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Augmentation of laminar flow heat transfer in annular flat tubes by means of helical screw-tape inserts in the inner tube

تحسين انتقال الحرارة لسريان رقائقي خلال مواسير مزدوجة مسطحه باستخدام ولائج من شريط حلزوني بأبعاد مختلفة داخل الماسورة الداخلية

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ملخص البحث:

تم في هذا البحث دراسة خصائص انتقال الحرارة ومعامل الاحتكاك داخل ماسورتين مسطحتين داخل بعض مع وضع شرائط حلزونية ذات نسبة لي مختلفة (طول من لية الواحدة إلى قطر اللية) وأيضا استخدام مسافات طولية بين اللي في كل حاله على حده داخل الماسورة الداخلية بأبعاد مختلفة . لقد استخدم ماء بارد وماء ساخن كموائع وسيطة في الماسورة الداخلية والخارجية بالترتيب . التجارب غطت مدى لرقم رينولدز من ١٣٠ إلى ٥٧٠. تم توضيح تغيير خصائص الشريط الحلزوني على انتقال الحرارة ومعامل الاحتكاك .

أظهرت النتائج أن معامل انتقال الحرارة ومعامل الاحتكاك تنقص بزيادة S و Y .ولقد تم مقارنة المواسير المسطحة مع المواسير الدائرية وكذلك تمت المقارنة مع الأعمال السابقة للمواسير الدائرية . ولقد وضعت النتائج في صورة لابعديه وقد تم استنتاج معادلة لهذة النتائج .

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ABSTRACT

The heat transfer characteristics and friction factor in horizontal double pipes of flat tubes with full length helical screw element of different twist ratio and helical screw inserts with different spacer length are investigated. Cold and hot water are used as working fluid in tube side and shell side respectively. The experiments covered a range of Reynolds numbers $5.7*10^2 \leq Re \leq 1.31*10^3$. The effect of spacer length and twist ratio on the heat transfer augmentation and friction factor on heat transfer have been presented separately.

The study shows that, Nusslet number (Nu) and friction factor (f) decrease with the increase of spacer length (S) and twist ratio (Y) for flat tube. The comparison between the data of present plain circular tube with that of previous plain circular tubes showed good agreement but the data of present plain flat tube showed a higher in heat transfer and pressure drop than that of plain circular tube. The correlations of average Nusselt number and friction factor with Re, S and Y are presented.

Keywords: Heat transfer, laminar flow, flat tubes, helical screw-tape inserts

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NOMENCLATURE

A_t: Tube cross section area, m^2

C_p: Specific heat capacity at constant pressure, KJ /Kg K

d : Diameter of the rod , mm

D_{eq}: Equivalent diameter, m

L: Length of tube, m

h: Heat transfer coefficient, Kw/m^2k

k: Thermal conductivity, Kw/mk

m: Mass flow rate, Kg/s

Nu: Nusselt number.

ΔP: Pressure drop, N/m^2

Pr: Prandtl number

V: Volume of water, m^3

Q: Heat transfer, Kw

Re: Reynolds number.

T: Temperature, K

Δt_m: Logarithmic mean temperature, K

S: Spacer length, mm

v: Velocity of water inside tube, m/s

y: Twist ratio.

ν: Kinematics viscosity of water, m/s^2

μ_a: Viscosity of fluid at bulk temperature , Kg/m.s

μ_b: Viscosity of fluid at wall temperature, Kg/m.s

ρ: Density of water, Kg/m^3

f: Friction factor

x: The ratio between S/d.

Subscripts

c: cold

h: hot

i: inside

o: outside

s: surface

w: water

INTRODUCTION

The technique of improving the performance of heat transfer system is referred to as heat transfer augmentation or intensification. This leads to reduce the size and cost of the heat exchanger. Among many types of heat exchangers, those constructed circular pipes have been used in many industries. In such a situation, the cross section of a pipe becomes a flat tube.

Heat transfer enhance technology has been developed and widely applied to heat exchanger applications; for example, refrigeration, automotives, process industry, chemical industry etc. Heat augmentation techniques play a vital role for laminar flow, since the heat transfer coefficient is generally low in plain tubes. Insertion of twisted tapes in tubes is one such augmentation technique. These details are discussed by Bergles [1,2] .

In the compound technique, two or more of active or passive techniques may be utilized simultaneously to produce an enhancement that is much higher than that of the techniques operating separately as presented by Yilmaz et al [3 and 4]. Ray and Date [5] studied numerically characteristics of laminar flow and heat transfer through square duct with twisted tape insert. The results showed that, for a fixed Reynolds number, the friction factor and Nusslet number increases with the decrease in twist ratio.

Rooyen and Kroeger [6] observed that, for laminar swirl flow heat transfer in a smooth tube subjected to axially constant tube wall temperature, the heat transfer rate increases considerably for a moderate increase in pressure drop. Local convective-condensation measurements for four refrigerant fluids: R134a, R410A, R125 and R32 in a micro-fin tube were presented by kedzierski and Goncalves [7]. This research showed that R32 exhibit the highest heat transfer performance due to its high thermal conductivity.

The turbulent mixed convection in a horizontal circular tube provided with inserted strip has been studied by Hsieh et al [8]. Sarma et al [9] proposed the generalized correlations for predicting friction factor and convective heat transfer coefficient for twisted tapes in a tube. Reasonable agreement was obtained from the comparison between the predicted results and the measured data.

Fahed et al. [10] compared the pressure drop and heat transfer coefficients obtained from a plain micro fin and twisted tape inserted tubes. Sarma et al [11] presented a new approach for predicting the convective heat transfer coefficient of a tube equipped with twisted tape at different pitch to diameter ratios. The predicted results were compared with empirical correlations. Kumara and Prasad [12] studied the heat transfer in a solar water heater with twisted tapes inserts.

Zimparov [13, 14] experimentally studied the heat transfer and isothermal friction pressure drop in two single-start and three-start spirally corrugated tubes combined with five twisted tape inserts with different relative pitches. Ebru [15] studied the effects on heat transfer, friction factor and dimensionless exergy loss by mounting helical wires of different pitchs in the inner pipe of heat exchanger. This research shows that, an augmentation of up to 2.64 times in Nusslet number and friction factor was about 2.74 times compared to the empty pipe.

Several investigations were carried out to determine the effect of the coiled wire or the twisted tape elements on heat transfer and friction factor for a long time [16–18]. This is because the wire coil or twisted tape inserts in a tube creates swirling flows that modify the near wall velocity profile due to the various vorticity distributions in the vortex core. The fluid mixing between the tube core and the near wall region is enhanced because of the swirl induced tangential flow velocity component. However, accompanying with the swirl induced heat transfer enhancement, the shear stress and pressure drag in a tube with the coiled wire or twisted tape insert are accordingly increased.

Using compound turbulators, Promvonge and Eiamsa-ard [19–21] investigated the effects of the conical-nozzle, conical-ring or V-nozzle together with a swirl generator (decaying swirl) on heat transfer and friction characteristics in a uniform heat flux tube and found that for using both enhancement devices, the increase in heat transfer rate is about 20–50% for twist ratio, and regularly spaced helical screw inserts with different spacer length.

The purpose of the present paper is therefore, two folds: (i) to provide reliable correlations for both frication factor and Nusslet number and (ii) to study the heat transfer and friction factor characteristics of laminar flow through a flat tube fitted with full length helical screw twisted tap inserts of various twist ratio, and regularly spaced helical screw inserts with different spacer length.

EXPERIMENTAL APPARATUS

The experimental set-up consists of a counter flow double pipe heat exchanger and water flow system with its accessories. As shown in Fig. 1, the hot water circuit is arranged as; hot water storage tank that provided with multiple electrical heaters with rheostat, a 0.5 hp centrifugal pump, flow meter, piping system with suitable valves and the inner tube of the heat exchanger. The storage tank was perfectly insulated by 50 mm layer of glass wool and covered by galvanized steel sheet. The tank was fitted with suction, return and vent pipes. The vent pipe has two functions, to ensure that the storage tank is filled continuously with water to save the heater elements and to prevent the accumulation pressure inside the storage tank that may result from temperature rise or back pressure. All hot pipes were insulated with suitable insulation thickness and secured with galvanized steel cover. Cold water is supplied from city water passing, via a cold water tank, through the calibrated flow meter to the inside tube of the heat exchanger. Therefore, the cold water flow rate could be changed while the hot water flow is constant and controlled by the bypass system using the back flow branch and the regulating valve shown in Fig. 1.

Two sets of tubes are used in the present experiments, circular tubes and flat tubes with the same surfaces area. For the set of circular tubes, the outer copper tube is 63.5 mm in diameter with a thickness of 2 mm while the inner copper tube is 38.1 mm, inside diameter with a thickness of 2 mm and both of them have a length of 1500 mm. For the second set of double tubes, which is flat tubes, the outer copper flat tube has a major axis of 68.94 mm and minor axis of 53.94 mm with 2 mm thickness. The inner copper flat tube has a major axis of 42 mm and a minor axis of 30 mm with a thickness of 2mm. The helical screw-tapes with different geometries are inserted separately inside the inner flat tubes. The helical tape inserts with various twist ratio is made by winding uniformly a strip of 6.5 mm width over a 6 mm rod to make a helical tape is swirl for flow. The geometrical configuration of helical screw-tape is shown in Figs.2. The helical screw-tape inserts have different twist ratios and different spacer lengths. The twist ratio "Y" is defined as the ratio of length of one twist to diameter of the twist and it is varied from 2.17 and 5 to give Y = 2.17, 3.33, 4.3, and 5 respectively as shown in Figs.2. This range was selected with reference to very widely reported literature [22] value between 2 and 7 for twisted inserts. In addition, the helical tape inserts with spacer lengths 100mm, 200mm, 300mm, and 400mm are used at constant Y in each run. The outer tube is perfectly insulated with glass wool of 40 mm thickness to minimize the heat loss.

Nine T-type thermocouples, previously calibrated, were inserted in test section and distributed regularly. They are used to measure cold water, hot water, and wall temperature at inlet, mid and exit of the test section, in addition to the hot water in the storage tank. The pressure drop across the inner tube was measured by using U- Tupe manometer. When steady state condition is established, the water flow rates, the pressure drop and all thermocouple temperatures have been recorded.

UNCERTAINTY ANALYSIS

The uncertainties in the present experimental results depend upon the accuracy of the individual measuring instruments and the manufacture accuracy of the flat tubes. Its minimum division (its sensitivity) limits the accuracy of an instrument. The uncertainties are estimated based on the differential approximation. The uncertainties are estimated based on the differential approximation method. For a typical experiment the uncertainties of temperature is 0.1 °C, the pressure uncertainty of about ±2 %, the flat tubes surface area is 3%, and the flow rate is 2 %, as well as those of geometrical parameters and consequences their physical parameters, the uncertainties of the average Nusselt number, friction factor and Reynolds number are estimated as a maximum of 3.2 % , 2.8 % and 4.1 % respectively.

DATA REDUCTION

Using water on both sides of the heat exchanger can perform with reasonable temperature difference between the hot and cold sides. The flow velocities and the Reynolds number are calculated from the measured flow rates based on the equivalent diameter. The tube equivalent diameter is calculated from the volume of water required to fill a given length of tubing as follows:

$$D_{eq} = (4V/\pi L)^{0.5} \quad (1)$$

The tube cross-sectional flow area A_t, is calculated by using the tube equivalent diameter ($A_t = \pi D^2_{eq} / 4$).

Reynolds number is calculated inside tubes as:

$$Re = (v\, D_{eq} / \nu) \quad (2)$$

The mean flow velocity inside tube is given as:

$$v = (m_c / (\rho\, \pi\, D^2_{eq} / 4) \quad (3)$$

The heat transfer rates of the two fluid streams are calculated as:

$$Q_c = m_c\, C_{pc}\, (T_{co} - T_{ci}) \quad (4)$$

$$Q_h = m_h\, C_{ph}\, (T_{hi} - T_{ho}) \quad (5)$$

The average heat transfer coefficient for inside tube, h_c, is calculated from the following equation:

$$h_c = Q_c / (\pi\, D_{eq}\, L\, \Delta t_{mc}) \quad (6)$$

Where Δt_{mc} is the logarithmic mean temperature difference defined as:

$$\Delta t_m = [((T_{ci} - T_{si}) - (T_{co} - T_{so})) / (\ln (T_{ci} - T_{si}) / (T_{co} - T_{so}))]$$

The average Nu number is calculated based on the equivalent diameter as:

$$Nu = h_c\, D_{eq} / k_c \quad (7)$$

The friction factor is calculated using Darcy's equation at Hewitt [16] as:

$$f = (D_{eq} / L)\, (\Delta P / 2\rho\, v^2) \quad (8)$$

Where ΔP is the pressure drop over length L

RESULTS AND DISCUSSION

Results of average Nusselt number (Nu):

Preliminary experiments have been performed on a smooth tube to check out the results with empirical correlation proposed by reference [23]. This correlation is stated as:

$$Nu = 1.86 [Re\, Pr\, D/L]^{0.33} [\mu_a/\mu_b]^{0.14} \qquad (9)$$

The experimental *Nu* results of plain flat tube are compared with the plain circular tube and the previous Eq. (9) of circular tube as shown in Fig. 3. This figure represents the variation of average Nusselt number (*Nu*) with Reynolds number for the three cases. The *Nu* of the present experimental results of the plain flat tube are higher than that of plain circular tube while that of plain circular tube are found to agree within 6 % with the previous Eq.(9) of plain circular tube approximately.

Figures 4 through 7 show the relation of *Nu* with *Re* of plain flat tube and a flat tube with inserted swirl sheet of different *Y* and *S*. These figures show that *Nu* increases with the increase of *Re* but it changes dramatically with *Y* and *S*. The same general shape of curves can be seen from these figures. In addition, these figures show that all *Nu* results of the flat tube with inserted swirl sheet with different *S* and *Y* are more than that of plain flat tube.

The effect of *Y* on the *Nu-S* relation at a constant of Re = 1297 is shown in Fig. 8. . This figure shows that *Nu* decrease with the increase of *S* and *Y*. This can be explained, that the behavior of flow inside the flat tube with the inserted swirl sheet is affected by increase of S or *Y*. The increase of *S* or *Y* leads to decrease in turbulence intensity of the flow inside the flat tube with the inserted swirl sheet and in turn a decrease of *Nu*. Thus, insert the helical screw sheet inside tube make swirls to the flow and generates periodic disruptions to the development of the viscous boundary layer. This disturbance of the main flow and the viscous boundary layers in the helical screw insert inside tube enhance the heat transfer.

Results of friction factor (f):

Preliminary experiments have been performed on a smooth tube to check out the results with the empirical correlations that proposed by reference [23]. This correlation is stated as:

$$f = 64 / \mathrm{Re} \tag{10}$$

The experimental f results of plain flat tube are compared with the plain circular tube and the previous equ.(10) of circular tube as shown in Fig. 9. This figure represents the variation of friction factor (f) with Reynolds number for the three cases. The f of the present experimental results of the plain flat tube are slightly higher than that of plain circular tube while the present results of plain circular tube is found to agree within 8 % with the equ.(10) of plain circular tube.

The effect of Y and S on the characteristics of friction factor inside flat tube with and without inserted swirl sheet will be discussed in the present section. Figures 10 through 13 show the relation of f with *Re* of plain flat tube and a flat tube with inserted swirl sheet of different *Y* and *S*. It is seen from the figures that the f decreases with the increase of *Re*, but it decreases with the increase of S. The same general shape of curves can be seen from all the figures.

The variation of f with S at different Y at constant Reynolds number is shown in Fig. 14. This figure shows that f decrease with the increase of Y and S. The behavior of flow inside the flat tube with inserted swirl sheet is affected by the increase of S or Y. The increase of S or Y leads to a decrease in turbulence intensity of the flow inside the flat tube with the inserted swirl sheet and in turn a decrease of f.

<u>**Correlation of the results**</u>

The experimental results are fitted, using power regression, to determine the following empirical correlations for a flat tube:

$$Nu = 6.11 Re^{0.199}(1+x)^{-0.064} Y^{-0.318} \quad (11)$$

And:

$$f = 54.41 Re^{-0.87}(1+x)^{-0.045} Y^{-0.146} \quad (12)$$

$5.86 * 10^2 \le Re \le 1.31 * 10^3$ & $0 \le x \le 66.67$ & $2.17 \le Y \le 5$

The calculated data from Eq. 11 and 12 of the Nu_{cal} and f_{cal} are plotted against experimental data of the Nu_{Exp} and f_{Exp} in Fig. 15 and 16 respectively. As shown from these figures the maximum deviation between the experimental data and correlations are ±10 % and ±8 % for Nu and f respectively.

<u>**CONCLUSION**</u>

Augmentation of laminar flow and heat transfer in flat tubes by means of helical screw-tape inserts is investigated experimentally. The friction factor in the horizontal flat tubes with full-length helical screw element of different twist ratio and helical screw inserts with different spacer length are investigated. Cold and hot water are used as working fluid in tube side and shell side respectively. The experiments covered a range of Reynolds numbers $5.7*10^2 \le Re \le 1.31*10^3$. The effect of spacer length on the heat transfer augmentation and friction factor and the effect of twist ratio on heat transfer augmentation and friction factor have been presented.

The results of the present investigation can be summarized as:

- The results indicate that the helical screw element of different twist ratio and helical screw inserts with different spacer length have an effect on the results of heat transfer coefficient and friction factor.
- The averaged Nusslet number, Nu increase with the increase in Reynolds number and with the decrease in twist ratio and spacer length. At spacer length equal 0, increased Nu by about 119 % compared with the empty tube.
- The data of plain flat tube is greater than that of plain circular tube and the data of flat tubes containing a helical screw tapes is better than plain tubes for all Re, Y and S.
- For a fixed Reynolds number, the friction factor (f) increases with the decrease in twist ratio and spacer length for the flat tubes.
- The correlations of average Nusselt number and friction

factor with Re, S and Y are presented.

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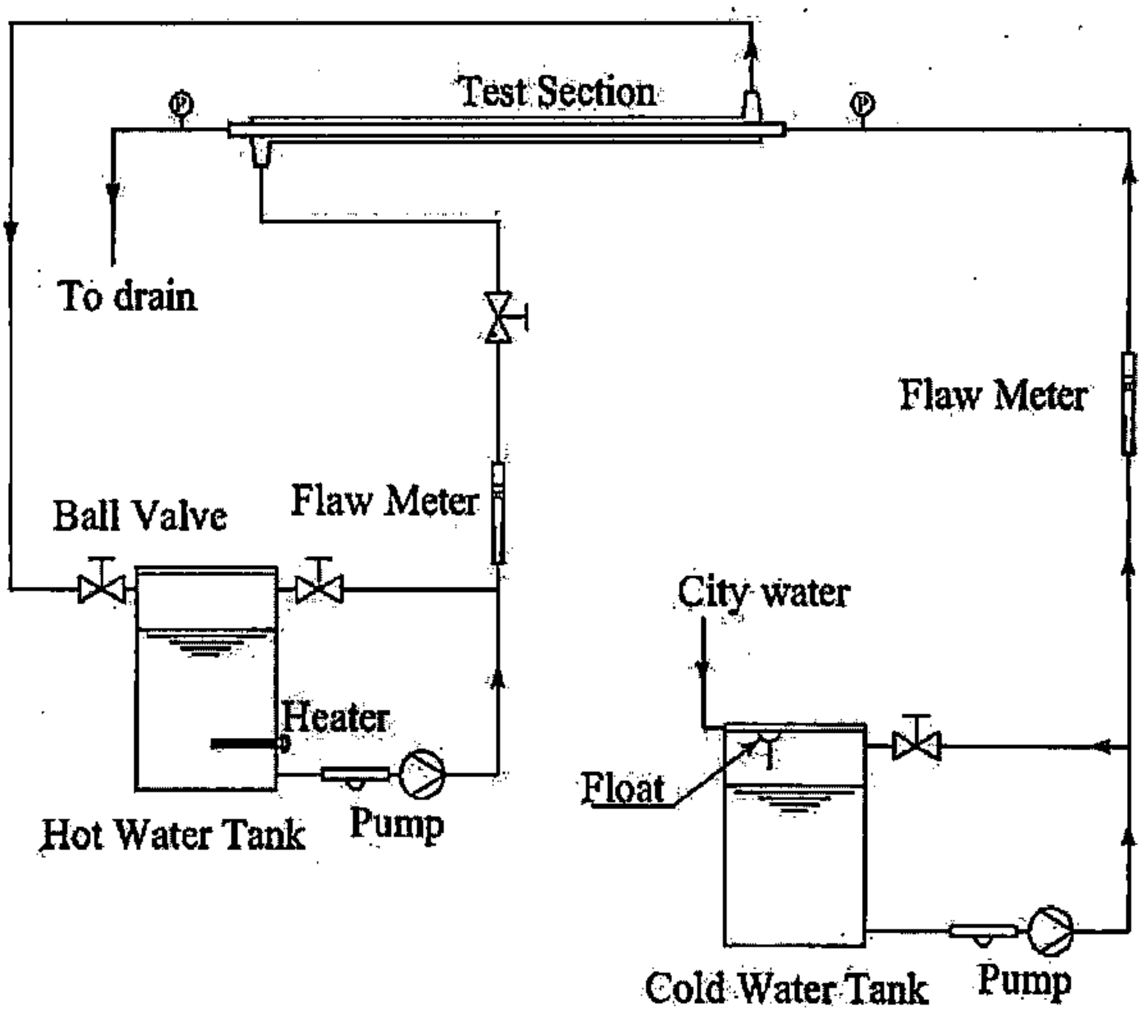


Fig.1a Diagram of experimental set up

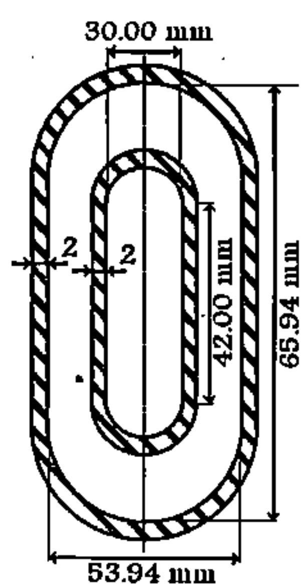
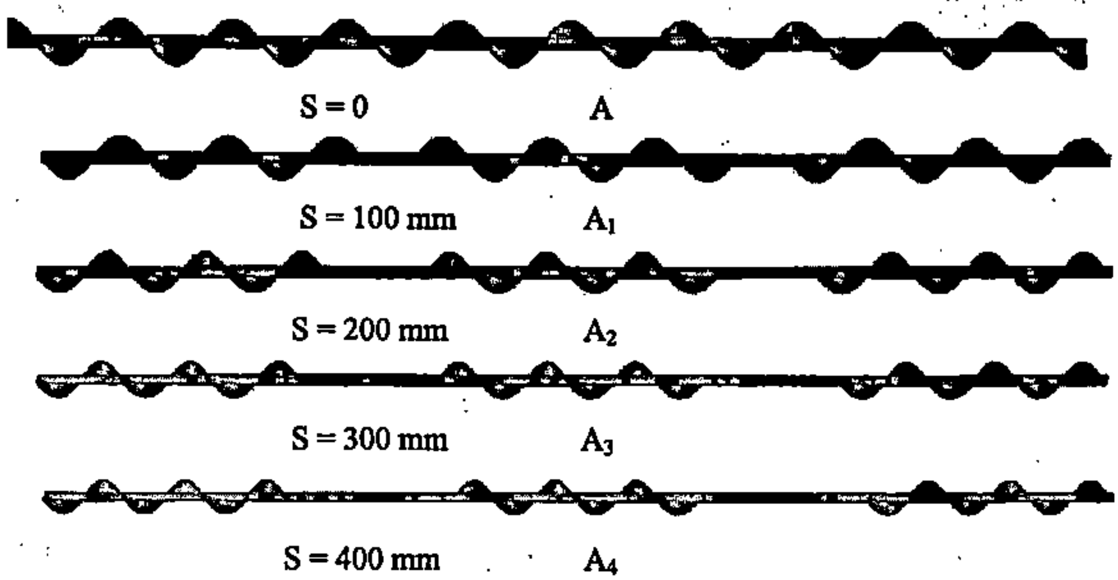
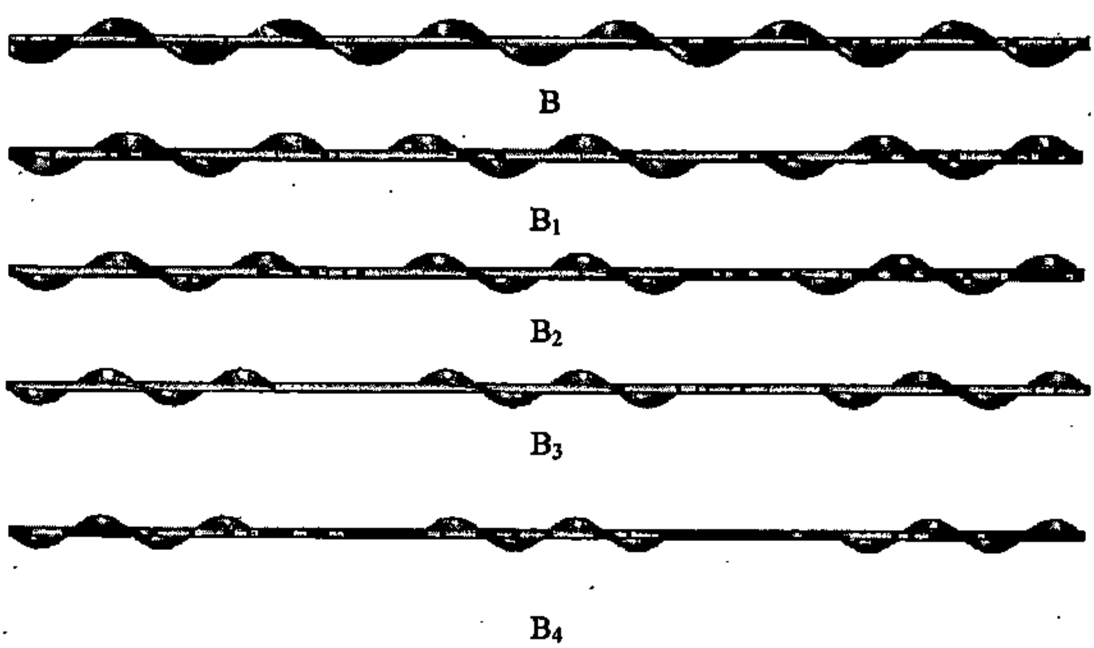


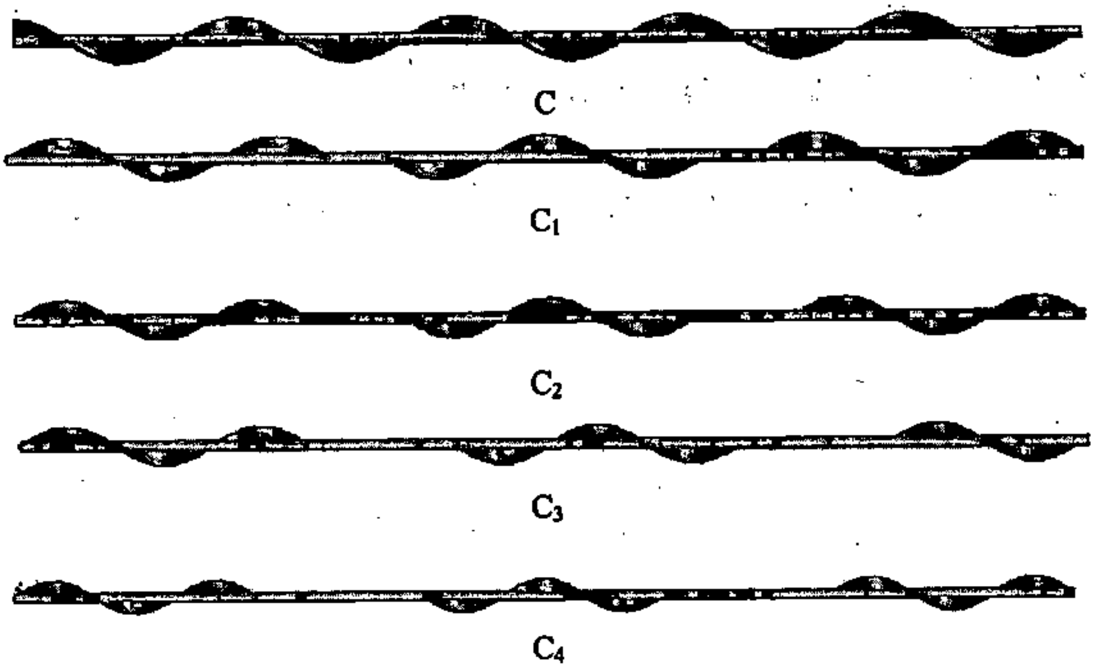
Fig. 1b Cross section area of double flat tubes



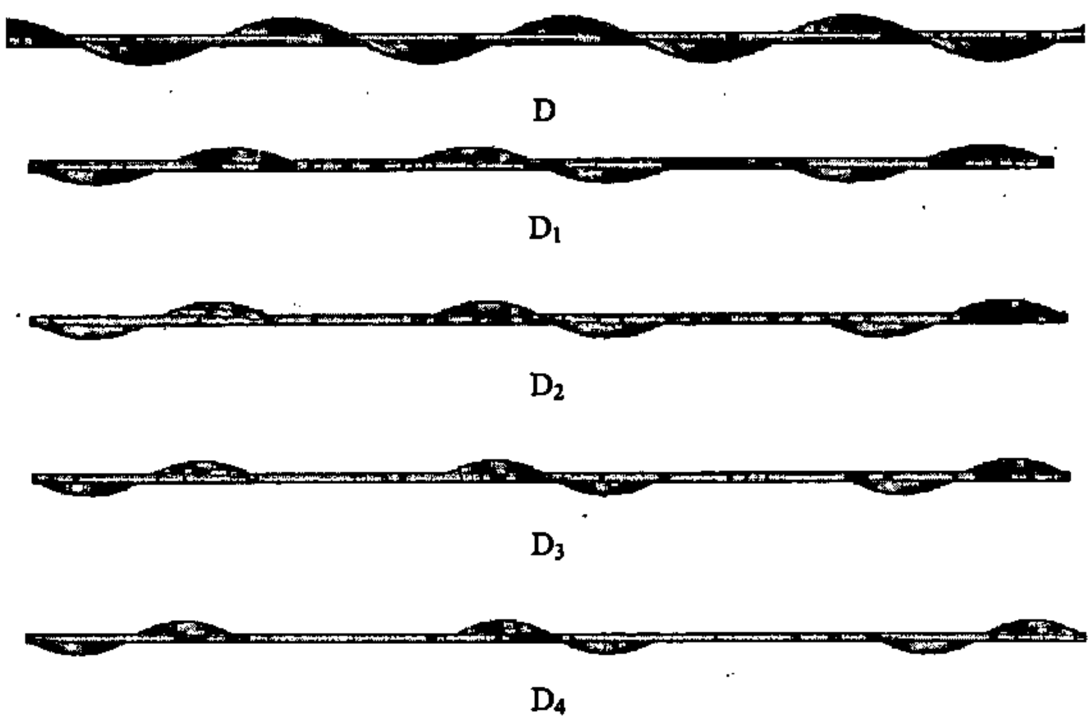
a. Helical tape inserts of twist ratio 2.17 and twist ratio with various spacer length.



b. Helical tape inserts of twist ratio 3.3 and twist ratio with various spacer length.



c. Helical tape inserts of twist ratio 4.3 and twist ratio with various spacer length.



d. Helical tape inserts of twist ratio 5 and twist ratio with various spacer length.

Fig.2 Helical tape inserts with various spacer length and variable twist ratio.

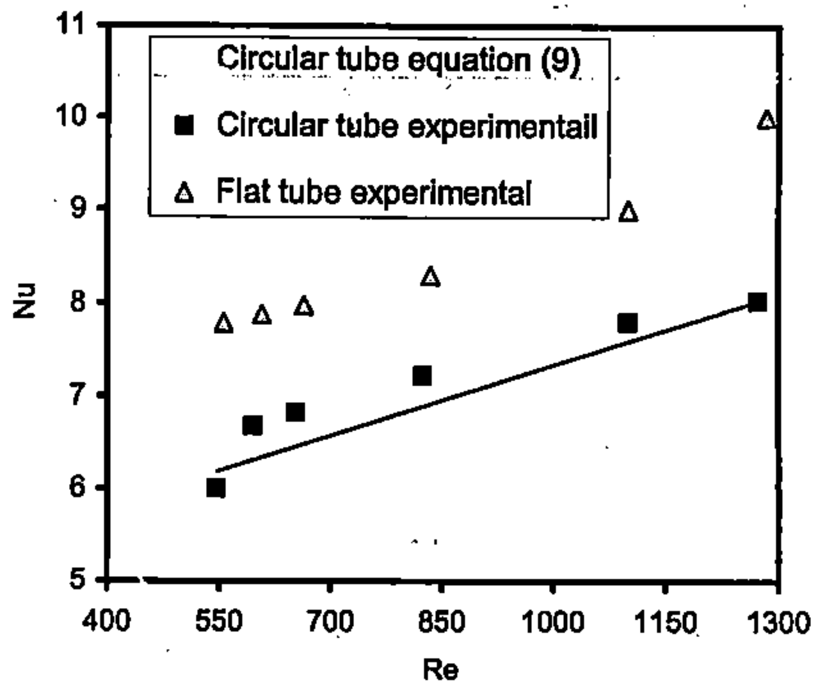


Fig.3 Data verification of Nusslet number for plain tube.

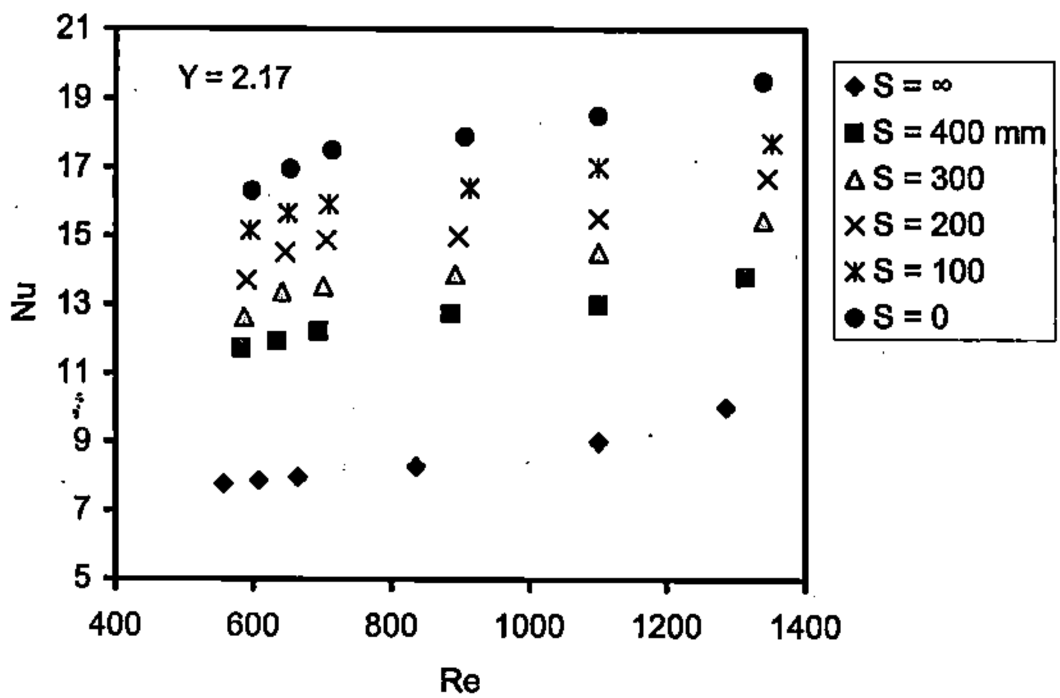


Fig.4 Nusslet number vs Reynolds number for helical screw of twist ratio 2.17 with different spacer length in case of flat tube

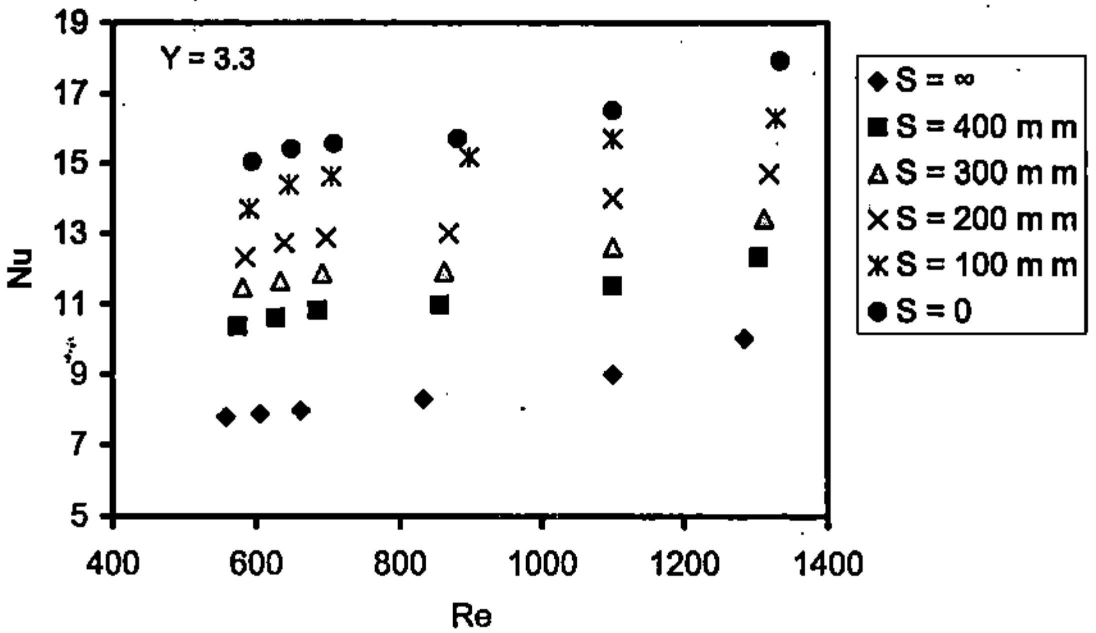


Fig.5 Nusslet number vs Reynolds number for helical screw of twist ratio 3.33 with different spacer length in case of flat tube

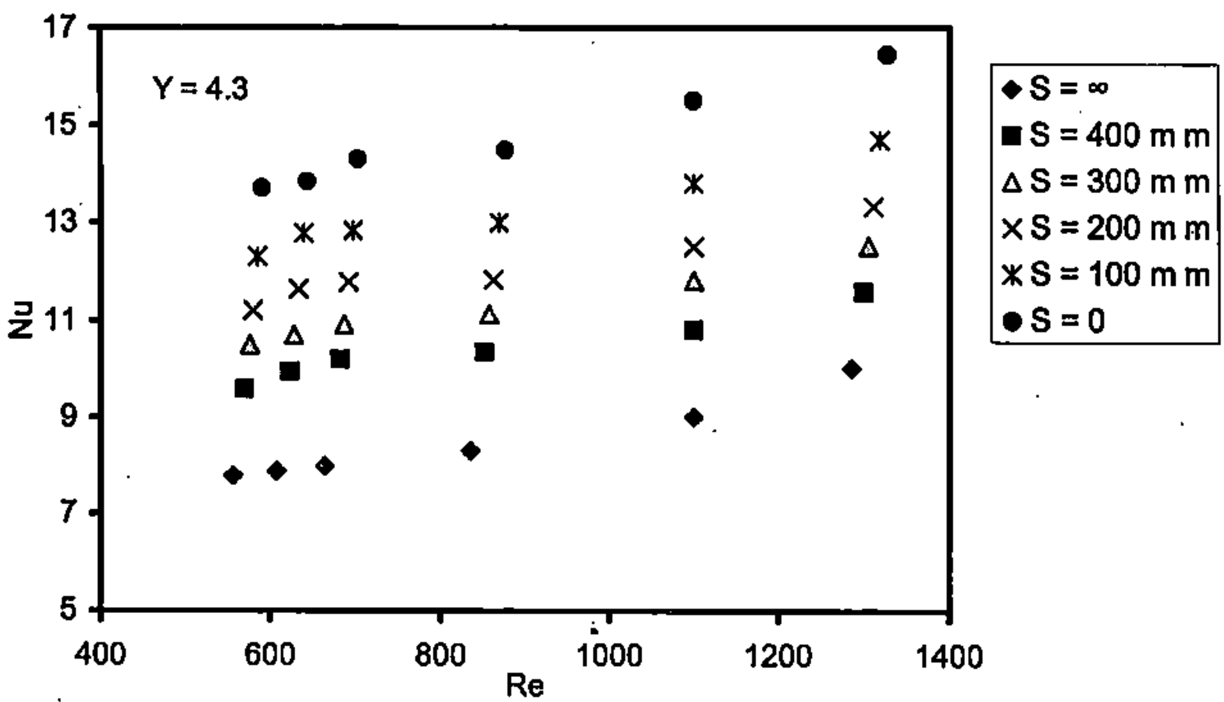


Fig.6 Nusslet number vs Reynolds number for helical screw of twist ratio 4.3 with different spacer length in case of flat tube

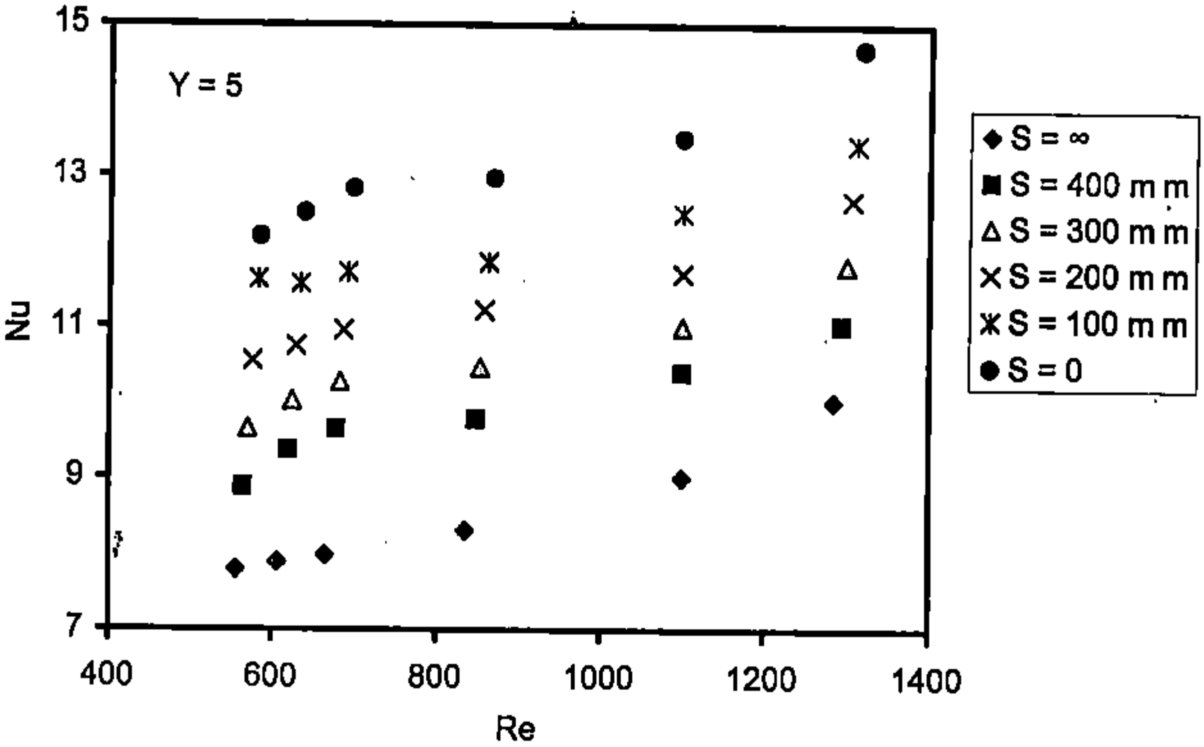


Fig.7 Nusslet number vs Reynolds number for helical screw of twist ratio 5 with different spacer length in case of flat tube

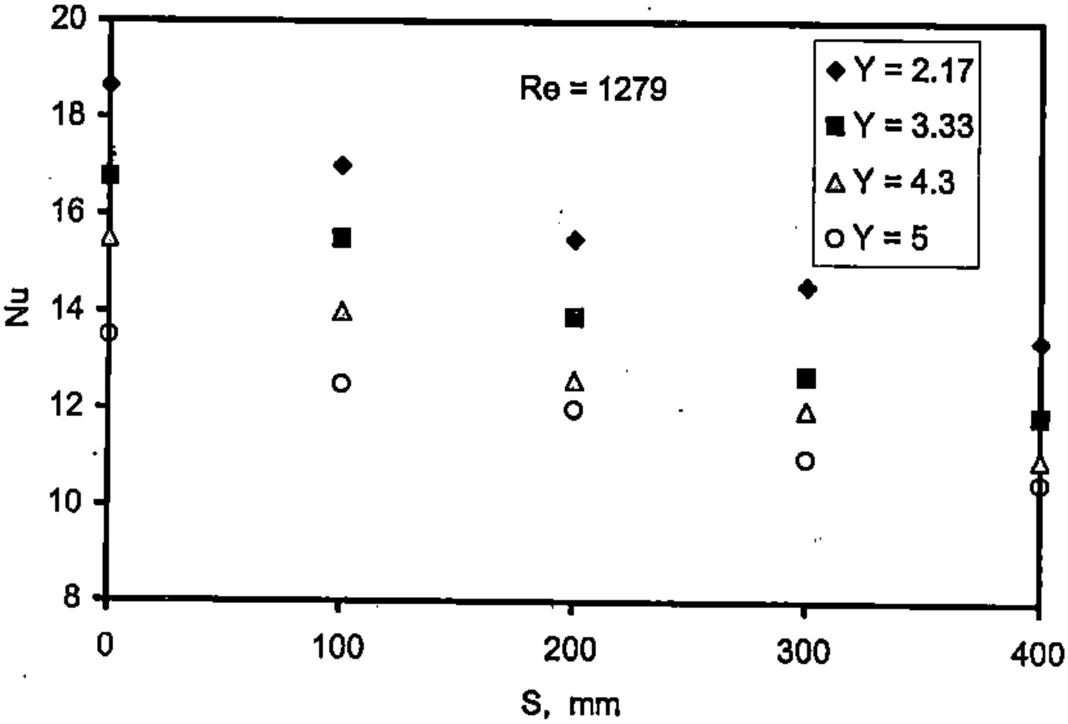


Fig.8 Nusslet number vs spacer length for helical screw with different twist ratio at constant Re in case of flat tube

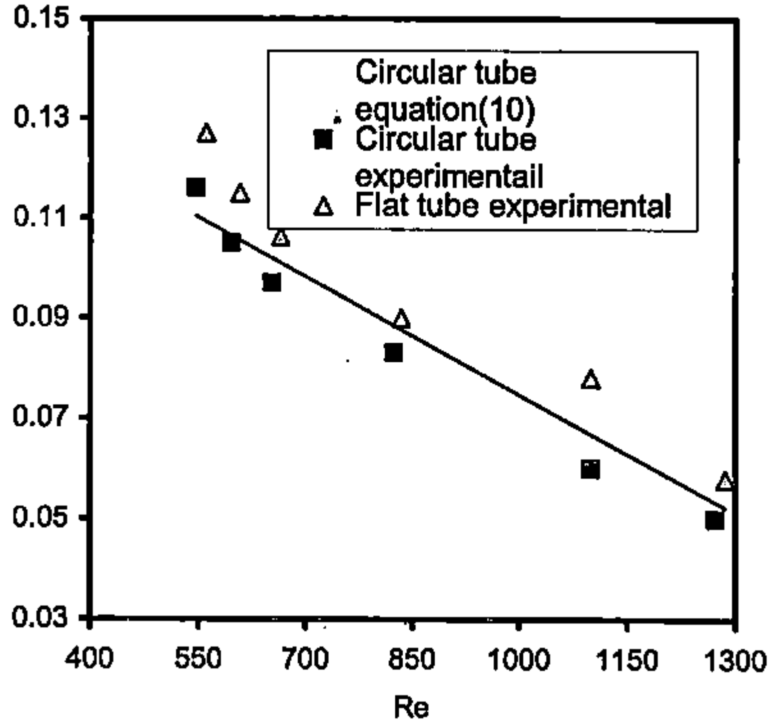


Fig.9 Data verification of friction factor for plain tube

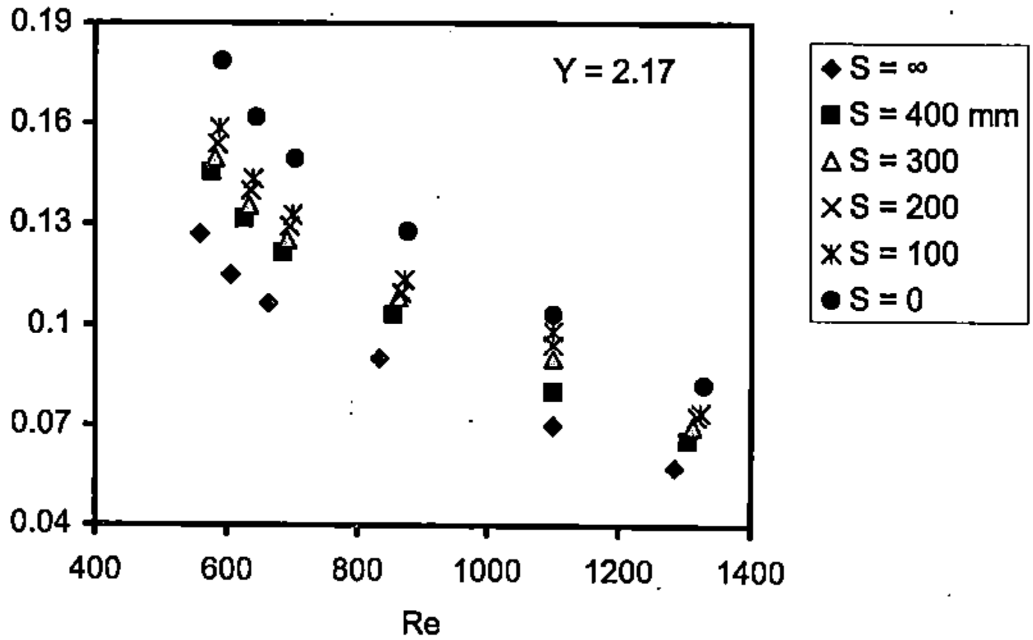


Fig.10 Friction factor vs Reynolds number for helical screw of twist ratio 2.17 with different spacer length in case of flat tube

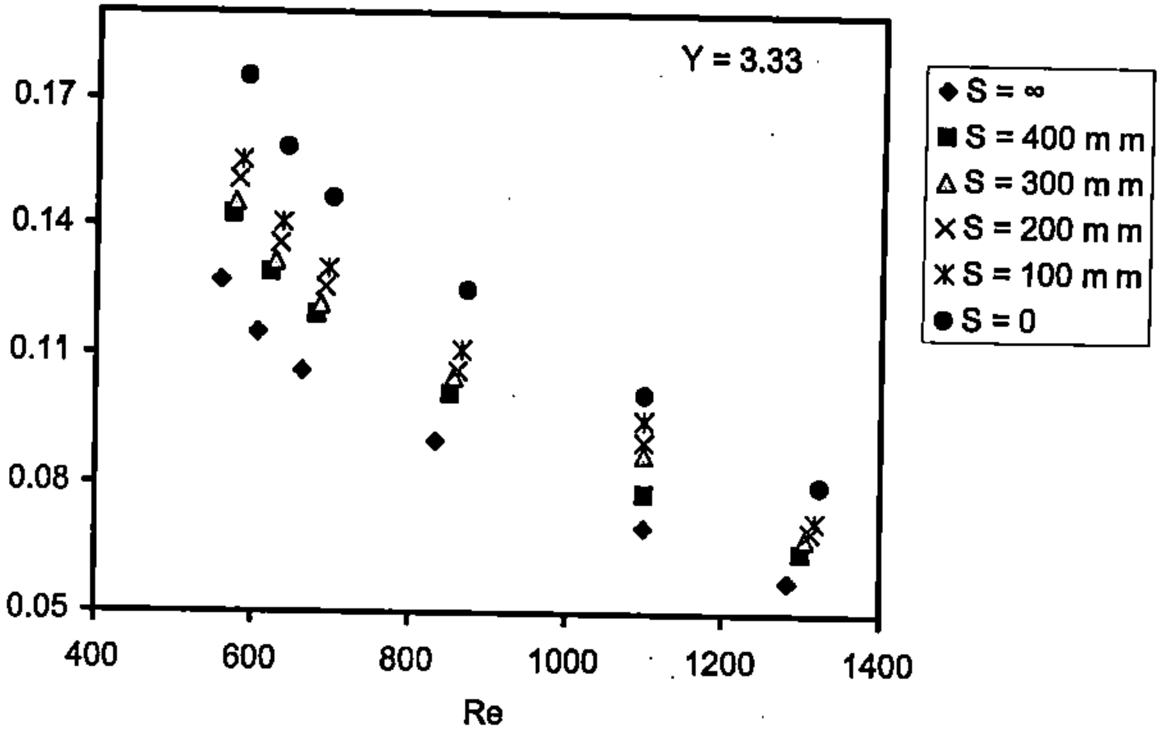


Fig.11 Friction factor vs Reynolds number for helical screw of twist ratio 3.33 with different spacer length in case of flat tube

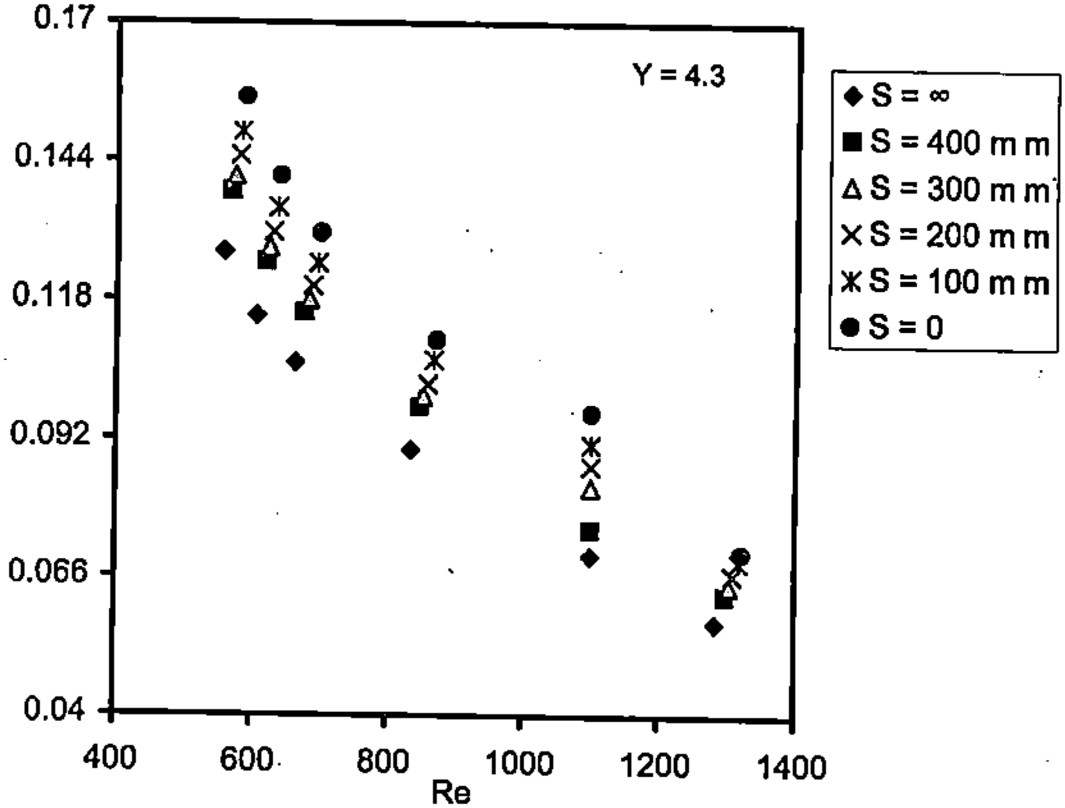


Fig.12 Friction factor vs Reynolds number for helical screw of twist ratio 4.3 with different spacer length in case of flat tube

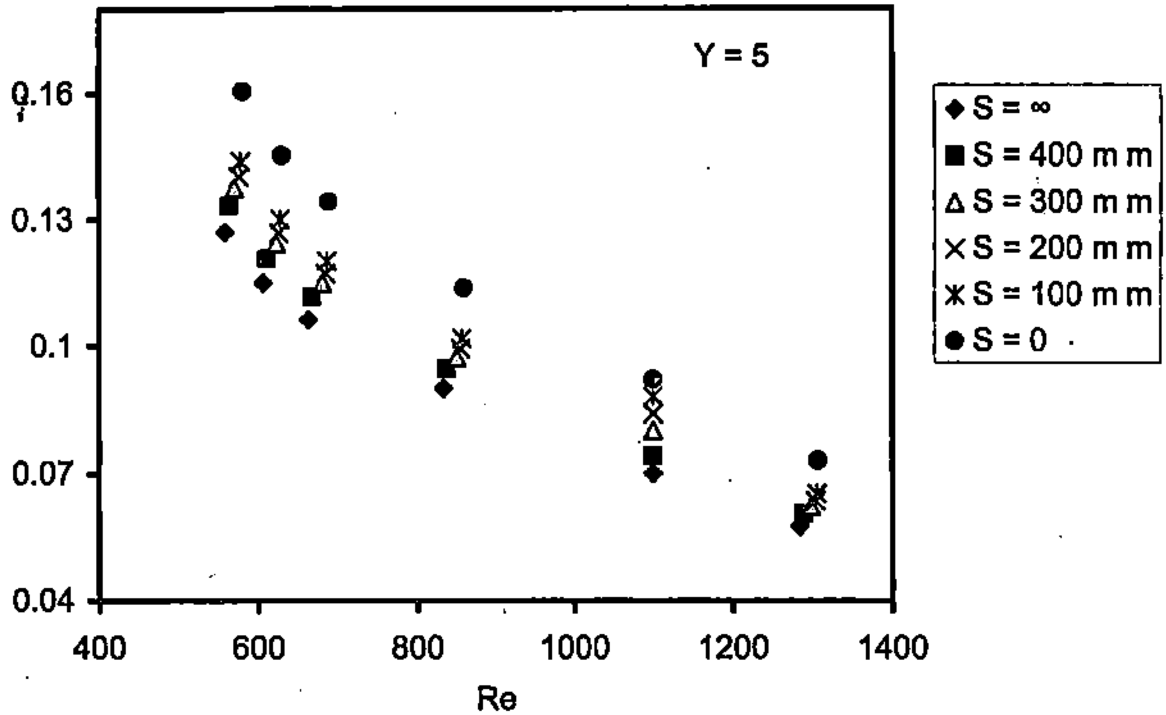


Fig.13 Friction factor vs Reynolds number for helical screw of twist ratio 5 with different spacer length in case of flat tube

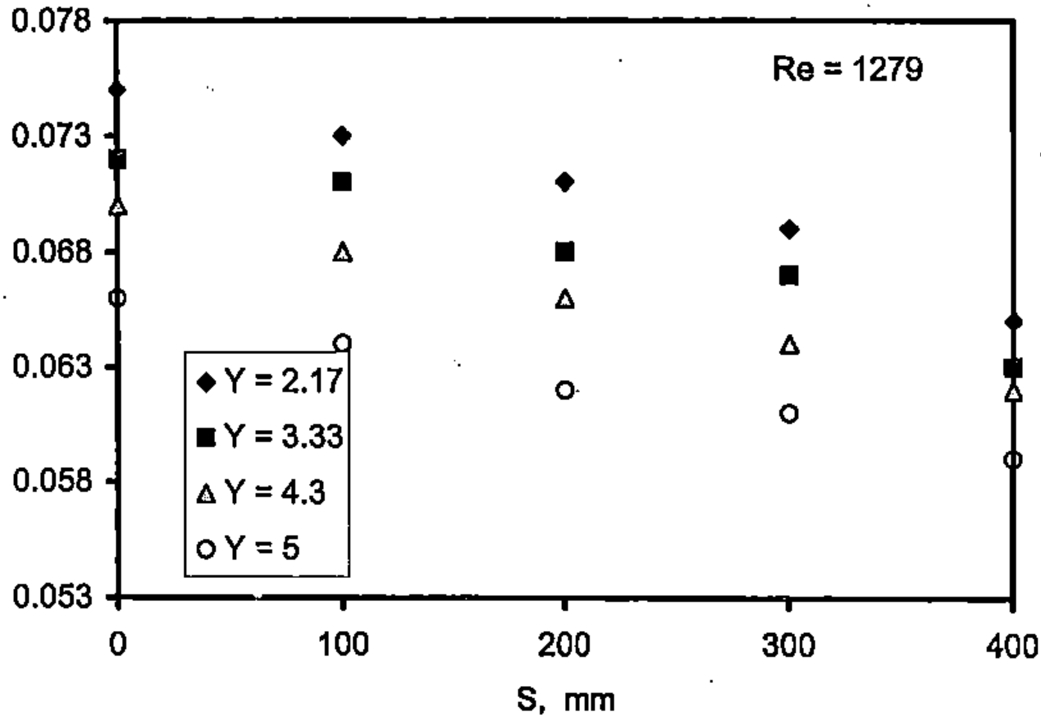


Fig.14 Friction factor vs spacer length for helical screw with different twist ratio at constant Re in case of flat tube

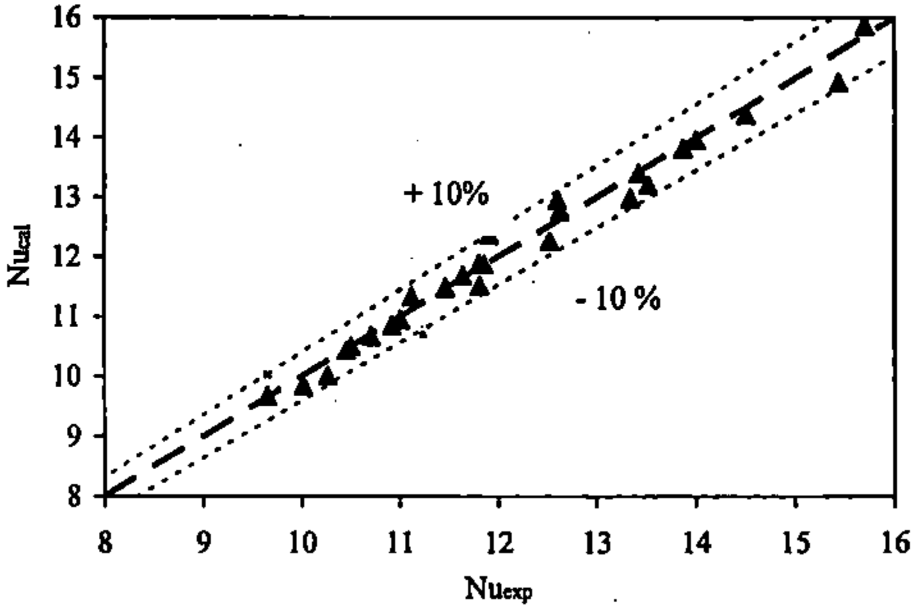


Fig. 15 Nu_{cal} against Nu_{exp} for flat tube

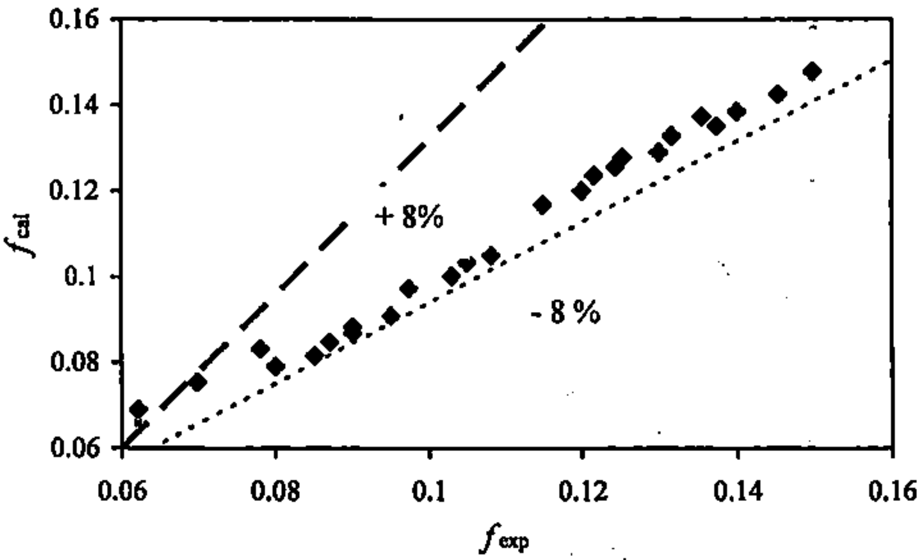


Fig. 16 f_{cal} against f_{exp} for flat tube