THE EFFECT OF TRIPPING WIRE RINGS ON THE AVERAGE HEAT TRANSFER FROM A CROSS CIRCULAR CYLINDER

تأثير حلقات سلك حلزوني على الانتقال الحراري المتوسط
من اسطوانة دائرية متعرضة

By

A.A. SULTAN, M.A.SHALABY and F.F. ARAID

Mechanical Power Engineering Department,
Faculty of Engineering, Mansoura University, Egypt.

الخلاصة – يتضمن هذا البحث الدراسة العملية لتأثيرات حلقات من سلك حلزوني ملفوفة حول أنبوبة من النحاس الأصفر ساخنة وموضوعة عرضيا مع اتجاه سريان تيار هوائي – وهذه الأنبوبة مثبتة في نفق هوائي حيث يتغير رقم رينولدز من ٧٧٢٠ الى ٥٩٧٠٠ . عدد حلقات السلك الحلزوني الملفوفة حول الانبوبة تتراوح بين ٩ حلقات و ٥٣ حلقة . وقد أوضحت الدراسة أن معامل الانتقال الحراري دالة في رقم رينولدز ونسبة الخطوة القطرية لحلقات السلك الحلزوني الملفوفة حول الانبوبة النحاسية . وقد صيغت النتائج في معادلة تجريبية لامكان حساب معامل الانتقال الحراري المتوسط . وقد أجريت التجارب أيضا على الانبوبة النحاسية بدون وجود حلقات السلك الحلزوني حتى يتمنى اجراء مقارنة توضح مدى تأثير وجود حلقات السلك الحلزوني على معامل الانتقال الحراري المتوسط . ولقد خلصت النتائج الى أنه بزيادة عدد حلقات السلك الحلزوني يزيد معامل الانتقال الحراري المتوسط وتتراوح الزيادة بين ١٣٪ و٦٢٪ .

ABSTRACT - Forced convection heat transfer experiments were provided to study the effects of tripping wire rings on the average heat transfer from a cross circular cylinder. The experimental data have been obtained when the test tube is covered by a number of tripping wire rings as well as the results obtained when the test tube is bare. The number of the tripping wire rings ranges between 0 and 53. The results have been presented in terms of Nusselt number and Reynolds number as well as the pitch-to-diameter ratio of the tripping wire rings. Wind tunnel experiments extended over the range of the free stream Reynolds numbers from 7.19×10^3 to 5.97×10^4. The pitch-to-diameter ratio varies between 0.172 and 1.131. The experiments are carried out for a constant rate of heat flux. The values of the average Nusselt number are fitted in a correlation. The correlation indicates that the Nusselt number is a significant function of Reynolds number and pitch-to-diameter ratio. The augmentation of heat transfer varies from 13% to 62% as the pitch to diameter ratio varies from 1.131 to 0.172.

INTRODUCTION

Tubes operating in crossflow are principal elements of heat exchangers employed in energetics, transportation, chemical and other industries. Continuing improvement of compact heat exchanger designs requires finding new methods for enhancement of heat transfer on such tubes.

With a cylinder in crossflow of a real fluid, a laminar boundary layer is formed on the front part as the result of the viscous forces. It is commonly accepted that, in the lower range of Reynolds number (Re), the cylinder is enveloped all around by a laminar boundary layer, which separates from its surface only at the rear stagnation point. An increase of Re leads to an increase in the effect of inertia forces, so that the laminar boundary layer separates the surface at a certain distance from the rear stagnation point, and a complex vortex structure is formed in the wake. With a further increase of Re, the boundary layer gradually becomes turbulent, and its separation point is shifted upstream. This complex fluid dynamic behaviour is reflected in the heat transfer between the cylinder and the fluid. The two processes have been the subject of both applied and fundamental studies for many years.

Early twentieth century publications on the subject show that there was a general tendency to study the lower range of Re. The earliest studies of heat transfer were performed in air [1-3], but these were soon followed by the first attempts to study flows of liquid [4,5]. Two major directions were covered in this stage: that of the average heat transfer from wires, so significant for thermal anemometry, and of the heat transfer tubes in crossflow.

The second direction of research was closely connected with the progress of boiler design. Along with single cylinder , studies were carried out with arrays, or bundles of cylinders [6].

Recently, the response of the Nusselt number for a circular tube to the angle at which the cylinder is yawed relative to crossflow has been studied by Sparraw and Yanez [7]. Their experiments were performed in air and encompassed ten yaw angles in the range between 0° and 60° and the free stream Reynolds number range between 9000 and 70,000. More recently, Shalaby and Araid [8] studied experimently the forced convection heat transfer from a cross circular yawed tube. Their work was performed in air and encompassed seven yaw angles in the range between 0° and 28° and 6-8 values of the free stream Re between 9000 and 25,000. in addition, supplementary experiments were carried out to study the response of the cylinder heat transfer to alterations of the upstream thermal and fluid flow conditions. As a further supplement to the heat transfer experiments, the pattern of fluid flow adjacent to the cylinder surface was visualized by means of the oil-lampblack technique.

The present experimental work is performed to introduce the average conductance over the entire surface (front and rear), when the air flow is considered normal to the circular cylinder. The experimental data have been obtained when the cylinder is covered by a number of tripping wire rings as well as the data obtained when the cylinder is bare. The number of tripping wire rings ranges between 0 and 53. The reults have been presented in terms of a Nusselt number based on the cylinder diameter and a Reynolds number based on cylinder diameter and thc upstream normal veloeity. The Reynolds number ranges from 7720 to 59700. The tripping wire ring are used around the cylinder to ensure that a turbulent boundary layer prevailed over the outer surface of the heated tube. The heated tube is made from brass of 32 mm outer diameter, 27 mm inside diameter and 300 mm long.

EXPERIMENTAL APPARATUS AND PROCEDURE

An experimental test rig was designed and constructed for the planned experiments, see Fig.(1). It consisted of a low-turbulence open circuit wind tunnel of a 305 mm square cross section, in whieh air from the laboratory room was drawn through the system, by downstream blower (13). The flow rate was controlled by a trottle value (15). The velocity of the air stream drawn through the system was sensed with the help of pitot tube (6) and an inclined alcohol manometer (26) at the centres of nine imaginary equal square areas into which its plane was divided, the pitot tube being situated 50 cm upstream. The air velocity was also measured by the hot-wire probe (5) located 75 cm upstream. The difference in velocity values by the two methods was about $\pm$ 1 %.

Figure (1) shows the general layout of the experimental apparatus with the associated air supply system and the heating tube. Figure (2) shows a schematic diagram of the test tube. A polished brass tube (9) having outer diameter (0) equal to 32 mm, inside diameter (d) equal to 27 mm and 300 long is used. On the brass tube test section, there are eight copper-constantan thermocouples (11) distributed as shown in Fig.(2) at mid length of the test tube. The copper-constantan thermocouples used are made from 30 gauge wires and are fixed in thin slots (1mm deep) cut on the outer surfaee of the tube. Thus the average of eight local temperatures is obtained as follows

$$T_w = [\, T_1 + T_2 + T_3 + T_5 + T_6 + (T_4 + T_8)/2 \,]\,/7 \qquad (1)$$

One may observe that the values obtained by the thermocouples of points 4 and 8 should be close equal to each other, because they are located in two symmetrical postions on the test tube. Thus, the mean value of the two readings is used. During this experimental work the difference between T_4 and T_8 was about $\pm$ 0.04 °C. The tube was heated electricaly by means of the main heater (3).This heater eonsisted of nickel chromium heating wire wound around a thermal brick tube (2) and situated in the test section brass tube as shown in Fig.(2). To prevent the heat loss from the main heater ends two guard heaters (8) were used. The combination of the guard heater is shown in Fig.(2). The guard heater is located in electrically insulated groove made on the test section brass end (5). The heat input to each of the main and guard heaters is controlled by using autotransformers (18), (21) and (28) as well as three voltmeters (19) and (22) and three ammeters (20) and (23),

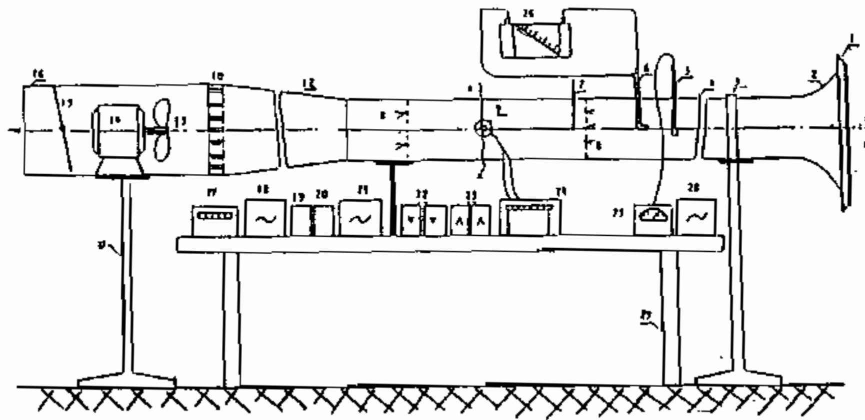
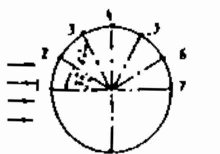


Fig. (1) Experimental Test rig.

1- stablizing plate , 2- inlet collector, 3&17- wind tunnel carriers, 4- stablizing section, 5- thermoanemometer probe, 6- Pitot tube, 7-thermometer, 8- copper-constantan thermocouples, 9- test tube, 10- honycomb, 11-test section duct, 12-connection section, 13-downstream blower, 14 - -electric motor, 15- throttle value, 16- outlet section, 18- main heater autotransformer, 19-main heater voltmeter, 20- main heater ammeter, 21&28- two autotransformers of guard heaters, 22-two voltmeters of guard heaters, 23- two ammeters of guard heaters , 24- temperature recorder, 25- thermoanemometers, 26- inclined tube manometer, 27- digital multimeter, 29-stand table.



Thermocouples position on the midlength of the brass tube test section

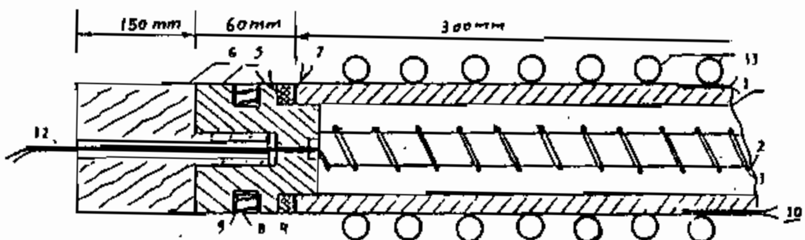


Fig.(2). Tube Test Section.

1- brass tube , 2- thermal brick tube , 3- electrical main heater, 4- tephlon ring, ,5- test section brass end , 6- wooden end , (7 and 11) copper - constantan thermocouples , 8- guard heater , 9- electrical insulation (mica ring) ,10- copper-constantan thermocouples, ,12- electrical heater power supply, 13 - tripping wire rings.

see Fig.(1). A tephlon ring (4), located in between the two ends of the brass main test tube (1) and the brass end of the test tube (5) of outside diameter 32 mm and inside diameter 27 mm and its length is 5 mm. The whole set is placed in between the two wooden ends (6). For a fixed main heater input, the guard heater input is regulated so as to maintain as small a temperature difference as possible less than 0.2 °C (less than 0.01 mv on the digital multimeter (27), Fig.(1)) across the tephlon ring (4), thereby ensuring that the heat flow from the test tube ends is negligible. Twelve-junction thermocouple used for this purpose had three junctions on eaeh side of the tephlon ring (4). The junctions are located at the midheight of the tephlon ring at three locations, each of them has 120° apart from the others. These thermocouples are connected to digital multimeter (27) with an accuracy of 0.001mv. Two wooden ends (6) are fixed at the test section ends. The total length of the test section assembly is 700mm. The heated test tube is covered with some tripping wire rings (13). Each ring is made of 0.8 mm nickel chromium wire diameter, 5.5 mm coil outer diameter and each ring has 70 turns. The experimental work has been made when the heated brass tube is covered by a number of tripping wire rings ranging between 0 and 53. The tripping wires were used around the heated test tube to ensure that a turbulent boundary layer prevailed over the tube surface.

The flow air stream temperatures before and after the test tube, and the wind tunnel surface temperature are measured by the set of thermocouples (8) located as shown in Fig.(1). The test tube assembly is supported in the wind tunnel duct, in which it can be ajusted perpendicular to the main air stream .

In order to find the heat lost by radiation (q_r) the average value of emissivity of 0.03 for polished brass tube is taken from [9], in which they reported that no significant dependence of emissivity on temperature was observed The values of q_r were of order of 0.23% of the input power to the main heater q_{in} neglecting the heat loss by conduction from the ends of the test tube (q_{en}). A steady state was usually achieved after about 3 hours. The average convective heat transfer coefficient was determined from the expression.

$$h = q/A \ (T_w - T_{\infty}) \qquad \ldots (2)$$

The probable error in finding the average heat transfer coefficient was estimated to be about ± 7%. The projected area of the test tube on a vertical-plane perpendicular to the tunnel axis was calculated as follows

$$\acute{A}_p = D \times L \qquad \ldots (3)$$

It is found that the blockage of the wind tunnel free stream cross section area is about 10%. Test and lessmann [10] have reported that their heat transfer results with and without blockage differ by a maximum of 7%. Therefore, the blockage does not affect the heat transfer results to some extent.

RESULTS, DISCUSSION AND CONCLUSIONS

For the determination of the heat transfer coeflcients and their correlation with air flow, some quantities were measured for each data run. The power input to the main heater, the heat lost by radiation, the average tube surface temperature, the air flow stream velocity and the free stream temperature were recorded. The heat lost due to unbalance between the main and guard heaters was neglected because the temperature difference on the sides of the tephlon rings was kept very small (Less than 0.2°C). Net rate of heat transferred by convection was used to calculate the average heat transfer coefficient from equation (2). The typical devlation of any thermocouple reading from the average wall temperature was 0.25% of ($T_w - T_\infty$). Similarly, the three free stream thermocouples were averaged to obtain T_∞, with typical percentage devlations being 0.11% of ($T_w - T_\infty$).

For the Reynolds number, account was taken of the fact that the presence of the cylinder results in the 10% blockage of the cross section of the wind tunnel. The blockage correction for the free stream velocity was based on equation (8a) of Morgan [6] which, when evaluated for the condition of the experiments, yielded

$$U^* = 1.052 \cdot U_\infty \qquad \ldots (4)$$

Then, the free stream Reynolds number followed as

$$R_e = U^*_\infty D/\nu \qquad \ldots (5)$$

The thermophysical properties appearing in equations (2) and (5) were evaluated at a reference temperature $(T_w + T_\infty)/2$.

The parameters varied independently during the course of experiments included the Reynolds number was varied from 7.19×10^3 to 5.97×10^4, the pitch-to-diameter ratio of the tripping wire rings (P/D) from 0.172 to 1.131, the main stream velocity from 4.5 to 30.5 m/sec. The temperature difference between the test tube surface and the oncoming air (Δt) was varied from 28.7 to 110.2°C at a given Reynolds number. In all, 102 data points were obtained and in addition, a number of experiments were repeated.

The results in terms of Nusselt number (Nu) versus Reynolds number (Re) for pitch-to-diameter ratios equal to 0.172, o.253, 0.300, 0.566 and 1.131 are shown in Fig.(3) for q = 250. W$\pm$ 1W and in Fig.(4) for T_w=76$\pm$ 0.2 °C.

The experiments performed to establish the generality of the results will now be described. The first set of experiments was carried out on the cylinder in cross flow without tripping wire rings. The cross flow Nusselt numbers are plotted in Fig.(3) as a function of the free stream Reynolds number Re. Also appearing in the figure is a curve representing the correlation of Zhukouskas (28) in Ref.[11]. Inspeetion of Fig.(3) shows that the Nusselt number value increases, in general, with Reynolds number. The experimental data obtained for the bare tube are found in a quite good agreement with the same obtained from correlation (28) of Ref.[11].

About the response of Nusselt number to the tripping wire rings surrounding the test tube, one may observe that the Nusselt numbers for each of the five investigated test tubes are plotted as a function of the free stream

Reynolds number in Figs. (3and4). From an overview of the figures, it is seen that Nusselt numbers increase,in general with the tripping wire rings number, and this increase is ordered in a regular manner (i.e. as the number of the tripping wire rings increases the Nusselt number value increases). One may also observe that. Nusselt number values increase, in general, with the free stream Reynolds number. The results displayed on the Fig.(3) are obtained at q = 250 W $\pm$ 1 W and for pitch-to-diameter ratios (P/D) equal to 0.172, 0.253, 0.3, 0.566 and 1.131, in which they correspond to the tripping wire rings number (N) equal to 9 , 17 , 31 , 37 and 53. On the other hand the results displayed on Fig.(4) are obtained at T_w = 76°C. During the course of this work the power input to the heated test tube was continuously readjusted (by the main heater autotransformer (18), in Fig.(1)) inorder to keep the average wall temperature constant and equal to 76°C $\pm$ 0.2°C. From Figs. (3and4) one may observe that the augmentation of heat transfer ranges between 13% to 62% data obtained for the case of the bare tube.

Fig.(5) shows the variation of Nusselt number versus pitch-to-diameter ratio (P/D). One may observe that the Nu value decreases as the pitch-to-diameter ratio increases to approach the data obtained for the bare tube.

Fig.(6) shows the variation of Nusselt number versus the pitch-to-diameter ratio for Reynolds number values equal to 13.5×10^3, 27×10^3 and 51×10^3 at q = 250 W $\pm$ 1 W. The results are displayed on log-log sheet, in which one may observe that the Nusselt number values increase with Reynolds number. It is also seen that the value of Nu decreases with the pitch-to-diameter ratio (P/D).

During the present experimental work, the tripping wire rings were not welded on the heated test tube surface. They were only fixed around the test tube under the triping wire strength. So that, the tripping wire rings around the test tube are considered as boundary layer disturbing means, not as an extended surface of the test tube.

CORRELATION

Finally, an attempt is made to correlate the results obtained in the present study. Such a correlation is quite useful from a designer's standpoint. The average Nusselt number is correlated with the other relevant governing parameters of the test tube, namely Reynolds number (Re) and pitch to diameter ratio (P/D). The following correlation is obtained.

$$Nu = 0.284\ R_e^{0.6}\ (P/D)^{-0.19} \qquad \ldots (6)$$

$$7.19\times10^3 \leqslant R_e \leqslant 5.97\times10^4$$
$$0.172 \leqslant P/D \leqslant 1.131$$

The correlation (6) predicts the values of the Nu which agree with results to within $\pm$5%, as shown in Fig.(7).

In correlation (6), the exponent of 0.6 on R_e clearly suggests the presence of a turbulent boundary layer. The equation shows also that the Nusselt number is a significant function of pitch to diameter ratio.

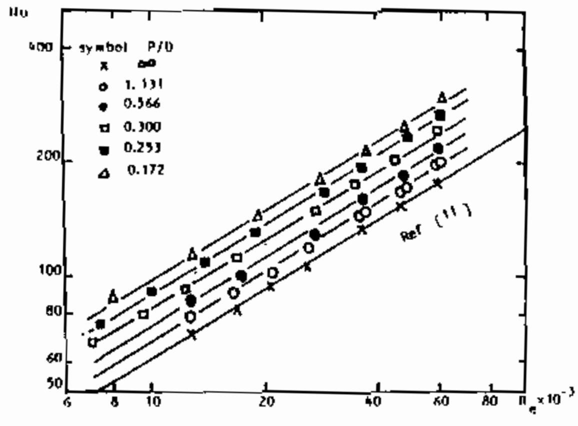


Fig. (3) The variation of Nusselt number versus free stream Reynolds number at Q = 250.5 W

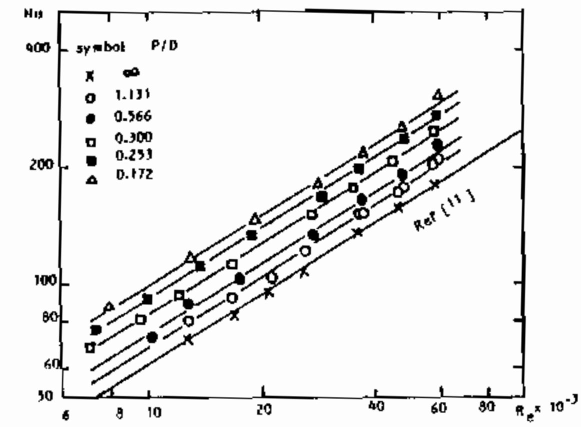


Fig. (4) The variation of Nusselt number versus free stream Reymolds number at T_w = 76 °C

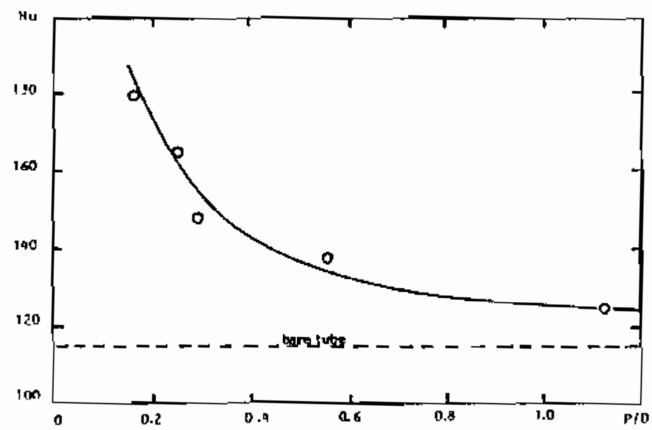


Fig.(5) The variation of Nusselt number versus pitch to diameter ratio at Re = 33500

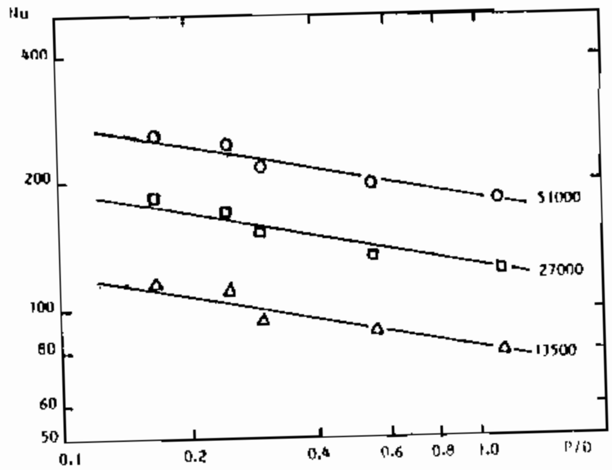


Fig.(6) The variation of Nusselt number versus the pitch to diameter ratio at Q=250 ± 1 W

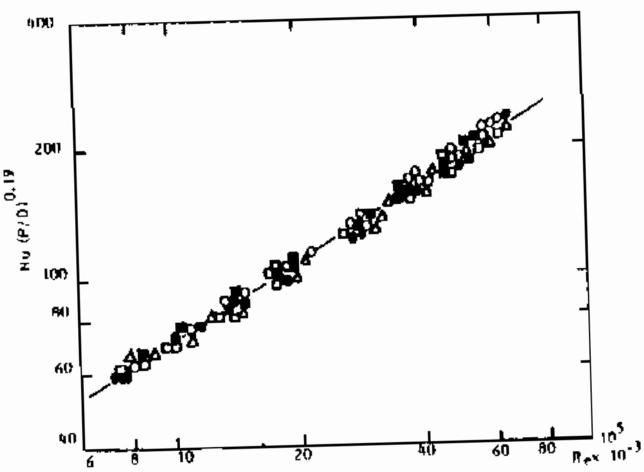


Fig. (7) Correlation between Re and Nu $(P/D)^{0.19}$

NOMENCLATURE

A	tube surface area, [m^2]
D	cylinder outside diameter, [m]
d	cylinder inside diameter, [m]
h	heat transfer coefficient, [$W/m^2.°C$]
k	thermal conductivity, [W/m.°C]
Nu	Nusselt number, [h D/K]
p	pitch of tripping wire rings, [m]
q	rate of heat transfer at cylinder test section, [W]
Re	free stream Reynolds number, [$U_\infty D/\nu$]
T_w	cylinder wall temperature, [°C]
T_∞	Free stream temperature, [m/s]
U_∞	Free stream veloeity, [m/s]
U^*	corrected free stream velocity, [m/s].

Greek symbols

ν	kinematic viscousity, [m^2/s]

REFERENCES

1- Kenelly, A.E., Wright, C.A., and Van Bylevet, I.S. "The Convection of heat from Small Copper Wires" Trans. Am. Inst. Elect. Eng. Vol.28, PP. 363-393, (1909).

2- King, L.W., "On the Convection of Heat from Cylinder in a stream of fluid". Phil. Trans. R. Soc. A21A, PP. 374-443, (1914).

3- Hughes, J.A. "On the cooling of eylinders in a stream of Air". Phil. Mag. Vol. 31, P. 181, (1916).

4- Worthington, H.V., and Malone, C.B. "Convection of Heat from Small Wires in water". J. Franklin Inst., Vol.184, P. 115, (1917).

5- Davis, A.H. "Convective Cooling of Wires in Streams Viscous Liquids". Phil. Mag. Vol. 47, P. 282, (1924).

6- Zukauskas, A.A. "Heat Transfer from Tubes in cross-flow". In Advances in Heat Transfer"., New York Academic Press. Vol.8, PP. 93-160, (1972).

7- Sparraw, E.M. and Yanez, A.A., "Effect of Yaw on Forced Conveetion Heat transfer from a circular cylinder"., Int. J. Heat & Mass Transfer. Vol. 30, No.3, PP. 427-435, (1987).

8- Shalaby, M.A. and Araid, F.F., "Forced Convection Heat Transfer from a Cross Circular Yawed Tube"., MEJ, Vol. 15, No.1, June (1990).

9- Eckert, E.R.G. and Robert M. Drake, JR., "Analysis of Heat and Mass Transfer"., McGraw-Hill Kogakusha, LTD., (1972).

10- Test, F.L., and Lessmann, R.C. "An Experimental Study of Heat Transfer During Foreed Convection over a rectangular body"., ASME Jr. of Heat Transfer, Vol. 102, PP. 146-151, (1980).

11- Mikheyev, M. and Mikheyeva, I., Fundamentals of Heat Transfer, Inergia, Moscow, P. 97 (1973).