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# AUGMENTATION OF NATURAL CONVECTION HEAT TRANSFER FROM HORIZONTAL CIRCULAR CYLINDER BY MEANS OF HELICAL-COIL-TURNS

تحسين معامل انتقال الحراره بالحمل الحر من اسطوانه
افقيه دائريه المقطع باستخدام لفات ملف حلزوني

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الخلاصه - يشتمل هذا البحث على دراسه عمليه لانتقال الحراره بالحمل الحر من اسطوانه نحاسيه افقيه دائريه المقطع قطرها الخارجي ٢١,٥ مم ملفوف حولها لفات ملف حلزوني، عدد لفات الملف الحلزوني يتغير من ٥٠ الى ١٩٦ لفه/متر و عدد الحلزونات في الملف يتغير من ١٧٨ الى ٨٣٣ حلزون/متر، في حين كانت نسبة قطر الاسطوانه الى قطر الملف الخارجي ثابته عند ٤,٢٢ ، الاسطوانه مسخنه بفيض حراري ثابت يعطي قيم رقم رايلي متغيره من ٢٠٠٠ الى ٥٢٠٠٠ في حين كان رقم برانتل ثابت تقريبا. اوضحت التجارب ان رقم نوسلت داله في رقم رايلي ، عدد لفات الملف الحلزوني و عدد الحلزونات في الملف. تتراوح نسبه التحسين في معامل انتقال الحراره باستخدام لفات الملف الحلزوني من ٥٦% الى ١٠٦% منسوبا للاسطوانه الملساء. تم صياغه النتائج في شكل معادلات تجريبيه لامكان حساب معامل انتقال الحراره.

## ABSTRACT

Augmentation of steady natural convection heat transfer from horizontal cylinder were studied experimentally. The test section consisted of a brass circular cylinder, of 301 mm length and diameters of 21.5/12.2 mm, heated with a constant heat flux supplied by an electrical resistive heater made of nickel-chrome wire coil. The experimental data have been obtained when the test cylinder is covered with eight type of helical-wire-coils wounded around it ,as well as when the cylinder is bare. The number of helix per meter length (Nc) varied from 178 to 833 and the number of turns of each helical-coil per meter length (Nt) varied from 50 to 196 at cylinder-to-helix diameters ratio (D/d) of 4.22. The tests were conducted in air in the range of Rayleigh number of 2000 to 52000 and Prandtl number of nearly 0.7.

The results of heat transfer coefficient are represented in the relation of average Nusselt number versus Rayleigh number . The relations indicate that (Nu) is a significant function of (Ra),(Nc) and (Nt). The augmentation of heat transfer in case of use the helical-wire-coils varied from 56 to 106 % compared with the bare cylinder.

## INTRODUCTION

Many techniques have been introduced in the past years to improve the effectiveness of heat exchanger involving natural convection and/or forced convection with or without phase change [1]. The use of flat-type circumferential fins to enhance natural convection heat transfer from horizontal tubes is standard practice in many heating and cooling applications, most notably for space heating ( e.g. baseboard heating ). Typically such heat transfer configurations consist of a horizontal tube to which is affixed a succession of uniformly spaced fin plates. The fins are either square or circular metal plates positioned perpendicular to the axis of the tube, with a central hole whose diameter corresponds to that of the tube. Thus the fins defined an array of parallel vertical channels. Each channel is open to the ambient at its lateral edges as well as at the top and bottom but is internally obstructed because of the presence of the tube.

Knudsen and Pan [2], studied the overall performance of the fin-tube natural convection heat transfer devices. Edwards and Chaddock [3], measured heat transfer coefficients for three fin-tube systems with different ratios of fin diameter to tube diameter. The coefficients included both fin and tube heat transfer contributions. It is clear that the heat transfer rate did not increase in proportion to the surface area as would be the case had the heat transfer coefficient been uniform. Rather, it is clear that the heat transfer rate increased more or less in proportion to the outer diameter of the fin. Sparrow and Bahrami [4], performed experiments to determine the natural convection heat flux distributions on the faces of simulated isothermal circular fins fixed to a horizontal adiabatic tube. A naphthaline sublimation mass transfer approach was used which suggested that, the highest coefficients occur at the fine periphery while the lowest occur at the fin base, which supports the finding of Ref. [3].

Some local heat transfer measurements have been made on transverse finned tubes in forced convection. Wong [5] used the naphthalene sublimation technique to obtain local mass transfer coefficients on concentrically and eccentrically finned cylinders in air. Jones and Russell [6] performed similar work but measured heat transfer coefficients using a transient technique, i.e. by suddenly injecting the model into an air stream and recording the local temperature variation. Tolpadi and Kuehn [7] numerically studied steady three dimensional conjugate natural convection heat transfer from a horizontal isothermal cylinder with infinitely large transverse isothermal fins.

The effect of cylinder dimensions, inter cylinder spacing, positions of the cylinder on the plate and Rayleigh number were studied. Kown and Kuehn [8] theoretically analyzed conjugate heat transfer by steady laminar natural convection from an infinitely vertical long conducting fin attached below a heated isothermal cylinder and developed heat transfer correlations that included the fin-cylinder boundary layer overlap. Kwon et al [9] extended this work to the case of a short vertical fin. The same numerical technique was used, but optical data were obtained to verify the numerical results. The influence of fin length on steady, conjugate, natural convection heat transfer from an isothermal cylinder with one vertical longitudinal conducting plate fin was studied in Ref. [10]. Tolpadi and kuehn [11], studied experimentally steady laminar natural convection heat transfer from horizontal isothermal cylinder in water with single nonisothermal longitudinal plate fin at various angles. Local fin temperature and heat transfer from the fin and cylinder are obtained using a Mach-Zehnder interferometer. Overall heat transfer measurements are made from the thermocouple and multi meter data for comparison. The total heat transfer rate does not change significantly from the bare cylinder value.

Helical-coil rings were used by Sultan et al [12] to augment forced convection heat transfer from circular cylinder in cross flow of air. The enhancement in heat transfer coefficient was varied from 13 to 62 %, in the range of Reynold's number from 7000 to 60000.

From the literature review, one may observe that most studies of augmentation heat transfer from horizontal cylinders have been obtained either by means of longitudinal plate fins or using transverse plate fins. The main object of the present work is to use the helical-coil turns as a new technique to enhance natural convection heat transfer from horizontal cylinder heated under constant heat flux and to compare the heat transfer coefficient in case of using the helical-coil turns with the smooth cylinder under the same operating conditions.

## EXPERIMENTAL APPARATUS AND PROCEDURES

The experimental apparatus is shown diagrammatically in Fig. 1. It consists essentially of brass cylinder (3) of length equal to 301 mm, outer diameter of 21.5 mm and inner diameter of 12.2 mm. Two teflon circular pieces (1), have the same outer diameter of cylinder, are fitted to the ends of the cylinder so that the outside surface of the teflon forms a continuation of the brass surface. Teflon was chosen because of its low thermal conductivity in order to reduce the heat lost from the cylinder ends.

The surface of the cylinder is heated by means of a helical-type electric heater. It consists of a nickel-chrome wire coil (9) uniformly wound to give constant heat generation per unit length. The coil is protected by an envelope of pyrex glass tubing (6). To avoid convection current between the glass tube and the cylinder, this gap is filled with mica tape (5) wound on the glass tube and asbestos powder (4) to fill the space between the mica tape and the inner surface of the brass cylinder. The ends of the heater wire are connected to circular porcelain pieces (8). The porcelain pieces are fitted in holes bored centrally in the teflon ends and therefore keep the concentricity of the heater.

To measure the surface temperature of the heated cylinder, six square cross-section grooves, 1 mm wide and 1 mm deep, were cut over the outer surface of the cylinder. A total of seven thermocouples of 0.25 mm copper-constantan are impeded inside the grooves to their places and kept in contact with the cylinder surface using glue and then thoroughly polished. Five thermocouples (11) were placed circumfurentially , 45° apart, at the middle of the cylinder . To make sure that the temperature of the cylinder circumference at any point in the same plane is the same ( i.e. no axial heat transfer), the remaining two thermocouples (2) are placed in horizontal plane cirtcumferentially 180° apart and 10 mm apart from both ends of the cylinder. No difference between the readings of the thermocouples at the same plane was noticed during the experiments.

The temperature of air is measured by a thermocouple placed outside the channel far from the effect of the hot cylinder.

To minimize radiation from the cylinder its surface is polished. The emissivity of the polished brass is taken as 0.03 [13]. The calculated heat lost by radiation varied between 1 % and 4 % of the total heat transfer.

The power consumed by the heater is measured by accurate ammeter and voltmeter , and a voltage stabilizer is used to ensure that the incoming main voltage is constant. A 12-point self switching digital temperature recorder of full scale of 200° capable of reading 0.01 °C was used to take the reading of the thermocouples.

The heat input to the heater is controlled by using an auto-transformer . Once the switch of the power source is closed, the heating system starts. During the course of this experimental work nearly 90 minutes are needed to reach the steady state condition. This condition is satisfied when the temperature reading does not record any change within a time periods about 15 minutes. Applying the principle of conservation of energy gives:

$$P = Q_{rad} + Q_{conv} \qquad (1)$$

The heat energy carried away by radiation ($Q_{rad}$) is calculated during the course of this work, and the heat energy convicted away from the cylinder surface ($Q_{conv}$) can be expressed using Newton's law of convective cooling as follows:

$$Q_{conv} = P - Q_{rad} = A\, h\, ( t_v - t_\infty ) \qquad (2)$$

The average coefficient of natural convection heat transfer can be calculated using the following correlation:

$$h = ( P - Q_{rad} ) \,/\, [\, A\, ( t_v - t_\infty ) \,] \qquad (3)$$

During the present experimental work, helical-coil-turns (10)were not welded on the heated test cylinder surface, they were fixed around the test cylinder under the helical-coil turns strength, Therefore they had not only a bad contact with the cylinder surface but also a small area of contact with it. So that, the helical-coil turns around the cylinder surface may be act partially as an extended surface of the test cylinder and/or as a boundary layer disturbing means.

## RESULTS AND DISCUSSION

This investigation concerns with the determination of natural convection heat transfer coefficients for horizontal brass circular cylinder, of diameters 21.5/12.2 mm, using eight type of helical-coil-turns made from copper wire of diameter 0.85 mm.

One smooth cylinder is used to standarize the experimental set-up and also to evaluate the increase in the average heat transfer coefficient using eight type of helical-coil-turns around the brass cylinder, relative to the smooth one.

For the determination of the heat transfer coefficients and correlation with Rayleigh numbers, some quantities were measured for each data run. The power input to the electric heater, the average cylinder surface temperature and the free stream temperature were recorded. The heat lost by radiation were calculated and the net rate of heat transfer by convection was used to calculate the average heat transfer coefficient from Eqn (3). The typical deviation of any thermo-couple reading from the average wall temperature was 1 % of ($t_v - t_\infty$).

The properties of air which are used in the calculations of the experimental results were determined at the arithmetic mean temperature of the mean surface temperature ($t_v$) and the free stream temperature ($t_\infty$). The values of the physical properties are taken from Appendix of Reference [13]. The ranges of experimental data are shown in Table 1.

Table 1. Ranges of experimental data

| Parameter | Range | Parameter | Range |
|---|---|---|---|
| $P_c$ (mm) | 1.2,1.87,2.8,5.0 | q ($W/m^2$) | 50:2750 |
| $P_t$ (mm) | 5.1,6,8,10,20 | $t_v$ ($^oC$) | 15.5:129 |
| $N_c$ (coil/m) | 833,535,357,178 | $t_\infty$ ($^oC$) | 14:21.6 |
| $N_t$ (turn/m) | 196,148,100,50 | Ra | 2000:52000 |
| $P_t/d$ | 1,1.33,1.96,3.92 | D/d | 4.22 |
| $P_c/d$ | 0.235,0.367,0.549,1.098 | | |

The heat transfer results ,for bare cylinder,are plotted in terms of the average Nusselt number (Nu) versus the average Grashof number (Gr) multiplied by Prandtl number (Pr) (i.e Rayleigh number Ra ) as shown in Fig (2). The experimental Nusselt values are about 8 % higher than that obtained by Mc Adams [14]. It is concluded from reference [15]that, the natural convection heat transfer coefficient under uniform heat flux was found to be 7 % higher than the same data obtained under constant temperature, so the present data are in good agreement with that of the literature and the average Nusselt number can be correlated with Rayleigh number of the heated cylinder under constant heat flux as follows:

$$Nu = 0.572\ Ra^{0.25} \qquad (4)$$

where: $4300 \leq Ra \leq 52000$

The correlation (4) predicts the values of average Nusselt number with a maximum error of $\mp$ 6 %.

About the response of Nusselt number to the helical-coil turns surrounding the test cylinder, one may observe that Nusselt number for each of the eight investigated test cylinders are plotted as a function of Rayleigh number in Figs (3-6). From the over view of the figures , it is seen that Nusselt number increases, in general, with the existence of the helical-coil turns. One may also observe that, Nusselt number values increase, in general, with the Rayleigh number.

The results desplayed on Figs (3,4) are obtained at a number of helix per meter length ($N_c$) of 833 , corresponding to coil pitch-to-diameter ratio ($P_c/d$) of 0.235, and a number of turns per meter length ($N_t$) of 50,100,148 and 196 in which they correspond to turns pitch-to-coil diameter ratio $(P_t/d)$ of 3.92,1.96,1.33 and 1 respectively. It is seen from the figures the the Nusselt number increases with the decrease of $(P_t/d)$ up to a value between 1.33 and 1 and then decreases with further decrease of $(P_t/d)$, which indicates that the helical-coil turns act not only as an extended surface but also as a boundary layer disturbing means.

Figs. (5,6) shows the relation between Nusselt number and Rayleigh number at a number of helix per meter length $(N_c)$ of 178,357,535 and 833 , corresponding to coil pitch-to-diameter ratio $(P_c/d)$ of 1.098,0.549,0.367 and 0.235 respectively, and a number of turns per meter length $(N_t)$ of 196 corresponding to turns pitch-to-coil diameter ratio $(P_t/d)$ of 1. The figures show that the Nusselt number increases with the decrease of $(P_c/d)$up to a value between 0.367 and 0.235 and then decreases with further decrease of $(P_c/d)$, which supports the finding of Figs (3,4) that the helical-coil turns act as a boundary layer disturbing means as well as an extended surface.

Finally it is shown from Figs (3,5) that the existence of the helical-coil turns augment natural convection heat transfer by about 56 % : 106 % of the bare cylinder.

CORRELATIONS

Finally, an attempt is made to correlate the results obtained in the present study. Such correlations are quite useful from the designer's standpoint. The average Nusselt number is correlated with the other relevant governing parameters, namely Rayleigh numbers $(Ra)$, coil pitch-to-diameter ratio $(P_c/d)$ and turns pitch-to -coil diameter ratio $(P_t/d)$. The following correlations are obtained:

$$Nu = 0.572\,[1+0.611(P_c/d)^{-0.484}\ (P_t/d)^{-0.566}\,]\ Ra^{0.25} \tag{5}$$

for: $2300 \le Ra \le 52000$ , $0.235 < P_c/d \le 1.098$ , $1.33 < P_t/d \le 3.92$ and $D/d = 4.22$

and:

$$Nu = 1.114\ Ra^{0.25} \tag{6}$$

for: $2000 \le Ra \le 5200$, $P_c/d = 0.235$ , $P_t/d = 1$ and $D/d = 4.22$

The correlations (5 and 6) predict the values of average Nusselt number with an error of ∓ 7 % and ∓ 6 % respectively as shown in Figs (7,8).

## CONCLUSIONS

Experimental investigation of a group of helical-coil turns used as a heat transfer augmentative device in the natural convection from a horizontal cylinder was carried out. The following conclusions can be drawn from results of this investigation:

1-The helical-coil turns can increase the natural convection heat transfer coefficient significantly. The ratio of the natural convection heat transfer coefficient with the helical-coil turns to that of the bare cylinder is ranging from 0.56 to 1.06 over the range of Rayleigh number from 4300 to 52000, coil pitch-to-diameter ratio ($P_c/d$) from 0.235 to 1.098, turns pitch-to-coil diameter ratio ($P_t/d$) from 1 to 3.92 and cylinder diameter-to-coil diameter ratio ($D/d$) of 4.22, and correlations are proposed for Nusselt number (Nu).

2-The helical-coil turns around the cylinder acts as a boundary layer disturbing means as well as an extended surface. so the heat transfer is affected by ($P_c/d$) and ($P_t/d$) The heat transfer coefficient increases with the decrease of ($P_c/d$) and ($P_t/d$) to a certain values, where the effect of helical-coil turns as a disturbing means is smaller , and then decreases with further decrease of ($P_c/d$) and ($P_t/d$).

## NOMENCLATURE

$A$ - cylinder outer surface area, [$m^2$]
$D$ - cylinder outside diameter, [m]
$d$ - coil diameter, [m]
$g$ - gravitational acceleration, [$m/s^2$]
$Gr$ - Grashof number, [$g\beta(t_v-t_\infty)D^3/\nu^2$]
$h$ - average heat transfer coefficient, [$W/m^2\ ^\circ C$]
$k$ - thermal conductivity of air, [$W/m\ ^\circ C$]
$N_c$ - number of helix per unit length, [coil/m]
$N_t$ - number of turns per unit length, [turn/m]
$Nu$ - average Nusselt number, [$hD/k$]
$P$ - electric power, [W]
$P_c$ - coil pitch, [m]
$Pr$ - Prandtl number,
$P_t$ - turns pitch, [m]
$Q_{conv}$ - convective heat flux, [W]
$Q_{rad}$ - radiative heat flux, [W]
$Ra$ - Rayleigh number, [$Gr\ Pr$]
$t_v$ - cylinder mean surface temperature, [$^\circ C$]
$t_\infty$ - free stream temperature, [$^\circ C$]

Greek symbols

$\beta$ - coefficient of volumetric thermal expansion, [$1/^\circ K$]
$\nu$ - kinematic viscosity, [$m^2/s$].

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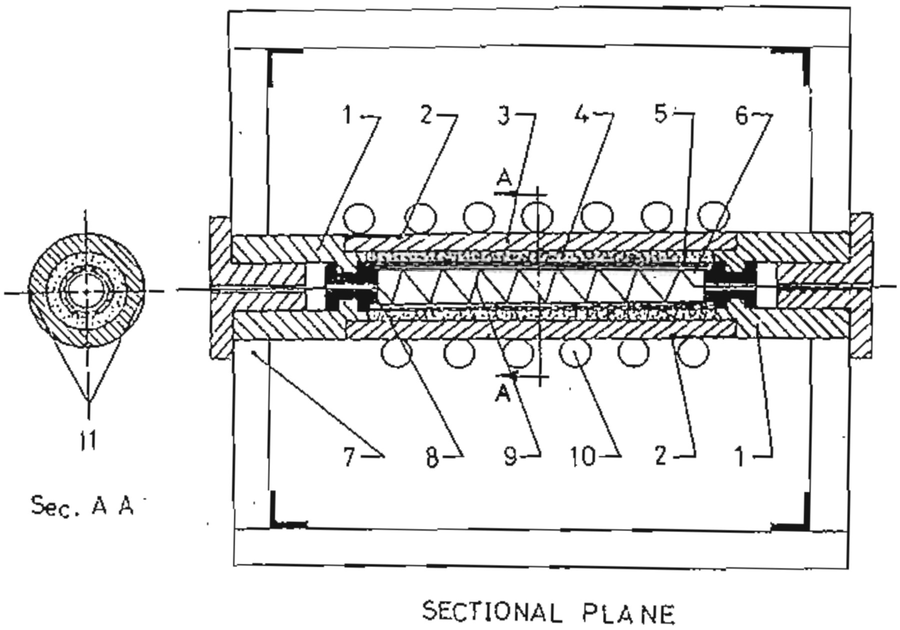


Fig. (1) Experimental Apparatus

| | |
|---|---|
| 1- teflon circular pieces, | 2- thermo-couple, |
| 3- brass test cylinder, | 4- asbestos bowder, |
| 5- mica tape, 6- pyrex glass tube, | 7- side plate, |
| 8- porecelain connection pieces, | 9- nickel-chrome coil, |
| 10-helical-coil-turns, | 11-thermo-couples. |

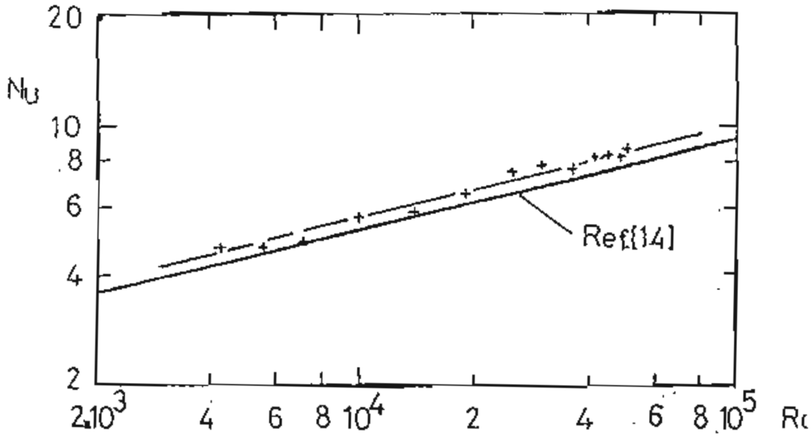


Fig. (2) Varation of average Nusseltnumber with Rayleigh number for bare cylinder.

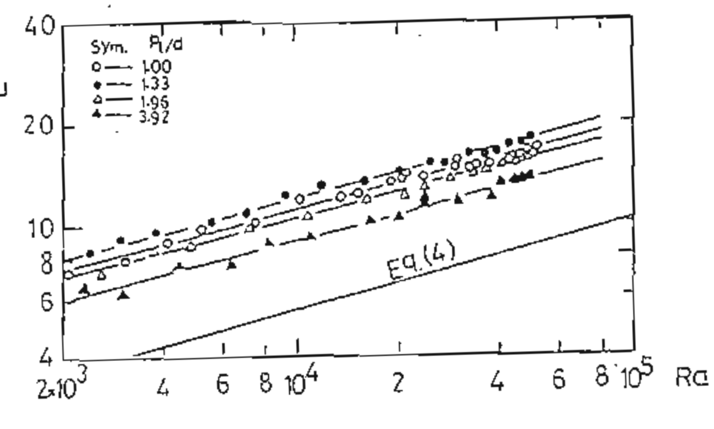


Fig. (3) Variation of average Nusselt number with Rayleigh number at (pc/d) equal to 0.235

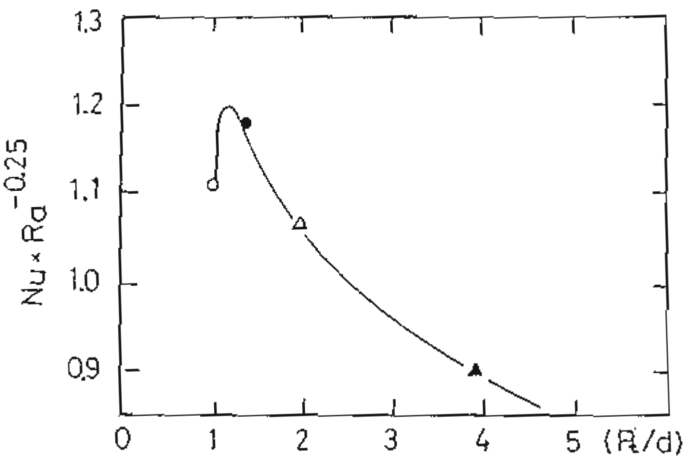


Fig. (4) Effect of (pl/d) on average Nusselt number at (pc/d) equal to 0.235

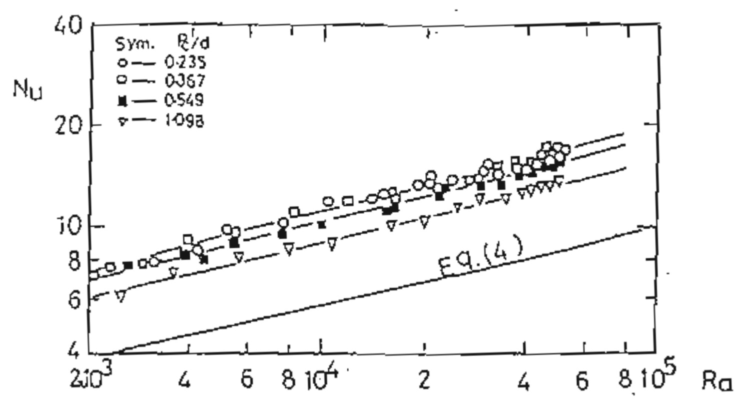


Fig. (5) Variation of average Nusselt number with Rayleigh number at (pl/d) equal to 1.

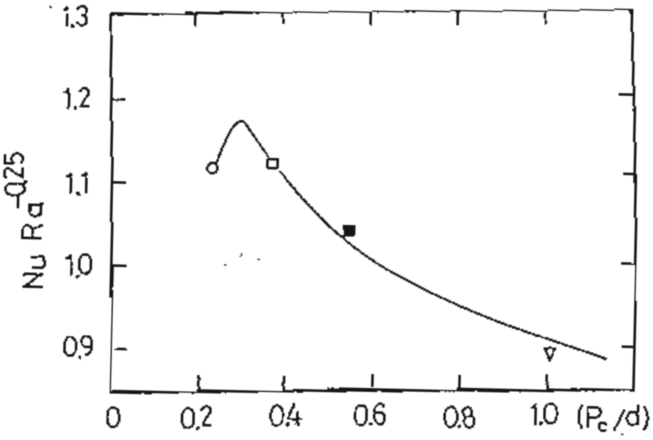


Fig. (6) Effect of (pc/d) on average Nusselt number at (pt/d) equal to 0.235

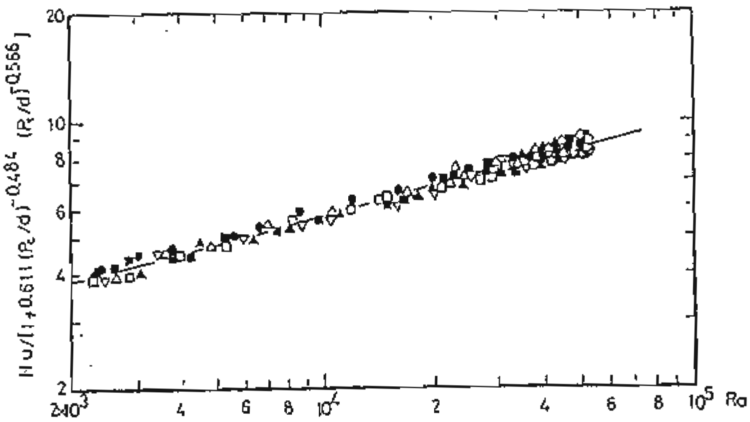


Fig. (7) Average nusselt number correlation for various values of (pt/d) and (pc/d)

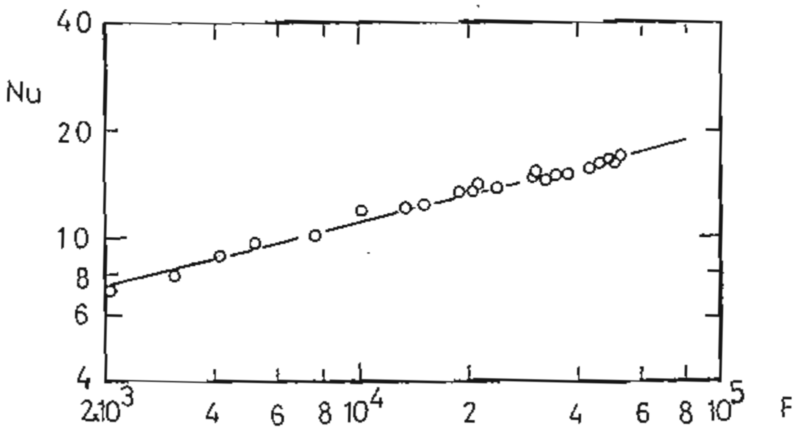


Fig. (8) Average nusselt number correlation at (pc/d) of 0.235 and (pc/d) of 1.