

Heat Transfer Enhancement Studies in a Circular Tube Fitted with Right-Left Helical Inserts with Spacer

P.K.Nagarajan and P.Sivashanmugam

Abstract—Experimental investigation of heat transfer and friction factor characteristics of circular tube fitted with 300 right-left helical screw inserts with 100 mm spacer of different twist ratio has been presented for laminar and turbulent flow.. The experimental data obtained were compared with those obtained from plain tube published data. The heat transfer coefficient enhancement for 300 R-L inserts with 100 mm spacer is quite comparable with for 300 R-L inserts. Performance evaluation analysis has been made and found that the performance ratio increases with increasing Reynolds number and decreasing twist ration with the maximum for the twist ratio 2.93. Also, the performance ratio of more than one indicates that the type of twist inserts can be used effectively for heat transfer augmentation.

Keywords—Heat transfer augmentation, right-left helical screw inserts with spacer, Twist ratio, Heat Transfer

I. 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. Heat transfer enhancement technology has been developed and widely applied to heat exchanger applications; for example, refrigeration, automotives, solar water heater process industry, chemical industry etc. Many techniques of active and passive techniques are available for augmentation. These details are discussed in detail by Bergles. Also heat augmentation techniques play a vital role for laminar flow, since the heat transfer coefficient is generally low in plain tubes. Bergles [1, 2] presented a comprehensive survey on heat transfer enhancement by various techniques. Among many techniques (both passive and active) investigated for augmentation of heat transfer swirl flow device) has been shown to be very effective, due to imparting of helical path to the flow. Vast Literature [3-24] is available on full length twisted tape and helical inserts staright and right-left inserts. This paper reports heat transfer augmentation in a circular tube fitted with 300R-L inserts with 100 mm spacer.

II. TECHNICAL DETAILS OF HELICAL SCREW TAPE INSERTS

The geometrical configuration of right-left helical inserts with spacer is shown in Fig.1. The helical screw tape inserts with various twist ratio is made by winding uniformly a strip

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of 8.5 mm width over a 8 mm rod. The twist ratio 'Y' defined as the ratio of length of one full twist (360 °) to diameter of the twist is varied from 2.93 to 4.89. The 300 R-L right-left helical inserts with 100 mm spacer were formed by joining 300 mm length of right twist and 300 mm length of left twist along with 100 mm spacer alternatively.

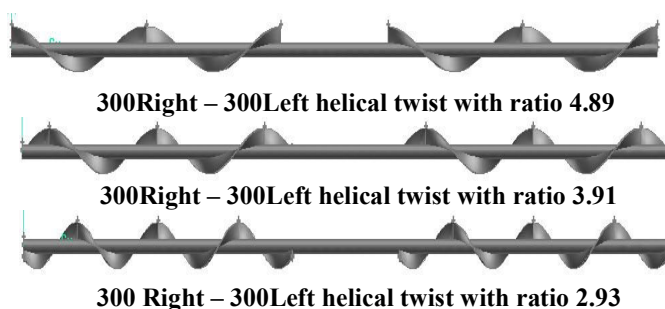


Fig. 1 Diagram of 300 right-300 left helical screw inserts with 100 mm spacer

III. EXPERIMENTAL SET UP

The experimental set up used for the present study is same as that used in previous paper and is explained in this section. The schematic diagram of the experimental set up is shown in Fig.2a. It consists of calming section, test section, Rota meters, inlet tank with the capacity 1 cubic meter for supplying water, outlet mixing section, water collection tank capacity 1 cubic meter for receiving water from test section. Calming section with the dimension 2000 mm long, 25.45 mm ID, 33.33 mm OD made of stainless steel tube is used to eliminate the entrance effect. The test section is of smooth stainless steel pipe with the dimension 1800 mm long with 25.45 mm ID, 33.33 mm OD. The outside surface of the tube is brazed with thermo well made of stainless steel having dimensions 6.36 mm ID, 1 mm thick, 120 mm length at distance of 150 mm all along the length in a straight line as shown in Fig.2b. The test section tube is wound with ceramic beads coated electrical SWG Nichrome heating wire of resistance 37 Ω . Over the electrical winding, two layers of asbestos rope tape is wound. Over the asbestos tape winding approximately 50 mm thickness of glass wool is lined and over which, another two layers of asbestos rope tape is wound to minimize heat loss. The terminals of the Nicrome wire are attached to the Auto-Transformer, by which heat flux can be varied by varying the voltage. The Auto-transformer is connected to the servo voltage regulator to minimize the voltage fluctuations. Calibrated RTD PT 100 type temperature sensors of 0.1 ° C accuracy with digital indicator are placed in

the thermowell to measure the outside wall temperatures of test section. One end of the electrical heating test section is attached with the calming section, and while the other end is attached with the mixing section (length 200 mm, ID 25.45 mm, OD 33.33 mm), where two baffles are provided inside the pipe at a distance of 100 mm from the flange connection for efficient mixing of outlet fluid. Flanges are used for attachments, and 50.90 mm thick non-conducting polypropylene disc is placed in-between the flanges to prevent the heat conduction flow to the calming section, and the mixing section. Two RTD PT 100 type temperature sensors one just before the test section and the other after the mixing section are placed to measure the inlet and outlet temperature of fluid. The inlet tank used for supplying water is fitted with portable agitator not shown experimental diagram for maintaining constant temperature.

Two calibrated Rotameters having flow ranges of (0.1 to $1 \times 10^{-3} \text{ m}^3$ per minute, and 1 to $12 \times 10^{-3} \text{ m}^3$ per minute) to cover the full laminar ranges are attached to the calming section to measure the flow. The water at constant temperature is being taken from the inlet tank through centrifugal pump as shown in Fig.2a. The by-pass valve attached to Rotameter is used to regulate the flow rate to the test section. The two pressure tapes one just before the test section and the another just after test section are provided, and attached to U tube manometer for pressure drop measurement.

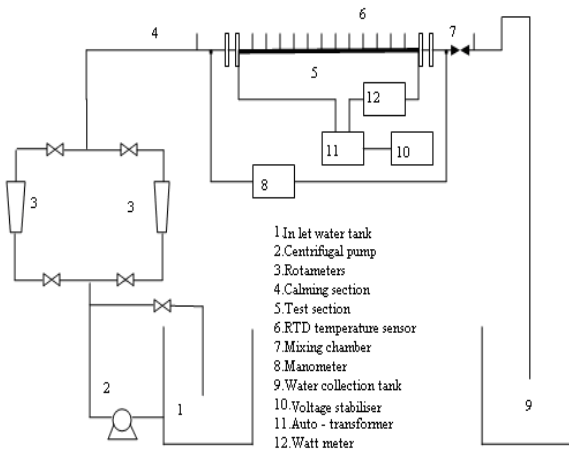


Fig. 2 a Diagram of Experimental set up



Fig. 2 b Diagram of test section

IV. EXPERIMENTAL PROCEDURE

To start with the centrifugal pump was switched on, and the water flow rate to the test section was adjusted using by-pass valve. The heat flux was set by adjusting the electrical voltage with the help of Auto-transformer, and the constant heat flux was allowed to continue till the steady state is attained. The steady state was obtained within 1 hour for the first run, and 25 minutes for the subsequent runs. The inlet, outlet

temperature of water, and the wall temperatures in all RTD temperature sensors were noted after steady state. The electrical heat flux was measured by calibrated Ammeter and Volt meter, and also with the help of watt meter. The flow rates to the test section and heat flux were varied, and readings were taken after attaining steady state. The flow rate was varied from $0.1 \times 10^{-3} \text{ m}^3$ per minute to $12 \times 10^{-3} \text{ m}^3$ per minute. Experiments were conducted for plain tube, and subsequently by inserting 300 R-L inserts with 100 mm spacer of different twist ratio. The pressure drop was measured for each flow rate with the help of manometer under isothermal condition of flow.

V. PRESSURE DROP CALCULATION

The pressure drop was determined from the differences in the level of manometer fluid. The fully developed friction factor was calculated from the following equation.

$$f = (D_i / L) (\Delta P / 2\rho u_m^2) \quad (1)$$

Where ΔP is the pressure drop over length L

VI. HEAT TRANSFER CALCULATION

The heat transfer rate in the test section was calculated using [25]

$$Q = V^2/R = m C_p (T_{out} - T_{in}) = U_o A_o (T_{wo} - T_f) \quad (2)$$

where

$$1 / (U_o A_o) = 1 / (h_i A_i) + \ln (D_o / D_i) / (2\pi k_w L) \quad (3)$$

The internal convective heat transfer coefficient, h_i was determined by combining Eq. (2) and (3)

The thermal equilibrium test showed that the heat supplied by electrical winding in the test section was 8 to 10% larger than the heat absorbed by the fluid. This was caused by thermal loss from the test section. The average value of heat transfer rate obtained by heat supplied by electrical winding, and heat absorbed by the fluid was taken for internal convective heat transfer coefficient calculation.

The Nusselt number was calculated using equation

$$Nu = h_i D_i / k \quad (4)$$

All the fluid thermo physical properties were determined at the average of the inlet and outlet bulk temperatures, T_f .

VII. RESULTS AND DISCUSSION

A. Plain tube data

The data obtained by the experiment for the plain tube has been verified with data reported by Sieder and Tata [26] and found that the experimental data are matching reported data with the discrepancy of less than $\pm 11 \%$ for both laminar and turbulent flow.

B. Effect of twist ratio on heat transfer augmentation in laminar flow

Fig. 3 depicts variation of Nusselt number with Reynolds number for 300 mm right-left helical twist with 100 mm spacer of different twist ratio. Nusselt number for the tube fitted with 300 mm right-left helical twist is higher than that for plain tube for a given Reynolds number attributing to heat transfer enhancement due to swirl flow. As the Reynolds number increases the Nusselt number increases due to increased convection. Also, as the twist ratio decreases the Nusselt number increases for a given Reynolds number and reaching a maximum for the twist ratio of 2.93 due to fact that as the twist ratio decrease, the intensity of swirl generated increases with the maximum intensity for the twist ratio 2.93. One can also observe that the Nusselt number for 300 mm right -left twist is almost close to 300R-L inserts but more than that for straight twist for a given twist ratio. This may be due to reason that repeated left-right movement of fluid during course of flow through tube attached with left-right twist will enhance the heat transfer by virtue of efficient mixing in the radial direction.

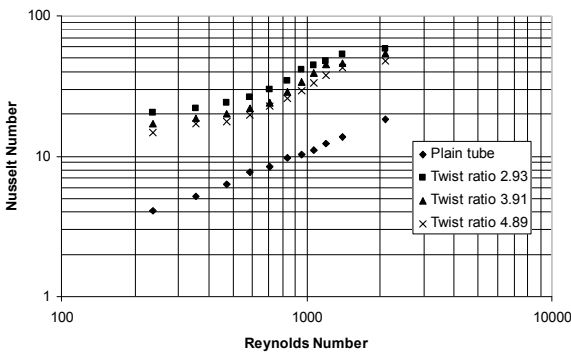


Fig. 3. Nusselt number vs Reynolds number for right-left helical insert 300 R and 300 L with 100 spacer under laminar flow

C. Effect of twist ratio on friction factor in laminar flow

Fig.4 shows the variation of friction factor vs Reynolds number for the tube fitted with right-left helical twist with 300 R – L inserts with 100 mm spacer. The friction factor for the tube fitted with right-left helical inserts 300 R-L is higher than that for plain tube and decreases with Reynolds number for a given twist ratio. However, the friction factor increases with twist ratio for a given Reynolds number and reaching maximum for the twist ratio 2.93. It is also observed from Fig.5.that the friction factor for right -left twist is more than that for straight twist and less than that for 300R-L inserts for a given twist ratio resulting from repeated right-left movement of fluid during course of flow through tube attached with right-left twist.

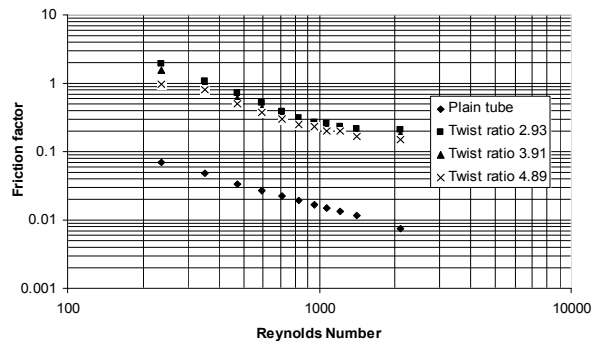


Fig. 4. Friction factor vs Reynolds Number for right-left helical insert 300 R and 300 L with 100 spacer in laminar flow

D. Effect of twist ratio on heat transfer augmentation in turbulent flow

Fig. 5 depicts variation of Nusselt number with Reynolds number for 300 mm right-left helical twist with 100 mm spacer of different twist ratio. One can observe from Fig.5 that the trend observed in laminar flow is seen here.

E. Effect of twist ratio on friction factor in turbulent flow

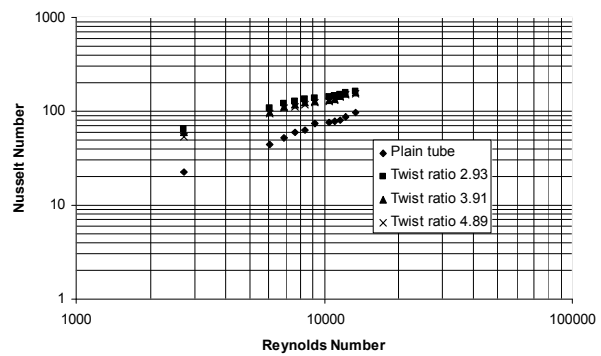


Fig. 5 Nusselt number vs Reynolds number of kenics insert 300 R and 300 L with 100 spacer under turbulent flow

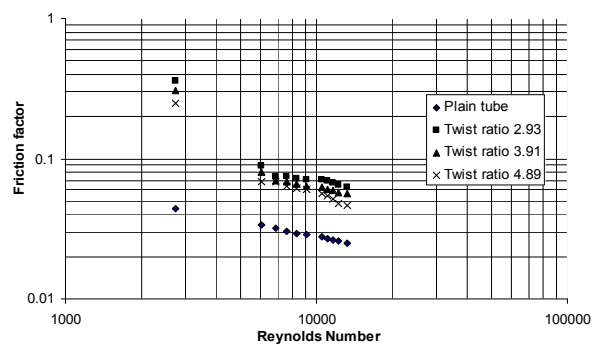


Fig.6 shows the variation of friction factor vs Reynolds number for the tube fitted with right-left helical twist with 300 R – L inserts with 100 mm spacer.

One can observe from Fig.6 that the trend observed in laminar flow is seen here.

F. Performance Evaluation Analysis

Bergles [27] have suggested several criteria for the performance evaluation of heat transfer enhancement devices. The performance of the any heat transfer enhancement device evaluated on the basis of the following two important criteria:

(i) Basic geometry fixed, pumping power fixed increase heat transfer

Friction factor vs Reynolds number for right-left helical insert 300 R and 300 L with 100 spacer in turbulent flow

(ii) Basic geometry fixed, heat duty fixed reduce pumping power

Usui and Sano [28] proposed a performance evaluation analysis for the same pumping power and this method used for present study for laminar flow, the performance ratio is defined as :

$$\text{Performance ratio } (\eta) = \frac{Nu_{\text{twist}}/Nu_{\text{plain}}}{(f_{\text{twist}}/f_{\text{plain}})^{0.1666}} \quad (5)$$

Whereas for turbulent flow

$$\text{Performance ratio } (\eta) = \frac{Nu_{\text{twist}}/Nu_{\text{plain}}}{(f_{\text{twist}}/f_{\text{plain}})^{0.291}} \quad (6)$$

The performance analysis was made by Eq 5 and 6 and the results are shown in Figs.7 and 8. It is observed from Figs.7 and 8 that the performance ratio increases with increasing Reynolds number and decreasing twist ratio with the maximum for the twist ratio 2.93 for both laminar and turbulent flow. Also, the performance ratio of more than one indicates that the type of twist inserts can be used effectively for heat transfer augmentation.

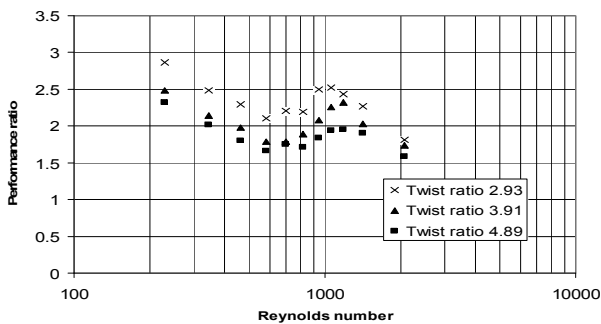


Fig. 7 performance analysis for right-left helical insert with 300 R and 300 L with spacer under laminar flow

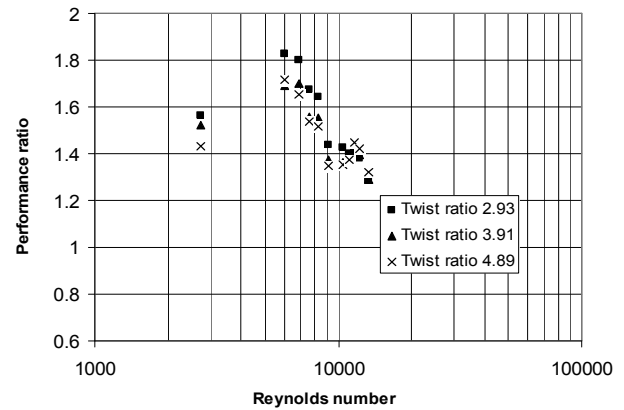


Fig. 8 Performance analysis for right-left helical insert with 300 R and 300 L with 100 spacer under turbulent flow

VIII. CONCLUSIONS

(i) Experimental investigation of heat transfer and friction factor characteristics of circular tube fitted with 300 right-left helical screw inserts with 100 mm spacer have been presented.

(ii) The experimental data obtained were compared with those obtained from plain tube published data.

(iii) The heat transfer coefficient enhancement for 300R-L inserts with 100 mm spacer is quite comparable with 300R-L inserts.

(iv) Performance evaluation analysis has been made and found that the performance ratio increases with increasing Reynolds number and decreasing twist ratio with the maximum for the twist ratio 2.93 both laminar and turbulent flow.

NOMENCLATURE

- A_i inside surface area of test section area, m^2
- A_o outside surface area of test section area, m^2
- C_p specific heat at constant pressure, $KJ/kg \text{ } ^\circ K$
- D_i inside diameter of test section, mm
- D_o outside diameter of test section, mm
- f friction factor, dimensionless
- f_{plain} friction factor for plain tube, dimensionless
- f_{twist} friction factor for twist, dimensionless
- h_i average convective heat transfer coefficient, $W/m^2 \text{ } ^\circ K$
- k thermal conductivity of fluid, $W/m \text{ } ^\circ K$
- k_w thermal conductivity of the tube wall, $W/m \text{ } ^\circ K$
- L length of the test section, m
- L_t left twist length, m
- Nu Nusselt number, dimensionless $Nu = h_i D_i / k$
- Nu_{plain} Nusselt number for plain tube, dimensionless
- Nu_{twist} Nusselt number for twist, dimensionless
- Q heat transfer rate, W
- Pr Prandtl number dimensionless $Pr = C_p \mu / k$
- R resistance of the heating element, ohm (ζ)
- R_t right twist length, m
- Re Reynolds number based on internal diameter of the tube, dimensionless
- T_f average of fluid temperature in the test section, $^\circ K$
- T_{in} inlet bulk temperature of fluid, $^\circ K$

T_{out} outlet bulk temperature of fluid, °K
 T_{wo} average wall surface temperature outside test section, °K
 u_m bulk average fluid velocity, m/sec
 U_o over all heat transfer coefficient, $W/m^2 \cdot ^\circ K$
 V voltage output from the Auto-transformer, V
 Y twist ratio (Length of one full twist (360°)/ diameter of the twist), dimensionless

Greek Letters

ρ density of fluid, kg/m^3
 μ viscosity of fluid, Ns/m^2
 ΔP pressure drop of fluid, N/m^2

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