available literature showed that the heat transfer process in the hydrodynamically fully developed region of duct is better than that in the developing region of this duct.

Key wards: Mixed convection, fully developed, annulus

دراسة عملية لانتقال الحرارة بالحمل المختلط للجريان المعاكس بتجويف حلقي مائل وعمودي

الخلاصة

تم أجراء دراسة عملية لأنتقال الحرارة بالحمل المركب بتجويف حلقي مائل بزاوية 60^0 وعمودي بزاوية 90^0 لجريان الهواء الطباقي المعاكس في منطقتي التشكيل وتمام التشكيل الحراري بأنبوب داخلي معزول وأنبوب خارجي مسخن تسخين منتظم ($r_1/r_2=0.41$) ولمدى عدد رينولدز من 450 الى 1000 وفيض حراري يتغير من 150 W/m^2 الى 780 W/m^2 . شرط التشكيل الهيدروديناميكي يحصل بأستخدام أنبوب حلقي يمثل مقطع الدخول (مقطع المخمد) يمتلك نفس الأبعاد التي يمتلكها مقطع الأختبار ($L/D_h \approx 40$) . منطقة الحمل المختلط تحدث بالأختبار المناسب لمديات رقم رينولدز والفيض الحراري، بحيث أن عدد ريجاردسن الحاصل يتغير تقريباً من 0.05 الى 0.97 . تم وضع معادلات تجريبية لنتائج معدل أنتقال الحرارة بمجموعة لا بعدية للوغارتم Nu_m ضد لوغارتم Ra/Re ، وقورنت بالاعمال السابقة وبينت أن عملية أنتقال الحرارة بمنطقة تمام التشكيل الهيدروديناميكي لقناة هي أفضل من تلك لمنطقة التشكيل لنفس القناة.

الكلمات الدالة: حمل مختلط،تمام التشكيل،تجويف حلقي

Nomenclatures

Cp Specific heat at constant pressure, $(J/Kg.^0C)$

D_h Hydraulic diameter, (m)

h Coefficient of heat transfer, $(W/m^2.{}^0C)$

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)

u_i Initial velocity (m/s)

λ Thermal conductivity $(W/m.^oC)$

τ temperature (^0C)

x Axial distance

Greek

Ψ annulus inclined angle, (degree)

ν Kinematics viscosity, (m^2/s)

ρ Air density at any point, (kg/m^3)

μ Dynamic viscosity, (Kg/m.s)

Abbreviation

TDF Thermally developing force convection

SDF Simultaneously developing force convection

Introduction

General

The convection heat transfer coefficient and fluid flow characteristics of duct flows are often affected by the presence of gravity forces, particularly at low or moderate flow rates. The orientation of the duct can have a considerable influence on the velocity and temperature profiles and the associated heat transfer in the duct. For horizontal tubes, the buoyancy forces are per pendicular to the main flow direction and they give rise to secondary currents in the cross section. For vertical tubes, the gravity forces are in the main flow direction , and axial symmetry is presented since there is no secondary flow in the cross section. In inclined tubes, however, buoyancy forces act in both the main flow and the cross-stream direction. In practice, this situation is commonly ecountered in heat exchanger equipment and in sollar

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 $\dfrac{u_i D_h}{\nu}$

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

Nu_x Local nusselt number $\dfrac{qD_h}{k(t_s - t_b)}$

G_Z Graetz number Re.Pr.D_h / x

Subscript

x Local

b Bulk

f Film

m Mean

i Inlet

fd,t Termally fullu developed

m Average

hy Hydrodynamically fully developed

w Wall

collectors (Choudhury and Patankar 1988)[1]. Other applications for mixed convection can be found in thermal control of electronic components, chemical vapour deposition, fluid flow in solar collector, passive cooling of nuclear reactors, various manufacturing processes, heat exchangers for viscous liquids in chemical and food processing industries, heat exchangers for biomedical applications and compact heat exchangers for gas flow(Grassi and Testi 2008)[2].

Pervious work

Many experimental and theoretical investigations have been conducted to study the mixed convection heat transfer inside horizontal, inclined and vertical annulus.(Kotake and Hattori 1985)[3] presented numerical study into combined convection flows in a horizontal annulus. Two cases of thermal boundary condition at the wall were considered for the secondary flow; constant heat flux and constant wall temperature, which lead to an appreciable difference in the flow behavior. The numerical results of the mean Nusselt number are compared with the experiment showing a good agreement. Steady – state, fully developed velocity and temperature fields in mixed convection through a horizontal annulus (ratio of outside to inside radius of 1.25) , with a prescribed constant heat flux on the inner cylinder and an adiabatic outside cylinder, were analyzed by (Kaviany 1986)[4]using finite difference approximations. The effects of the buoyancy driven lateral flow on of the inner surface temperature were studied in detail. The results show that, as the buoyancy potential (Rayleigh number) increases, the lateral flow structure changes from one cell (on each side) to two cells. (Hanzawa,et al 1986)[5] performed experiments to study the mixed convection of upward gas flow in a vertical annulus of radius ratio range from 0.39 to 0.63 and hydraulic diameter to heating section length range from 0.34 to 1.4.The effects of operation conditions on the temperature profiles, flow pattern and heat transfer coefficient were investigated. (Falah 1993)[6] performed experiments to study the local and average heat transfer by mixed convection to a simultaneously developing upward air flow in a vertical, inclined, and horizontal concentric cylindrical annulus with a radius ratio of 0.41 and the inner cylinder with a heated length of 0.85m and outer cylinder were subjected to the ambient temperature. The results show that the heat transfer process improves as the angle of inclination deviates from the vertical to the horizontal position. (Khalid, et al 2006)[7] studied experimentally the local and average heat transfer by mixed convection in an inclined annulus with aradius ratio of 0.555 and uniformaly heated inner cylinder with a length of 1.2m while the outer cylinder is subjected to the ambiant temperature. The investigation covers Reynolds number range from 154 to 845, heat flux varied from 96 W/m^2 to 845 W/m^2 . Results show that the local Nusselt number increases as heat flux and Reynolds number increases and as the angle of inclination moves from the vertical to the horizontal position. (Ihsan and Akeel 2009)[8] performed numerical calculations 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 uniformaly heated inner cylinder and adiabatic outer cylinder. The range 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. (Gada 2009)[9] studied simultaneously developing assisting and opposing laminar mixed convection heat transfer in the entrance region of inclined concentric annuli with radius ratio of 0.555.The investigation covered Reynolds number range ($383 \leq Re \leq 1500$) and Rayligh number range ($1.005\times10^5 \leq Ra \leq 1.52158\times10^5$). Results show that the average Nusselt number increases as the angle of inclination deviates from vertical to horizontal position for both aiding and opposing flow cases.

Main purpose of this study

One of the purposes of this work is to give a correlation able to predict the behavior of heat transfer process for laminar opposing flow of air in the thermally developing region of vertical and inclined annulus with uniformly heated outer tube and adiabatic inner tube where the flow is hydrodynamiclly fully developed at the thermal section by using calming section. In addition, to study the effect of Richardson number, Rayliegh number, Reynolds number, angle of inclination and calming section on the heat transfer process and comparing the results with the available previous works.

Experimental Apparatus

Flow open circuit

The experimental apparatus of the present study is shown schematically in Fig.(1). The air flow rate is circulated around an open loop through the calming and test section using centrifugal fan (B) and regulated using flow control valve (C). The heated air is exhausted to the atmosphere. The test section is mounted on a cast iron frame with a horizontal spindle so that the annulus angle of inclination could be changed. The flow rate is measured by using H_2O manometer (E). The test section (K)is made of cast iron inner tube and aluminum outer tube with 30mm hydraulic diameter and 1200mm long.
There are three Teflon pieces; the first is a bell mouth (G), which is fixed at the begging of entrance section (G), the other Teflon piece (G) lies between calming and test section, the last Teflon piece (G) represents the test section exit. The air passes through the test section, is hydrodynamically fully developed by using another annulus with the same material and dimensions as test section as shown in Fig. (1).

Test section set-up

The aluminum outer tube of test section is heated electrically by using an electrical heater as shown in Fig.(2). It consists of 0.5mm in diameter nickel-chrome wire (L) electrically isolated by ceramic beads, wounded uniformly along the outer tube as a coil with 15mm pitch in order to give uniform wall heat flux. An asbestos rope was used as a 15mm spacer to secure the winding pitch. The outside of the outer tube test section was then thermally insulated by asbestos (Q) and gypson (S) layers, having thicknesses of 60mm and 6.5mm respectively. The cast iron inner tube is insulated from its inside surface by fiber glass with outside diameter of 5.7mm. The outer tube surface temperatures were measured by twenty 0.2mm-asbestos sheath alumel-chromel (type K) thermocouples, fixed along the outer tube surface. All thermocouple wires and heater terminals have been taken out the test section. The inlet bulk air temperature was measured by one thermocouple placed in the chamber, while the outlet bulk air temperature was measure by two thermocouples (N) located in the test section exit as shown in Fig.(2). The local bulk air temperature was calculated by fitting straight line-interpolation between the measured inlet and outlet bulk air temperatures since the constant wall heat flux is
pplied. To evaluate the heat losses through the test section laggining, eight thermocouples were inserted in the lagging as two thermocouples at four stations along the heated section 285mm apart as shown in Fig.(2). By using the average measured temperature and thermal conductivity of the lagging, the heat losses through the lagging can be determined. The heat losses from the ends of test section can be evaluated by inserting two thermocouples in each Teflon piece connecting with the outer tube. By knowing the distance between these thermocouples and the thermal conductivity of the Teflon, the end

losses could be calculated.

Evaluation of measurements

Local convective heat transfer coefficients are obtained from an energy balance applied to sections of the tube, using local outer wall and bulk temperature as follows:

$$h = \frac{q_w}{(T_w)-(T_b)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

T_w in equation (1) is the local outer wall temperature, T_b the bulk temperature, and q_w the uniform heat flux density at the outer heated tube.

The heat transfer coefficients are expressed as local Nusselt number defined in the usual manner:

$$Nu_x = \frac{h_x \cdot D_h}{\lambda} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

with the thermal conductivity λ and the equivalent diameter D_h=30mm.

The physical properties are evaluated at the local average of outer wall and bulk temperatures. Physical properties of air are given in the reference (Incropera and Dewitt 2003)[10].

Average Nusselt number is calculated by:

$$Nu_m = \frac{1}{L}\int_o^L Nu_x\,dx \quad \ldots\ldots\ldots\ldots\ldots\ldots(3)$$

Where x the axial direction.

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. 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 ±3.5 % in heat transfer coefficient (Nusselt number) and minimum error of ± 1.35% in Reynolds number and ± 1.41%, in Rayliegh number.

Checking the assumptions

Being preoccupied with the exactitude of the assumptions that we chose at the beginning, we will start a series of handlings with the aim of checking if we obtain a parabolic velocity profile in the tube for sure and, if such is the case, to determine the vertical x-coordinate to which this type of profile is present. In fact, we want to check if the experimental installation, as we conceived it, does not introduce itself of the disturbances into the tube. Theoretically, to have a hydrodynamically developed flow, the hydrodynamic entry length of the flow is a function of Re for low Reynolds number flow as found by solving the complete set of Navier-Stokes Equation.(Shah and London 1978)[11] gave two equations for the hydrodynamic entry length as follows:

$$\left(\frac{L_{hy}}{D_h}\right) = 0.59 + 0.056\,\mathrm{Re} \quad \ldots\ldots\ldots\ldots\ldots(4)$$

$$\left(\frac{L_{hy}}{D_h}\right) = \frac{0.6}{0.035\,\mathrm{Re}+1} + 0.056\,\mathrm{Re} \quad \ldots..(5)$$

The L_{hy} predicted by Eq.(4) is somewhat higher than L_{hy} observed experimentally, where the definitions of both L_{hy} are the same (a dimensionless duct length required to achieve u_{max} as 0.99 $U_{max,fd}$).So, Eq.(4) is in better agreement with the experimental values

(Shah and London 1978) [11]. But, as we have only a limited annular tube length, we will set a hydrodynamic entry length (L_{hy}/D_h ≈40) such as to deduct the maximum Reynolds number that we can use to carry out the experiments. From Eq.(5), the following Equation can be deduced:

$$0.00196\,\text{Re}^2 + (0.056 - 0.035\frac{L_{hy}}{D_h})\,\text{Re} + (0.6 - \frac{L_{hy}}{D_h}) = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots(6)$$

But $L_{hy}/D_h \approx 40$

∴Re=713.852

Therefore, from a theoretical point of view, we should carry out the experiments at Reynolds number lower than 714, so that the flow may be hydrodynamically developed. But, is not enough for us. To avoid this constraint and to increase the maximum Reynolds number that we can use, a stabilization settling chamber of 300mm length has been added at the beginning of calming section. Later on, to see wether the stabilization chamber helps us to have a hydrodynamically developed flow even for Reynolds number higher than 714, we carried out experiments for the Reynolds number range from 450 to1000.

Results

The variation of the outer wall temperatures along the annulus length may be affected by several variables such as heat flux, Reynolds number (these variables give Richardson number), angle of inclination, and the flow direction (upward or downward). Fig.(3) shows the effect of Richardson number and annulus position on variation of outer tube temperature distribution. As can be shown from this figure that the values of temperature for vertical annulus position $\psi=90^o$ are higher than that for inclined position $\psi=60^o$ for the same Richardson number because the net body force has two components in downward components in downward inclination; one normal to the main flow direction (driving the secondary flow within the cross-section), and the other component acts opposite to the main flow direction. The second component would influence the velocity and velocity and temperature profiles in the heated section and may give rise to flow reversal in the upper part of the cross section leading to reducing the temperature. While in vertical position there is only one component of body force in opposite direction with the main flow giving high level of temperature. It is noticed from this figure also that for the same annulus position, the temperature values increase as Richardson number increases because of the free convection is the dominating factor in the heat transfer process. This figure reveals that the surface temperature increases at the annulus cylinders entrance and attains a maximum point after which the surface temperature begins to decrease. The rate of surface temperature rises at early stage is directly proportional to the wall heat flux. This can be attributed to the increasing of the thermal boundary layer faster due to buoyancy effect as the heat flux increases for the same Reynolds number. The point of maximum temperature on the curve represents actually the starting of thermal boundary layer fully developed. The region before this point is called the entrance of annulus cylinder. The behavior and trend of the local Nusselt number with the dimensionless axial distance (inverse Greatz number) as shown in Fig.(4) are opposites of that for the temperature distribution because the heat transfer coefficients (i.e, $Nu_x=h_xD_h/\lambda$) is in reverse proportional with the temperature ($h_x=q/T_x-T_b$). In the other words, the heat transfer process in an inclined position ($\psi=60^o$) is better than that in a vertical position ($\psi=90^o$).Figs.(5&6) show the variation of average Nusselt number with the dimensionless axial distance (Gz^{-1}) for difference values of Richardson number and for $\psi=60^o$ & 90^o, respectively. It is obvious that the values of Nu_m increase as Richardson number decreases because of the forced convection domination in the heat transfer process and a little effect of buoyancy force for high Re. Figs (7&8) show the variation of Nu_m with Rayligh number for various values of Richardson number and for $\psi=60^o$ &90^o, respectively. It can be seen from these two figures that the values of Nu_m increase as Rayliegh number increase for same Richardson number because the dominating factor in the heat transfer process is the natural convection which is stronger as the heat flux increases (i.e, Rayliegh number increases). The comparison between the presnt work and (Kays and Grawford) work is shown in Fig.(9) which shows the variation of the local Nusselt number with (Gz^{-1}). As it can be shown, the values of local Nusselt number for simultaneously developing forced convection flow (SDF) upstream are higher than that in the temperature developing forced convection flow (the work of Kays and Grawford) then they are in close to each other and equal down stream because the flow in (TDF) becomes hydrodynamical and thermally fully developed downstream and identical to that in (SDF). The values of Nu_x for the present work lie between them upstream and are approximately higher than them downstream with the same trend. Fig(10) shows the variation of value of $[Nu_m(60^o)/ Nu_m(90^o)]$ with the values of $[Ra^{1/3}/Re^{1/2}]$ and compared the obtained trend with that in (Bohne and Obemeier)[12] work to give the same behavior but relatively lower because the low range of Gr in the present work($2000\leq Ra\leq 10000$) compared with ($10^5\leq Ga\leq 10^8$) that taken by (Bohne and Obemeier)[12]. The values of the Nu_m are correlated in empirical equation for each position ($\psi=60^o$ & $\psi=90^o$) and are plotted in Figs.(11&12) respectively, in form of $\log Nu_m$ against log(Ra/Re) for the range of Re from 450 to 1000 and Ra from 2000 to 10000. Both can by represented by the formula:

$$Nu_m=c(Ra/Re)^m \quad \text{……………..…...(7)}$$

All the points as can be seen are represented by straight lines of the following equations:

$$Nu_m=1.543(Ra/Re)^{-0.032} \text{ for } \psi=60^o \quad \text{...(8)}$$

$$Nu_m=1.321(Ra/Re)^{-0.043} \text{ for } \psi=90^o \quad \text{...(9)}$$

The values of (m) represent the slope of each carve.

Conclusions

1. The heat transfer process for mixed convection opposing flow in $\psi=60^o$ (inclined) is better than in that $\psi=90^o$ (vertical).
2. The values of average Nusslt number increase as Richardson number decreases because of the forced convection domination in the heat transfer process and as Ra increases for the same Ri because of the natural convection domination in the heat transfer process.
3. The behavior and trend of the local Nusselt number with the dimensionless axial distance (Gz^{-1}) are relatively similar to that of (Kays and Grawford) work.
4. The behavior and trend of $[Nu_m(60^o)/ Nu_m(90^o)]$ against $[Ra^{1/3}/Re^{1/2}]$ are the same as that in the work of(Bohne and Obemeier) but with lower level because the high difference in Rayliegh number between them.

5. An empirical equations have been deduced for the average Nusselt number as a function of Ra/Re for both $\psi=60^o$ (inclined) and $\psi=90^o$ (vertical).

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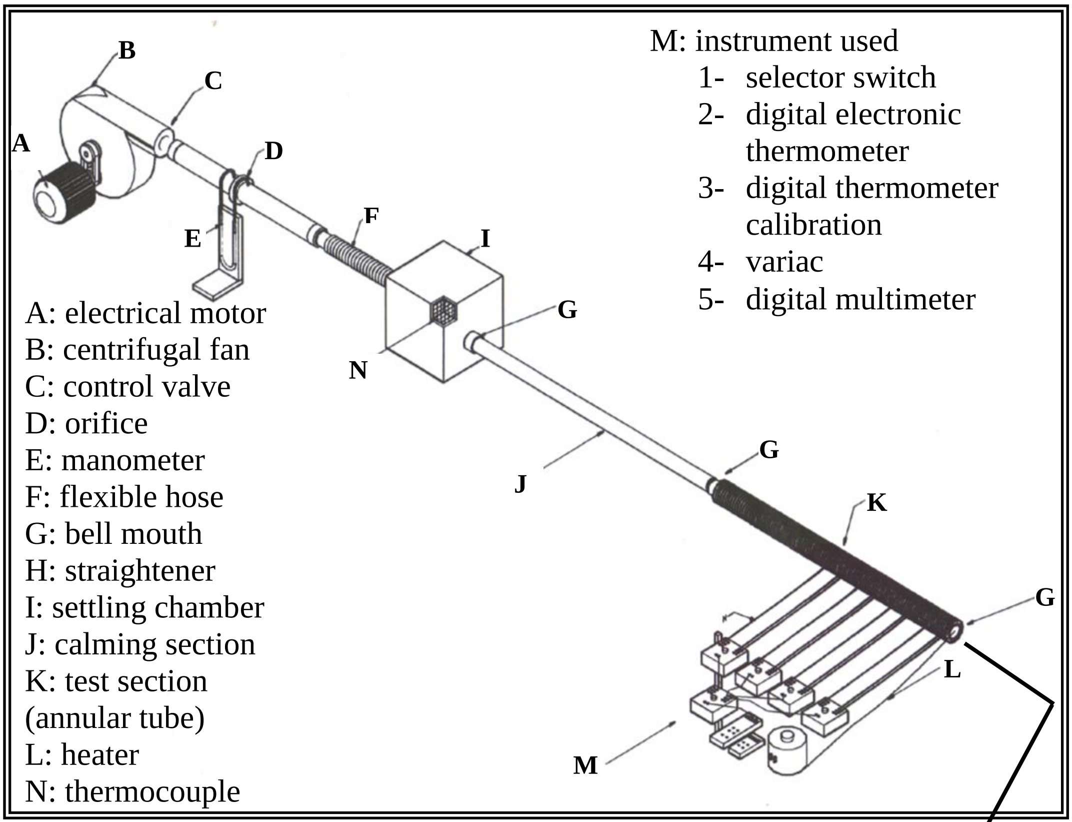

Fig.(1) Diagram of Experimental Arrangement

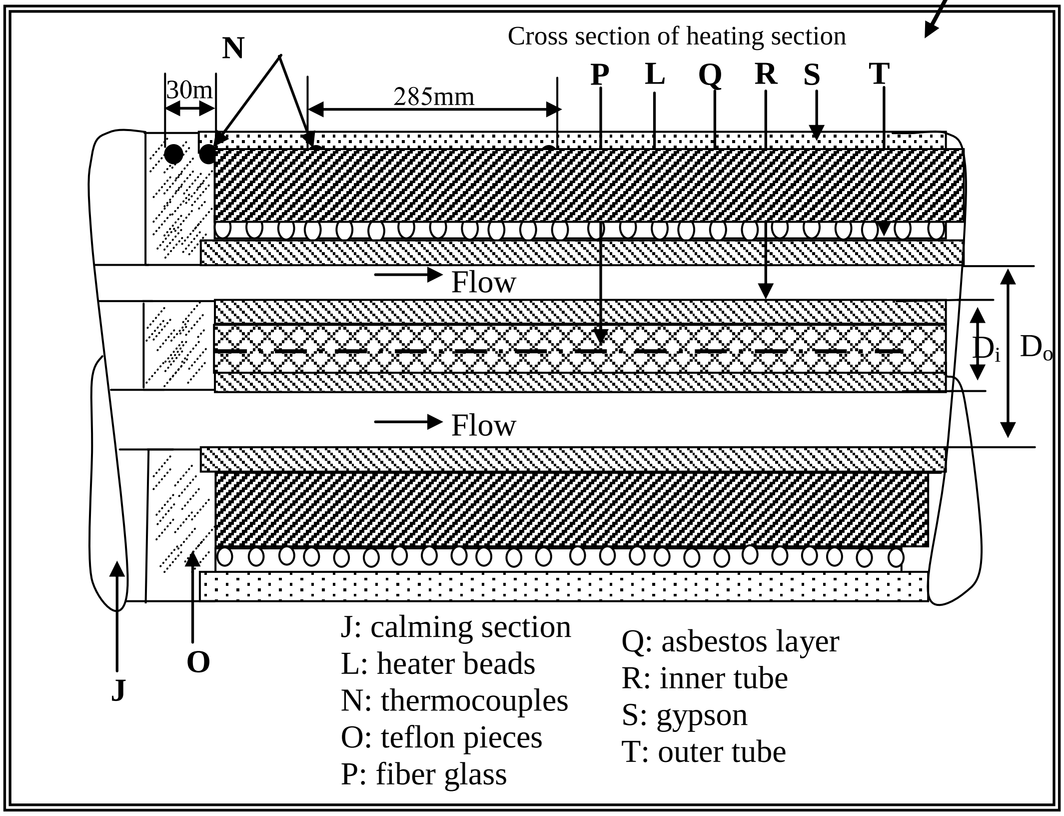

Fig. (2) Heating Element Arrangement

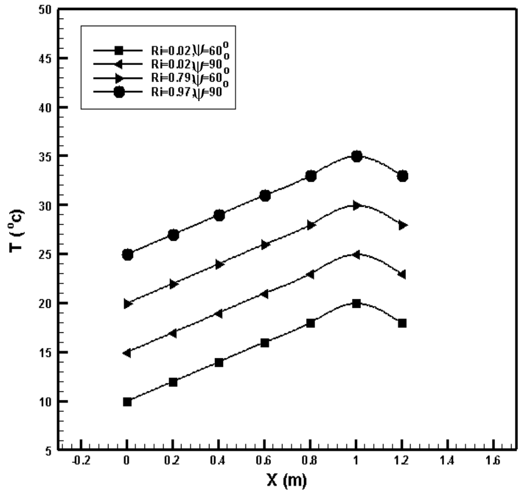

Fig.(3): Variation of surface temperature with axial distance along the outer tube showing the effect of annulus orientation

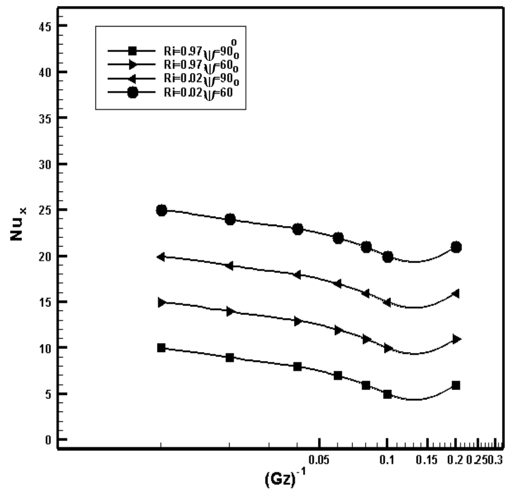

Fig.(4): Variation of local Nusselt number versus dimensionless axial distance for inclined and vertical

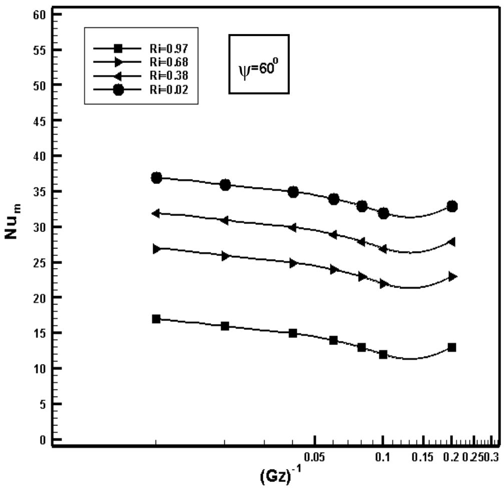

Fig.(5): Variation of average Nusselt number versus dimensionless axial distance for various Ri, and ψ=60°

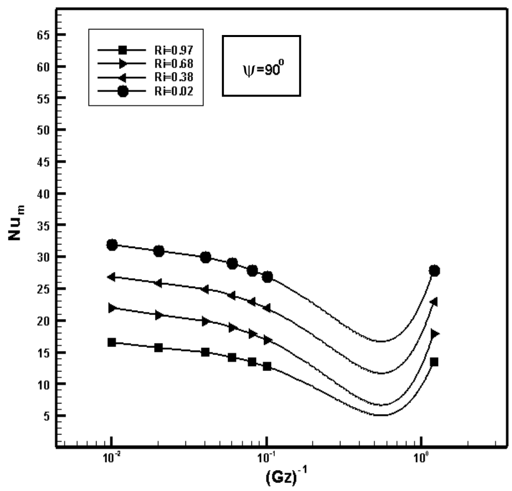

Fig.(6): Variation of average Nusselt number versus dimensionless axial distance for various Ri, and ψ=90°

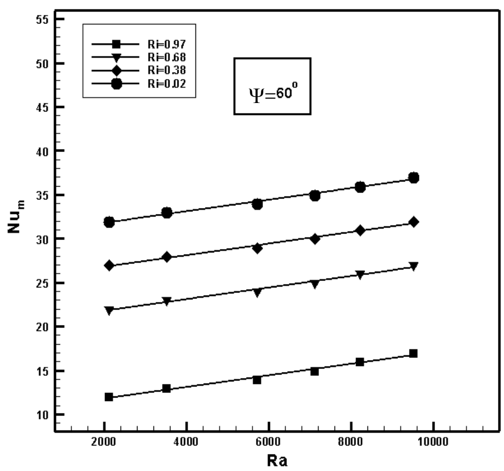

Fig.(7): Average Nusselt number versus Rayliegh number for different value of Ri, and ψ=60°

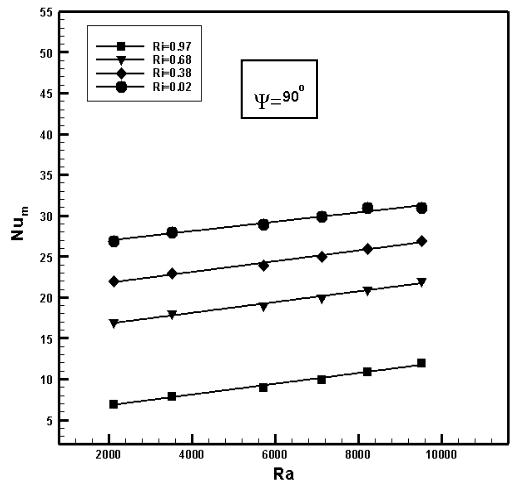

Fig.(8): Average Nusselt number versus Rayliegh number for different value of Ri, and ψ=90°

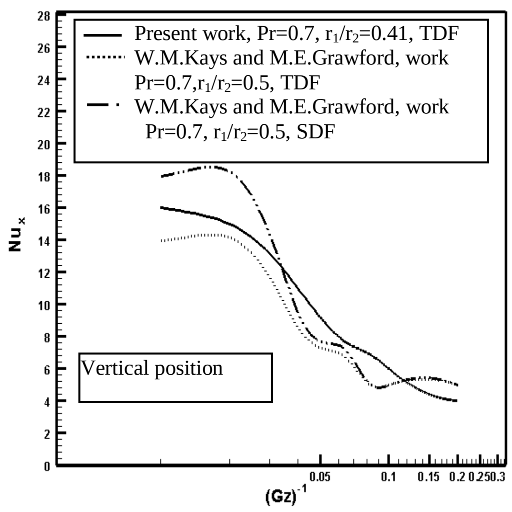

Fig.(9): Comparison of the present work with previous works showing the local Nusselt number versus dimensionless axial distance.

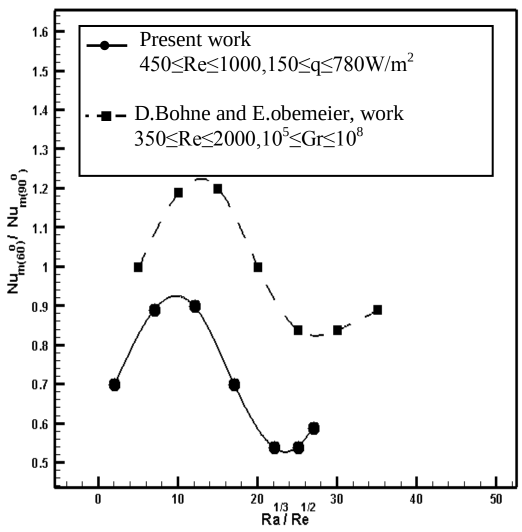

Fig.(10): Comparison the present work with Bohne and Qbemeier work showing [$Nu_m(60^o)$/ $Nu_m(90^o)$] versus [$Ra^{1/3}/Re^{1/2}$].

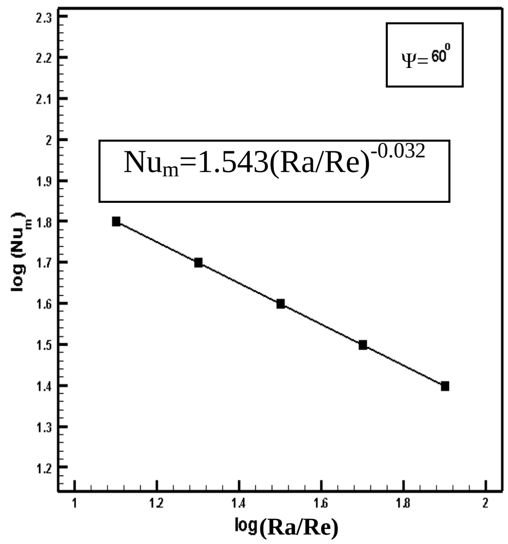

Fig.(11): Logarthmic average Nusselt number versus Logarithmic (Ra/Re)for ψ=60^o.

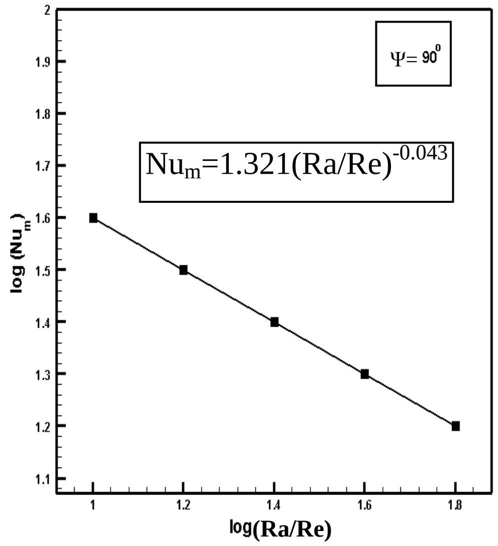

Fig.(12): Logarthmic average Nusselt number versus Logarithmic (Ra/Re)for ψ=90^o.