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An Experimental Investigation of Natural Convection heat transfer Through Horizontal Open Ended Four Sided Tubes With Different Aspect Ratios

دراسة تجريبية لانتقال الحرارة بالحمل الحر خلال أنابيب أفقية مفتوحة الطرفين ذات مقطع رباعي بنسب جانبية مختلفة

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ملخص البحث:

دراسة معملية لانتقال الحرارة بالحمل الحر داخل أنابيب رباعية الاوجة ذات نسب جانبية مختلفة فى وجود تسخين منتظم ، تغيرت النسب الجانبية من 0.25 الى 1 . اجريت التجارب لعدد رايلى يتراوح من 2.02×10^5 إلى 4.6×10^6 . وتم حساب معاملي انتقال الحرارة الموضعي والمتوسط وعدد ناسيلت لمختلف الإبعاد الجانبية عند قيم مختلفة لرقم رايلى وأوضحت نتائج التجارب ان اختلاف الإبعاد الجانبية يؤثر ذلك على معاملي انتقال الحرارة الموضعي والمتوسط . لوحظ زيادة درجات الحرارة بزيادة المسافة الأفقية من طرفى الأنبوبة حتى تصل لأعلى قيمة في منتصف الأنبوبة. أيضا يزيد عدد ناسيلت بزيادة النسب الجانبية عند نفس مسافات المحور الافقى . تم ربط النتائج المعملية مع نظيرتها من المتاحة للأنابيب الأفقية.

Abstract

Experimental investigation of free convection heat transfer from the inside surface of horizontal four sided tubes of different aspect ratio with a uniformly heated surface is presented. The aspect ratios is changed from 1 to 0.25. The experiments covered a range of Rayleigh number, Ra from 2.02×10^5 to 4.6×10^6. The local and average heat transfer coefficients and Nusselt numbers are estimated for different aspect ratios at different Rayleigh numbers. The experimental results showed that the aspect ratio has a significant effect on the local and average heat transfer coefficients. It is noticed that temperature distributions increase with the increase of axial distance from both ends of the tube upto a maximum value at the middle of the tube. Also, the local Nu increases with the increase of aspect ratio at the same axial distance . The results obtained were correlated with the available data for the horizontal four sided tubes

Keywords: Natural convection; Constant heat flux; Horizontal four sided tubes; Different aspect ratio

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NOMENCLATURE.

A : Area. (m^2)

a: height of the four sided tubes. (m)

b: width of the four sided tubes. (m)

AR: aspect ratio, a/b

C: Constant.

Cp : Specific heat of air at constan pressure. (Kj/kg.K)

g : Gravity acceleration. (m/s^2)

Gr : Grashof number, $g\,\beta\,q\,D^4/v^2k$.

h_x : Local heat transfer coefficient, $q/(Ts - T\infty)$ W/m^2K

h_m: mean heat transfer coefficient, $q/(Tms - T\infty)$ W/m^2K

k: Thermal conductivity of air. (W/m K)

L: Length of the four sided tubes. (m)

Nu_x :Local Nusselt number , h D/k

Nu_m :Mean Nusselt number , hm D/k.

Pr : Prandtl number , $C_p\mu/k$

Q : Total heat flow. (W)

q : Heat flux. Q/A (W/m^2)

Ra: Rayleigh number, Gr Pr.

T_∞: Ambient air temperature. (K)

T_{mf} : mean film temperature. (K)

T_{ms}: mean surface temperature. (K)

T_s: upper surface temperature for tube (K)

X: local distance. (m)

ΔT_s: local temperature difference. $(T_s - T_\infty)$ (K)

ΔT_{ms}: mean temperature difference. $(T_{ms} - T_\infty)$ (K)

ß :volumetric coefficient of thermal expansion (1/K)

μ :dvnamic viscosity of air (kg/m.s)

ν :kinematic viscosity of air, (m^2/s)

1- Introduction

Free convection from inside four sided tubes is receiving growing interest in the last few decades because of its use in many practical fields in the areas of energy conservation, design of solar collectors, heat exchangers ,nuclear engineering, cooling of electrical and electronic equipment and many others. Heat transfer studies of free convection from four sided tubes are necessary for a better thermal design of industrial applications. As far as the tube itself is concerned, four sided tubes had been found in favor of reducing the flow resistance and enhancing the heat transfer inside or from the outside surfaces of the four sided tubes. It is considered to be useful to review the literature of circular tubes as a special case of four sided tubes. Free convective heat transfer from circular cylinders has been studied intensively in the past. Qing et al [1] studied Experimentaly heat transfer of natural convection in vertical rectangular channels with large aspect ratio.

Experimental results demonstrated that air natural convection heat transfer characteristics in three channels were similar. The wall temperatures were not always increasing along the flow direction, but reached a maximum value at an upstream point near the channel exit. Khaled Khodary et al. [2] studied natural convection from square cylinder in vertical channel . Steady laminar natural convection from horizontal hot square cylinder under uniform surface temperature, and positioned between two vertical adiabatic parallel plates is investigated numerically and compared with experiment. The problem is quantitatively as well as qualitatively examined for air as a working medium, channel height to square side length of 20, aspect ratio A varied from 0.1 to 0.8 and different lateral locations of the cylinder. Results were obtained for values of Grashof number ranged from 1 to 3×10^{6}. K.Elshazly et al.[3] studied experimental free convection heat transfer from the inside surface of inclined and vertical elliptic tubes of axis ratio 2:1 with a uniformly heated surface. The orientation angle α is changed from 0° to 90° with step of 15°, and the inclination angle θ is changed from 15° to 75° with step of 15° (inclined case). The inclination angle θ =90° (vertical case) is considered a special case of the inclined case and is studied separately. The experiments covered a range of Rayleigh number Ra from 2.6×10^{6} to 3.6×10^{7} . M. Moawed et al·[4] studied experimentally free convection heat transfer from the inside surface of vertical and inclined elliptic tubes of different axis ratios and different orientation angles with a uniformly heated surface. The axis ratio has a significant effect on the local and average heat transfer coefficients. The local Nusselt number (Nu) increases with the increase of axial distance from the lower end of the tube until a maximum value is attained near the top of the tube. Also, the local Nu increases with the increase of α, θ and aspect ratio at the same axial distance. Omara et al. [5] studied experimentally the heat transfer by natural convection from the inside surface of the horizontal elliptic tube of axis ratio 2:1 with a uniformly heated surface. The angle of attack was changed from 0° to 90° with steps of 15°. The experiments covered a range of Rayleigh number, Ra from 1.45×10^{6} to 1.78×10^{7}. It was found that the angle of attack has a significant effect on the local and average heat transfer coefficients. Abdul-Aziz [6] studied the heat transfer by natural convection from the inside surface of uniformly heated tubes at different angles of inclination. The experiments covered a range of Ra from 1.44×10^7 to 8.85×0^8, L/D from 10 to 31.4 and angle of inclination (measured from vertical position) from 0° to 75°.The results showed that the average Nu was maximum when the tube was vertical. Khamis [7] and Al-Arabi et al.[8] investigated theoretically and experimentally natural convection heat transfer inside vertical annuli, the theoretical study for radius ratios of 0.26, 0.5 and 0.9 with two cases of heating the inner or outer tube, while the other tube was adiabatic. Shehata [9], studied natural convection inside vertical, inclined and horizontal annuli of radius ratio 0.73 with heated inner tube and outer tube is adiabatic. Sarhan et al. [10] presented an experimental study of natural convection inside open-ended horizontal and vertical annuli with different aspect ratios. Correlations of the dimensionless group of average Nu– Ra × (D/L) were presented for the horizontal and vertical cases. Yasin Varol [11] numerically studied Natural convection heat transfer in a triangle enclosure with flush mounted heater on vertical wall. Governing parameters on heat transfer and flow fields are aspect ratio of triangle, location of heater, length of heater and Rayleigh number. It is observed that the most important parameter on heat transfer and flow field is the position of heater which can be a control parameter for the present system.

As can be seen from the previous work, to the authors knowledge, there is no available data on natural convection inside four sided tubes. The present work has been conducted to provide experimental data on natural convection heat transfer in open ended horizontal four sided tubes with a uniformly heated wall for different aspect ratios.

2- Experimental apparatus and procedure

The experimental setup, is shown in Fig. (1). It consists of a horizontal four sided tube mounted on a frame. Four sided copper tubes with different aspect ratios and thicknesses of 1 mm and 500 mm lengths are used separately as test sections. The tested four sided tubes that are used in these experiments have aspect ratios (a/b) equal to 1, 0.75, 0.5 and 0.25.

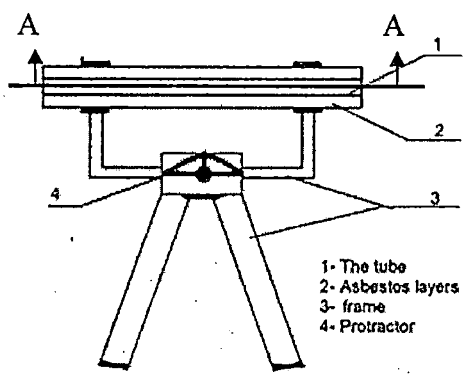
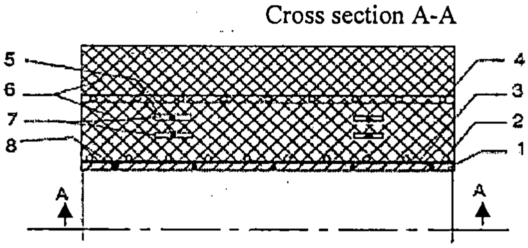


Fig (1) experimental setup

The outer surface of the tested four sided tubes is covered with an electric insulating tape on which nickel-chrome wire of 0.3 mm diameter is uniformly wound to form the main heater as shown by Fig. (2). The main heater is covered with an asbestos layer of 55mm thickness, on which another nickel-chrome wire of 0.3 mm is uniformly wound to form a guard heater. The guard heater is covered with a 45mm thick asbestos layer. four pairs of thermocouples are installed in the asbestos layer between the main heater and the guard heater. The thermocouples of each pair are fixed on the same radial line. The input to the guard heater is adjusted so that, at steady state, the readings of the thermocouples of each pair became practically the same. Then, all the heat generated by the main heater is flowing inward to the four sided tubes.



1.Four sided tubes. 2.Main heater.
3.Electric insulation. 4.Guard heater.
5.Pair of thermocouples.
6.Asbestos layers.
7.Aluminium strips. 8.Thermocoples.

Fig (2) Electric heater arrangement

The inner surface temperature of the four sided tubes is measured by 28 copper-constantan thermocouples of 0.4 mm diameter soldered in slots milled along the axis and circumferential directions. The distribution of thermocouples is located at seven measuring stations for each face at axial distance of 50, 120, 190, 250, 310, 380 and 450 mm from one end of the four sided tubes as shown in Fig.(3).

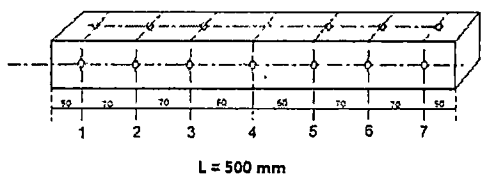


Fig (3) Distribution of the thermocouples on the four sided tube.

The ambient air temperature was measured by two thermocouples fixed at the four sided channel ends. All temperatures are measured by digital thermometer capable of reading ±0.1°C. The input electric power was regulated by AC power variance and is measured by a digital wattmeter with a resolution of ±0.1W.

The experiments were conducted with the test section in a closed room 2×2 m of plastic transparent walls to prevent currents of air, and the measuring instruments are mounted outside of this room. The input electric power to the main heater is controlled and changed by the AC variac for each experiment. The steady state condition for each run was achieved after 3 to 4 hours approximately. The steady state condition was considered to be reached when the temperature reading of each thermocouple was not changed by more than ±0.5°C within 20 min. When the steady state condition was reached, the readings of all thermocouples, the input power, the inside surface temperature of four sided channels and the ambient air temperature were recorded.

3. Uncertainty analysis

Generally, the accuracy of the experimental results depends upon the accuracy of the individual measuring instruments and the manufactured accuracy of the four sided tubes. Also, the accuracy of an instrument is limited by its minimum division (its sensitivity). In the present work, the uncertainties in both the heat transfer coefficient (Nusselt number) and Rayleigh number were estimated following the differential approximation method. For a typical experiment, the total uncertainty in measuring the main heater input power, temperature difference (ΔT), and the four sided tubes surface area were ±0.46%, ±0.11%, and ±0.54%, respectively. These were combined to give a maximum error of ±3.2% in heat transfer coefficient (Nusselt number) and a maximum error of ±4.5% in Rayleigh number. , the uncertainty in the result can be given as [12]

$$U_h = \left[\left(\frac{\partial h}{\partial Q}U_Q\right)^2 + \left(\frac{\partial h}{\partial \Delta T}U_T\right)^2 + \left(\frac{\partial h}{\partial A}U_A\right)^2\right]^{1/2} \quad (1)$$

4. Data reduction

The local heat transfer coefficient between the inner surface of the four sided tube and the air inside the four sided tube is calculated from.

$$h_L = q / (T_s - T_\infty) \quad (2)$$

where q is the average heat flux transferred by free convection from the inside surface of the four sided tubes, and it is equal to the electric power. It was found that the heat transfer by radiation from the ends of the four sided tubes are less than 3% and can be neglected

$$Q_r = \sigma\, \varepsilon\, A_{ends} \; (T_s^4 - T_\infty^4) \quad (3)$$

$$q = Q/A \quad (4)$$

The corresponding local Nusselt number, Nu is calculated from Eq.(5)

$$Nu_L = h\; L/k \quad (5)$$

The average heat transfer coefficient between the inner surface of the four sided tube and the air inside the four sided tube is calculated from Eq. (6)

$$h_m = q / \Delta T_{ms} \qquad (6)$$

where

$$\Delta T_{ms} = (T_{ms} - T_\infty)$$

$$T_{ms} = {}^{1}/_{L} \int_0^L t_s \ dx. \qquad (7)$$

The corresponding average Nusselt number, Nu_m is calculated from Eq.(8)

$$Nu_m = h_m \times L/k \qquad (8)$$

The Rayleigh number, Ra is calculated from

$$Ra = Gr \times Pr \qquad (9)$$

The physical properties are evaluated at the mean film temperature, Ref. [12], as given by

$$T_{mf} = (T_{ms} + T_\infty) /2 \qquad (10)$$

5. Results and discussion

The present experimental data of a horizontal four sided tubes with different aspect ratios are discussed in this section. The inner surface of the four sided tubes is subjected to constant heat flux. The effect of aspect ratios and Ra on the heat transfer results is discussed in this section. The results obtained in this work include the temperature distribution on the inner surface of the four sided tubes, local Nu and average Nu_m. The present experimental data covered ranges of Ra from 2.02×10^5 to 4.6×10^6, of $\theta = 0°$ (horizontal cases) and for AR from 1 to 0.25.

5.1. Temperature distribution

The experimental results showed that the circumferential variation of the surface temperature of the four sided tubes measured at any axial location is small for all cases. Generally, the maximum circumferential variation in temperature is ±1°C under constant heat flux conditions. This can be attributed to the high thermal conductivity of the four sided tubes and low Prandtl number of the air. The samples of the inside surface temperature difference $(T_s - T_a)$ of the four sided tubes along the axial length for the different heat fluxes employed, different aspect ratios in Figs. (4) to (7).The $[(T_s - T_a)\ X]$ curves for all cases have the same general shape.

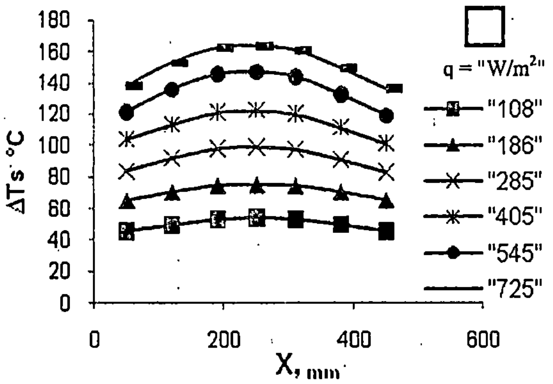


Fig (4) Variation of wall temperature difference ΔT_s with axial distance X , AR =1

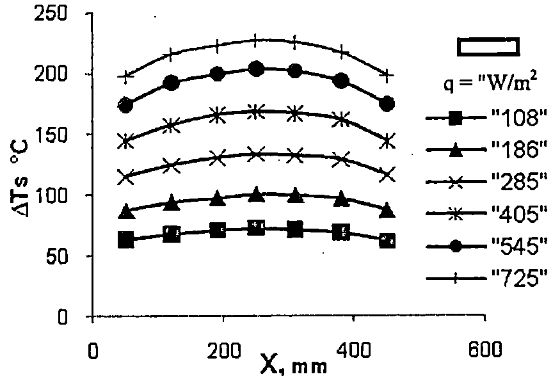


Fig (7) Variation of wall temperature difference ΔT_s with axial distance X , AR =0.25

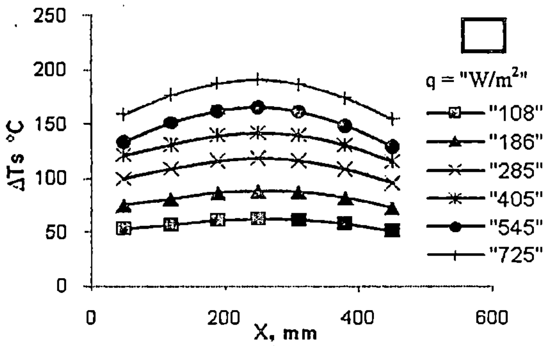


Fig (5) Variation of wall temperature difference ΔT_s with axial distance X, AR =0.75

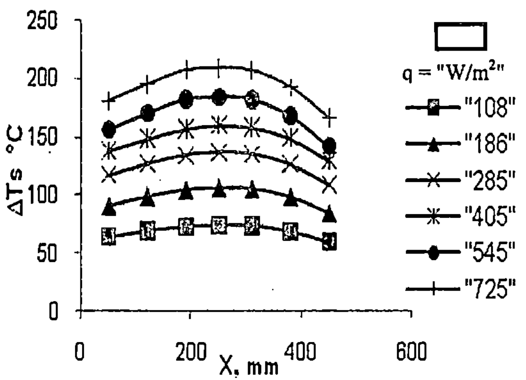


Fig (6) Variation of wall temperature difference ΔT_s with axial distance X , AR =0.5

The surface temperatures gradually increas from both ends of the tube until it reache a maximum value at the middle section, half way through, the horizontal tube. The variation is apparently a symmetrical around the middle section of the tube. Also, at the same axial distance the surface temperature difference increases with increase of the heat flux . This shape of curves is believed to be due to a growing boundary layer of the air flow from both ends of the tube. Due to the buoyancy force, a flow of air is drawn at the lowest point from both ends where it merges through the tube length until the middle of the tube. The stream of air , however, flows back at the upper point from same end.

Fig. (8) shows the variation of mean temperature difference (ΔTs_m) with X at aspect ratio (AR) ,where AR change from 1 to 0.25. As shown by this figures, The aspect ratio has a significant effect on temperature difference. The mean temperature difference (T_w-T_a) decreases with the increase of aspect ratio (AR) at the same axial distance.

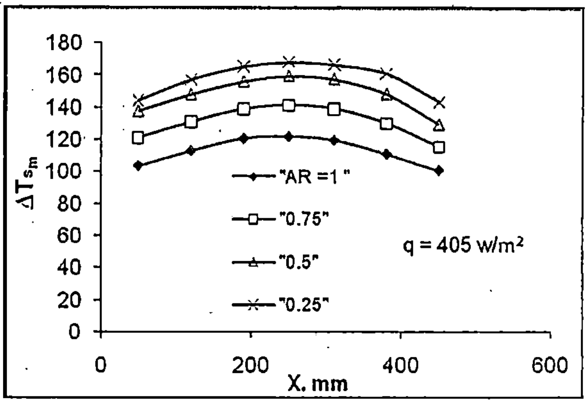


Fig.(8) Variation of mean temperature difference ΔT_{sm} with axial distance X , q = 405 W/m²

5.2. Nusselt number

Figures. 9 to 12 show sample data of the axial variation of local nusselt number, Nu_L of the horizontal tubes with the dimensionless length (X/L) for different heat fluxes and different aspect ratio (AR) . As shown in these figures, the local nusselt number (Nu_L) has a maximum value at both ends of the horizontal tube and it decreases gradually until it reaches a minimum value at the middle of the tube. The two halves of the horizontal tubes around the midpoint are symmetrical. This can be explained as follows : when cold air is drawn from both ends of the tube it causes a high rate of heat transfer and, consequently ,greater heat transfer coefficient at both ends. This can be explained by the growth of the boundary layer from both ends toward the middle of the four sided channel causing the decrease of the convective heat transfer coefficient and consequently decreases the Nu_x towards the middle of the four sided channel. Corresponding to Ref. (5)

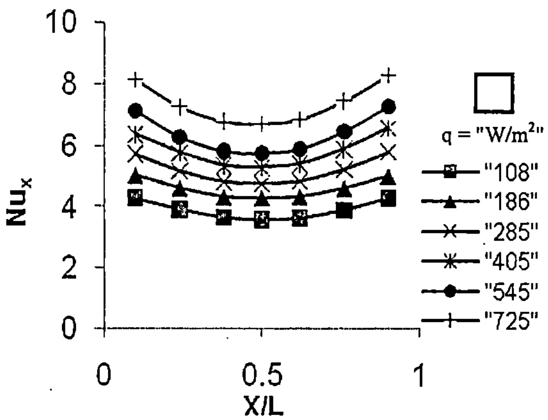


Fig (9) Variation of the local Nusselt Nu_x with axial distance along the tube X/L AR=1

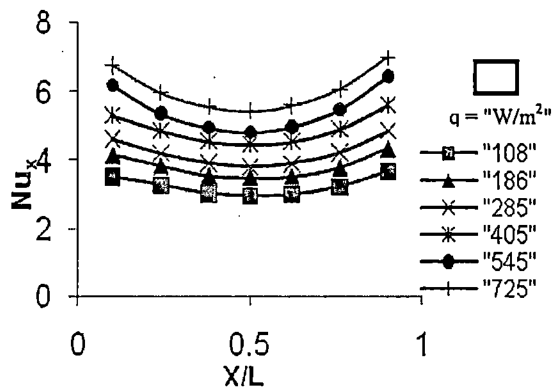


Fig (10) Variation of the local Nusselt Nu_x with axial distance along the tube X/L AR=0.75

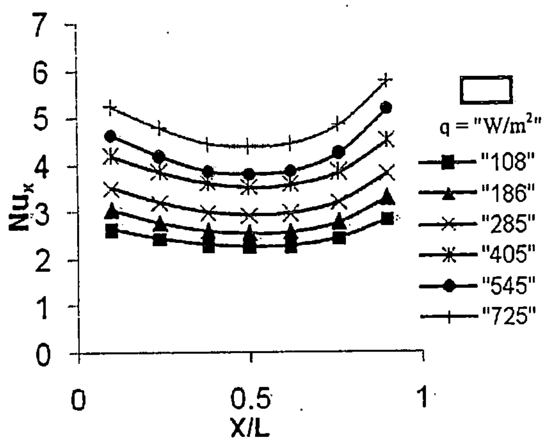


Fig.(11) Variation of the local Nusselt Nu_x with axial distance along the tube X/L AR=0.5

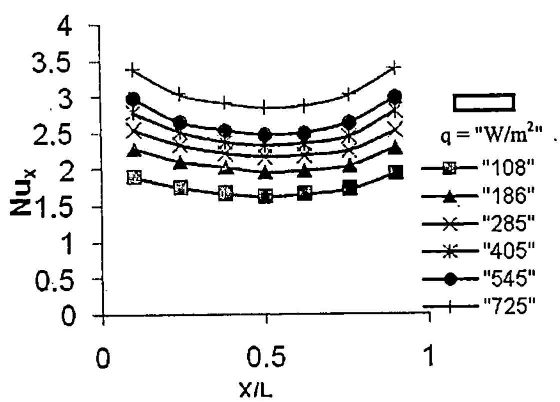


Fig (12) Variation of the local Nusselt Nu_x with axial distance along the tube X/L AR=0.25

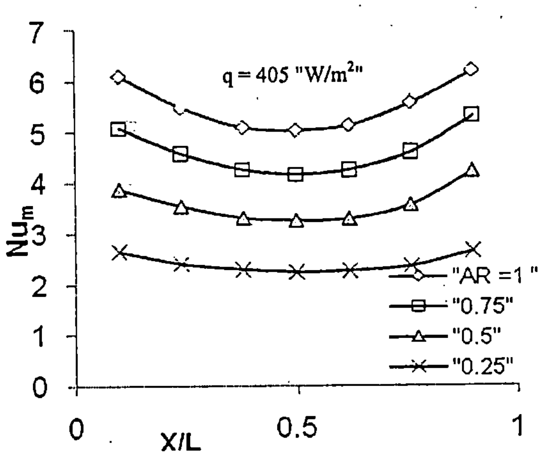


Fig (13) Variation of Nusselt mean Nu_m with axial distance X/L , q = 405 W/m²

Figure (13) shows that aspect ratio has a significant effect on mean Nusselt number. The mean Nusselt number (Nu) increases with aspect ratio (AR) at the same axial distance.

5.3. Correlation of the results :

As explained before that the experimental data of the tubes are classified as the horizontal case from various aspect ratio of tubes .

The general correlation of the horizontal tube of Nu_m as a function of Ra can be written in the following form:

$$Nu_m = c\, Ra^n \times (AR)^m \qquad (A)$$

Where: c , n and m are constant.

The experimental results were fitted using power regression to determine the constants .The resulting empirical correlation are :

$$Nu_m = 0.0016\,(Ra)^{0.57}\,(AR)^{0.75}$$

$$0.25 \le AR \le 1$$

$$2.02\times10^5 \le Ra \le 4.6\times10^6 \qquad (11)$$

Equation (11) represents the general equation for natural convection inside horizontal four sided tubes. The calculated data from Eq.

(11) of the average Nusselt number (Nu_{mCal}) is plotted against experimental data of the average Nusselt number (Nu_{mExp}) in Fig. (14). As shown in this figure the maximum deviation between the experimental data and the correlation of the horizontal four sided tube is ±16%.

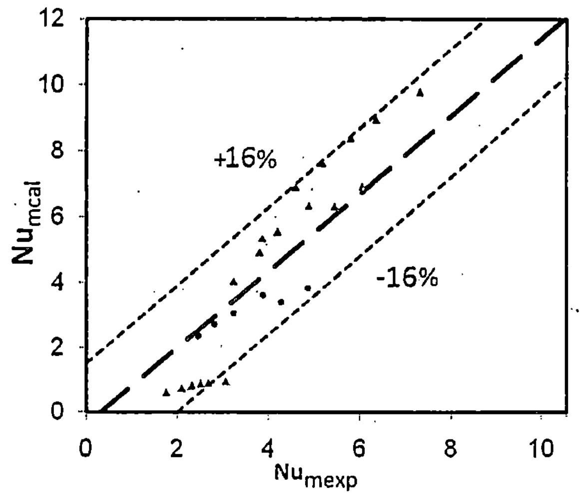


Fig. 14. Average Nu_{mcal} vs Average Nu_{mexp} for horizontal four sided tubes.

6.Comparison with previous work

As mentioned before, to the author's knowledge, there is no published information about natural convection heat transfer inside horizontal four sided tubes that can be compared with the present work. The only available information that can be compared with the present work was established by Sarhan et al. [10] and Omara et al [5]. The comparison of the present work with the work of Sarhan et al. [10] and Omara et al [5], is shown in Fig. (15). As shown in this figure, the present results of natural convection inside horizontal tube are lower than the results of Sarhan et al. [10] and Omara et al [5]. Flow in a four sided tube is that where there are very right corner angel the thickness of the boundary layer can become large relative to the distance between adjacent wall surface. This leads to heat transfer coefficient very much lower in the right angle corners. So natural convection inside four sided tube are lower than elliptic and and circular tube

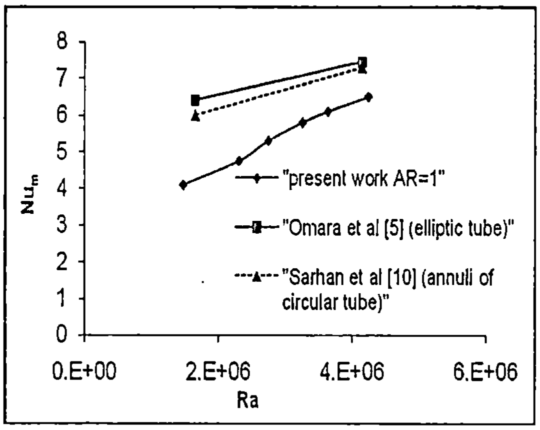


Fig (15) comparison between present work (AR=1) & Sarhan et al. [10] and Omara et al [5] at Θ= 0°

7.Conclusions

Free convection heat transfer from the inside surface of horizontal tubes of different aspect ratios with a uniformly heated surface are investigated experimentally. The aspect ratio is changed from 1 to 0.25 . The experiments covered a range of Rayleigh number, Ra from 2.02×10^5 to 4.6×10^6. The local and average heat transfer coefficients and Nusselt number are estimated for different aspect ratios at different Rayleigh numbers.

The following main points can be drawn from this study:

- the aspect ratio has a significant effect on the local and average heat transfer coefficients. The local Nusselt number (Nu) increases with aspect ratio (AR) at the same axial distance .
- The temperature distributions increase with axial distance from both ends of the tube until a maximum value at the middle of the tube.
- The local nusselt number (Nu_L) has a maximum value at both ends of the horizontal tube and it decreases gradually until it reaches a minimum value at the middle of the tube.
- Correlations of the mean Nusselt number (Nu_m) with Rayleigh number (Ra) for natural convection inside open-ended horizontal tubes at different aspect ratios (AR) are presented Eq [10] .

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