EXPERIMENTAL SYUDY OF MIXED CONVECTION HEAT TRANSFER FOR A THERMALLY DEVELOPING AIR FLOW IN AN INCLINED ANNULUS

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ABSTRACT

The investigation-included experiments were conducted to investigate the local and average mixed convection heat transfer coefficient for thermally developing and fully developed hydrodynamically laminar airflow in an inclined annulus. The experimental setup, using an aluminum annulus having a radius ratio of (0.41) and the inner tube with a heated length of (0.85m) with constant wall heat flux boundary conditions. The investigation covered Reynolds number range from (332 to 1128) and heat flux varied from (82w/m^2 to 545w/m^2). The air is developed hydrodynamically by using the entrance section annulus (calming section) having the same inner and outer diameter as a test section but with variable lengths. The entrance sections included a long calming section with a length of 90 cm (L/D_h =20.93) and short calming section with lengths of 60 cm (L/D_h =13.95). The results obtained represent the temperature distribution along the inner tube surface, the local and average Nusselt number distributions with the dimensionless axial distance (Z^+). For all annulus orientations and all entrance sections, the results show an increase in the Nusselt number values as the heat flux increases and as the angle of the inclination moves from the vertical to the horizontal position. A comparison was made between the experimental results and the available previous work of Mataira, D. and Subba Raju, K. (1975), for the local Nusselt number distribution with the dimensionless axial distance (Z^+). The results obtained showed a good agreement and has the same trend that obtained in the previous work.

الخلاصة

يتضمن هذا البحث إجراء تجارب عملية لبحث معامل انتقال الحرارة الموقعي و المعدل بواسطة الحمل المختلط لجريان الهواء الطباقي المتطور حرارياً وتام التطور هيدروديناميكياً في أنبوب حلقي مائل. يتكون الجهاز العملي من أنبوب حلقي مائل نسبة نصف القطر له (0.41) و أنبوب داخلي مسخن و بطول (0.85m) تحت الشرط الحدي ثبوت الفيض الحراري. يتراوح رقم رينولدز في هذا البحث من 332 إلى 1128 ، وكذلك يتغير الفيض الحراري من (82w/m^2 إلى 545w/m^2). يكون الهواء تام التطور باستعمال أنابيب مدخل حلقية(أنابيب تهدئة) لها نفس قطر الأنبوب الداخلي والخارجي لجزء الاختبار واكن بأطوال مختلفة. تتضمن أنابيب المدخل أنبوب طويل بطول (90cm)(L/D_h=20.93) وكذلك أنبوب مدخل قصير بطول(60cm)(L/D_h=13.95). النتائج المستخرجة وضحت توزيع درجة الحرارة على سطح الأنبوب الداخلي

، وكذلك توزيع رقم نسلت الموقعي و المعدل مع المسافة المحورية اللابعدية(Z^{+}). لكل زوايا ميل الأنبـــوب الحلقي و لكلا أنبوبي المدخل المستخدمين ، بينت النتائج إن قيم رقم نسلت تزداد كلما ازداد الفيض الحراري وكذلك يزداد رقم نسلت كلما زاوية ميل الأنبوب الحلقي تتغير من الموقع العمودي إلى الموقع الأفقي. أجريت مقارنة بين الدراسة العملية والعمل السابق للأدبيات المتـــوفرة لعمـــل Mataira, D. and Subba Raju, K. (1975) لتوزيع رقم نسلت الموقعي مع المسافة المحورية اللابعدية (Z^{+}) حيث أظهرت النتائج موافقة جيدة و تم الحصول على نفس الشكل الذي حصل عليه الباحث.

KEY WORDS
Experimental mixed convection, heat transfer, thermally developing air flow, inclined annulus.

INTRODUCTION

In convective heat transfer problems, the flow usually is classified as forced convection flow, in which the flow is caused by external forces such as pumps or fans, or free convection flow in which the flow is created by the fluid density variations due to the wall to fluid temperature difference under the influence of body forces Shai, I. and Barnea, Y. (1986). In most physical applications, the buoyancy forces have negligible effects because they have usually a considerable smaller magnitude than those accompanying the forced flow. But in certain practical situations, however, the magnitude of the two forces may be of the same order and both may then be expected to influence the flow significantly. Therefore, when the free convection superimposed on the forced convection heat transfer process gives rise to a new field of study called mixed (combined) convection Shai, I. and Barnea, Y. (1986). Therefore, the flow can be divided as pure forced convection, combined (forced and free) convection or pure free convection depending upon the relative magnitude of these two forces. A Practical instance of this (mixed convection) occurs in loss of cooling accident (LOCA) events of nuclear reactors where emergency conditions result in a low forced flow superimposed on the natural convection in the reactor core. Frequently, the prediction of the heat transfer by forced convection alone without considering the secondary flow can cause considerable errors. The interaction of the free and forced convection currents can be very complex because it depends not only on all the parameters determining both forced and free convection relative to one another but sometimes also on a large number of interacting parameters including the relative direction of the natural and forced convection to each other, the geometry of the arrangement, the velocity profile at annulus entrance and the heating surface boundary conditions Cony, J.E.R. (1975). Laminar flow combined convection heat transfer in annulus is encountered in a wide variety because of importance in industrial engineering applications. The following examples can be cited: heating or cooling of heat exchangers for gas flows Cony, J.E.R. (1975), cooling of electronic equipment Chen, Y.C. (1998), the cooling core of nuclear reactors Hashimoto, K. (1986). The full understanding of the prevailing velocity and temperature fields, as well as, the pressure drop and heat transfer coefficient, are necessary for the proper design. In a High Temperature Gas Cooled Reactor (HTGR), the reactor core is designed to a chieve a high outlet temperature. Helium gas coolant flows throw annular channels between fuel rods and inner walls of holes hexagonal graphite blocks. Fuel element wall temperature is very high because of the relatively inferior heat transfer characteristics of a gas. It is therefore necessary to investigate the effect of large temperature difference between wall and fluid on heat transfer and flow behavior in annular passage. In normal conditions, helium gas flows down ward in the reactor core .In abnormal conditions such as emergency cooling, the direction of flow might be reversed owing to buoyant force. For the "HTGR" core safety part of view, it is, therefore, of practical importance to investigate combine forced-free convection heat transfer in vertical annulus reactors Hashimoto, K. (1986).

Ciampi, M. et al (1986) carried out experiments to study the superposed free and forced convection for water in annulus with constant inner wall heat flux. For Reynold number ranged from (200 to 2200), the laminar flow Rayleigh number ranged from ($4.4*10^{6}$ to $1.6*10^{9}$). An equation was suggested to fit the experimental data for laminar and turbulent flow regions:

$$\overline{Nu} = 0.115 * Ra^{0.4} (D_i / D_h)^{0.8} \quad -----(1)$$

$$\overline{Nu} = 0.0202 * Ra^{0.34} * Re^{0.08} * (D_i / D_h)^{0.51} \quad -----(2)$$

eq. (1) correlated 80% While eq. (2.2) correlated 90%.

Zysina et al (1973) performed experiments to study laminar and transition air flow in an annulus. This work use air as working fluid. The Reynolds number ranged over the limits of ($10^{3} \leq Re \leq 10^{5}$), while the Grashof numbers range was ($3*10^{3} \leq Gr. \leq 3*10^{6}$), the ratio of Δt (the temperature difference between the opposite walls) to the temperature difference (δt) between the wall and flow was in the range of ($0.2 \leq \frac{\delta t}{\Delta t} \leq 10$). Variation of (Nu) number was depicted.

Bohne and Obermeir (1986) studied upward and downward fluid flow in an internally heated concentric annulus in a vertical, inclined and horizontal position have been by. The annulus equivalent diameter was 16 mm and a heated core of (1 m) length, water and water-glycol mixture are used as working medium. Experimental parameters covered were (Re) number from (350 to 20000), (Gr) number varied from (10^{5} to 10^{8}), and (Pr) varied from (2.3 to 75).The results show an increases of Nusselt number in the laminar flow range as the annulus inclination deviates from vertical to horizontal position .The measured heat transfer data also presented in from of correlation equation.

Ogunba and Barrowac (1979) used a Hydrogen bubble cluster to study visually the laminar water flow in an internally heated annulus. The annulus parameters were radius ratio equal to (0.25), heat flux varied from (0 W/m^2 to 2340 W/m^2) and Reynolds number equal to (445). The development of the axial velocity profile was shown and the local heat transfer coefficient has obtained in a length of (25) equivalent diameter.

Lundberge et al (1962) performed experiments to study hydrodynamically developed laminar upward air flow in an annulus. Four annulus radius ratio were used (0.132,0.25,0.375 and 0.5). The thermal condition of the inner wall was isothermal and the outer wall was adiabatic, while Reynolds number varied from (800 to 1500). Variation of the inner surface temperature and the heat transfer coefficient along the annulus were depicted. Also they presented theoretically a complete analysis for the thermal problem in hydrodynamically fully developed upward airflow in a vertical annulus. This includes evaluation of the four fundamental solutions in the thermal energy regions by solution of the eigen value problem.

Huetz and Petit (1974) performed experiments to study a mixture of sodium and potassium (Pr=0.2), flow in a horizontal concentric annulus of radius ratio (0.5). The thermal condition of the inner wall was constant heat flux by using thermoelectric effect of junction liquid metal steel at the wall while the thermal condition of the outer wall was adiabatic. Experiments were carried out in case of low axial velocity and ($Gr > 10^{7}$) in an annulus having a length of 5 m. Results show three zones of Nusselt number variation as a function of Peclet number.

Hanzawa et al (1986) studied upward gas flow in an annulus of radius ratio range from (0.39 to 0.63). Apart of the inner tube was isothermally heated while the outer tube was kept adiabatic. The study covered Grashof number range from ($1.5*10^{5}$ to $2.6*10^{8}$), (Re) number from (20 to 1000) and the length of heating section (D_b/L) range from (0.34 to 1.4). Results of temperature gradient in radial direction and the heat transfer coefficient were depicted a long the heating parts.

Sherwin and Wallis (1972) conducted experiment on water flowing in a vertical annulus of radius ratio (3) with uniform heated inner surface. Experimental result confirm that the Nusselt number of mixed convection increase. Flow visualization by dye injection was used to determine the onset of flow instability. It was found that the instabilities were associated with large radial components of velocity. It was found that unstable flow occurs when the parameter (Gr_q /Re=209) "critical value" gradually reduced with developing flow until reaching a value appropriate to fully developed flow. Maitra, D and Subba (1975) conducted an experimental study for water upflow in an annulus having (L/D_h =77.74) under constant inner wall heat flux. The range of Re number from (1200 down to 200) and for Raleigh number up to ($2.5*10^5$) were obtain for water flowing in an annuls. Local Nusselt number were plotted according to the $(Z/D_h * Pe)$. The variation of (Nu) with (Ra) also plotted.

The purpose of the present work is to determine experimentally the effect of annulus inclination angle and the effect of buoyancy forces on the heat transfer process under mixed convection situation for hydrodynamically fully developed and thermally developing laminar airflow under constant inner wall heat flux boundary condition in an inclined annulus.

EXPERIMENTAL APPARATUS

Open Air Circuit

An open-air circuit was used including small centrifugal fan (B), rotameter(C), settling chamber (D), test section with calming section, and flexible hoses (E, F) as shown in **Fig. (1).**

The centrifugal fan driven electrically via-fine control variable resistance so that its power can be regulated accurately. The air was drawn by the fan in to the settling chamber and test section through the entrance section and then enters the rotameter through flexible hose (E) and then the air leaves the rotameter to the centrifugal fan through another flexible hose (F). Then the heated air was exhausted to the atmosphere. The settling chamber was carefully designed to reduce the flow fluctuations and to get a uniform flow at the entrance section by using flow straightener (G).

The air then passes through calming section and test section. Asymmetric flow and a uniform velocity profile produce by a well designed Teflon bell mouth (H) which is fitted at the annulus outer copper tube "calming section" (I) and bolted inside the settling chamber (D). For all entrance length tests, the air passes through the test section, is fully developed hydrodynamically by using calming section "the same outer copper tube but with same diameter and inner aluminum pipe having same diameter as inner pipe but with variable length as entrance sections". The outer tube for calming and test section is the same but the inner tube calming section is connected with the test section by a (2 mm) thickness, (300 mm) outside diameter and (250 mm) long Teflon connection piece (N) bored with the same inside diameter of the test section and entrance section.

The test section consisted of 2mm-wall thickness, (300 mm) outside diameter and (900mm) long aluminum tube (K) located centrally in 2 mm thickness, (730 mm) inside diameter and (900 mm) long copper tube, the test section exit fitted with Teflon ice (M). An O ring (P) is used to hold a support the aluminum tube (K) with the Teflon piece (N) centrally inside the settling chamber by adjustable screws (Q). The Teflon was chosen because its of low thermal conductivity in order to reduce the heat loss from the aluminum tube ends. The inlet bulk air temperature was measured by one thermocouple (G) placed in the beginning of the entrance section (calming section), while the outlet bulk air temperature was measured by three thermocouples (Z) 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 as shown in **Fig. (1)**.

Heating System

The inner tube is heated electrically by using an electrical heater as shown in **Fig. (2)**, section (A-A). It consists of a (0.5) mm in diameter nickel-chrome wire (R) electrically isolated by ceramic beads, wound as coil and surrounded by a Pyrex-glass tube (S). The latter is covered by a (2-mm) thickness asbestos layer (T), and the space between the asbestos and the inner tube wall is filled with a fine grade sand (U) to avoid heat convection in it and to smooth out any irregularities in the heat flux. The hole apparatus is designed with a view to obtain a good concentricity of the core tube and the containing tube.

The inner tube surface temperatures were measured by twenty-five (0.2) mm-glass-covered alumel-chromel (type K) thermocouples, fixed along the inner tube. The measuring junctions (which were made by fusing the ends of the wires together by means of an electric spark in an atmosphere free from oxygen) embed in grooves in the wall normal to the annulus axis as shown in **Fig. (2)**, section (A-A). The thermocouples were fixed by drilling twenty-five holes (V) of (1.6 mm) diameter and approximately (2mm) deep and along the inner tube wall while the ends of the holes chamfered by a (2mm) drill then the measuring junctions were secured permanently in the holes by sufficient amount of high temperature application Defcon adhesive (X). Another seventh thermocouples were used to measure the inner surface temperature of the annulus outer tube (I). Thermocouples position at the outer surface of copper tube were located and then a (1mm) deep pits were drilled in which the thermocouples were fixed by devcon adhesive. To evaluate the heat losses from the ends of the test 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.

Experimental Procedure

The procedure employed to carryout a certain experiment was as follows:

1- The required calming section length was fitted with the test section.
2- The annulus orientation was adjusted as required.
3- The centrifugal fan was then switched on to draw the air through the test section.
4- The electrical heater was switched on and the heater-input power then adjusted to give the required heat flux.
5- 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 multimeter 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 and for same Reynolds number.

The subsequent runs for other Reynolds number and annulus orientation ranges were conducted in the same procedure.

6- During each test run, the following readings were recorded:

a- The length of calming section in (cm).
b- The annulus orientation angle in degree.
c- The reading of the rotameter (air flow rate) in (m^3/hr).
d- The heater current in ampere.
e- The heater voltage in volts.
f- The readings of all thermocouples in mV.

Data Analysis Method

The following simplified steps were used to analyze the heat transfer process for the airflow in an annulus when its inner tube surface was subjected to uniform wall heat flux boundary condition.

The total input power supplied to inner tube can be calculated:

$$Q_t = V''*I \qquad ---(3)$$

The convection and radiation heat transferred from the inner tube surface: -

$$Q_{cr} = Q_t - Q_{cond.} \qquad ---(4)$$

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

Th convection heat flux can be represented by:

$$q_{cr} = \frac{Q_{cr}}{A_1} \qquad ---(5)$$

Where $A_1 = 2\pi * r_1 * L$

The convection heat flux, which is used to calculate the local and average heat transfer coefficient is obtained after deduce the radiation heat flux from q_{cr} value.

$$q_{r_x} = \frac{\sigma}{1/\varepsilon_1 + (1/\varepsilon_2 - 1) * N} * \left[(t_{sx} + 237)^4 - (\bar{t}_2 + 273)^4 \right] \qquad ---(6)$$

Hence the convection heat flux at any position is:

$$q_{conv.} = q_{cr} - q_r \qquad ---(7)$$

The local heat transfer coefficient can obtain as:

$$h_x = \frac{q_{conv.}}{t_{sx} - t_{ax}} \qquad ---(8)$$

Where: t_{sx}= local inner tube surface temperature.

t_{ax}= local bulk air temperature which was calculated with the effect of radiation was neglected.

$\bar{t}_2$ = Average outer tube temperature.

All the air properties were evaluated at the mean film temperature (Holman, J.P. (1992))

$$t_{fx} = \frac{t_{sx} + t_{ax}}{2} \qquad ---(9)$$

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

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

$$Nu_x = \frac{h_x . D_h}{k_x} \qquad ---(10)$$

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

$$\bar{t}_s = \frac{1}{L} \int_{x=0}^{x=L} t_{sx}\, dx \qquad ---(11)$$

$$\bar{t}_a = \frac{1}{L} \int_{x=0}^{x=L} t_{bx}\, dx \qquad ---(12)$$

$$\bar{t}_f = \frac{\bar{t}_s + \bar{t}_a}{2} \qquad ---(13)$$

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

$$\overline{Nu} = \frac{qD_h}{k(\bar{t}_s - \bar{t}_a)} \qquad ---(14)$$

$$\overline{Gr} = \frac{g\beta D_h^{\,3}(\overline{t_s} - \overline{t_a})}{\nu^2} \quad ---(15)$$

$$\overline{Ra} = \overline{Gr} * Pr \quad ---(16)$$

Where: $A = \pi(r_2^2 - r_1^2)$, $u_i = \dfrac{\dot{m}}{A}$, $\beta = \dfrac{1}{(273 + t_f)}$ All the air physical properties (ρ, μ, ν and κ) were evaluated at the average mean film temperature ($\overline{t_f}$).

RESULTS AND DISCUSSION

A total of (120) test runs were conducted to cover five annulus orientations, horizontal ($\theta = 0^o$), inclined ($\theta = 30^o$, & $\theta = 60^o$) and vertical ($\theta = 90^o$) with the annulus having tow different length (calming section annulus) of 60 cm ($L/D_h = 13.95$) and 90 cm ($L/D_h = 20.93$). The range of heat flux used varied from (82 W/m^2 to 545 W/m^2) and Reynolds number varied from (332 to 1128).

SURFACE TEMPERATURE

Horizontal Position

Generally, the variation of the surface temperature along the annulus inner tube may be affected by many variables such as heat flux, flow velocity, the annulus inclination. The temperature distribution in the horizontal annulus for selected runs is plotted in **Figs. (3&4)**. The general shape for all curves in tow figures is as follows at test section entrance the inner tube surface temperature gradually increases up to a specific axial position the temperature reaches a maximum value after which the inner tube surface temperature decreases at annulus exit.

Fig. (3) show the effect of (Re) number variation on the inner tube surface temperature for heat flux (241W/m^2). It is obvious that the increasing of (Re) number reduces the inner tube surface temperature, as the heat flux is kept constant. It is necessary to mention that as heat flux increases the inner tube surface temperature increases because the free convection is the dominating factor in the heat transfer process.

Fig. (4) show the variation of the surface temperature along the annulus inner tube for different heat flux, for Re=598 and for calming section length equal to 60 cm ($L/D_h = 13.95$). This figure reveals that the inner tube surface temperature increases at annulus entrance reaching a maximum value after which the inner tube surface temperature decreases earlier for higher heat flux. This can be attributed to the developing of the thermal boundary layer faster due to buoyancy effect as the heat flux increases for the same (Re) number.

The inner tube surface temperature variation for the second calming section with length equal to 90 cm ($L/D_h = 20.93$) for Re=598, show a similar trend as mentioned above for ($L/D_h = 13.95$).

Inclined Position

The variation of the inner tube surface temperature along the axial distance for inclined annulus ($\theta = 30^o$, $\theta = 45^o$ & $\theta = 60^o$) is plotted for selected runs and have the same general shape similar to that mentioned for horizontal annulus.

Vertical position

The variation of the inner tube surface temperature along the axial distance for inclined annulus ($\theta = 90^o$) is plotted for selected runs and have the same general shape similar to that mentioned for horizontal annulus and inclined annulus.

The effect of heat flux and Reynolds number variation on the inner tube surface temperature distribution is the same as that mentioned for the horizontal and inclined positions. But it necessary to mention that in vertical annulus the direction of the buoyancy forces in the same direction as that of the main forced flow creating a case of one dimensional flow and as the heat flux increases the inner tube surface temperature increases because the free convection is the dominating factor in the heat transfer process. The extreme of mixed convection depends on the magnitude of Reynolds number and the heat flux for same annulus orientation. When the heat flux and Reynolds number are kept constant, the effect of buoyancy forces in a horizontal annulus is larger than that in other annulus orientations. So that, it can be expected at the same conditions of flow rate and input heat flux, the inner tube surface temperature distribution along the annulus decreases as the annulus orientation changes from vertical to horizontal position. The inner tube surface temperature variation, for ($\theta = 90^{\circ}$), for the second calming section with (L/D_h=20.93), shows a similar trend as mentioned above for the (L/D_h=13.95).

Inclination Angle Effect on the Temperature Variation

The effect of annulus inclination on the inner tube surface temperature distribution for Re=332 and q=241 W/m^2, is shown in **Fig. (5)** .The figure show that the inner tube surface temperature distribution for a vertical position is higher than that for a horizontal position. This behavior can be attributed to that when heat is transferred through the wall of a horizontal annulus, the density of the air near the wall decreases. The body forces acting due to this density change cause the hotter air to climb upward along the annulus inner tube circumference while the fluid of higher density (lower temperature) moves downward along the annular outer tube inner surface. So, a longitudinal vortices is created which is superimposed on the main forced flow. This secondary flow assists the main flow in removing heat from the inner tube surface and improving the heat transfer process because of the large velocity near the inner tube wall, shows in Kaviany, M (1986).
Also, in vertical annulus, the velocities due to buoyancy flow and forced flow are in the same direction (assistant flow), thus the rotational symmetry is retained. From the previous explanation, it is evident that, at low Reynolds number the free convection is the dominating factor in the heat transfer process, as a result, the surface temperature of the inner tube reduces as the annulus inclination moves from vertical to horizontal position.
Fig. (6) shows the influence of the annulus orientation on the inner tube surface temperature for Re=1128, q=545 W/m^2. It was observed that for all angles of inclination at the specific axial distance (x=0.35 m) near the annulus entrance the inner tube surface temperature for the vertical position is lower than that in inclined and horizontal positions. Then beyond this axial distance a reverse trend take places and the inner tube surface temperature for the horizontal position becomes lower than that in inclined and vertical positions. This behavior can be explained that as the flow at annulus entrance, the effect of free convection is small and forced convection is dominant in this region causes the inner tube surface temperature in vertical position less than in the inclined and horizontal positions which is similar the trend shown in **Fig. (3)**. But after a certain axial distance a significant reduction in the inner tube surface temperature as the annulus orientation moves from the vertical to the horizontal position due to the effect of free convection which begins to dominate the flow and reduces the inner tube surface temperature (Zysina, L.M. (1973), Remash (1986)).

Annulus Inclination Effect on (Nu_x)

The results for Re=332, q=241 W/m^2 is shown in **Fig.(7)** and this figure indicates that for all (Z^+) values, the (Nu_x) value for horizontal position is higher than that for inclined and vertical positions. As explained that the free convection is dominant, for horizontal annulus creates along the annulus a longitudinal vortex with the vortex intensity reduces as annulus orientation changes from horizontal to vertical position. High vortices intensity improves the heat transfer process in the horizontal annulus (Ogunba, V. (1979), Bohne, D. (1986)).

Fig.(8) shows the (Nu_x) results for Re=1128 and high heat flux q=545 W/m^2 which indicates that the (Nu_x) values for horizontal annulus is slightly higher than that of inclined and vertical positions. This behavior continues until a certain Z^+ value (0.029) where beyond it the (Nu_x) value for horizontal annulus is higher than that for other annulus positions. This is related to the small buoyancy effect at annulus entrance and a forced convection is being the dominating factor in the heat transfer process. But in the annulus downstream the secondary flow becomes more effective which improves the heat transfer. The heat transfer appears to be higher for the horizontal position and its effect is reduced as annulus orientation moves toward the vertical position.

Comparison with Previous Work

The work of Mataira, D. and Subba Raju, K. (1975) was carried out with inner tube constant heat flux vertical annulus of radius ratio equal to (0.377). Water was used as a working fluid. The variation of the Nusselt number in the presented experimental work has the same trend as that obtained by Mataira, D. and Subba Raju, K. as shown in **Fig. (9)**.

CONCLUSIONS

The following general conclusions can be drawn from the experimental work conducted in the present investigation as follows:

1- The variation of the surface temperature along the heated inner tube annulus for all annulus orientations has the same general shape. This variation is affected by many factors summarized in the following points:

a- The inner tube surface temperature increases as the heat flux increases, for the same (Re) number and same annulus orientation.

b- The inner tube surface temperatures for low (Re) number is higher than for high (Re) number, for the same heat flux and same annulus orientation, because of the free convection domination.

c- The inner tube surface temperatures for low Re no. and high heat flux decrease as the annulus orientation moves from vertical position to horizontal, and that revels the free convection domination. But the inner tube surface temperature for high Re no. and low heat flux increases as the annulus orientation move from vertical position to horizontal position when forced convection is dominant and that reach forced convection domination .

2- The variation of (Nu_x) with (Z^+), for all annulus orientations has the same trend. This variation is affected by many factors summarized as follows:

a- For same (Re) number the (Nu_x) value increases with the increase of the heat flux.

b- For the same heat flux, low (Re) number, for all annulus orientations the (Nu_x) value moves toward the right of the (Nu_x) predicted for (TPFC), because of natural convection is dominant.

c- For the same heat flux, high (Re) number, for all annulus orientations the (Nu_x) value moves toward the left of the (Nu_x) predicted for (TPFC), because of forced convection is dominant.

d- For the same heat flux and low (Re) number, the (Nu_x) value increases as the annulus angle of inclination moves toward horizontal position. The lower value of (Nu_x) occurs in the vertical annulus and the higher value occurs in the horizontal annulus when the free convection is the dominating factor on the heat transfer process.

e- For the same heat flux and high (Re) number, the (Nu_x) value increases as the annulus angle of inclination moves toward vertical position. The lower value of (Nu_x) occurs in the horizontal annulus and the higher value occurs in the vertical annulus when the forced convection is the dominating factor on the heat transfer process.

3- The free convection effects tended to decrease the heat transfer at low (Re) number and to increase the heat transfer for high (Re) number at constant heat flux.

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NOMENCLATURE

The symbols used have the following meanings, unless otherwise stated in the text.

Symbol		Description	Unit
A	=	Annulus area	m^2
A_1	=	Outside surface area of the inner tube	m^2
C_p	=	Specific heat at constant pressure	J/kg.°C
D_h	=	Hydraulic diameter	m
g	=	Gravitational acceleration	m/s^2
h	=	Heat transfer coefficient	$W/m^2.°C$
I	=	Heater current	Amp.
k	=	Thermal conductivity	W/m.°C
L	=	Annulus length	m
$Q_{cond.}$	=	Conduction heat loss	W
$Q_{cr.}$	=	Heat loss by convection and radiation	W
$q_{cr.}$	=	Convection and radiation heat flux	W/m^2
$q_{r.}$	=	radiation heat flux	W/m^2
Q_t	=	Total heat input	W
R	=	Radial coordinate	m
r_1	=	Outer radius of inner tube	m
r_2	=	Inner radius of outer tube	m
R	=	Dimensionless radial coordinate $= r-r_1/r_1-r_2$	---------
t	=	Temperature at any point	°C
t_m	=	Mixing cup temperature over any cross section	°C
V"	=	Heater voltage	Volt
x	=	Axial coordinate	m
X	=	Dimensionless axial coordinate $= x/r_2$	---------

Greek Symbols

β	=	Thermal expansion coefficient	1/K

θ	=	Annulus orientation	Degree
μ	=	Dynamic viscosity	kg/m.s
ν	=	Kinematics viscosity	m^2/s
ρ	=	Air density	kg/m^3

Dimensionless Group

Gr	=	Grashof number	$= g\beta(t_s - t_a)D_h{}^3/\nu^2$
Gz	=	Graetz number	$= Re.Pr.D_h/L$
Nu	=	Nusselt number	$= h.D_h/k$
Pe	=	Peclet number	$= Re.Pr$
Pr	=	Prandtl number	$= \mu C_p/k$
Ra	=	Rayleigh number	$= Gr.Pr$
Re	=	Reynolds number	$= \rho.v.D_h/\mu$
Z^+	=	Axial distance	$= x.Re.Pr/D_h$

Superscript

a	Air
b	Bulk
calm.	Calming section
f	Film
i	Annulus inlet
s	Surface
t	Total
x	Local

Superscript

Average

Abbreviation

TPFC Theoretical Pure Forced Convection

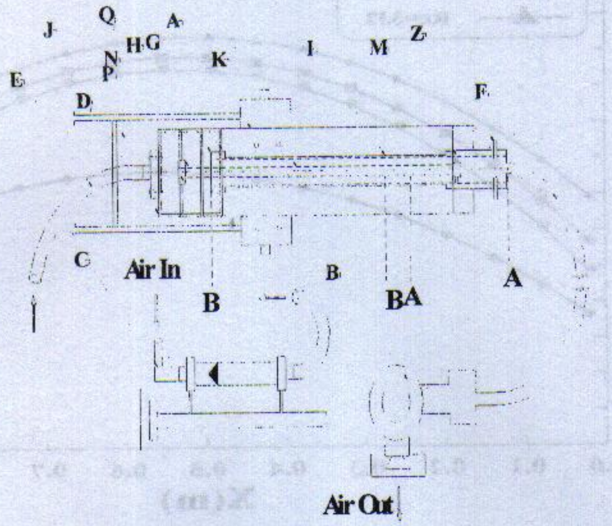

Fig. (1) Diagram of Experimental Apparatus

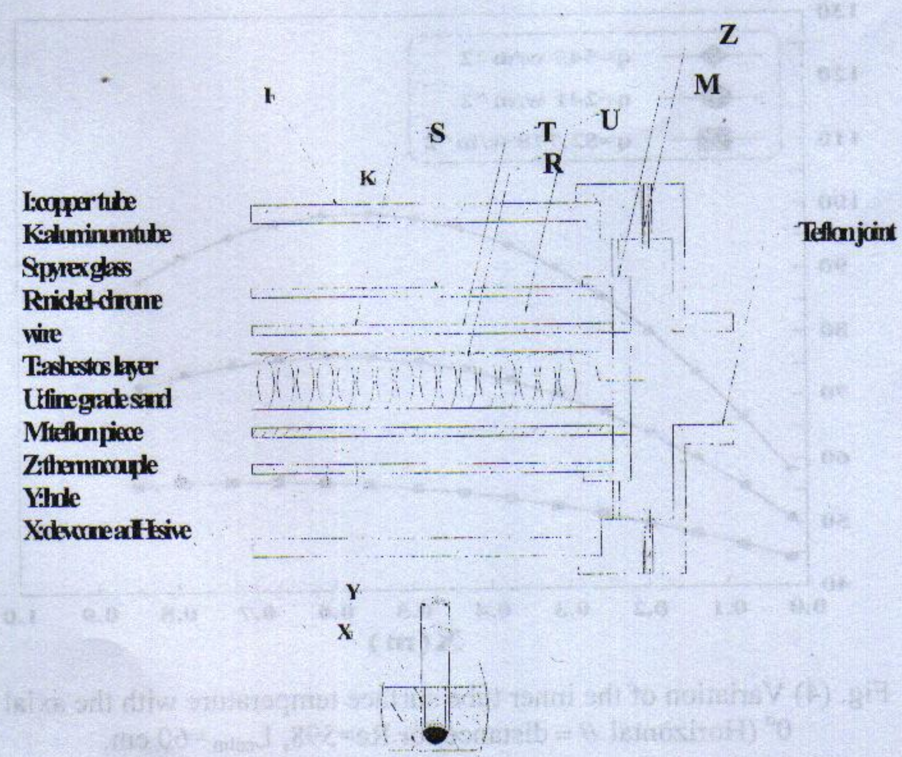

Fig. (2) section(A-A)

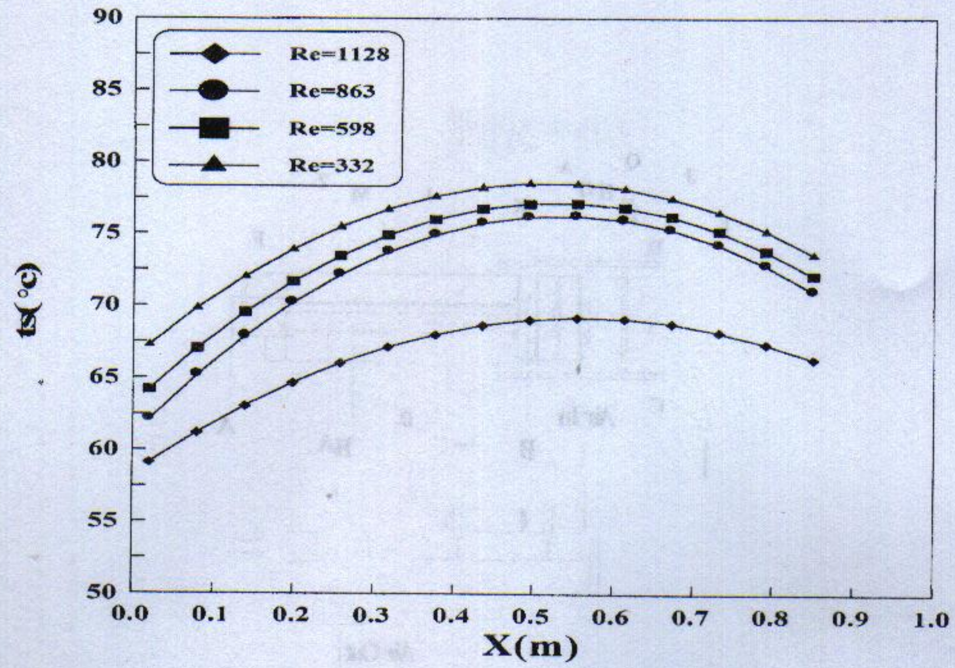

Fig. (3) Variation of the inner tube surface temperature with the axial distance for q=241 W/m^2,$L_{calm.}$=60, $\theta = 0^o$ (Horizontal).

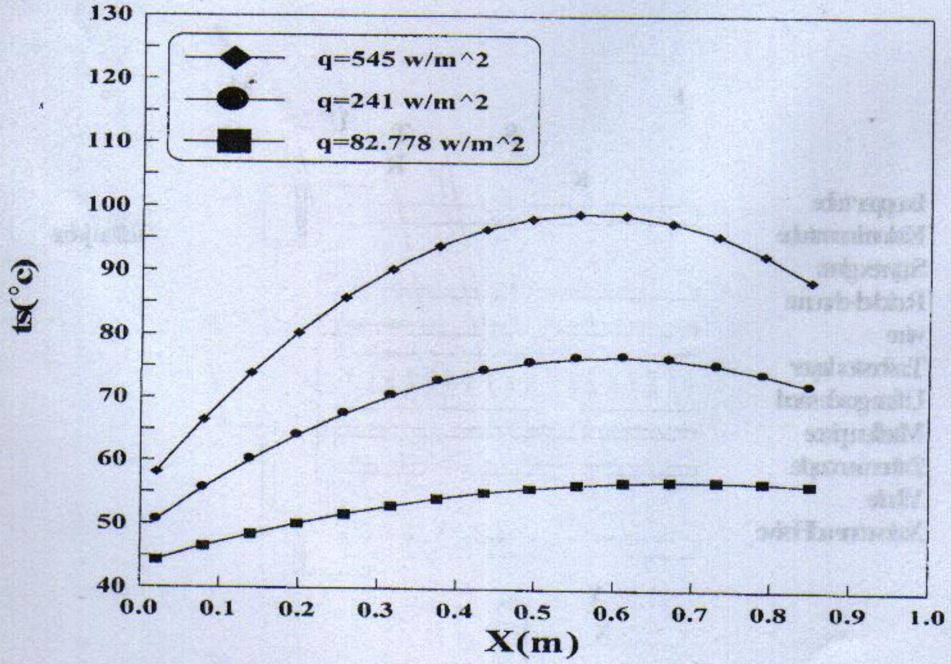

Fig. (4) Variation of the inner tube surface temperature with the axial 0^o (Horizontal $\theta =$ distance for Re=598, $L_{calm.}$=60 cm,

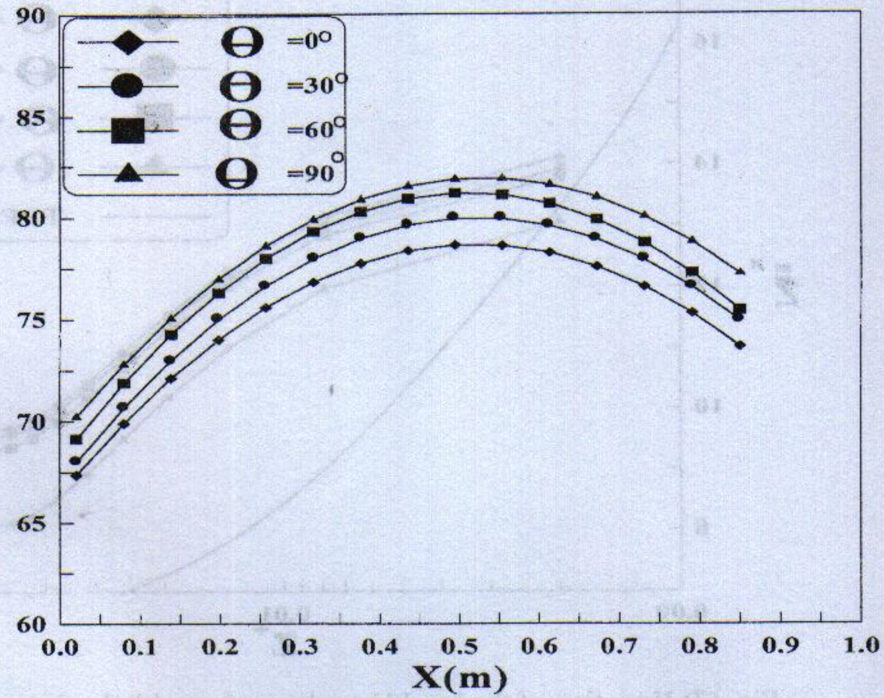

Fig. (5) Variation of the surface temperature with the axial distance for Re =332 and q=241 W/m^2, $L_{calm.}$ =60 cm.

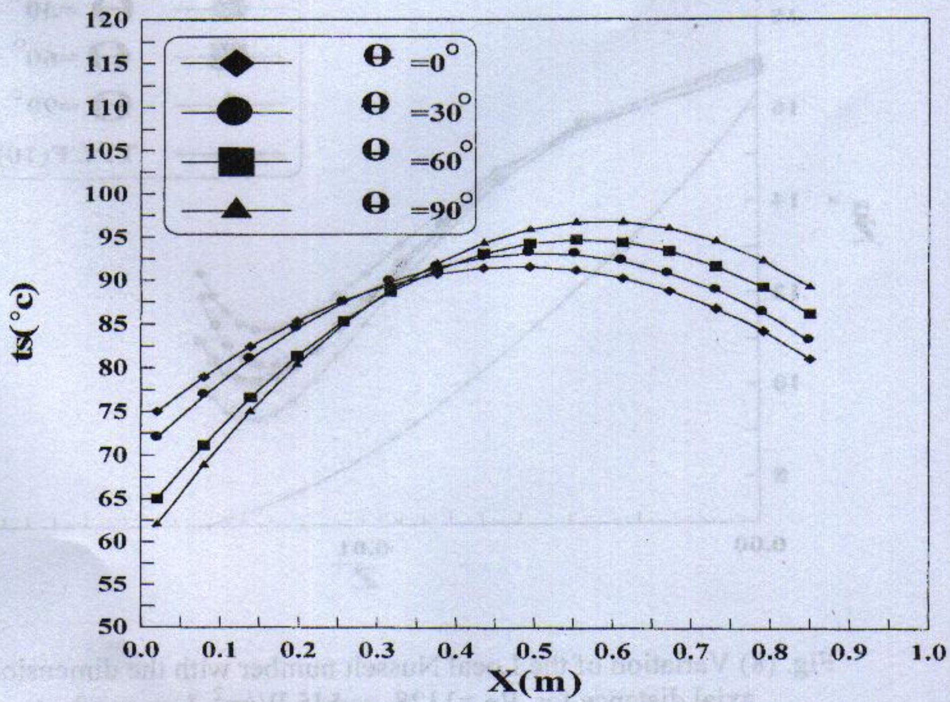

Fig. (6) Variation of the surface temperature with the axial Distance for q =545 w/m^2 and Re=1128, $L_{calm.}$= 60 cm.

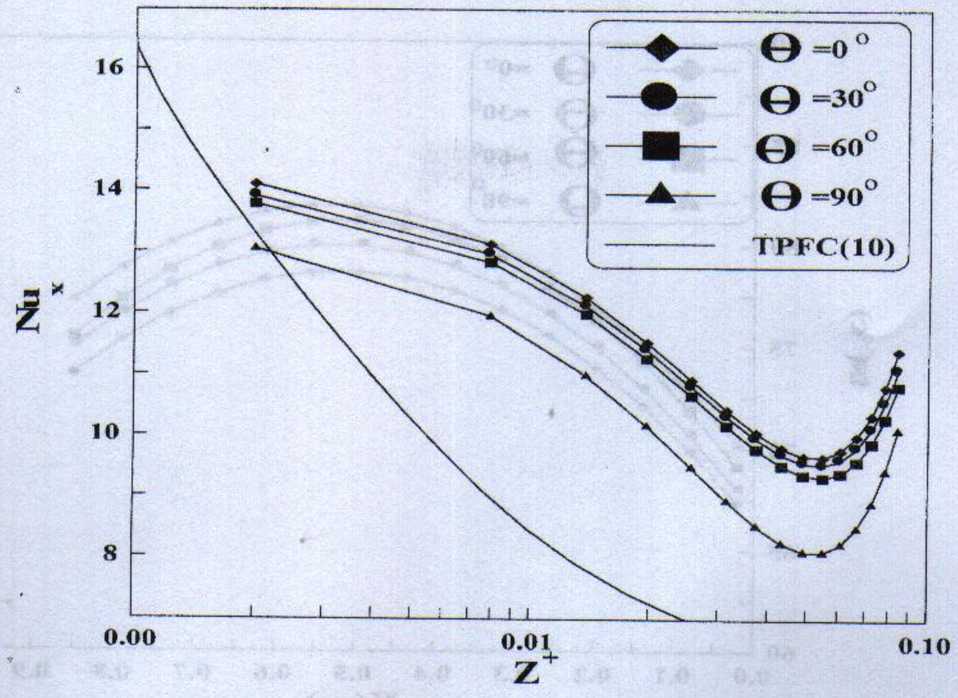

Fig. (7) Variation of the local Nusselt number with the dimensionless axial distance for Re = 332 , q=241 W/m^2, $L_{calm.}$= 60 cm

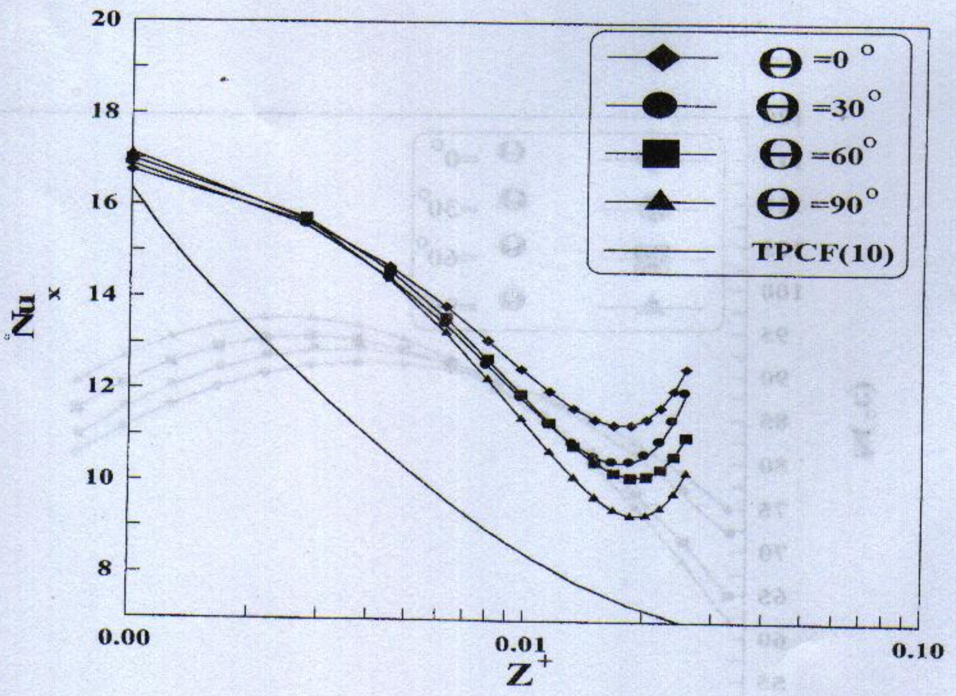

Fig. (8) Variation of the Local Nusselt number with the dimensionless axial distance for Re =1128, q=545 W/m^2, $L_{calm.}$= 60 cm.

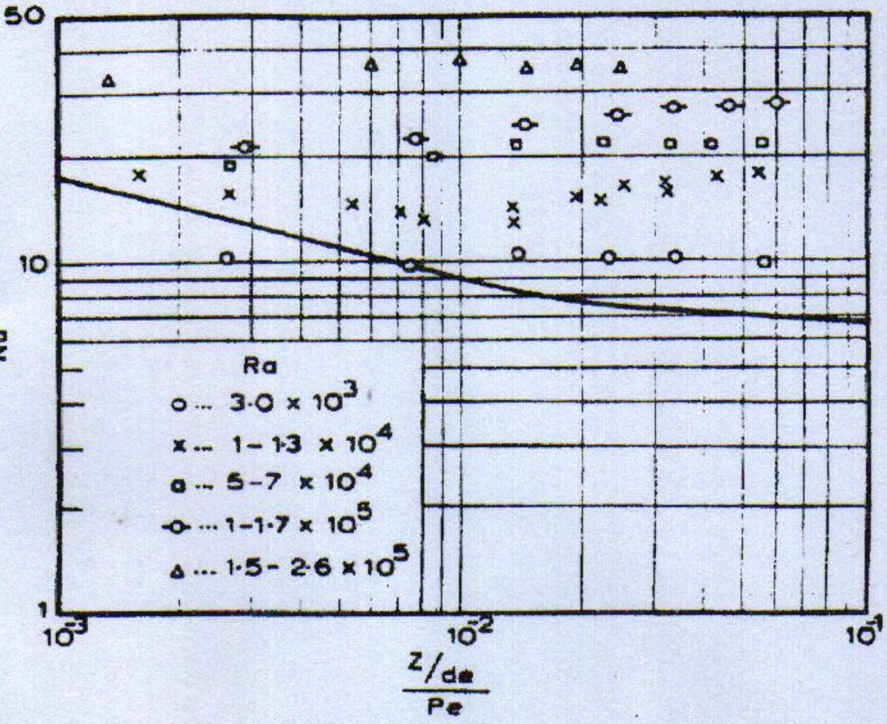

Fig. (9) Local Nusselt number of (Maitra, D and Subba, K. Raju, (1975)