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Friction and Forced Convection Heat Transfer Characteristics in Tubes by Use of Helical-Wire-Coil Inserts.

Ahmed Sultan
Lecturer at Mechanical Power Engineering Department., Faculty of Engineering., El-Mansoura University., Mansoura., Egypt.

FRICTION AND FORCED CONVECTION HEAT TRANSFER CHARACTERISTICS IN TUBES BY USE OF HELICAL-WIRE-COIL INSERTS.

خصائص الأحتكاك و انتقال الحراره بالحمل الجبرى
في الأنابيب باستخدام ولائج ملف سلك حلزوني

By

SULTAN A. A.
Mechanical Engineering Department,
Faculty of Engineering, Mansoura
University, Mansoura, Egypt.

الخلاصه – يتضمن هذا البحث دراسه عمليه لانتقال الحراره بالحمل الجبرى و مفاقيد الضغط الناتج عن الاحتكاك لتيار هوائي يمر داخل أنبوبه نحاسيه أفقيه دائريه المقطع قطرها الداخلي ١٣,٥ مم ذات ولائج من السلك الحلزوني، قطر السلك ١,٤ مم و زاويه الحلزون تتراوح ما بين ٦٥° الى ٨٦,٧° (عدد لفات السلك الحلزوني تتراوح ما بين ٧١٤ الي ٨٩ لفه / دقيقه)، الماسوره النحاسيه مسخنه ببخار تحت الضغط الجوى و تتغير سرعه الهواء داخلها ليعطي رقم رينولدز من ٦٠٠٠ الى ١٢٠٠٠٠ - اوضحت التجارب زياده معامل انتقال الحراره باستخدام ولائج السلك الحلزوني و تتراوح الزياده بين ٣٠% و ١٤٥% مع زياده مقابله في معامل الاحتكاك، اوضحت النتائج ايضا ان معاملي انتقال الحراره و الاحتكاك داله في رقم رينولدز و زاويه الحلزون.

ABSTRACT

This work summarizes the results of experimental investigation of convective heat transfer and friction loss characteristics of air flow in a horizontal circular copper tube of diameters 13.5/22 mm., using helical wire-coils ,of diameter 1.4 mm , inserted in. The helix angle (α) of the helical-wire-coil is varied from 65° to 86.7°. The outer surface of the test tube is heated by superheated steam (degree of superheat 2:3 °C) at atmospheric pressure. During the experimental work, Reynold's number is changed from 6000 to 120000 and Prandtl number is about 0.7. The heat transfer enhancement of the inserted helical-wire-coil is ranging from 30 % to 145 %. However, A quantitatively increase in hydrodynamics resistance is noticed. Correlations of Nusselt number and friction factor has been made. It was found that the nusselt number and friction factor were dependent on both Reynold's number and helix angle.

INTRODUCTION

The trend in heat exchanger design continues to be in the direction of higher heat transfer rates per unit volume. In the usual commercial fluid-to-fluid heat exchangers, there is an abvious economic incentive to reduce equipment size. This accomplished by introduction of more surface area and/or augmenting heat transfer coefficients. Other devices, such as microwave power tubes, high field electromagnets, and electron accelerator targets, involve high heat generation rates which must be dissipated through relatively small surface areas. In these high heat flux systems, the surface-to-fluid temperature differences must be kept moderate to avoid melting or other structural failure of the heated surface. These situations have simulated interest in development of techniques to augment or enhance heat transfer.

A variety of argumentative techniques (surface treatment , displaced promoters, vortix flow, surface vibration, fluid vibration, electrostatic fields, and additives) has been investigated. The use of rough surfaces is one of several enhancment techniques reported by Bergles [1], Dipprey and Sabersky [2] and Nikuradse [3], through which it is possible to achieve a two fold objections of obtaining the maximum heat transfer rates with a minimum frictional pressure drop. These devices can be employed either to increase the heat transfer rate or to reduce the pumping power or heat transfer area. Considerable work has been also reported on turbulent promoters, such as transverse rib-roughned tubes [4-6], spiraly corrugated tubes [7,8] and converging diverging tubes [9-10]. However, very limited work has been published on the thermohydraulic performance of helical wire inserted tubes, especially for convective heat transfer applications. Sethumadhavan and Rao [11] investigated experimentally the heat transfer in a 25 mm inside diameter copper tube fitted with helical-wire-coil inserts of varying pitch (P), helix angle (α) and wire diameter (d). They correlated their results in an imperical formula and they made an optimization study on the basis of maximization of the heat transfer rate and also minimization of pumping power and heat exchanger frontal area to identify the most efficient tube within the matrix of data. Sethumadhavan and Rao [11] study gave a considerable increase in heat transfer rate without a significant increase in friction power, with water and 50 % glycerol as working fluid and hot water as heating medium. Nag and Rao [12] studied the friction and heat transfer performances, and developed a suitable correlations for momentum and heat transfer roughness functions, based on friction and heat transfer similarity laws. The working fluid in their study was R-12 and the test tube was heated electrically. The experimental studies of Uttrawar and Raja Rao [13] have been carried out on isothermal pressure drop and convective heat transfer to servotherm medium grade oil in laminar flow in seven wire-coil inserted tubes of varying wire diameter and pitch of wire coil. Their results indicated that as much as four fold improvement can be obtained in laminar flow heat transfer coefficient using these tubes.

From the review of literature, one may observe that most studies of flow inside horizontal tubes, using helical-wire-coil inserts, have been heat transfer and friction measurements for high Prandtl number fluids ($5.2 < Pr < 32$) heated electrically or using hot water. The helix angle of the helical-wire-coil inserts was in the range $30^{\circ} < \alpha < 75^{\circ}$.

The present work was therefore undertaken:

(i) To obtain turbulent flow heat transfer and friction results for air (Pr nearly equal to 0.7) flowing through helical-wire-coil inserted tubes and to develop correlation for predicting heat transfer and friction coefficients for helix angle in the range $65^{\circ} \leq \alpha \leq 86.7^{\circ}$.

(ii) To compare the heat transfer and pressure drop performances of helical-wire-coil inserted tubes with the performance of the smooth tube under similar operating conditions.

EXPERIMENTAL APPARATUS AND OPERATING PROCEDURE

Helical-wire-coil inserts were developed for testing in circular tube test sections. The tubes are of copper , with 13.5 mm inside diameter and 22 mm outside diameter. The helical-wire-coil inserts are prepared from commercially available copper wires, which is used in the electrical transformers, of 1.4 mm in diameter. This study concerns with the determination of heat transfer rates and friction losses on eight helical-wire-coils of pitch-to-wire diameter (P/d) varies from 1.0 to 8.0 .The characteristic parameters, which define the roughness geometry of the eight helical wire-coil inserted tubes are given in Table 1.

Table 1 : Characteristic dimensions of helical-wire-coils

P (mm)	1.40	1.96	2.52	2.80	4.20	5.60	8.40	11.20
P/d	1.0	1.4	1.8	2.0	3.0	4.0	6.0	8.0
α (deg)	86.7	85.4	84.0	83.4	80.2	78.0	70.9	65.2
T/m	714	510	397	357	238	178	119	89

One smooth tube was used to standardize the experimental set-up and also to evaluate the increase in tube side heat transfer coefficient and friction factor in the eight helical-wire-coil inserted tubes, relative to the smooth one.

Fig. (1) shows a schematic diagram of the experimental test-rig. The actual test section consisted of a 980 mm long double pipe heat exchanger, the inner tube of which was either the smooth tube or helical-wire--coil inserted tubes under test. The outer tube of the test rig is 50 mm inside diameter made of galvanized iron pipe, was fitted with a tube of an insulated material and the set was insulated with a glass wool of 50 mm thickness, having two ports one of them was for the inlet superheated steam and the other was for the outlet of the condensate.

The experimental test rig fig. (1) was an open loop, the air which is coming from the laboratory room was drawn through the system by a downstream blower (13).The flow rate was controlled by a valve (2) and measured by a caliparated orifice meter (21). The pressure drop through the test tube (20) was measured by a pressure transducer (5) connected to digital readout (6). Slightly superheated steam at atmospheric pressure was used as a heating medium in the annulus of the heat exchanger so as to give constant metal wall temperature. The steam was generated in an electric boiler (17) and superheated by means of an electric superheater (11). The steam flowed from the boiler to the heat exchanger (27) through a steam line, thermally insulated by glass wool. The condensate was discharged in a collector (9) in order to be able to calculate the heat rejected from the steam , i.e the heat added to the air flow . The temperatures of the air flow at the test section inlet and outlet were measured by a 0.25 mm diameter copper-constantan thermocouples (18,23). The temperatures of the inner surface of the test tube were measured by thermocouples at four points (T), distributed along the tube axis. The used thermocouples connected to a 12-point self switching temperature recorder, having a full scale of 200 °C. In order to measure the pressure and temperature in the boiler (17), a calibrated pressure gauge (16) and a mercury thermometer (12) with scale divisions of 0.1 °C, wereused. An augmentation coil (22) is inserted inside the test tube (20) as shown in Fig.(1-b). The augmentation coil have pitch-to-coil diameter ratio (P/d) ranges between 1 to 8,.corresponding to a range of helix angle (α), which is varied between 86.7° to 65°.

There were two unheated straight tubes (8,3) , each of length 750 mm. These tubes were located before and after the test section (4), in order to stabilize the fluid flow. The two stabilizing tubes were connected to the test tube by two teflon flanges (24), in order to avoid the back conduction effect.

The air flow velocity in the test tube was calculated on the basis of the bare tube diameter. The physical properties of air were taken at the arithmetic mean flow stream temperature. The heat gained by the working fluid was calculated from the enthalpy change of the air and checked by the enthalpy change of the condensed steam. The difference between the two quantities was about 5 %.

RESULTS AND DISCUSSION

Friction Factor The fanning friction factor was calculated from the following equation:

$$f = \frac{\Delta p}{(L/D)\ (\rho v^2/2)} \qquad (1)$$

The nonisothermal friction factor data were obtained for Re= 6000 : 120000 and Pr nearly equal to 0.7, for both the smooth tube and the eight helical-wire-coil inserted tubes. The smooth tube nonisothermal friction factor data for the turbulent flow of air agrees, within $\pm$ 8 %, with the isothermal friction factor given by Blasius [14], as shown in Fig.(2).

The turbulent flow friction factors for the helical-wire-coil inserted tubes were found to increase substantially when these tubes replaced the smooth tube.Fig. (2) shows the variation of (f) with (Re) for the eight helical-wire-coil inserted tubes and the smooth one. It is clear from the figure that the increase in friction factor ranges from 50 % for tube 1 (fitted with a helical-wire-coil of pitch 1.4 mm), to as high as 250 % for tube 4 (fitted with helical-wire-coil of 2.8 mm pitch) compared to the smooth tube. The other tubes produced friction factor enhancment intermediate between 50 and 250 %.This order of increase in the friction factor was also reported earlier by Sethumadhavan and Raja Rao [11].

Fig.(3) shows the relation between coil pitch to diameter ratio (p/d) , and friction factor (f), it is shown from the figure that, as the value of p/d increases the friction factor increases rapidly and reaches a maximum value nearly at p/d = 2 (α = 83.4°) and then slightly decreases with further increase in p/d . The friction factor variation may be due to the effect of increase of surface area, the effect of increase of the disturbance in the main core of the flow and the effect of swirl flow generated in both the main core of the flow and the laminar sub layer . The combination of these effects explains the above mentioned variation of friction factor with (p/d).

Heat Transfer. The tube side heat transfer coefficient was calculated by the following equation:

$$Q = h\ A\ (\bar{T}_w - \bar{T}_f) \qquad (2)$$

and an average of inlet and outlet bulk temperatures was used to calculate the property values of test fluid.

The tube side heat transfer coefficient h , calculated in terms of Nusselt number (h D /k), was plotted against Reynolds number in Fig. (4) for the smooth tube and the eight helical-wire-coil inserted tubes for turbulent flow of air. The smooth tube experimental values of Nusselt number for turbulent flow of air agrees, within $\pm$ 6 %, with the Mekheyev [15] heat transfer equation for turbulent flow of air.

$$Nu=0.018(Re)^{0.8} \tag{3}$$

From fig. (4), it can be seen that the enhancement in Nusselt number for air in case of helical-wire-coil inserted tubes was between 30 to 145 percent, depending on the coil pitch-to-diameter ratio (p/d) (i.e. helix angle α). The increase in heat transfer coefficient can not only be accounted for by the temperature profile effect in the wall region due to the location of the helical-wire-coil inserts relative to the wall where laminar sub layer exists, but also for the increase in heat transfer surface area.It is evident from the Fig. (4) that, the Nusselt number for the helical-wire-coil inserted tube is a function of Reynolds number and coil pitch-to-diameter ratio (p/d).

Fig.(5) shows the relation between Nusselt number and the pitch-to-diameter ratio (p/d). From the figure one may observe that, as the value of (p/d) decreases Nusselt number increases up to a value of (p/d) equal to 3 (corresponding to an (α) equal to 80.2^{o}) and then decreases with further decrease in (p/d).

The basic data for the Nusselt number and friction factor that have been presented here can be used as input to the computations of the enhancement characteristics of swirl affected pipe flows. Such enhancement evaluations may be performed for a wide variety of constrains (e.g. fixed heat duty, fixed mass flow rate, fixed transfer surface area, fixed pumping power, etc.). The actual execution of the enhancement evaluations is beyond the scope of this paper.

The present heat transfer and friction factor results could not be compared with those of the available literatures, because it was not possible to find a common ground for the characterization of the swirl.

CORRELATIONS

An attempt was made to correlate the results obtained in the present study, such a correlation is quite useful from the designers stand point. The average Nusselt number and the friction factor were correlated with the other relevant governing parameters, namely Reynolds number (Re) and Helix angle (α). The following correlations were obtained:

Heat transfer correlation.

$$Nu = C1\ Re^{0.8} \tag{4}$$

where:

$60\ 00 \le Re\ \ 120000, \quad 65^\circ \le \alpha \le 86.7^\circ \quad$ and $d/D = 0.138$

The above correlation predicts the values of Nu which agrees with results within $\pm$ 6 % .

Friction Factor Correlation.

$$f = C2\ Re^{-0.205} \tag{5}$$

where:

$6000 \le Re \le 120000, \quad 65^\circ \le \alpha \le 86.7^\circ, \quad$ and $d/D=0.139$

The above correlation predicts the value of friction factor (f) which agrees with the experimental results within $\pm$ 8 %.

The factors C1 and C2 of equations (4) and (5) for various helix angle (α) are plotted in Figs. (5) and (3) and listed in table (2).

Table (2) Values of factors C1 and C2

P/d	1.0	1.4	1.8	2.0	3.0	4.0	6.0	8.0
C1x100	2.31	2.99	3.60	3.85	4.06	3.35	2.84	2.69
C2x10	2.85	4.82	5.94	6.22	5.63	5.16	4.62	4.51

CONCLUSIONS

Experimental investigation of a group of helical-wire-coils used as heat transfer augmentative device in the heating of air flow was carried out. The following conclusions can be drawn from the results of this investigation:

1- The helical-wire-coil inserts can increase the tube side heat transfer coefficient significantly. Nusselt number increases with the increase of helix angle (α) up to a value of 83.4° (corresponding to p/d of 3) and then decreases with further increase in (α).

2- The ratio of the heat transfer coefficient of the air flow in a helical-wire-coil inserted tubes to that of the bare tube was ranging from 1.3 to 2.45 in the range of Reynolds number between 6000 and 120000 and helix angle between 65° and 86.7°.

3- The friction factor increases with the increase of helix angle up to a value of (α) of 80.2° (corresponding to p/d of 2) and then decreases with further increase in (α).

4- The ratio of the friction factor for the helical-wire-coil inserted tubes to that of the bare one was ranging from 1.5 to 2.5 in the range of Reynolds number between 6000 and 120000 and helix angle between 65° and 86.7°.

NOMENCLATURE

C1 factor defined by equation (4),
C2 factor defined by equation (5),
Cp specific heat of fluid flow at constant pressure, [kJ /kg °C]
D inside diameter of the bare tube, [m]
d coil wire diameter, [m]
h tube inside heat transfer coefficient, [W /m^2 K]
k thermal conductivity, [W /m k]
L tube length, [m]
Nu Nusselt number base on bare tube inside diameter, [hD/k]
p/d coil pitch-to-diameter ratio,
Pr Prandtl number, [μCp/k]
Re Reynolds number based on bare tube inside diameter, [vD/ν]
T temperature, [°C]
T/m number of turns per meter, [1/m].

Greek symbols

α helix angle, [Degree]
μ dynamic viscosity, [N m / S^2]
ν kinematic viscosity, [m^2 /s].

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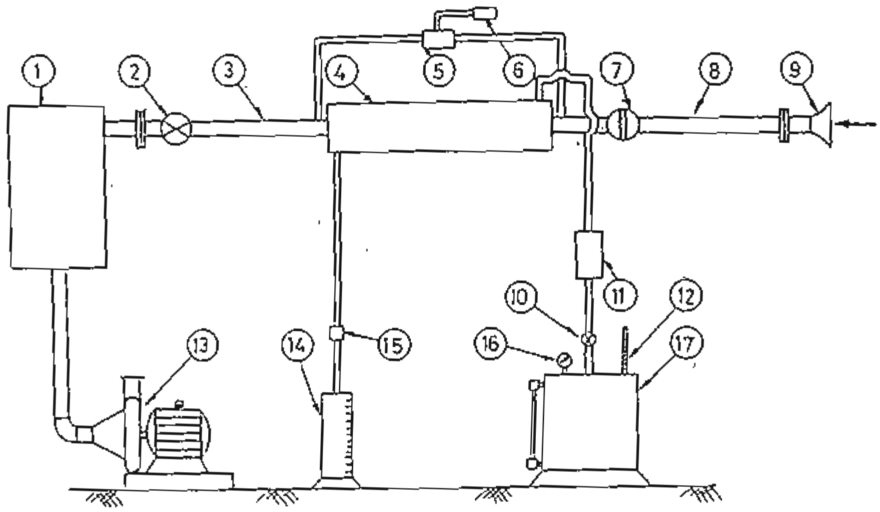
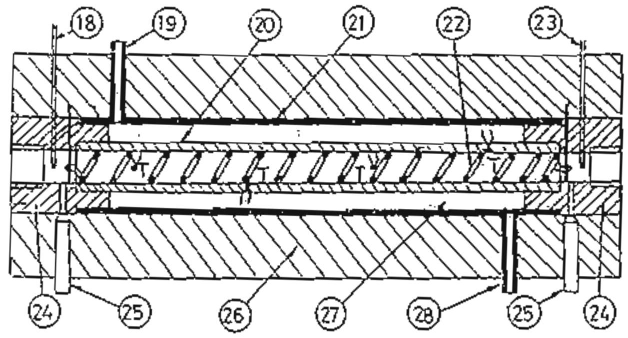


A)-Schematic diagram



B)-Test tube assembly

Fig. (1) Experimental Test Rig.

1- damping chamber, 2- air control valve, 3- outlet stabilizing tube, 4- test section, 5- pressure transducer, 6- digital readout, 7- orifice meter, 8 -inlet stabilizing tube, 9- collector, 10- steam control valve, 11- superheater, 12- thermometer, 13- air blower, 14- condensate collector, 15- steam trap, 16- pressure gauge, 17- electric boiler, 18,23 and T- thermocouples, 19- steam inlet, 20-test tube, 21- galvanized iron pipe, 22- augmentation coil, 24- teflon flange, 25- pressure tap, 26- glass wool insulation, 27- heat exchanger, 28- condensate outlet.

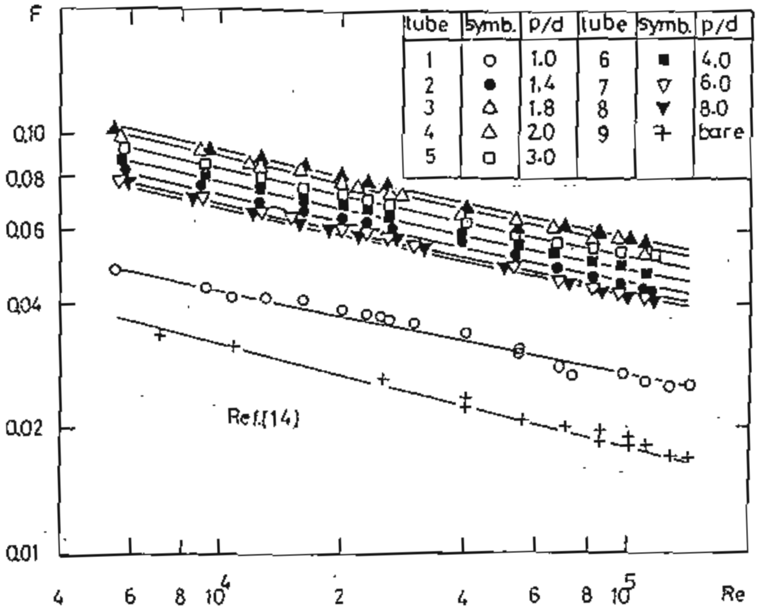


Fig. (2) Relation between friction factor (f) and Reynold's number (Re) for different coil pitch-to-diameter ratio (p/d).

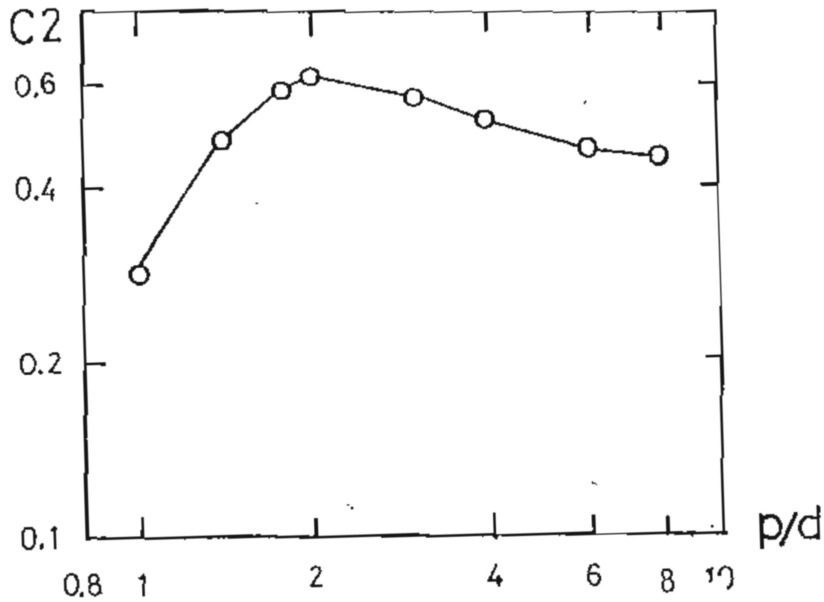


Fig. (3) Relation between factof C2 ($f/Re^{-0.205}$) and coil pitch-to-diameter ratio (p/d).

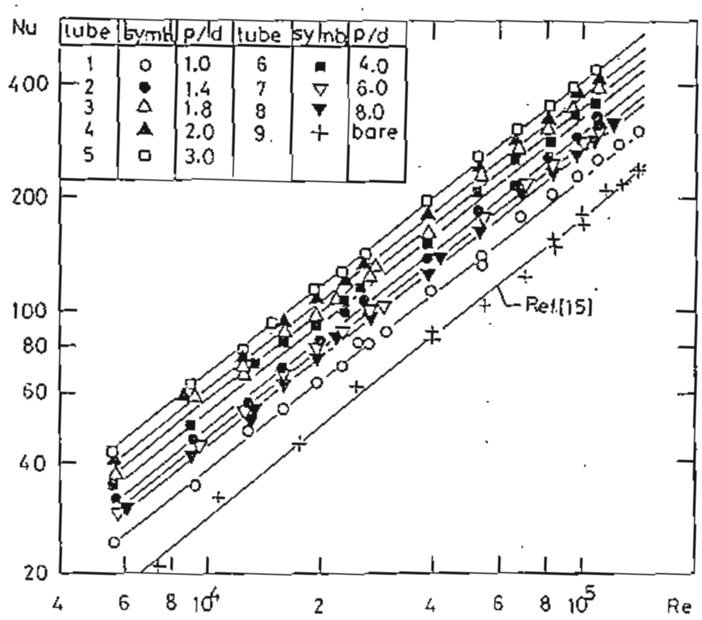


Fig. (4) Relation between average Nusselt number (Nu) and Reynold's number for different coil pitch-to-diameter ratio (p/d).

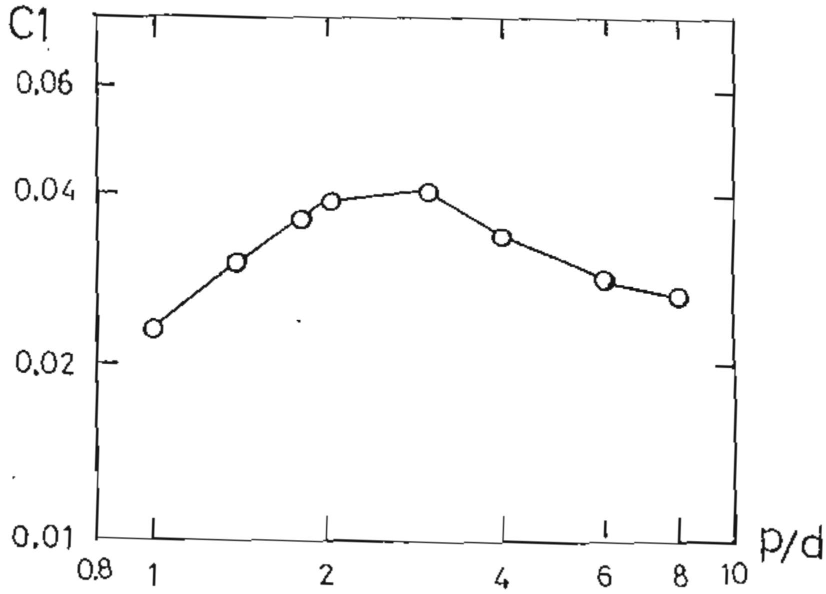


Fig. (5) Relation between factor C1 ($Nu/Re^{0.8}$) and coil pitch-to-diameter ratio (p/d).