EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION USING “TWISTED ALUMINIUM TAPER CLIPS” AND “TWISTED TAPES” AS INSERTS

A THESIS SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENTS FOR THE DEGREE OF

Bachelor of Technology
in
Chemical Engineering

By

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CERTIFICATE

This is to certify that the thesis entitled, "EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION USING TWISTED ALUMINIUM TAPER CLIPS AND TWISTED TAPES" submitted by Sri Vivek Chandan Mohapatra and Debashis Sahu in partial fulfillments for the requirements for the award of Bachelor of Technology Degree in Chemical Engineering at National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by them under my supervision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University / Institute for the award of any Degree or Diploma.

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ABSTRACT

Heat exchangers have several industrial and engineering applications. The design procedure of heat exchangers is quite complicated, as it needs exact analysis of heat transfer rate and pressure drop estimations apart from issues such as long-term performance and the economic aspect of the equipment. Whenever inserts are used for the heat transfer enhancement, along with the increase in the heat transfer rate, the pressure drop also increases. This increase in pressure drop increases the pumping cost. Thus it's highly essential not to allow the pressure drop to go beyond a specified value while going for heat transfer enhancement techniques using inserts.

The present paper includes various heat transfer augmentation techniques. A literature review of heat transfer augmentation using twisted tapes has been included. Experimental work on heat transfer augmentation using a new kind of insert called TWISTED ALUMINIUM TAPER CLIP (TATC) is carried out. Inserts when placed in the path of the flow of the liquid, create a high degree of turbulence resulting in an increase in the heat transfer rate and the pressure drop. The work includes the determination of friction factor and heat transfer coefficient for various TATC having different twist ratios. The results are compared with twisted tapes having different twist ratios. Four TATC and two twisted tapes having different twist ratios are used in the study. The performance evaluation criterion R1 is found out to clearly depict the enhancement in the heat transfer rate.

The results are aptly supported by observation tables and figures.

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION:- (1)

Heat exchangers have several industrial and engineering applications. The design procedure of heat exchangers is quite complicated, as it needs exact analysis of heat transfer rate and pressure drop estimations apart from issues such as long-term performance and the economic aspect of the equipment.
The major challenge in designing a heat exchanger is to make the equipment compact and achieve a high heat transfer rate using minimum pumping power.

Techniques for heat transfer augmentation are relevant to several engineering applications. In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. Furthermore, sometimes there is a need for miniaturization of a heat exchanger in specific applications, such as space application, through an augmentation of heat transfer. For example, a heat exchanger for an ocean thermal energy conversion (OTEC) plant requires a heat transfer surface area of the order of 10 000 m2/MW. Therefore, an increase in the efficiency of the heat exchanger through an augmentation technique may result in a considerable saving in the material cost.

Furthermore, as a heat exchanger becomes older, the resistance to heat transfer increases owing to fouling or scaling. These problems are more common for heat exchangers used in marine applications and in chemical industries. In some specific applications, such as heat exchangers dealing with fluids of low thermal conductivity (gases and oils) and desalination plants, there is a need to increase the heat transfer rate. The heat transfer rate can be improved by introducing a disturbance in the fluid flow (breaking the viscous and thermal boundary layers), but in the process pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, several techniques have been proposed in recent years and are discussed in the following sections.
Chapter 2 deals with the literature review that has been done with the various techniques used for enhancement. It gives in detail the observations that were made using twisted tapes as inserts.

Chapter 3 includes the present experimental work which was done with a new kind of insert called twisted Al taper clip along with twisted tapes. It also includes the fabrication of the TATCs.

Chapter 4 deals with the results and discussion about the work done with the inserts. A comparison of friction factor and heat transfer coefficient is made with the smooth tube.
All the observations that were recorded are given in the Appendix.

CHAPTER 2

Literature Review

2.1 CLASIFICATION OF ENHANCEMENT TECHNIQUES (2,3,4)

Heat transfer enhancement, augmentation or intensification deals with the improvement of thermohydraulic performance of heat exchangers.Different enhancement techniques have been identified which can be broadly classified as ***passive*** and ***active techniques***. The primary distinguishing feature is that unlike active methods passive techniques do not require direct input of external power. They generally use surface or geometrical modifications to the flow channel, or incorporate an insert, material, or additional device. Except for extended surfaces, which increase the effective heat transfer area, these passive schemes promote higher heat transfer coefficients by disturbing or altering the existing flow behavior. This is however accompanied by an increase in the pressure drop. In case of active techniques the addition of external power essentially facilitates the desired flow modification and the concomitant improvement in the heat transfer rate. The use of two or more techniques (passive and/or active) in conjunction with the purpose of further enhancement of heat transfer, constitutes ***compound enhancement.***

The effectiveness of any of these methods is strongly dependent on the mode of heat transfer (single-phase free or forced convection, pool boiling, forced convection boiling or condensation, and convective mass transfer) and type and process application of the heat exchanger. The ***passive techniques*** are as follows:

- *Treated surfaces* are heat transfer surfaces that have a fine scale altercation to their finish or coating. The altercation could be continuous or discontinuous, where the roughness is much smaller than what affects single-phase heat transfer and they are primarily used for boiling and condensation duties.
- *Rough surfaces* are generally surface modifications that promote turbulence in the flow field , primary in single phase flows and do not increase the heat transfer surface area. Their geometric features range from random sand-grain roughness to discrete three dimensional surface protuberances.
- *Extended surfaces* more commonly referred to as finned surfaces provide an effective heat transfer surface area enlargement. Plain fins have been used routinely in many heat exchangers. The newer developments , however have led to modified finned surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.
- *Displaced enhancements devices* are inserts that are used primarily in confined forced convection and they improve energy transport indirectly at the heat exchange surface by "displacing " the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.
- *Swirl flow devices* produce and superimpose swirl or secondary recirculation on the axial flow in a channel. They include helical strip or cored screw-type tube inserts ,twisted ducts, and various forms of altered (tangential to axial direction) flow arrangements , and they can be used for single –phase as well as two- phase flows
- *Coiled tubes* are what the name suggests, and they lead to relatively more compact heat exchangers. The tube curvature due to coiling produces secondary flows or Dean vortices, which promote higher heat transfer heat coefficients in single phase flows as well as in most regions of boiling.
- *Surface tension devices* consist of wicking or grooved surfaces, which direct and improve the flow of liquid to boiling surfaces and from condensing surfaces.

- *Additives for liquids* include the addition of solid particles, soluble trace additives, and gas bubbles in single–phase flows, and gas bubbles in single –phase flows, and trace additives, which usually depress the surface tension of liquid, for boiling systems.
- *Additives for gases* include liquid droplets or solid particles, which are introduced in single-phase gas flows in either a dilute phase (gas-solid suspensions) or dense phase (fluidized beds).

Descriptions for the various ***active techniques*** are as follows:

- *Mechanical aids* are those that stir the fluid by mechanical means or by rotating the surface. The more prominent examples include rotating tube heat exchangers and scraped-surface heat and mass exchangers.
- *Surface vibration* has been applied primarily, at either low or high frequency, in single phase flows to obtain higher convective heat transfer coefficients.
- *Fluid vibration* or fluid pulsation, with vibrations ranging from 1.0 Hz to ultrasound (~1.0 MHz), used primarily in single–phase flows, is considered to be perhaps the most practical type of vibration enhancement technique.
- *Electrostatic fields* which could be in the form of electric or magnetic fields, or a combination of the two, from dc or ac sources, can be applied in heat applied in heat exchange systems involving dielectric fluids. Depending on the application ,they can promote greater bulk fluid mixing and induce forced convection (corona “wind”) or electro-magnetic pumping to enhance heat transfer.
- *Injection,* used only in single–phase flow, pertains to the method of injecting the same or a different fluid into the main bulk fluid either through a porous heat transfer interface or upstream of the heat transfer section.
- *Suction* involves either vapor removal through a porous heated surface in nucleate or film boiling, or fluid withdrawal through a porous heated surface in single- phase flow.
- *Jet impingement* involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface. Single or multiple jets (in clusters or staged axially along the flow channel) may be used in both single –phase and boiling applications.

Furthermore, as mentioned earlier, any two or more of these techniques (passive and/or active) may be employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by any of those techniques separetly. This simultaneous utilization is termed ***compound enhancement.*** Some promising applications, for example, are in heat or mass exchangers where one technique may preexist; this is particularly so when the existing enhancement is from an active method.

2.2 Performance evaluation criteria:

In most practical applications of enhancement techniques, the following performance objectives, along with a set of operating constraints and conditions, are usually considered for optimizing the use of a heat exchanger:

1. Increase the heat duty of an existing heat exchanger without altering the pumping power (or pressure drop) or flow rate requirements.
2. Reduce the approach temperature difference between the two heat-exchanging fluid streams for a specified heat load and size of exchanger.
3. Reduce the size or heat transfer surface area requirements for a specified heat duty and pressure drop

4. Reduce the process stream's pumping power requirements for a given heat load and exchanger surface area .

It may be noted that objectives 1,2 and 4 yield savings in operating (or energy) costs, and objectives 3 lends to material savings and reduced capital costs. These objective functions have been described by many different performance evaluation criteria (PEC) in the literature (Bergles et al., 1974 a,b; Webb, 1981,1994;).

The heat exchanger performance is represented by two dependent variables; heat transfer rate Q and pressure drop Δ p or pumping power P, which can be expressed as

$$\mathbf{Q = (UA)\ \Delta Tm}$$

$$\mathbf{\Delta p = 2f\, LG^2/di\, \rho \quad and \quad P = \Delta p\, G\, Ac/\rho}$$

Here the primary independent operating variables are the approach temperature difference ΔTi ($\Delta Tm = \varphi(\Delta Ti)$) and the mass flow rate m, and in the case of the tubular geometry, the design variables (heat transfer surface area A or exchanger size) are the diameter di and length L of tubes and number of tubes per pass. PEC are established for the process stream by selecting one of the operational variables for the performance objective and applying the design constraints on the remaining variables.

For single phase flow heat transfer inside enhanced and smooth tubes of the same envelope diameter, PEC for 12 different cases outlined by Bergles (1998) and Webb (1994) are listed in the table 2.1.

<u>Table 2.1 PEC criteria</u>

Case	geometry	m	P	Q	Δ Ti	objective
FG-1a	N,L	x			x	Q **increase**
FG-1b	N,L	x		x		**ΔTi decrease**
FG-2a	N,L		x		x	Q **increase**
FG-2b	N,L		x	x		**ΔTi decrease**
FG-3	N,L			x	x	P **decrease**
FN-1	N		x	x	x	L **decrease**
FN-2	N	x		x	x	L **decrease**
FN-3	N	x		x	x	P **decrease**
VG-1	_	x	x		x	(NL)^a **decrease**
VG-2a	(NL)^a	x	x		x	Q **increase**
VG-2b	(NL)^a	x	x	x		**ΔTi decrease**
VG-3	(NL)^a	x		x	x	P **decrease**

1.<u>FG criteria</u>- the area of flow cross section (N and di) and tube length L are kept constant.this would typically be applicable for retrofitting the smooth tubes of an existing thereby maintaining the same basic geometry and size (N, di, L). the objectives then could be to increase the heat load capacity Q for the same approach temperature **Δ** Ti and mass flow rate m or pumping power P; or decrease **Δ**Ti or P for fixed Q and m or P; or reduce P for fixed Q.

2.<u>FN criteria</u>- the flow frontal area or cross section (N and di) is kept constant, and the heat exchanger length is allowed to vary. Here the objectives are to seek a reduction in either the heat transfer area (A → L) or the pumping power P for a fixed heat load.
3.<u>VG criteria</u>- the flow frontal area or cross section (N and L) are kept constant , but their diameter can change. A heat exchanger is often sized to meet a specified heat duty Q for a fixed process fluid flow rate m. Because the tube side velocity reduces in such cases so as to accommodate the higher friction losses in the enhanced surface tubes, it becomes necessary to increase the flow area to maintain constant m. This is usually accomplished by using a greater number of parallel flow circuits.

For the quantitative evaluation of these PEC, algebric expression can be obtained that relate the enhanced surface performance (Nu or j and f. Re) with that of an equivalent smooth duct. For a specified tube bundle geometry (N, L, di) in a shell and tube heat exchanger, the heat transfer coefficient h and pumping power P can be expressed as

$$h = C_p\, j\, G/ Pr^{2/3}$$

$$P = \frac{fAG^3}{\rho^2} \times \frac{2L}{d_i}$$

Thus the performance of enhanced tubes can be related to that of equivalent smooth tubes (N, L, di same) as

$$\frac{(h/h_o)(A/A_o)^{\frac{1}{3}}}{(P/P_o)^{\frac{1}{3}}} = \frac{j/j_o}{(f_a/f_o)^{\frac{1}{3}}}$$

given either j(or Nu) and f data, or correlations for both the enhanced and smooth duct, evaluation of the objectives for the PEC is rather straight forward. One of the groupings hA/h_oA_o, (P/P_o), (A/A_o) becomes the objective function with the other two set as 1.0 for the corresponding operating constraints which also provide the mass flux ratio G/G_o, required to satisfy the above equation.
A common enhancement objective for many batch processing applications in the chemical and process industry is to increase thermal performance of a given heat exchanger by using enhancement techniques but without changing the pumping power and approach temperature difference requirements. This corresponds to the FG-2a criterion of the above table which can be implemented by expressing the ratio of the heat transfer rate for the enhanced and smooth duct respectively as

$$Q/Q_o= (Nu/Nu_o)_{N,L,di,}\Delta Ti,P$$

The constraint of fixed pumping power can be expressed as

$$(f.Re^3)= (f_o.Re_o^3)$$

to establish the relationship between the Reynolds numbers (or mass fluxes G and Go) for the enhanced and smooth flow passages.

In designing new enhanced surface heat exchangers for a specified heat duty, approach temperature difference, and pressure drop, a reduction of the required heat transfer surface area is often primary objective. The consequent FN-1 criterion of the table can be stated as

$$A/A_o = (Nu_o/Nu)_{N,di,Q,}\ \Delta Ti,\ \Delta p$$

In this case the Reynolds numbers or mass fluxes in the enhanced and smooth duct geometries are related by the fixed **Δp** requirements as follows

$$(f.Re^2) = (f_o.Re_o^2)$$

Some additional considerations and variations of these PEC for single phase which are based on a systems of complete heat exchanger unit approach have also been considered. Bergles (1974b), Nelson and Bergles (1986) and Webb(1994) provide the necessary guidance and details of their algebric development. Furthermore in a very recent study Zimparov(2000) has outlined the application of PEC to compound enhancement that involves use of twisted tape inserts in spirally corrugated(rough) tubes.

The present work deals with the use of swirl flow devices as inserts as a form of passive technique in enhancing the heat transfer rate.

2.3Swirl flow devices:-

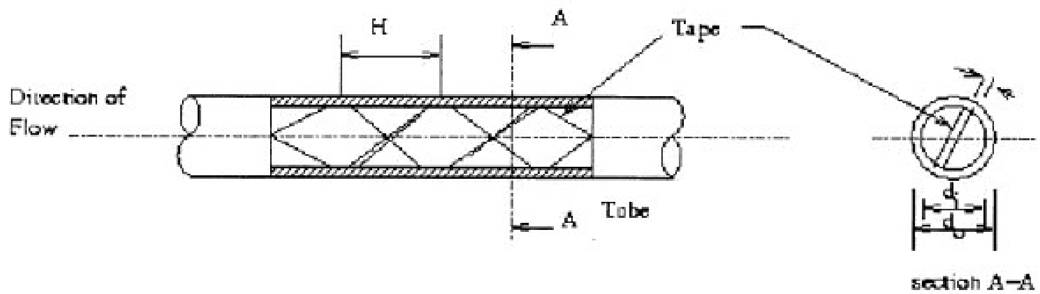
Swirl flow devices generally consist of a variety of tube inserts (twisted tubes, wire coils, ribs and dimples), geometrically varied flow arrangements, and duct geometry modifications that produce secondary flows. Typical examples of each of these techniques include twisted-tape inserts, periodic tangential fluid injection, and helically twisted tubes. Of these, twisted- tape inserts have received considerable attention in the literature, and their thermal hydraulic performance in single phase, boiling, and condensation forced convection, as well as design and application issues, have been discussed in great detail(Manglik and Bergles; 2002a)

2.4 Single phase flow:-

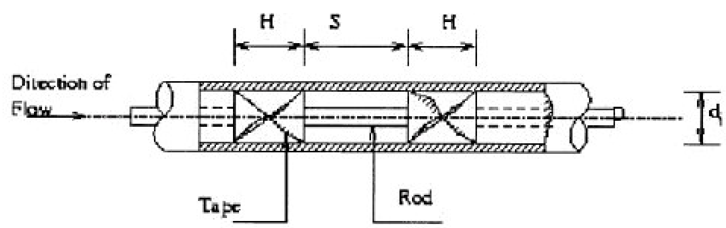
Perhaps the most effective and widely used swirl flow device for single-phase flows is the twisted-tape insert, which has design and application literature dating back more than a century. It has been shown to increase significantly the heat transfer coefficient with a relatively small pressure drop penalty. Frequent use of twisted tapes is in retrofit of existing shell and tube heat exchangers to upgrade their heat duties. Also when employed in a new exchanger for a specified heat duty, significant reduction in size can be achieved. The ease of fitting multitube bundles with twisted tape inserts and their removal, makes them particularly useful in fouling situations, where frequent tube side cleaning is required.

Heat transfer enhancement in a tube flow by inserts such as twisted tape, wire coils, ribs and dimples is mainly due to flow blockage, partioning of the flow and secondary flow. Flow blockage increases the pressure drop and leads to increased viscous effects because of a reduced free flow area. Blockage also increases the flow velocity and in some cases lead to significant secondary flow. Secondary flow further provides a better thermal contact between the surface and the fluid because secondary flow creates swirl and the resulting mixing of fluid improves the temperature gradient, which ultimately leads to a high heat transfer coefficient.

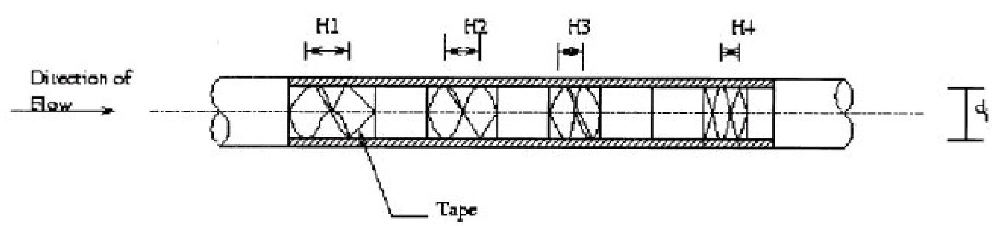
Twisted tape generates a spiral flow along the tube length. A wire coil insert in a tube flow consists of a helical coiled spring which functions as a non integral roughness. Figure 2.4.1 shows three typical configurations of twisted tape with 180^0 twisted pitch, and figure 2.4.2shows a typical configuration of a wire coil. In a turbulent flow , the dominant thermal resistance is limited to a thin viscous sublayer,. The wire coil insert is more effective in a turbulent flow compared with a twisted tape, because wire coil mixes the flow in the viscous sub layer near the wall quite effectively, whereas a twisted tape cannot properly mix the flow in the viscous sub layer. For a laminar flow, the dominant thermal resistance is limited to a thicker region compared with a turbulent flow. Thus a wire coil insert is not effective in a laminar flow because it cannot mix the bulk flow well, and the reverse is true for a twisted tape insert. Hence, twisted tapes are generally preferred in a laminar flow.



(a)



(b)



(c)

Fig 2.4.1 Example of (a) full-length twisted tape, (b) regularly spaced twisted tape and (c) smoothly varying (gradually decreasing) pitch full-length twisted tape

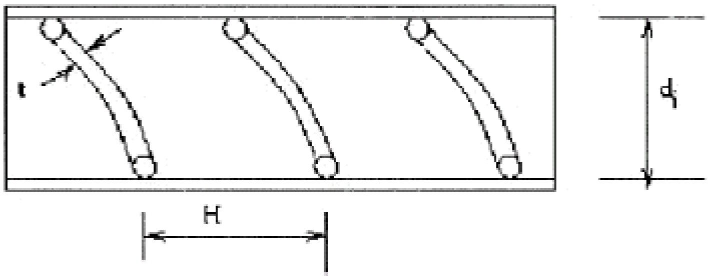


Fig 2.4.2 Example of wire coil insert

2.5 Laminar and turbulent flow through a circular tube:-

For fully developed laminar flow in a circular tube without any insert, the Nusselt number has a constant value under the condition of constant wall temperature, Nu = 3.657, and the friction factor for this flow is given as f = 16/Re. for fully developed turbulent flow in a smooth circular tube, the Nusselt number can be predicted by the Dittus- Boelter correlation, $Nu = 0.023\ Re^{0.8}\ Pr^{0.4}$, and the friction factor can be obtained by the correlation
$f = 0.079\ Re^{-0.25}$.

For Re<2100 the value of friction factor is calculated using the equation

$$f = \frac{16}{\text{Re}}$$

For the calculation of heat transfer coefficient:-

For Re<2100

Nu = f(Gz)

For Gz < 100

$$Nu = 3.66 + \frac{0.085Gz_1}{1+0.047Gz_1^{\frac{2}{3}}}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

(b) 2100<Re<10000(HOUGEN's EQUATION)

$$\frac{h_i d_i}{k} = 0.116\ (\text{Re}^{\frac{2}{3}} - 125)(\text{Pr}^{\frac{1}{3}})[1 + (\frac{d_i}{L})^{\frac{2}{3}}][\frac{\mu_b}{\mu_w}]^{0.14}$$

(c) Re>10000(DITTUS BOILTER EQUATION)

$$\frac{h_i d_i}{k} = 0.023\text{Re}^{0.8}\ \text{Pr}^{0.33}$$

2.6 Review :-

An extensive literature review of all types of heat transfer heat augmentation techniques using external inserts up to 1985 has been discussed by Bergles[1]. In the following subsections, literature involving recent work on passive heat transfer augmentation techniques involving twisted tapes, wire coils, dimples, ribs, and fins as an insert has been reviewed.

2.7 TWISTED TAPE IN LAMINAR FLOW:- (3,5,6)

A summary of important investigations of twisted tape in a laminar flow is represented is presented in table 1. Twisted tape increases the heat transfer coefficient with an increase in the pressure drop. Several researchers have studied various configurations of twisted tapes, such as full-length twisted tape, short length twisted tape, full length twisted tape with varying pitch and regularly spaced twisted tape. This section discusses which configuration of twisted tape is suitable for laminar flow.

Whitman[2] studied heat transfer enhancement by means of a twisted tape insert way back at the end of the nineteenth century. Date and Singham[3] numerically investigated heat transfer enhancement in laminar flow, viscous liquid flows in a tube with a uniform heat flux boundary condition. They idealized the flow conditions by assuming zero tape thickness, but the twist ratio and fin effects of the twisted tape were included in their analysis.

Saha et al.[4,5,12] reported experimental data on a twisted tape generated laminar swirl flow friction factor and Nusselt number for a large Pradntl number ($205 < Pr < 518$) and observed that , on the basis of a constant pumping power , short length twisted tape is a good choice because in this case swirl generated by the twisted tape decays slowly downstream which increases the heat transfer coefficient with minimum pressure drop, as compared to full length twisted tape. Regularly spaced twisted tape decreases the friction factor and reduces the heat transfer coefficient because the spacing of twisted tape disturbs the swirl flow. Hong and Bergles[6] reported heat transfer enhancement in laminar, viscous liquid flows in a tube with uniform heat flow boundary conditions, but their correlation has limited applicability as it is valid for a high Prandtl number (approx 730). The circumferential temperature profile for swirl flow is related to tape orientation. Tariq et al.[7] found that in a laminar flow the introduction of turbulent parameters, such as as internally threaded tube, is not efficient compared to a twisted tape insert on the basis of the overall efficiency.

Depending upon the flowrate and tape geometry, the enhancement in heat transfer is due to the tube partitioning and flow blockage, the large flow path and secondary fluid circulation. Manglik and Bergles [8] considered all these effects and developed laminar flow correlations for the friction factor and Nusselt number, including the swirl parameter, which defines the interaction between viscous,convective inertia and centrifugal forces. These correlations pertain to the constant wall temperature case for fully developed flow, based on both previous data and their own experimental data. The heat transfer correlation as proposed by them is

$$\mathbf{Nu = 4.162[6.413 \times 10^{-9} (SW\ \ Pr^{0.391})^{3.385}]^{3.385}\,]^{0.2} (\mu_b / \mu_w)^{0.14}}$$

where Sw is the swirl parameter and is defined as

$$\mathbf{Sw = Re/y^{0.5}} .$$

Based on the same data, a correlation for the friction factor was developed by Manglik and Bergles [8] and is given as

$$\mathbf{f_s\,Re_s = 15.767\ \underline{[\pi + 2 - 2(\delta/d)]^2}\ (1 + 10^{-6}\,Sw^{2.55})^{1/6}\ [\pi - 4(\delta/d)]}$$

where, d and **δ** are the tube inner diameter and the thickness of the twisted tape respectively. Date and Ray [9] derived a correlation for the friction factor and Nusselt number for a square duct from the predicted data. They compared the correlation for the friction factor with experimental data and the agreement was found to be within $\pm$10 per cent.

Lokanath and Misal [10] studied the performance of a plate heat exchanger and augmented shell and tube heat exchanger for different fluids. They found that twisted tapes of tighter twists are expected to give higher overall heat transfer coefficients in the augmented shell and tube heat exchanger. Shah and Skiepko [11] investigated the existence of entropy generation extrema and a relationship between the extrema and the heat exchanger effectiveness. Saha et al. [12] found that pinching (placing of a twisted tape exactly at the centre of the tube) of twisted tape in a tube performs better than a twisted tape inserted by a loose fit. They further showed that a non-zero phase angle in-between the segmented twisted tape gives poor results because the swirl will break easily in-between the two segmented twisted tapes. Furthermore, reduction in the width of the twisted tape is not effective, compared with a twisted tape of width equal to the inside diameter of the tube.

AI-Fahed et al. [13] observed that, for a low twist ratio (Y = 5.4) and high pressure drop, a loose fit is recommended for design of the heat exchanger, since it is easier to install and to remove for cleaning purposes.

Other than this twist ratio, a tight-fit twisted tape provides better performance than a loose-fit twisted tape. Liao and Xin [14] reported experimental data on the compound heat transfer enhancement technique and concluded that the enhancement of heat transfer in a tube with three dimensional internal extended surfaces by replacing continuous twisted tape with almost segmented twisted tape inserts results in a decrease in the friction factor but with a comparatively small decrease in the Stanton number. The Stanton number is defined as the ratio of heat transfer rate to the enthalpy difference and is a measure of the heat transfer coefficient.

Ujhidy et al. [15] have studied laminar flow of water in coils and tubes containing twisted tapes and helical static elements and proposed a modified Dean number that takes into account the curvature of the spherical line cut out by the helical element from the tubular housing. The Dean number is a measure of the magnitude of the secondary flow. Suresh Kumar et al. [16, 17] investigated the thermohydraulic performance of twisted tape inserts in a large hydraulic diameter annulus.

The thermohydraulic performance in laminar flow with a twisted tape is better than the wire coil for the same helix angle and thickness ratio. This is probably due to the fact that, in laminar flow, the dominant thermal resistance is not limited to a thin wall region but extends over the entire cross-section. Thus, a twisted tape insert mixes the bulk flow and is probably effective..

Saha and Chakraborty [19] observed laminar flow heat transfer and pressure drop characteristics in a circular tube fitted with regularly spaced twisted tape and concluded that there is a drastic reduction in the pressure drop, more than the corresponding reduction in heat transfer. Thus, it appears that, on the basis of a constant pumping power, a large number of turns may yield improved thermohydraulic performance compared with a single turn on twisted tape. Lokanath [20] represented experimental data on laminar flow of water through a horizontal tube under uniform heat flux condition and fitted with half-length twisted tape. He found that, on the basis of unit pressure drop and unit pumping power, half-length tapes are more effective than full-length tapes.

Table 1 Summary of important investigations of twisted tape in laminar flow

	Authors	Fluid	Configuration of twisted tape	Type of investigation	Observations	Comments
1	Saha and Dutta [4]	Water with ($205 < Pr < 518$)	(a) Short length (b) Full length (c) Smoothly varying pitch (d) Regularly Spaced	Experimental in a circular tube	(1) Friction and Nu low for short length tape (2) Short length tape requires small pumping power (3) Multiple twist and single twist has no difference on thermohydraulic performance (4) Uniform pitch (Fig. 1a) performs better than gradually decreasing pitch	Friction factor and Nusselt number increase with insertion of any passive augmentation technique Of several passive techniques studied in last decade, it was observed that twisted tape is effective in laminar flow
2	Bergles and Hong [6]	Water ($3 < Pr < 7$) ($83 < Re < 2460$) Ethylene Glycol ($84 < Pr < 192$) ($13 < Re < 390$)	Full-length twisted tape	Experiment in circular tube	(1) *Nu* is function of twist ratio, *Re* and *Pr* (2) Friction is affected by tape twist only at high *Re* (3) *Nu* is 9 times that of empty tube	Further, twisted tape has been used as full-length twisted tape, half-length twisted tape and varying-pitch twisted tape
3	Tariq *et al.* [7]	Air ($1300 < Re < 10^4$)	No insert	Experiment in internally threaded tube	(1) Efficiency lower for threaded tube than twisted tape, but better than some fluted geometries (2) Good promoter of turbulence (3) Heat transfer coefficient in internally threaded tube approximately 20 per cent higher than that in smooth tube	It was observed that short-length twisted tape is better than full-length one on basis of thermohydraulic performance Thermohydraulic performance for square tube with twisted tape is better than that of twisted tape in circular tube
4	Saha and Bhunia [5]	Servotherm medium oil ($205 < Pr < 512$, $45 < Re < 840$)	Twisted tape (twist ratio $2.5 \leqslant Y \leqslant 10$)	Experiment in circular tube	(1) Heat transfer characteristics depend on twist ratio, *Re* and *Pr* (2) Uniform pitch performs (Fig. 1a) better than gradually decreasing pitch (Fig. 1c)	Uniform pitch twisted tape (Fig. 1a) performs better than gradually varying pitch twisted tape (Fig. 1c)
5	Manglik and Bergles [8]	Water ($3.5 < Pr < 6.5$) and ethylene glycol ($68 < Pr < 100$)	Three different twist ratios: 3, 4.5 and 6	Experiment in isothermal tube	(1) Proposed correlation for friction and Nusselt number (2) Physical description of enhancement mechanisms	Pinching of twisted tape gives better results compared with connected thin rod
6	Ray and Date [9]	Water ($100 < Re < 3000$, $Pr \leqslant 500$)	Full-length twisted tape with width equal to side of duct	Numerical work for square duct	(1) Proposed correlations for friction and *Nu* (2) Higher hydrothermal performance for square duct than circular one (3) Local Nusselt number peaks at cross-sections where tape aligned with diagonal of duct	
7	Lokanath and Misal [10]	Water ($3 < Pr < 6.5$ and lube oil (*Pr* 418)	Twisted tape	Experiment in plate heat exchanger and shell and tube heat exchanger	(1) Large value of overall heat transfer coefficient produced in water-to-water mode with oil-to-water mode	
8	Shah and Skiepko [11]	Water	No insert	Experiment in heat exchanger	(1) Irreversibility minimum for energy conversion process thermal system and for heat exchanger irreversibility should be intermediate	

(*Table continued*)

2.8 Twisted tape in turbulent flow:--- (3,5)

The important investigations of twisted tape in turbulent flow are summarized in Table 2. In turbulent flow, the dominant thermal resistance is limited to a thin viscous sublayer near the wall. The following section discusses the performance of twisted tape inserts in turbulent flow. A tube inserted with a twisted tape performs better than a plain tube, and a twisted tape with a lower twist ratio provides an improved heat transfer rate at a cost of increase in pressure drop for low Prandtl number fluids. This is because the thickness of the thermal boundary layer is small for a low Prandtl number fluid and a tighter twist ratio disturbs the entire thermal boundary layer, thereby increasing the heat transfer with increase in the pressure drop, as discussed by Royds [21]. Smithberg and Landis[22] gave an analytical model of the tape-generated swirl mechanism.

Because of swirl, the ratio of maximum velocity to mean velocity is smaller in a tube with a twisted tape compared with that in a straight flow (i.e. without a twisted tape). This creates a centrifugal force and aids convective heat transfer [23–25]. Twisted tape is also effective in high Prandtl number fluids because for such fluids it provides high heat transfer with less pressure drop increase compared with other inserts [26]. Lopina and Bergles [27] observed that the difference between isothermal and heated flow friction factors for the swirl flow of liquids is substantially less than the corresponding difference for a plain tube. In turbulent flow, insertion of a twisted tape increases the heat transfer, but the pressure drop also increases significantly [28, 29]. Short-length twisted tape (25–45 per cent of the tube length) performs better than full-length twisted tape [30–33]. Date [34] reviewed available friction factor and Nusselt number correlations for flow in a tube containing a twisted tape and pointed out that existing correlations deviate from measurements by 30 per cent. Studies of Klepper [35] and Kidd Jr [36] suggest that short-length twisted tape is more useful in a gas-cooled nuclear reactor compared with full-length tape. Date [37] formulated and solved numerically the problem of fully developed, uniform property flow in a tube containing a twisted tape. He compared existing experimental data with his own numerical prediction. Bolla et al. [38] observed that in the turbulent flow region a transverse rib performs better than a twisted tape insert in a duct flow. Zozulya and Shkuratov [39] found that a smooth decrease in pitch of a twisted tape results in an improved heat transfer rate. Huang and Tsou [40] studied free swirl flow in a pipe. Blackwelder and Kreith [41] recommended that an optimum design of heat exchanger with twisted tape induced swirl flow should consist of continuous and decaying swirl flow. Backshall and Landies [42] studied the boundary layer characteristics of twisted tape generated incompressible swirl flow. Watanable et al. [43] observed that the maximum thermal stress appears near the piping connecting parts of the cover plate of the plate fin heat exchanger channels. Genis and Rautenbach [44] studied the thermohydraulic characteristics of a high-velocity water flow in short tubes with twisted tape inserts. Budov et al. [45] predicted a loss of head owing to hydraulic resistance in a twisted tape generated swirl flow. Beckermann and Goldschmid [46] and Yamada et al. [47] observed that, in addition to convection, radiative heat transfer from the relatively hot (1260 8C) cross-twisted tape in the flue-way of a water heater plays an important role in the heat transfer to the tube wall and this phenomenon cannot be neglected.

Gupte and Date [48] evaluated semi-empirically the heat transfer coefficient and friction factor for twisted tape generated swirl flow in an annulus. Twisted tape generated swirl flow in a vertical test section (contrary to the usual practice of a horizontal test section) was experimentally studied by Filipak [49]. Donevski and Kulesza [50] predicted frictional losses from the combined effects of axial and tangential boundary layer flow coupled with an additional vortex mixing effect. Rao [51] performed an experimental study of augmentation of heat transfer in the axial ducts of electrical machines. Fomina et al. [52] verified the first pilot plant having an economizer with twisted tape fins. Algifri and Bharadwaj [53] and Algifri et al. [54] presented a series of solutions to governing equations for short-length twisted tape generated decaying swirl. Burfoot and Rice [55] found that the surface roughness of a tape insert affects the thermohydraulic characteristics. Kumar and Bharadwaj [56] obtained theoretically the heat transfer and pressure drop correlations using the Kreith and Sonju [32] solution for the velocity vector, which decays along the axis of the tube. Kumar and Prasad [57] investigated experimentally the performance of a tape-inserted solar water heater. Fujita and Lopez [58], in an experimental investigation, found no evidence of tape fin effects in the heat transfer characteristics of a snugly fitted stainless steel tape insert when compared with a Teflon tape insert.

AI-Fahed et al. [59] observed that there is an optimum tape width, depending on the twist ratio and Reynolds number, for the best thermohydraulic characteristics. Saha et al. [60] have shown that, for a constant heat flux boundary condition, regularly spaced twisted tape elements do not perform better than full-length twisted tape because the swirl breaks down in-between the spacing of a regularly twisted tape. Rao and Sastri [61], while working with a rotating tube with a twisted tape insert, observed that the enhancement of heat transfer offsets the rise in the friction factor owing to rotation. Sivashanmugam and Sundaram [62] and Agarwal and Rao [63] studied the thermohydraulic characteristics of tape-generated swirl flow. Peterson et al. [64] experimented with high-pressure (8–16 MPa) water as the test liquid in turbulent flow with low heat fluxes and low wall–fluid temperature differences typical of a liquid–liquid heat exchanger.

Naumov et al. [65, 66] used a mathematical model for an adiabatic cross-section to study the pulsed asymmetric heating of pipes by an external heat flux and calculated the thermophysical parameters of cooled pipes containing twisted tapes within the collector of deviated ions in the T-15 tokamak injection system. Yokoya et al. [68] demonstrated a novel use of twisted tapes in controlling the flow in a continuous casting mould and refining process. Hijikata et al. [69], while carrying out experiments in a vertical test section, observed that the radiation between the pipe wall and the twisted tape increases the heat transfer rate by about 50 per cent. Chung and Sung [70] performed a direct numerical simulation for turbulent heat transfer in a concentric annulus, and they observed that the thermal structure is more effective near the outer wall than near the inner wall. Yang and Hwang [71] observed that a porous type of baffle is good for thermohydraulic performance and there is an optimum height of the baffle. Manglik and Bergles [72] developed correlations for both turbulent flow and laminar flow. For an isothermal friction factor, the correlation describes most available data for laminar, transitional and turbulent flows within 10 per cent. However, a family of curves is needed to develop correlation for the Nusselt number on account of the non-unique nature of laminar– turbulent transition. Their correlations are as follows

$$\mathbf{fRe^{0.25}=0.079\{\pi/(\pi-4\delta/d)\}^{1.75}\{(\pi+2-2\delta/d)^{1.25}/(\pi-4\delta/d)^{1.25})\}\{1+(2.752/y^{1.29})}$$

$$\mathbf{Nu=0.023Re^{0.8}Pr^{0.4}\{\pi/(\pi-4\delta/d)\}^{0.8}\{(\pi+2-2\delta/d)/(\pi-4\delta/d)\}^{0.2}(1+0.769/y)(\mu_b/\mu_w)^n}$$

where n=0.18 for heating and n=0.3 for cooling.

Table 2 Summary of important investigations of twisted tape in turbulent flow

	Authors	Fluid	Configuration of tape	Type of investigation	Observations	Comments
1	Royds [**21**]	Water	(a) Full length (b) Full length with tighter twist ratio	Experimental in circular tube	Tighter twist ratio results in greater heat transfer and pressure drop	In turbulent flow, short-length twisted tape is more effective than full-length twisted tape (Fig. 1a) based on thermohydraulic performance
2	Smithberg and Landis [**22**]	Water	Full length twisted tape	Analytical	Proposed mathematical model for tape generated swirl mechanism	
3	Cresswell [**23**]	Water	Twisted tape	Experiment in circular tube	Ratio of maximum velocity to mean velocity is smaller in swirl flow compared with straight flow	As Prandtl number increases, heat transfer coefficient is enhanced by twisted tape
4	Kreith and Margolis [**24**]	Water	Twisted tape	Experiment in circular tube	Centrifugal force aids convection when fluid is heated up	All configurations (Figs 1a to c) of twisted tape lead to high friction factor, which is due to fact that twisted tape disturbs entire flowfield
5	Thorsen and Landis [**25**]	Water	Twisted tape	Experiment in circular tube	Centrifugal force aids convection when fluid is heated up and inhibits convection when fluid is cooled	Overall enhancement ratio increases with tighter twist ratio and decreases with increase in Reynolds number
6	Gambill and Bundy [**26**]	Water	Twisted tape	Experiment in circular tube	Twisted tape also important for high Prandtl number	If pressure drop is not important, twisted tape is good option to use in turbulent flow
7	Lopina and Bergles [**27**]	Water	Twisted tape	Experiment in circular tube	Isothermal friction factor for swirl flow of liquids is substantially less than for a plain tube	
8	Colburn and King [**30**]	Water	Inserts like baffled tube and short-length twisted tape	Experiment in circular tube	Short-length twisted tapes more effective than full-length twisted tapes	
9	Seigel [**28**]	Water	Twisted tape	Experiment in horizontal tube	In horizontal tube inserted twisted tape increases heat transfer coefficient with increase in pressure drop	
10	Koch [**29**]	Water	Twisted tape	Experiment in circular tube	Increase in heat transfer coefficient accompanied by a comparable increase in pressure drop	
11	Seymour [**31**]	Water	Twisted tape	Experiment in heat exchanger	Short-length twisted tape performs better than full-length twisted tape	
12	Kreith and Sonju [**32**]	Water	Twisted tape	Experiment in circular tube	Short-length (25–45 per cent of tube length) tapes perform better than full-length tapes	
13	Date [**34**]	Water	Twisted tape	Experiment in circular tube	Friction and *Nu* for flow in tube containing tape deviate 30 per cent	
14	Klepper [**35**]	Water	Short-length tape	Experiment in circular tube	Usefulness of tape in gas-cooled nuclear reactor	
15	Kidds Jr [**36**]	Nitrogen	Short-length tape	Tube flow	Effectiveness of twisted tape in gas cooled nuclear reactor	
16	Date [**37**]	Water	Twisted tape	Numerical	Solved numerically problem of fully developed flow	

(*Table continued*)

Table 2 *Continued*

	Authors	Fluid	Configuration of tape	Type of investigation	Observations	Comments
17	Bolla *et al.* [**38**]	Water	Transverse ribs and twisted tape	Experiment in circular tube with both ribs and tape	Transverse rib performs better than tape	
18	Zozulya and Shkuratov [**39**]	Water	Twisted tape	Experiment in circular tube	Smooth decrease in pitch of twisted tape has significant influence on heat transfer	
19	Huang and Tsou [**40**]	Water	Twisted tape	Experiment in circular tube	Studied free swirl flow	
20	Blackwelder and Kreith [**41**]	Water	Twisted tape	Experiment in circular tube	Recommended that optimum design of heat exchanger with tape-induced swirl flow must consider combination of continuous and decaying swirl flow	
21	Backshall and Landies [**42**]	Water	Twisted tape	Experiment in circular tube	Studied boundary layer characteristics of incompressible twisted tape generated swirl flow	
22	Watanable *et al.* [**43**]	Water	Twisted tape	Experiment in heat exchanger	Maximum thermal stress appears near piping that connects parts of cover plate of plate fin heat exchanger channels	
23	Genis and Rautenbach [**44**]	Water	Twisted tape	Experiment in circular tube	Studied thermohydraulic characteristics of high-velocity water flow in short tubes	
24	Budov and Zamyatin [**45**]	Water	Twisted tape	Experiment in cylindrical channel	Loss of head due to hydraulic resistance in twisted tape generated swirl flow	
25	Beckarmann and Goldschimd [**46**]	Water	Twisted tape	Flue way of water heater	Hot cross tapes in flue way of water heater play important role in heat transfer to tube wall	
26	Yamada *et al.* [**47**]	Water	Wall radiation	Experiment in shell and tube heat exchanger	Made similar observation as above	
27	Gupte and Date [**48**]	Air	Twisted tape	Numerical study in annulus	Semi-empirically evaluated friction and heat transfer data for tape-generated swirl flow in annulus	
28	Filipak [**49**]	Water	Twisted tape	Experiment in vertical tube	Made experimental study of tape-generated swirl flow in vertical section	
29	Donevski and Kuleshza [**50**]	Water	Twisted tape	Experiment in circular tube	Predicted frictional losses coupled with additional vortex mixing effect	
30	Rao [**51**]	Water	Twisted tape	Experiment in axial duct flow	Performed experimental study on augmentation of heat transfer in axial ducts of electrical machines	
31	Fomina *et al.* [**52**]	Water	Twisted tape fins	Economizer	Checked first pilot plant having economizer with twisted tape fins	

(*Table continued*)

Table 2 *Continued*

	Authors	Fluid	Configuration of tape	Type of investigation	Observations	Comments
32	Algifri and Bharadwaj [53]	Water	Twisted tape (Short length)	Experiment in circular tube	Presented series of solutions to governing equations for short-length twisted tape generated decaying swirl flow	
33	Algifri *et al.* [54]	Water	Twisted tape	Experiment in circular pipe	Presented series of solutions to governing equations for short-length twisted tape generated decaying swirl flow	
34	Burfoot and Rice [55]	Water	Twisted tape	Experiment in circular pipe	Found that surface roughness of tape insert affects thermohydraulic characteristics	
35	Kumar and Bharadwaj [56]	Water	Twisted tape	Experiment in circular tube	Obtained theoretically heat transfer and pressure drop correlations	
36	Kumar and Prasad [57]	Water	Twisted tape	Experiment in solar water heater	Investigated performance of tape inserted solar water heater	
37	Fujita and Lopez [58]	Water	Twisted tape and teflon tape	Experiment in circular tube	Did not find any evidence of tape fin effects in heat transfer characteristics of snugly fitted stainless steel tape insert when compared with Teflon tape insert	
38	Al-Fahed and Chakroun [59]	Water	Full-length twisted tape	Experiment in horizontal isothermal tube	There is optimum tape width, depending on twist ratio and *Re*, for best thermohydraulic performance	
39	Saha *et al.* [60]	Water	Regularly spaced tape	Experimental study in circular tube	On basis of constant heat flux, regularly spaced tape does not perform better than full-length twisted tape	
40	Rao and Sastri [61]	Water	Twisted tape	Experimental study in rotating twisted tape	Enhancement of heat transfer offsets rise in friction factor due to rotation	
41	Sivanshanmugam and Sunduram [62]	Water	Twisted tape	Experiment in circular tube	Studied thermohydraulic characteristics of tape-generated swirl flow	
42	Agarwal and Raja rao [63]	Water	Twisted tape	Experiment in circular tube flow	Studied thermohydraulic characteristics of tape-generated swirl flow	
43	Peterson *et al.* [64]	Water	Twisted tape	Heat exchanger	Experimented with high-pressure water as test liquid in turbulent flow with low heat fluxes and low wall–fluid temperature differences typical of liquid-to-liquid heat exchanger	
44	Naumov and Semashko [65]	Water	Twisted tape	Experimental study in circular tube	Used mathematical model for adiabatic cross-section to study pulsed asymmetric heating of pipe by external heat flux and calculated thermophysical parameters of cooled pipes containing tapes within collector of seviated ions in T-15 tokamak injection system	

(*Table continued*)

Table 2 *Continued*

	Authors	Fluid	Configuration of tape	Type of investigation	Observations	Comments
45	Naumov *et al.* [66]	Water	Twisted tape	Experimental study in circular tube	Same as above (finite difference method)	
46	Naumov *et al.* [67]	Water	Twisted tape	Experimental study in circular tube	Same as above (finite difference method) with modification	
47	Yokoya *et al.* [68]	Water	Twisted tape	Experimental study	Demonstrated novel use of tapes in controlling flow in continuous casting mould and refining process	
48	Klaczak [33]	($1300 < Re < 8000$)	Shor-length twisted tape	Experimental study in circular tube	Found usefulness of short-length twisted tapes	
49	Hijikata *et al.* [69]	Water		Experiment in vertical test section	Radiation between pipe wall and tape increase heat transfer rate by 50 per cent	
50	Chung and Sung [70]	Air	Transverse curvature	Annulus pipe flow (direct numerical simulation)	Turbulent structure more effective near outer wall compared with inner wall	
51	Yang and Hawang [71]	Air ($1 \times 10^4 < Re < 5 \times 10^4$)	Porous baffles and solid baffles	Numerical in rectangular channel	Porous type baffle is good for thermohydraulic performance, performance does not depend on baffle height	
52	Manglik and Bergles [72]	Water ($3.5 < Pr < 6.5$) and ethylene glycol ($68 < Pr < 100$)	Three different twist ratios 3, 4.5, 6.	Experiment in isothermal tube	(1) Proposed correlation for friction and Nusselt number (2) Physical description of enhancement mechanism	

2.9 OBSERVATIONS:----

2.9.1 Twisted tape in laminar flow**:- (6)**

A twisted tape insert mixes the bulk flow well and therefore performs better in a laminar flow than any other insert, because in a laminar flow the thermal resistance is not limited to a thin region near the wall. However, twisted tape performance also depends on the fluid properties such as the Prandtl number. On the basis of a constant pumping power, short-lengthtwisted tape is better than full-length twisted tape. In the design of a compact heat exchanger for laminar flow, twisted tape can be used effectively to enhance the heat transfer.

2.9.2 Twisted tape in turbulent flow:- (3,5)

Twisted tape in turbulent flow is effective up to a certain Reynolds number range but not over a wide Reynolds number range. Compared with wire coil, twisted tape is not effective in turbulent flow because it blocks the flow and therefore the pressure drop increase is more. Hence, the thermohydraulic performance of a twisted tape is not good compared with wire coil in turbulent flow. Therefore, it may be concluded that, for compact heat exchanger design, wire coil is a good choice in turbulent flow. However, a short-length twisted tape yields good thermohydraulic behaviour compared with full-length twisted tape in turbulent flow.

In the present work experiment for heat transfer rate and friction factor were conducted with twisted tapes and Twisted Al taper clips for turbulent flow. The heat transfer rate and friction factor for TATCs were compared with twisted tapes.

CHAPTER 3

Present experimental work

5.1 PRESENT EXPERIMENTAL WORK

The experimental study on passive heat transfer augmentation using twisted aluminium tapered clips(TATC) and twisted tapes(TT) were carried on in a double pipe heat exchanger having the specifications as listed below:-

specifications--
inner pipe ID = 22mm
inner pipe OD=25mm
outer pipe ID =53mm
outer pipe OD =61mm
material of construction= Cu.
heat transfer length= 2.43m
pressure tapping to pressure tapping length = 2.825m

Water at room temperature was allowed to flow through the inner pipe while hot water (set point 65^{o}C) flowed through the annulus side in the counter current direction.

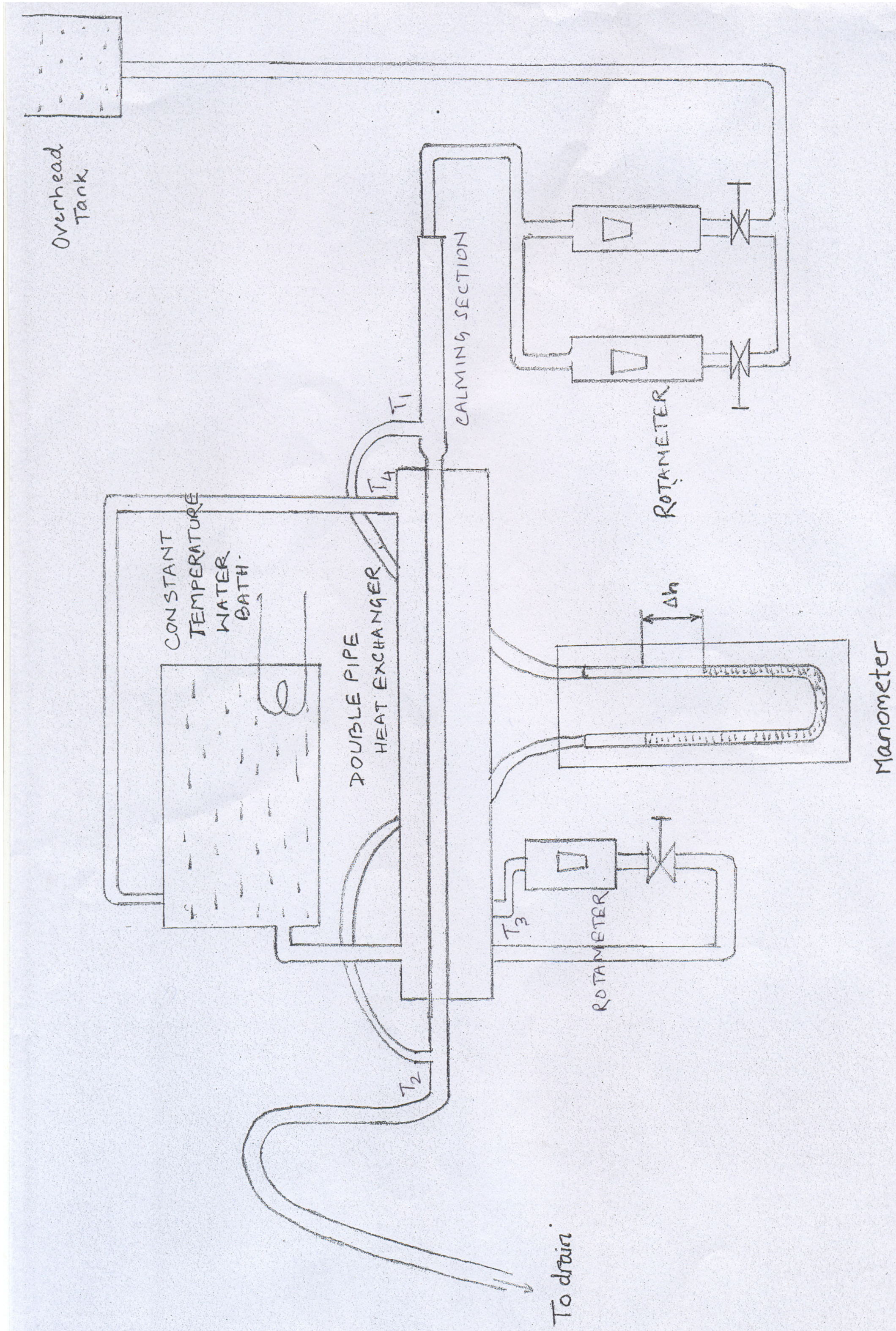
5.2 About the inserts...

The insert used for the experiment are twisted aluminium tapered clips and stainless steel twisted tapes. While much literature can be found about passive heat transfer augmentation using twisted tapes as mentioned earlier, tapered aluminium clips are a new kind of insert where no such experiments have been done thus giving us ample room for experimental studies.
The present work deals with finding the friction factor and the heat transfer coefficient for the TATC's with various twist ratios and comparing those results with that of twisted tapes of varying twists and finally finding the heat transfer enhancement as compared to a smooth tube.

The tapered aluminium clip was chosen for its wide availability and low cost. These tapes are used for plywood partitioning and is available in any plywood shop. Each tapered aluminium clip originally measured 12ft long and 20mm wide. While the twisted tapes were 20mm wide.

5.3 Fabrication..

The aluminium tapered clips(length 12ft) were first cut into 3 equal sizes .Holes were drilled at both ends of each tape so that the two ends could be clamped. Lathe was used to give the tapes the desired twist. One end was kept fixed on the tool part of the lathe while the other end was given a slow rotatory motion by holding it on the chuck side, while the tape was kept under tension by applying a mild pressure on the tool part side, to avoid its distortion, thus creating the required twist in the tapes. Three tapes with varying twist ratios were fabricated (Y=8.73,Y=5.23,Y=3.27) as shown in fig 3.1 to fig 3.4. The end portions of the fabricated tapes were cut and holes drilled for joining the two tapes. Each twisted tapered clip measured 1.15m. Three tapes with the same twist ratio and twist in the same direction were joined, thus giving a total length of 3.45m, sufficient for the double pipe heat exchanger used for the experimental study.
The twisted tapes (Y=5.05, Y=3.25) are shown in fig 3.5 and fig 3.6
The cross sectional view of the Al Taper Clip is shown in fig 3.7.



SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET UP

Experimental Set Up

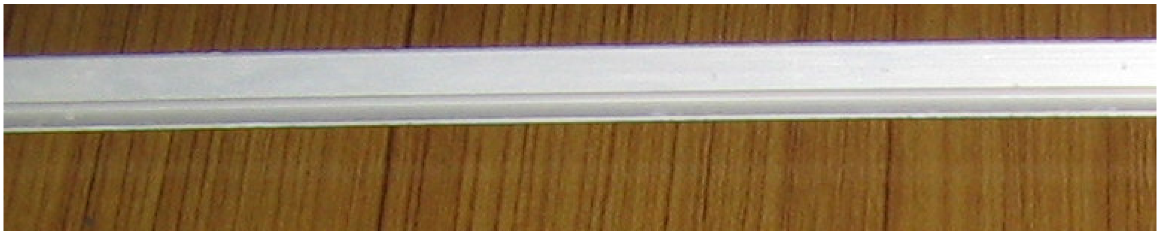


Fig 3.1 TWISTED Al TAPER CLIP (SMOOTH TAPE)(Y=0.0)

Fig 3.2 TWISTED Al TAPER CLIP(Y=8.73)

Fig 3.3 TWISTED Al TAPER CLIP(Y=5.23)

Fig 3.4 TWISTED Al TAPER CLIP(Y=3.27)


Fig 3.5 TWISTED TAPE(Y=5.05)


Fig 3.6 TWISTED TAPE(Y=3.25)

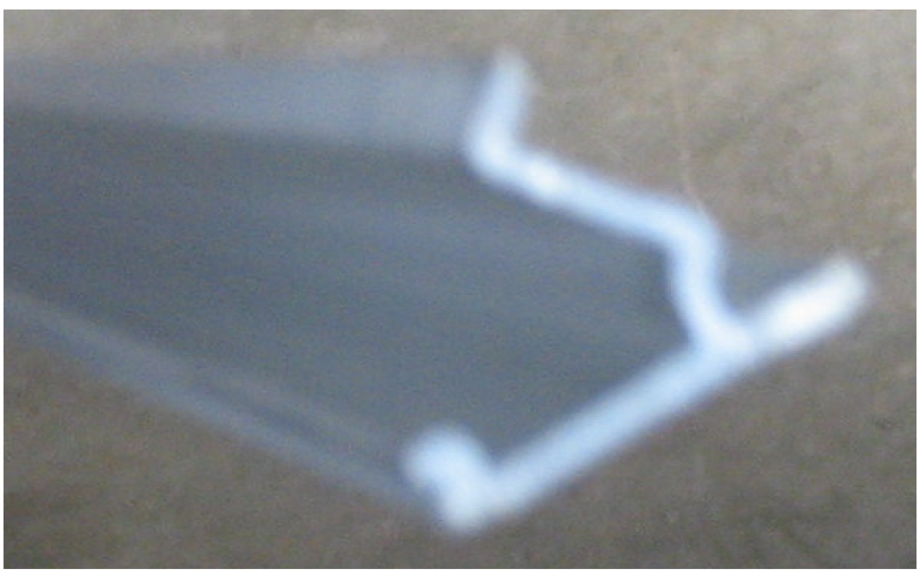


Fig 3.7 CROSS SECTIONAL VIEW OF PLAIN Al TAPER CLIP

5.4 Procedure

1. All the rotameters used were calibrated first.(Table no-A1.1, A1.2)
2. RTD used for measuring the temperatures were calibrated.(Table no-A1.3)
3 .For friction factor determination-

a. The manometer used contained ccl4 as the manometric liquid and a little of bromine crystals was added to it to impart a colour to it. The manometer was first adjusted so that the liquid level in both the limbs were equal.

b. After the manometer had been set, water(at room temperature) was allowed to flow through the inner pipe of the exchanger.

c. The manometer reading was noted for each of the flow rates.

d. the above procedures were followed for each of the inserts used.

e. Standardization of smooth tube:-

Before starting the experimental study on heat transfer augmentation using inserts, standardization of the smooth tube (without insert) has to be done so that the % difference between the theoretical frictional factor value and the actual value can be obtained.(Table no-A2.1)

Theoretical friction factor (f_0) was calculated using the formulas:-

(a) Re< 2100

$f = 16/\mathrm{Re}$

(b) Re>2100

$f = 0.046\ \mathrm{Re}^{-0.2}$

4. For heat transfer coefficient determination-

a. Standardization of the smooth tube:-

Before starting the experimental study on heat transfer augmentation using inserts, standardization of the smooth tube (without insert) has to be done so that the % difference between the theoretical heat transfer coefficient and the actual heat transfer coefficient can be obtained. (Table no-A3.1).

$(h_i)_{theoreical}$ was calculated only for smooth tube using the formulas:-

(a) For Re<2100

Nu = f(Gz)

For Gz < 100

$$Nu = 3.66 + \frac{0.085 Gz_1}{1 + 0.047 Gz_1^{\frac{2}{3}}} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

(b) 2100<Re<10000(HOUGEN's EQUATION)

$$\frac{h_i d_i}{k} = 0.116\ (\mathrm{Re}^{\frac{2}{3}} - 125)(\mathrm{Pr}^{\frac{1}{3}})[1 + (\frac{d_i}{L})^{\frac{2}{3}}][\frac{\mu_b}{\mu_w}]^{0.14}$$

(d) Re>10000(DITTUS BOILTER EQUATION)

$$\frac{h_i d_i}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.33}$$

b. Water to be flown throw the annulus side was first heated to a set temperature of 64^{o}C in a constant temperature water bath of capacity 500 lts.

c. When the set point was attained, the water at room temp was allowed to flow through the inner pipe of the exchanger.

d. Hot water flowed through the annulus side at a constant flow rate of (1250kg/hr) thus exchanging heat with the cold water on the tube side.
e. For each of the flow rates of the cold water, the temperatures T1, T2, T3, T4 were noted down only after the temperatures attained a constant value.
f. The steps (4b-4e) were followed for each of the inserts used.

CHAPTER 4

Sample calculations

6.1 SAMPLE CALCULATIONS

6.1.1ROTAMETER CALIBRATION

SMALL ROTAMETER (Table No.A1.1,)

For SL.NO-3

Observation1:	Wt= 10.5(Kg)
	Time= 219(secs)
	m1 = 0.0479(Kg/s)
Observation2:	Wt = 10.9(Kg)
	Time = 226(secs)
	m2 = 0.0482(Kg/s)
Observation3:	Wt = 10.3(Kg)
	Time = 214(secs)
	m3 = 0.0481(Kg/s)

$$m_{avg} = \frac{m_1 + m_2 + m_3}{3} = 0.0481(\text{Kg/s})$$

6.1.2 Pressure drop and Friction factor calculations:-

For Y=8.73(TATC) (Table No A2.3)
SL.NO. -8
m = 0.1081(Kg/s)
h= 0.132(m)

$$A = \frac{\pi}{4} d_i^{\,2} = 3.8\times10^{-4}$$

$$V(m/s) = \frac{m}{A\times\rho} = \frac{0.1081}{3.8\times10^{-4}\times1000} = 0.2845$$

$$\Delta P = (\rho_{ccl4} - \rho_w)\times g\times h = (1603-1000)\times9.81\times0.132 = 780.84 N/m^2$$

$$f_{actual} = \frac{\Delta P\times d_i}{2\times\rho\times L\times V^2} = \frac{780.84\times0.022}{2\times1000\times2.825\times(0.2845)^2} = 0.0371$$

$$\mu = 0.85cp$$

$$\text{Re} = \frac{4\times m}{\pi\times d_i\times\mu} = \frac{4\times0.1081}{\pi\times0.022\times8.5\times10^{-4}} = 7364$$

$$f_{theoretical} = 0.046\times\text{Re}^{-0.2} = 0.046\times(7364)^{-0.2} = 0.0078$$

$$\frac{f_a}{f_o} = \frac{0.0371}{0.0078} = 4.76$$

6.1.3 Heat Transfer Coefficient Calculation (Table no. A3.3)

m= 0.1081

	Actual Temp	Corrected Temp
T1=31.5°C	31.0	31.5
T2=38.2 °C	38.2	38.2
T3=53.6 °C	53.6	53.6
T4=51.5 °C	51.3	51.5

$$\frac{T1+T2}{2} = 34.4\ ^{o}\text{C} \qquad \frac{T3+T4}{2} = 52.6\ ^{o}\text{C}$$

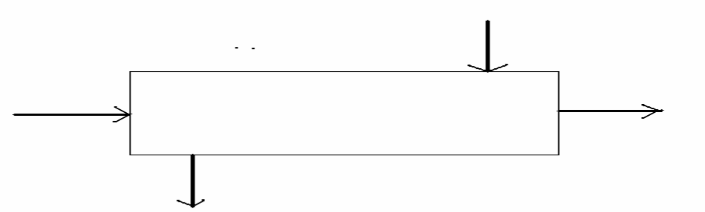


Fig4.1

$$Q_1 = m \times C_p \times (T2 - T1) = 0.1081 \times 4186 \times (38.2 - 31.5) = 3031.79 Watt$$
$$Q_2 = m \times C_p \times (T3 - T4) = 0.1081 \times 4186 \times (53.6 - 51.5) = 3052.10 Watt$$
$$Q_{avg} = \frac{Q_1 + Q_2}{2} = 3041.95 Watt$$
$$\% diff = \frac{Q_2 - Q_1}{Q} \times 100 = 0.67$$

T3-T2=15.4°C

T4-T1=20.0°C

$$L.M.T.D = \frac{(T3 - T2) - (T4 - T1)}{\ln \frac{(T3 - T2)}{(T4 - T1)}} = 17.6\ ^{o}\text{C}$$
$$A = \pi \times d_i \times L = \pi \times 0.022 \times 2.43 = 0.16795 m^2$$
$$U_i = \frac{Q}{A \times L.M.T.D} = \frac{3041.95}{0.16795 \times 17.6} = 1029\ W / m^2\ ^{o}\text{C}$$
$$\text{Re}_c = \frac{4 \times m}{\pi \times d_i \times \mu} = 7364$$

$$\Pr_c = 0.0111T^2 - 0.852T + 21.401$$

(Table A4.7)

$$where T = \frac{T1 + T2}{2} = 34.9$$

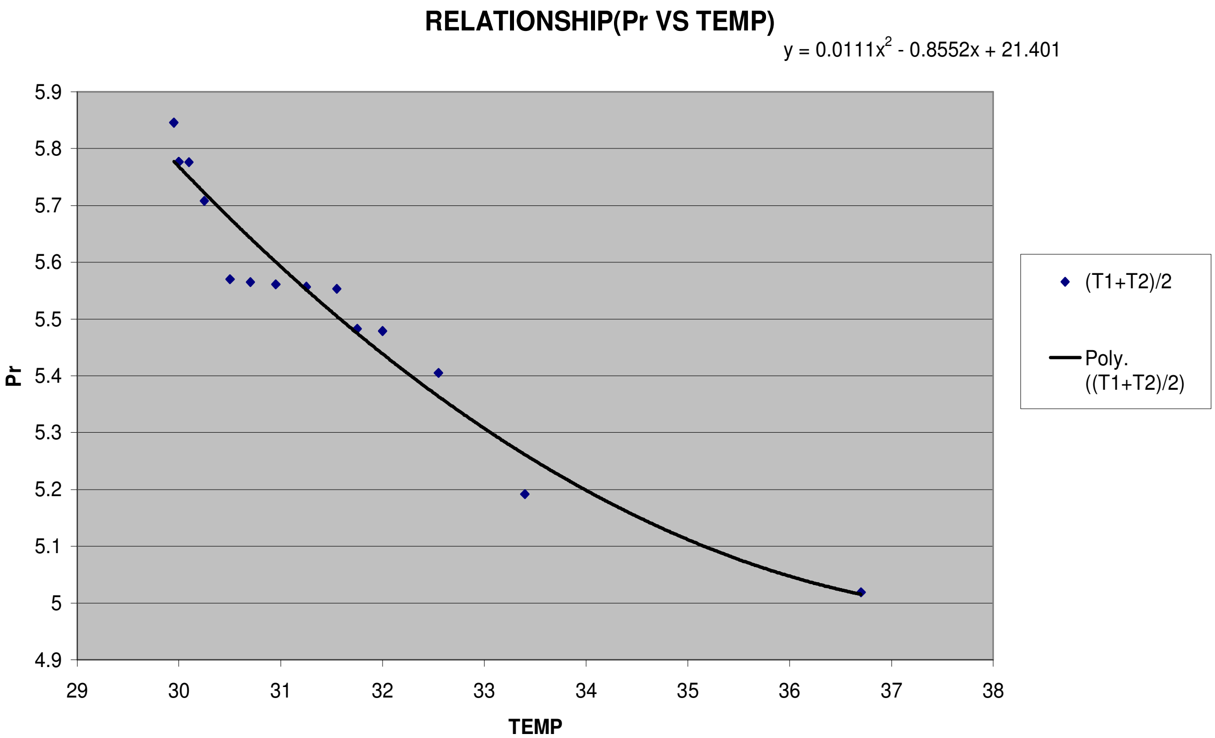


Fig 4.2

$$\Pr_c = 5.078$$

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{d_i}{d_o \times h_o} + \frac{x_w \times d_i}{k_w \times d_L} + R_m$$

Which reduces to

$$\frac{1}{U_i} = \frac{1}{(h_i)_{\exp t}} + K = \frac{1}{c(\mathrm{Re})^{0.8}}$$

K is to be found from the Wilson Chart as the intercept on the y-axis.

For Re>10,000

$$hi = 0.023\frac{k}{d_i}(\Pr)^{\frac{1}{3}}(\mathrm{Re})^{0.8} = c(\mathrm{Re})^{0.8}$$

$$\frac{1}{U_i} = \frac{1}{c(\mathrm{Re})^{0.8}} + K$$

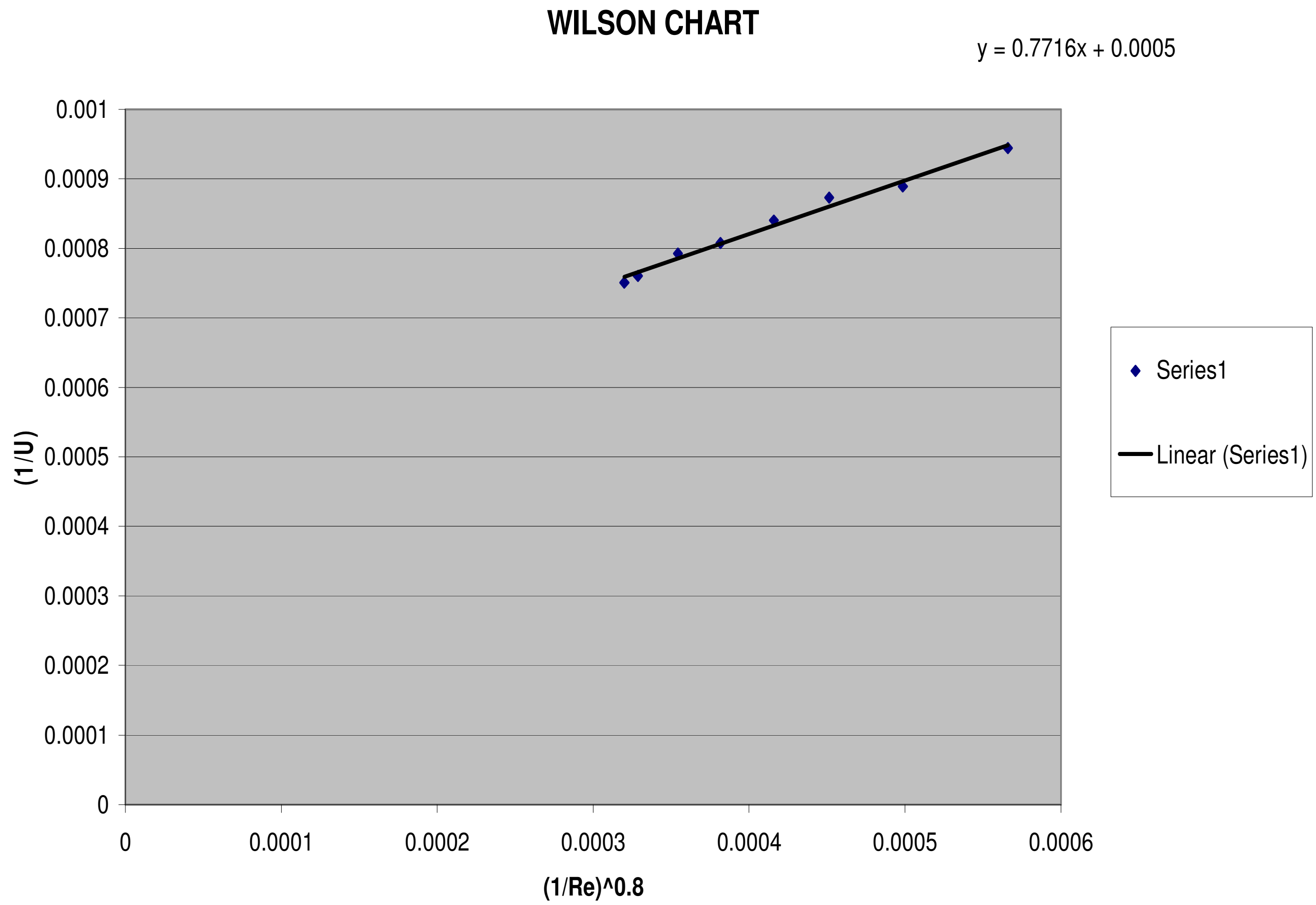


Fig 4.3

K=0.0005 m^2oC$/W$

$$\frac{1}{(h_i)_{\exp t}} = \frac{1}{U_i} - K = \frac{1}{1029} - 0.0005 = 4.718 \times 10^{-4}$$

$$(h_i)_{\exp t} = 2120 W/m^2\ {}^{o}\mathrm{C}$$

Performance evaluation criterion R_1 is found to be

$$R_1 = \frac{(h_i)_{\exp t}}{(h_i)_{theoretical\ forsmoothtube}} = \frac{2120}{1474} = 1.44$$

CHAPTER 5

Results and discussion

5. Results and Discussion

5.1 friction factor results

In Fig no.5.1.1 it was observed that for Re> 8000, $f = f_o$. For very low Re(Re = 1000) the observed height difference in the manometer was very small, resulting in the deviation. For Re in the range of 2000 to 6000, the flow of water is in the transition zone. The friction factor relation used was $f = 0.046\,Re^{-0.2}$.

The f vs Re graph is as shown below in fig no 5.1.2. As is clearly seen the actual friction factor(f) is lower than the theoretical friction factor (f_0) , thus the deviation.

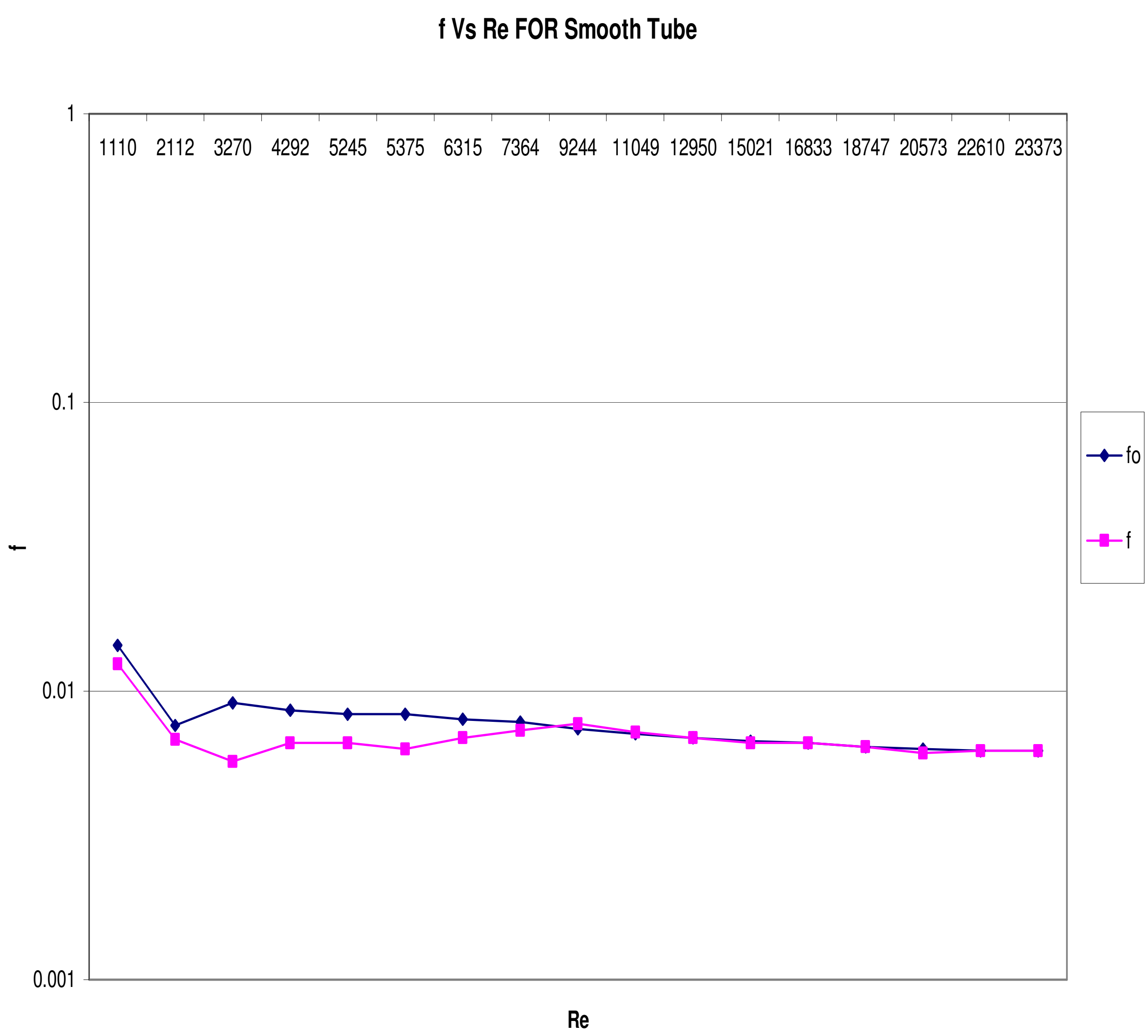


Fig 5.1.1

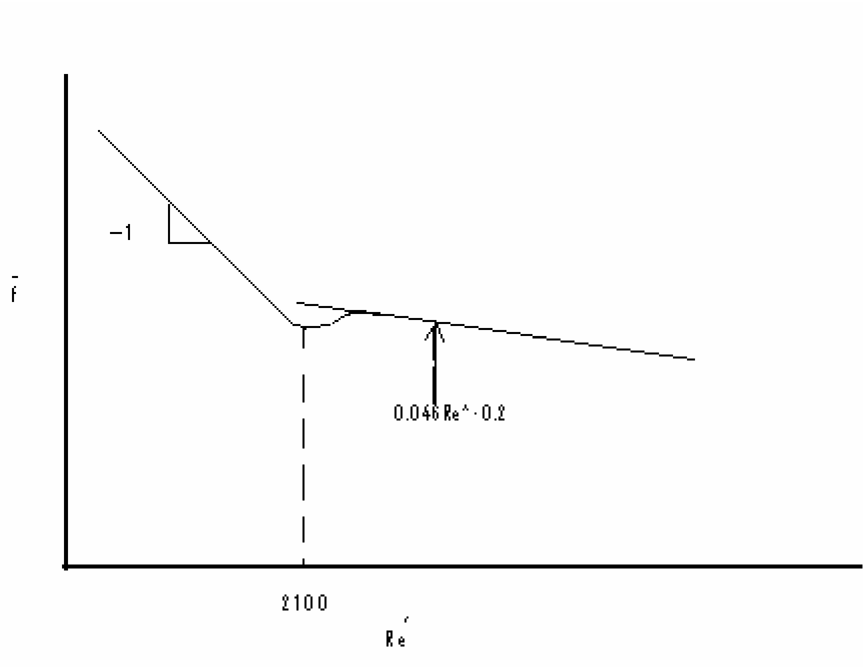


Fig 5.1.2

Re	**f**
2100	0.00762
2500	0.00962
3000	0.00928
4000	0.00876

The value of f calculated using the above equation is higher than the theoretical value.

In Fig no.5.1.3 friction factor increases as the degree of twist in the Twisted Al Taper Clip(TATC) increases which is indicated by a lower value of twist ratio. Hence the pressure drop increases accordingly i.e.,
f(smooth) < f(Y=0.0 or straight tape) < f(Y=8.73) < f(Y=5.23) < f(Y=3.27).

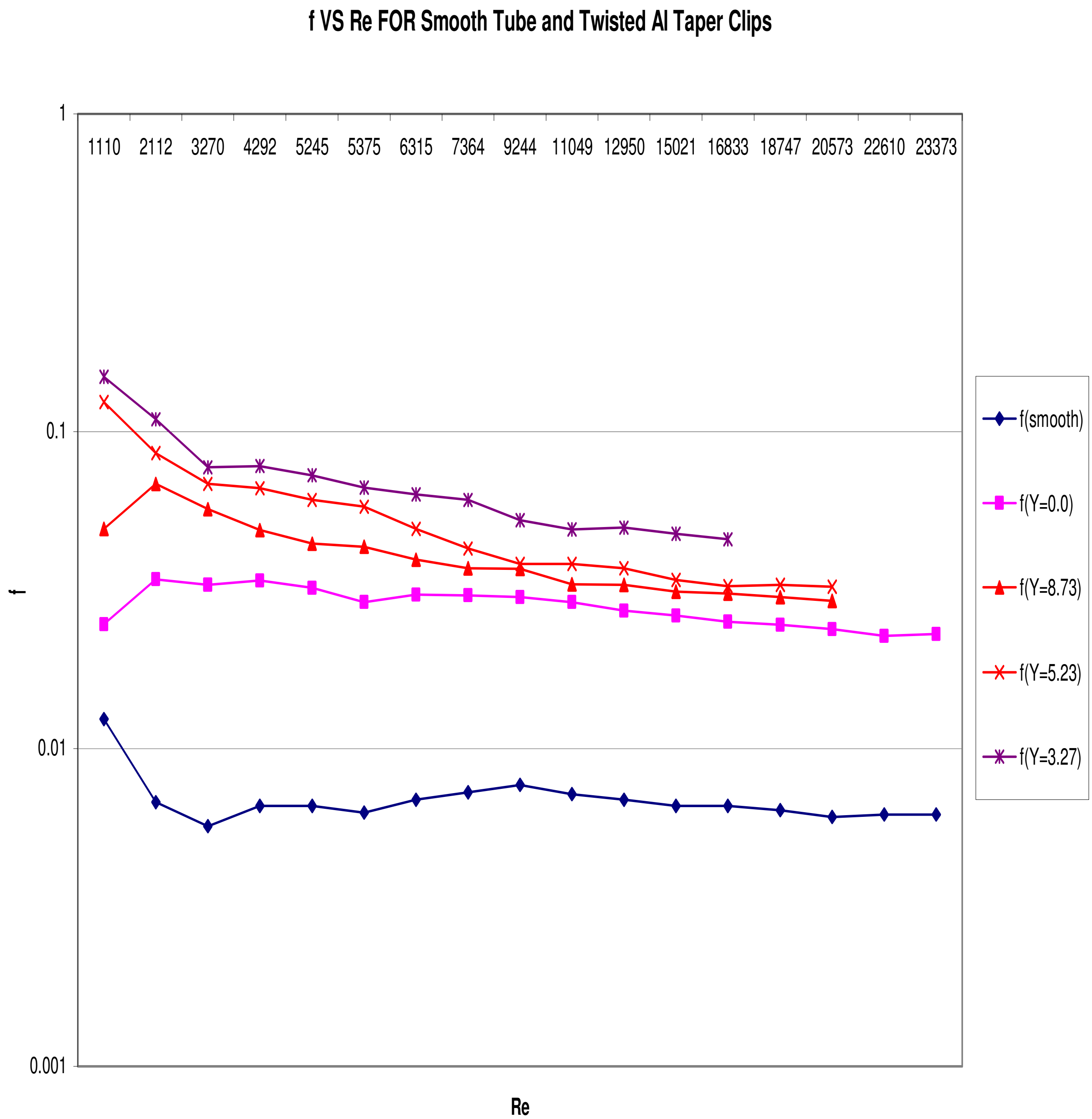


Fig 5.1.3

The same trend as mentioned above is observed in the case of Twisted Tapes fig 5.1.4 i.e., f(smooth) < f(Y=5.05) < f(Y=3.25)

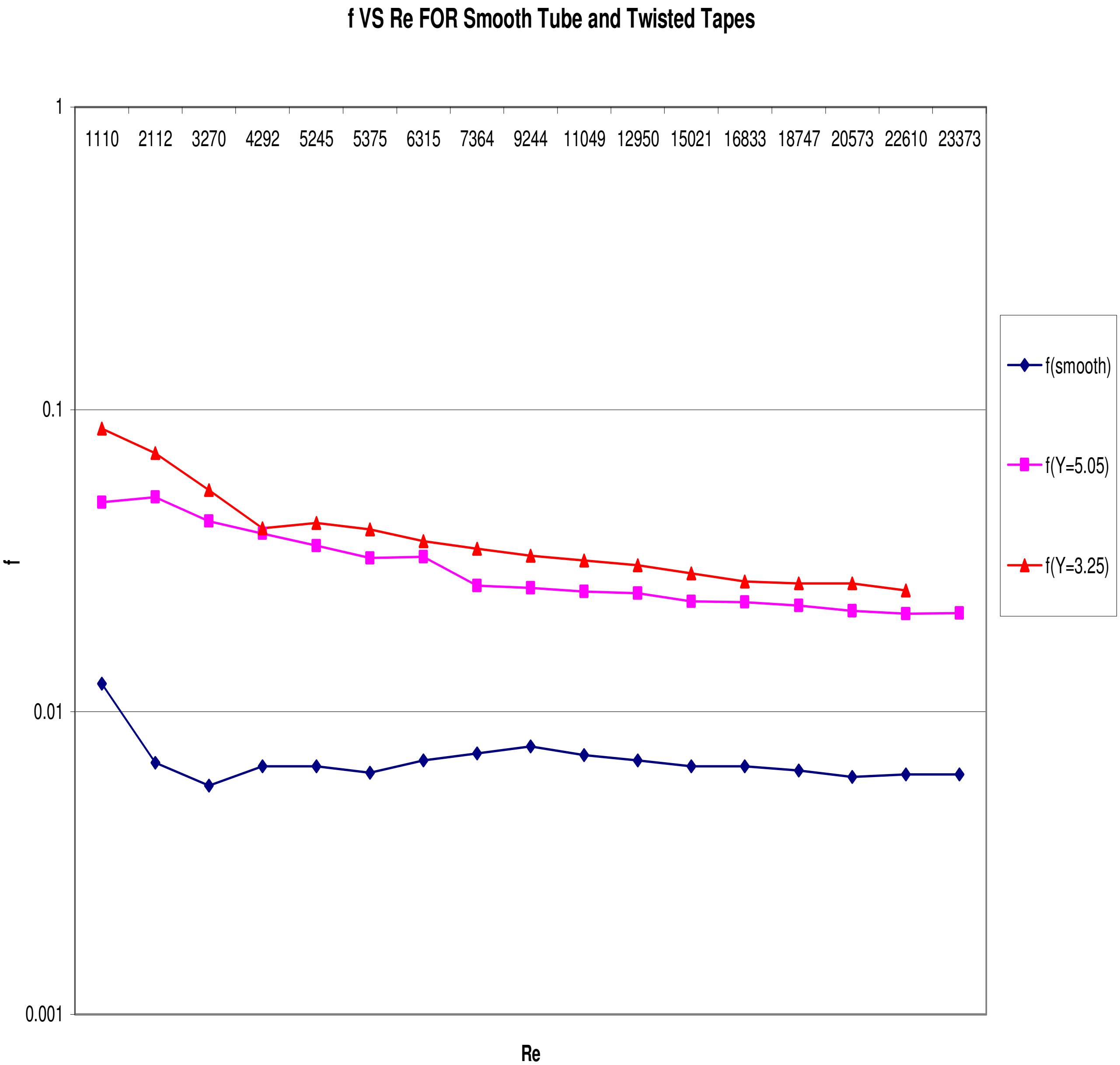


Fig 5.1.4

Fig no 5.1.5 shows the plot of f vs Re for all the inserts used in the experiment satisfying the above discussed trend. The TATC's used have twist ratios as Y = 0.0, Y = 8.73, Y=5.23,Y=3.27. The twisted tapes have twist ratios as Y=5.05, Y=3.25. It is noticed that for TATC and Twisted Tape having almost the same twist ratio, the TATC show a greater increase in the friction factor for the same Re.

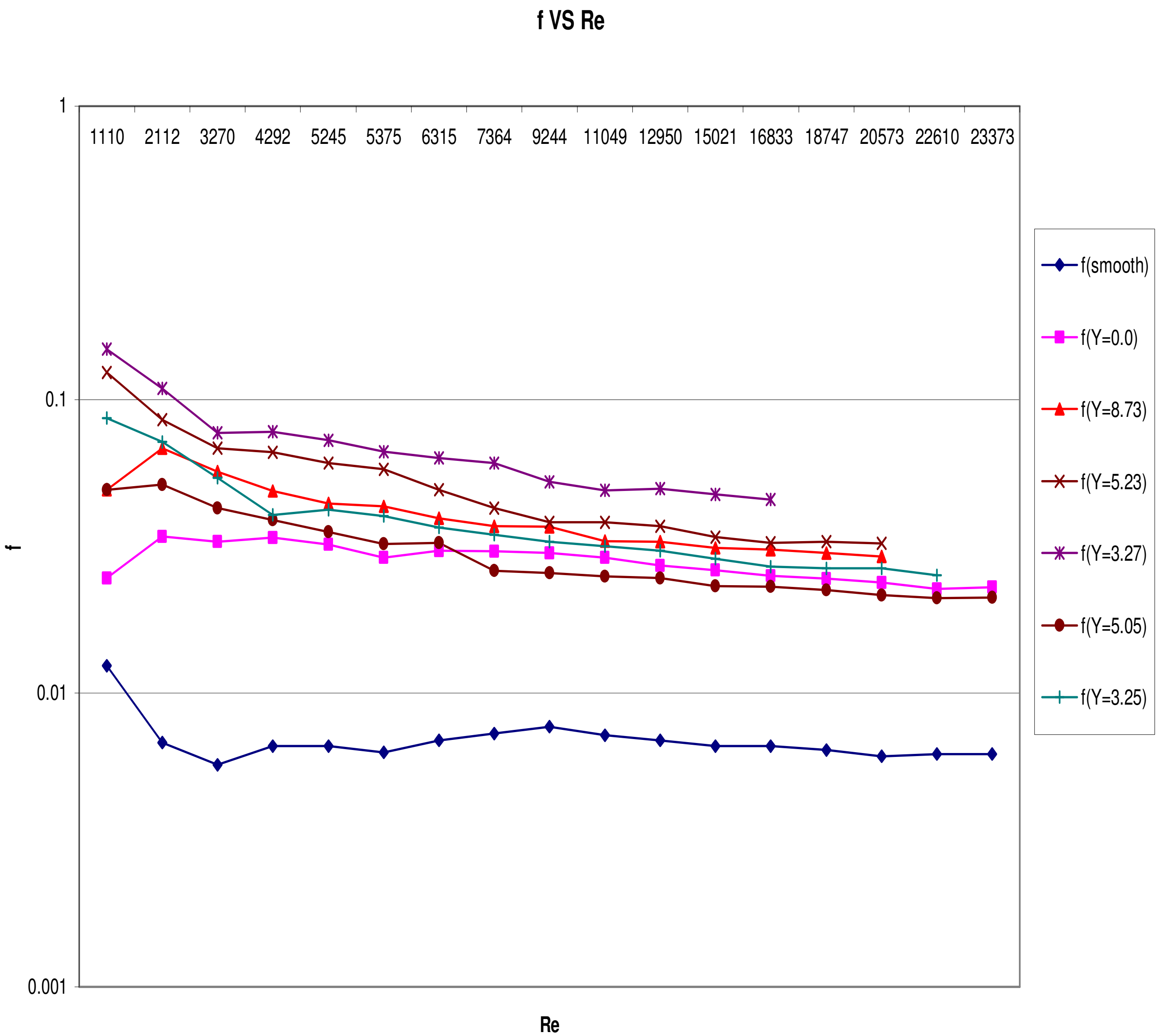


Fig 5.1.5

The graph between f_a/f_o vs Re (Fig no.5.1.6) shows an interesting trend for Re = 2112 where there is an abrupt increase in the value of f_a/f_o. For all other values of Re, f_a/f_o goes on increasing as the degree of twist increases.

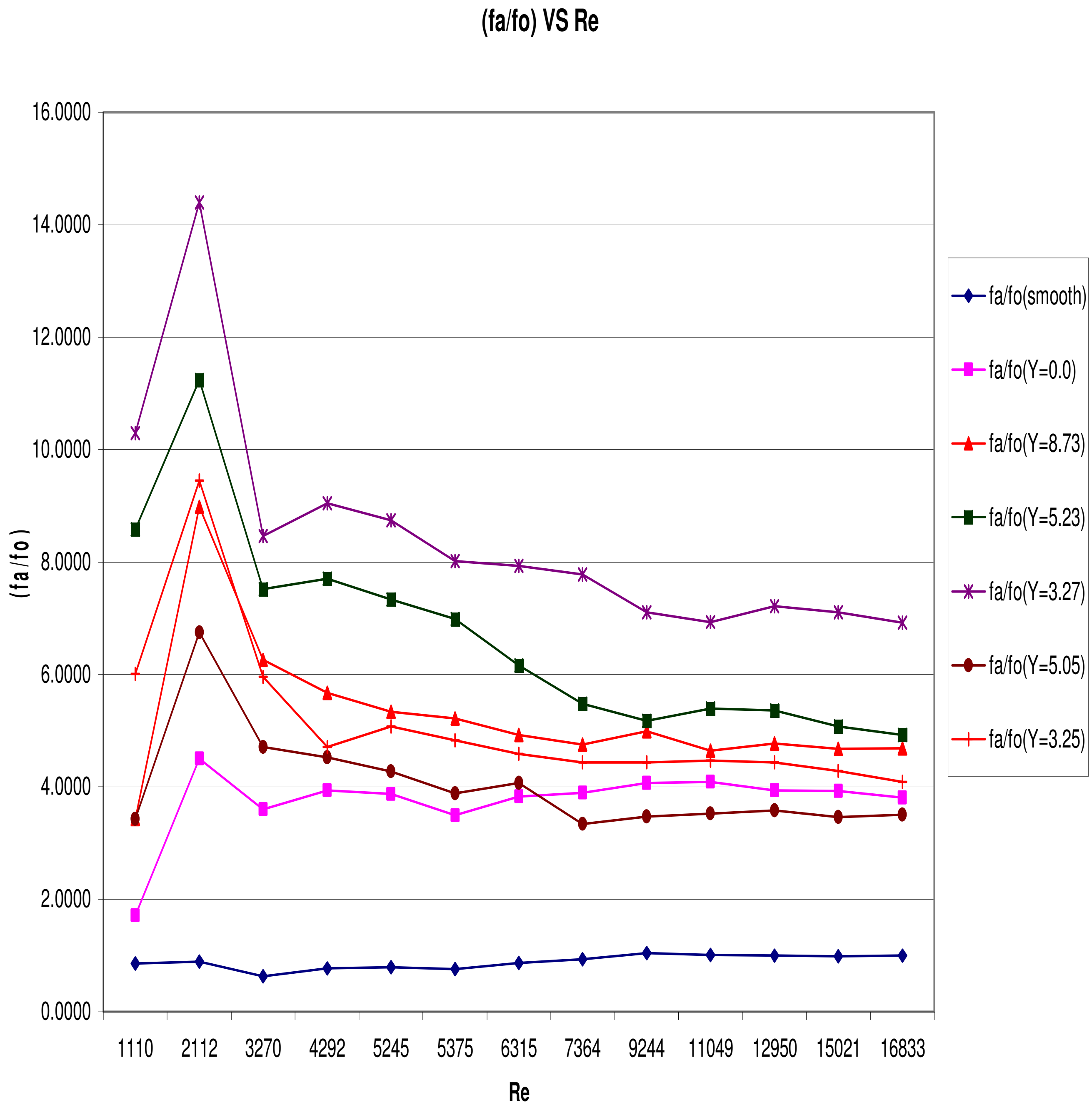


Fig 5.1.6

5.2Heat transfer coefficient results

As seen in the Fig no.5.2.1 the deviation between hi(expt) and hi(thero) for low Re upto about 6000, is because of the phenomenon of natural convection along with forced convection.

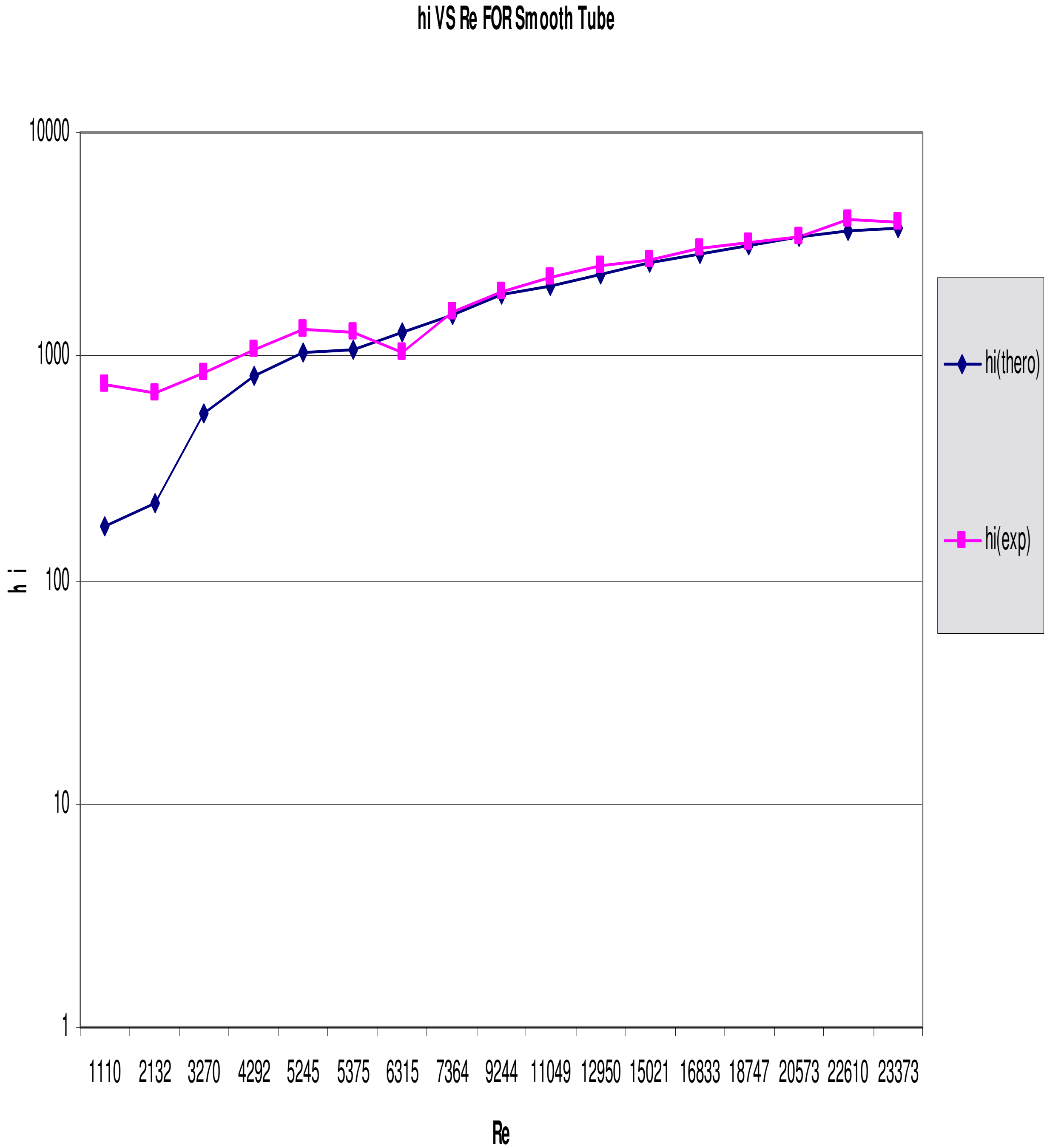


Fig 5.2.1

In Fig no. 5.2.2 and Fig no 5.2.3 the heat transfer coefficient increases as the twist in the inserts increases. The collective result for all the inserts used are shown in Fig 5.2.4. As in the case of friction factor, for the TATC and Twisted Tape having almost the same twist ratio the TATC show a greater increase in the heat transfer coefficient.

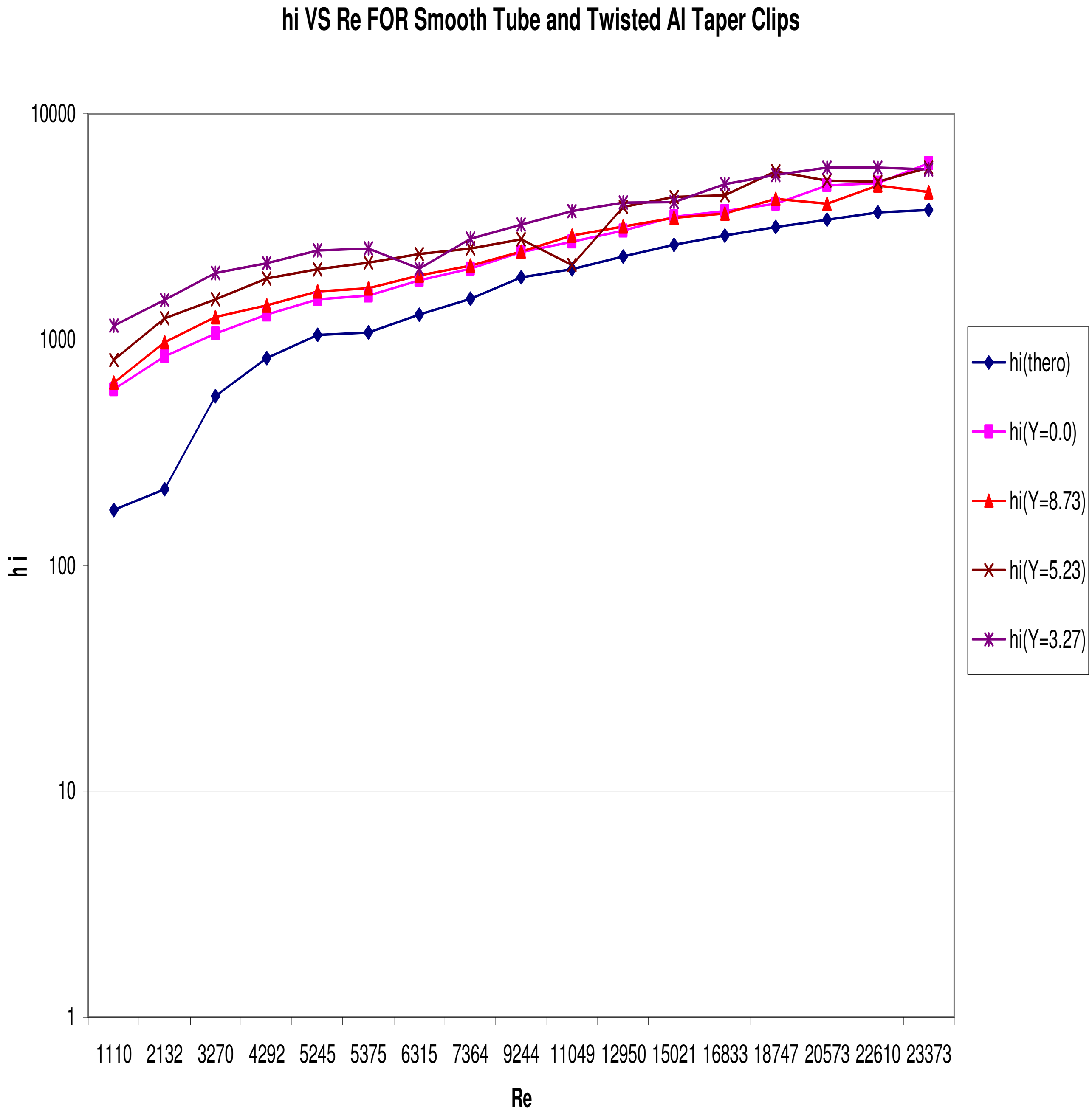


Fig 5.2.2

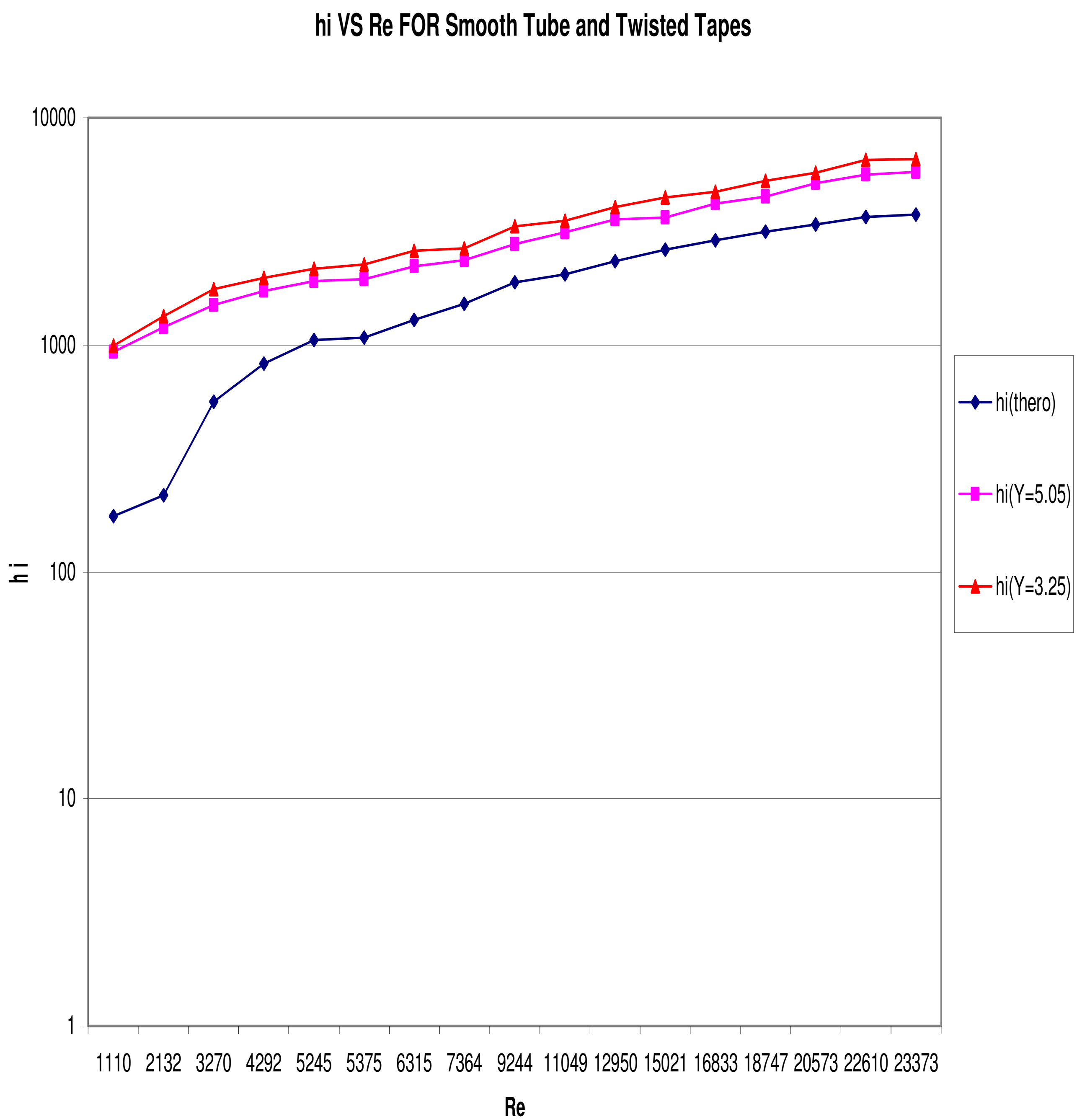

hi VS Re FOR Smooth Tube and Twisted Tapes
10000
1000
100
10
1
h i
1110 2132 3270 4292 5245 5375 6315 7364 9244 11049 12950 15021 16833 18747 20573 22610 23373
Re
hi(thero)
hi(Y=5.05)
hi(Y=3.25)

Fig 5.2.3

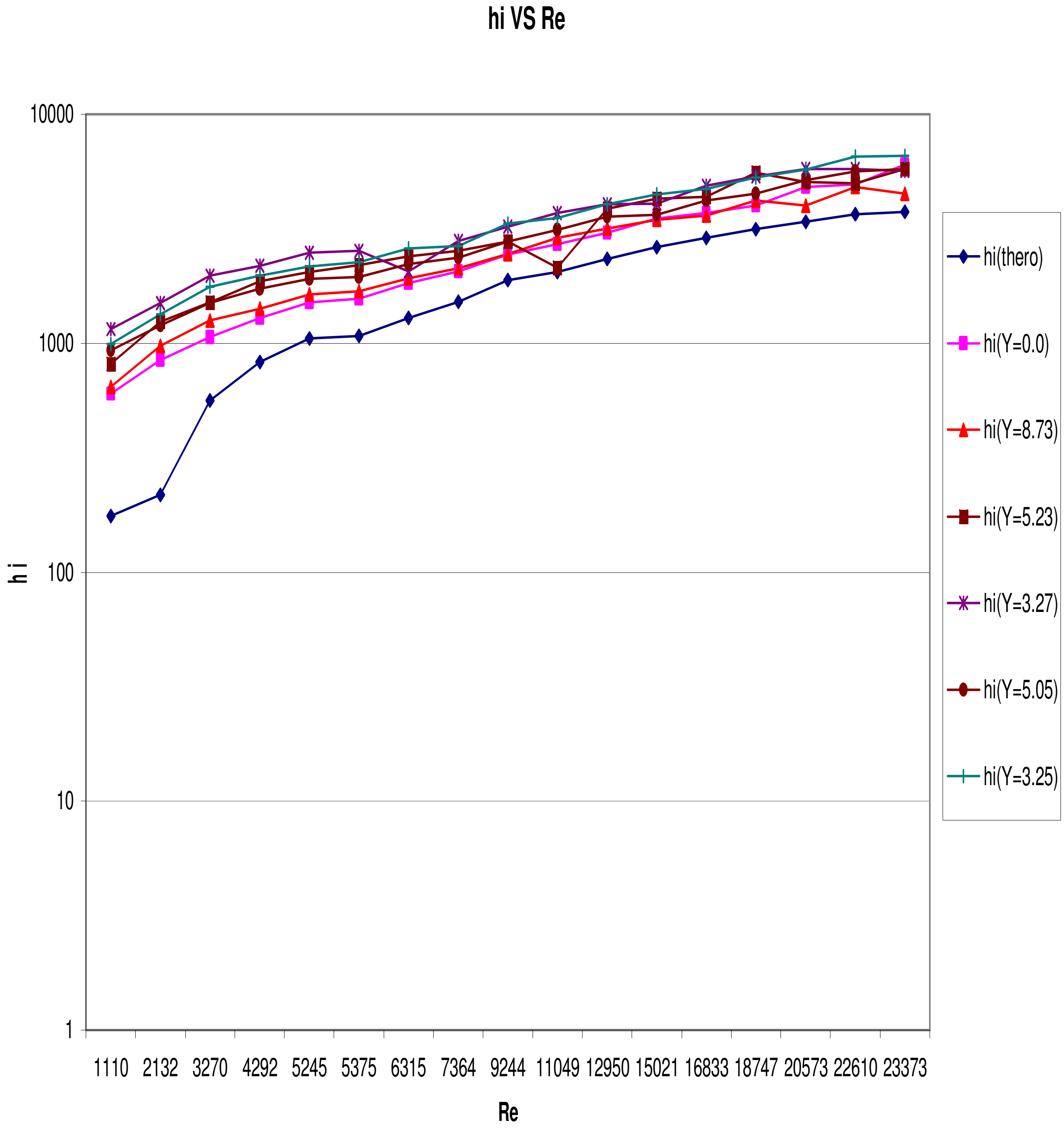

hi VS Re
10000
1000
100
10
1
h i
1110
2132
3270
4292
5245
5375
6315
7364
9244
11049
12950
15021
16833
18747
20573
22610
23373
Re
hi(thero)
hi(Y=0.0)
hi(Y=8.73)
hi(Y=5.23)
hi(Y=3.27)
hi(Y=5.05)
hi(Y=3.25)

Fig 5.2.4

The R1 vs Re plot Fig 7.1.2.5 is shown below. The fig clearly shows the fractional increase of htc for each of the inserts used.

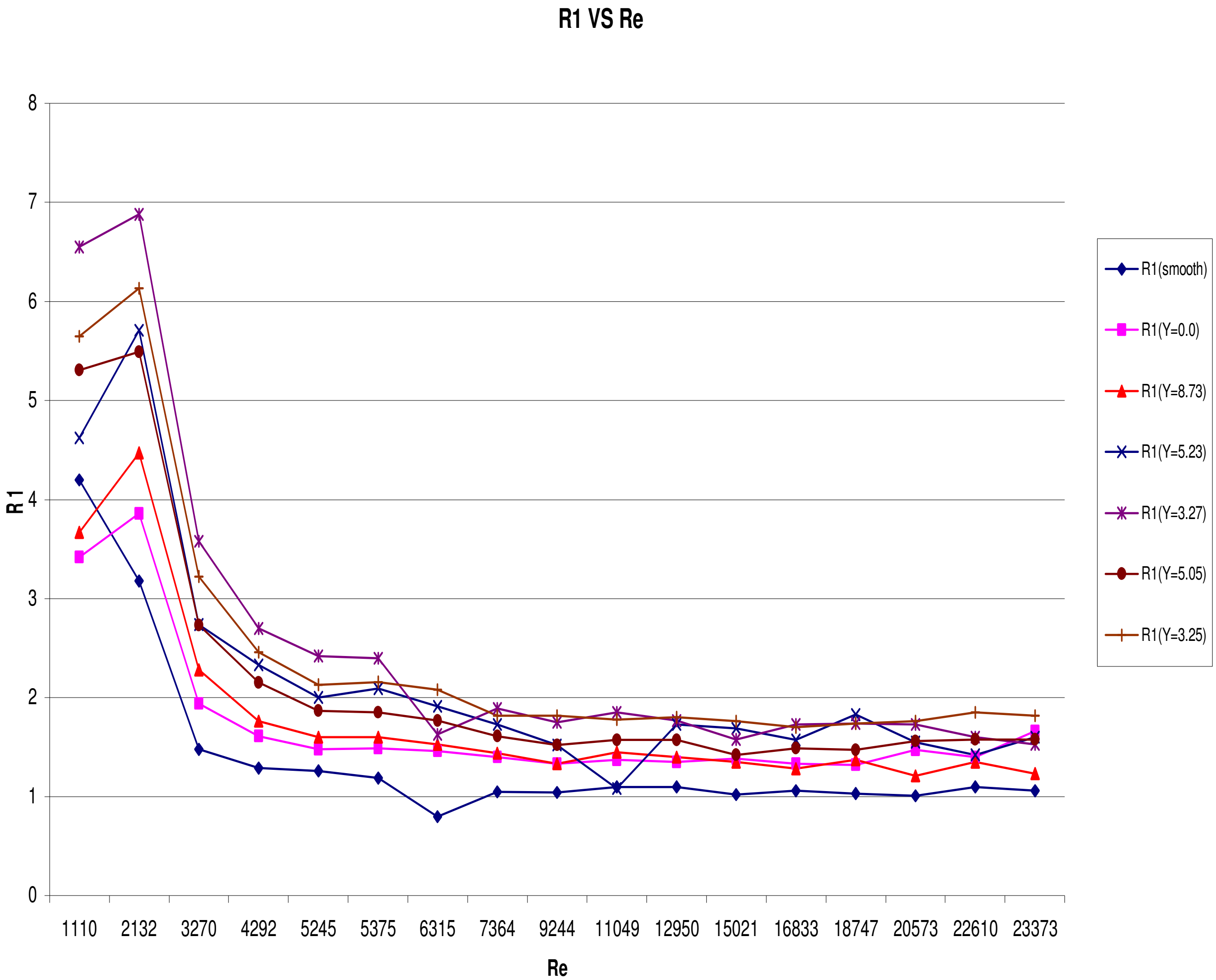


Fig 5.2.5

CHAPTER 6

Conclusion

Conclusion:-

1. The difference in the heat transfer coefficient for the actual and the theoretical values for low Reynolds number (upto 6000) in the smooth tube can be attributed to the natural convection which occurs along with the forced convection. This phenomena is prominent in the case of low Re. In case of higher Re, natural convection is negligible as compared to forced convection.
2. The pressure drop and the heat transfer coefficient increase as the degree of twist in the tapes goes on increasing.
3. For almost the same twist ratio, twisted aluminium taper clips show greater friction factor and heat transfer coefficient than the twisted tapes, because of higher degree of turbulence generated.

The range of f_a/f_o and R1 values for the TATC's and twisted tapes are shown in the table 6.1 and table 6.2 respectively.

Table 6.1

Sl.No	Y	Range of f_a/f_o	Range of R1
1	0.0(straight tape)	1.71 to 4.5	1.32 to 3.86
2	8.73	3.43 to 8.98	1.21 to 4.47
3	5.23	4.92 to 11.23	1.08 to 5.71
4	3.27	6.92 to 14.39	1.53 to 6.88

Table 6.2

Sl.No	Y	Range of f_a/f_o	Range of R1
1	5.05	3.34 to 6.75	1.42 to 5.49
2	3.25	4.09 to 9.44	1.70 to 6.13

In a heat exchanger, while the inserts can be used to enhance the heat transfer rate, they also bring in an increase in the pressure drop. When the pressure drop increases, the pumping power cost also increases, thereby increasing the operating cost. Since we have to keep the operating cost to a minimum, there has to be a proper balance between the heat transfer rate and the pressure drop. While there is a need for the heat transfer rate to be increased, the pressure drop can't be allowed to go beyond a certain specified limit.

So depending on the requirement, one of the above mentioned inserts can be used for heat transfer augmentation. As per the performance evaluation criteria R1,the taper aluminium twisted tape gives better performance as compared to the twisted tape having the same twist ratio.

NOMENCLATURE:--

A_c	inside surface area, m^2
A_i	heat transfer surface area, m^2
C_p	specific heat of fluid, J/kg.k
d_i	inside diameter of the tube, m
d_o	outside diameter of the tube, m
f	Fanning friction factor, dimensionless
f_a	augmented friction factor,dimensionless
f_o	theoretical friction factor, dimensionless
Gz	Gratz number,dimensionless
h	difference in level of mercury in the manometer, cm
hi	inside htc, W/m^2 °C
hi(expt)	experimental inside htc, W/m^2 °C
hi(theo)	theoretical inside htc, W/m^2 °C
H	linear distance of the tape for 180^0 rotation, m.
k_w	thermal conductivity of the tube wall, w/ m ^{0}c
L_h	heat transfer length, m
L_p	pressure taping to pressure taping length, m
L.M.T.D	log mean temperature difference, k
m	mass flow rate, kg/s
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
Δp	pressure drop, N/m^2
Q	heat transfer rate, W
Re	Reynolds number
R1	performance evaluation criteria, dimensionless
Sw	Swirl parameter
T	temperature in ^{0}c
Ui	overall htc based on the inside, W/m^2 °C
V	velocity of water, m/s
w	width of twisted tape insert, m
Wt	weight of water taken, Kg
y	twist ratio, dimensionless defined by H/d_i.

Greek letters

δ	width of the twisted tape, m
ρ	fluid density in kg/m^3
μ	dynamic viscosity N/m^2s

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(1) Proc. Institution of Mechanical Engineers Vol. 218 Part A: Journal of Power and Energy.

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(3) Bergles, A.E. Techniques to augment heat transfer. In Handbook of Heat Transfer Applications(Ed.W.M. Rosenhow), 1985, Ch.3(McGraw-Hill, NewYork).

(4) Saha, S. K. and Dutta, A. Thermo-hydraulic study of laminar swirl flow through a circular tube fitted with twisted tapes. Trans. ASME, J. Heat Transfer, 2001, 123, 417–421.

(5) Manglik, R. M. and Bergles, A. E. Heat transfer and pressure drop correlations for twisted tape insert in isothermal tubes. Part 1: laminar flows. Trans. ASME, J. Heat Transfer, 1993, 116, 881–889.

(6) Saha, S. K. and Bhunia, K. Heat transfer and pressure drop characteristics of varying pitch twisted-tape-generated laminar smooth swirl flow. In Proceedings of 4th ISHMT–ASME Heat and Mass Transfer Conference, India, 2000, pp. 423–428 (Tata McGraw-Hill, New Delhi).

Appendix

1. CALIBRATION

1.1 SMALL ROTAMETER

	Observation1			Observation2			Observation3			
Rotameter Reading (lph)	Wt	Time	m	Wt	Time	m	Wt	Time	m	Average m
1	10.5	645	0.0163	10.2	625	0.0163	10.25	631	0.0162	0.0163
2	10.6	337	0.0315	10.5	336	0.0315	11.2	359	0.0312	0.0313
3	10.5	219	0.0479	10.9	226	0.0482	10.3	214	0.0481	0.0481
4	11.7	187	0.0626	13	207	0.0628	11.4	179	0.0637	0.0630
5	11.2	144	0.0778	10.8	139	0.0777	11.35	146	0.0777	0.0777

1.2 LARGE ROTAMETER

	Observation1			Observation2			Observation3			
Rotameter Reading (lph)	Wt	Time	m	Wt	Time	m	Wt	Time	m	Average m
300	12.7	160	0.0794	11.1	139	0.0799	11.95	154	0.0776	0.089
350	12.5	135	0.0926	12.3	133	0.0925	11.15	120	0.0929	0.0927
400	11.3	105	0.1076	12	111	0.1081	11.3	104	0.1087	0.1081
500	12.55	93	0.1349	13.8	100	0.138	12.2	91	0.1341	0.1357
600	13.4	83	0.1614	11.9	73	0.1630	12.5	77	0.1623	0.1622
700	12.35	65	0.19	12.65	66	0.1917	11.5	61	0.1885	0.1901
800	13.4	61	0.2197	11.75	53	0.2217	11.45	52	0.2202	0.2205
900	12.85	52	0.2471	12.15	49	0.2480	12.8	52	0.2462	0.2471
1000	13.7	50	0.274	11.8	43	0.2744	12.2	44	0.2773	0.2752
1100	12.35	41	0.3012	12.5	41	0.3049	12	40	0.3	0.3020
1200	12.6	38	0.3316	13.3	40	0.3325	12.6	38	0.3316	0.3319
1250	12.4	36	0.3444	13	38	0.3421	12	35	0.3429	0.3431

1.3 RTD CALIBRATION

TEMPERATURE	ACTUAL TEMP	CORRECTION	CORRECTED TEMP
T1	15.3	+0.5	15.8
T2	15.8	0.0	15.8
T3	15.8	0.0	15.8
T4	15.6	+0.2	15.8

2.1 STANDARDISATION – SMOOTH TUBE (f vs Re)

SL.NO.	m	h	▲P	f	fo	%diff	Re
1	0.0163	0.1	5.92	0.0124	0.0144	-16.57	1110
2	0.031	0.2	11.83	0.0068	0.0076	-10.85	2112
3	0.048	0.4	23.66	0.0057	0.0091	-59.90	3270
4	0.063	0.8	47.32	0.0066	0.0086	-30.44	4292
5	0.077	1.2	70.99	0.0066	0.0083	-24.79	5245
6	0.0789	1.2	70.99	0.0063	0.0083	-30.39	5375
7	0.0927	1.8	106.48	0.0069	0.0080	-16.18	6315
8	0.1081	2.6	153.80	0.0073	0.0078	-6.07	7364
9	0.1357	4.3	254.36	0.0077	0.0074	3.43	9244
10	0.1622	5.8	343.09	0.0072	0.0071	1.30	11049
11	0.1901	7.6	449.57	0.0069	0.0069	-0.24	12950
12	0.2205	9.8	579.71	0.0066	0.0067	-1.53	15021
13	0.2471	12.2	721.68	0.0066	0.0066	-0.11	16833
14	0.2752	14.7	869.57	0.0064	0.0064	-0.86	18747
15	0.302	17	1005.62	0.0061	0.0063	-3.10	20573
16	0.3319	20.8	1230.41	0.0062	0.0062	0.13	22610
17	0.3431	22.3	1319.14	0.0062	0.0062	1.12	23373

2.2 (f vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=0.0

SL.NO.	m	h	▲P	fa	fo	fa/fo	Re
1	0.0163	0.2	11.83	0.0247	0.0144	1.7153	1110
2	0.031	1	59.15	0.0342	0.0076	4.5000	2112
3	0.048	2.3	136.05	0.0328	0.0091	3.6044	3270
4	0.063	4.1	242.53	0.0339	0.0086	3.9419	4292
5	0.077	5.8	343.09	0.0321	0.0083	3.8675	5245
6	0.0789	5.5	325.35	0.0290	0.0083	3.4940	5375
7	0.0927	8	473.23	0.0306	0.0080	3.8250	6315
8	0.1081	10.8	638.87	0.0304	0.0078	3.8974	7364
9	0.1357	16.9	999.71	0.0301	0.0074	4.0676	9244
10	0.1622	23.2	1372.38	0.0290	0.0071	4.0845	11049
11	0.1901	29.9	1768.71	0.0272	0.0069	3.9420	12950
12	0.2205	39	2307.02	0.0263	0.0067	3.9254	15021
13	0.2471	46.6	2756.59	0.0251	0.0066	3.8030	16833
14	0.2752	56.8	3359.964	0.0246	0.0064	3.8438	18747
15	0.302	66	3904.184	0.0238	0.0063	3.7778	20573
16	0.3319	76.2	4507.558	0.0227	0.0062	3.6613	22610
17	0.3431	82.3	4868.399	0.0230	0.0062	3.7097	23373

2.3 (f vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=8.73

SL.NO.	m	h	▲P	fa	fo	fa/fo	Re
1	0.0163	0.4	23.66	0.0494	0.0144	3.4306	1110
2	0.031	2	118.31	0.0683	0.0076	8.9868	2112
3	0.048	4	236.62	0.0570	0.0091	6.2637	3270
4	0.063	5.9	349.01	0.0488	0.0086	5.6744	4292
5	0.077	8	473.23	0.0443	0.0083	5.3373	5245
6	0.0789	8.2	485.07	0.0433	0.0083	5.2169	5375
7	0.0927	10.3	609.29	0.0394	0.0080	4.9250	6315
8	0.1081	13.2	780.84	0.0371	0.0078	4.7564	7364
9	0.1357	20.7	1224.49	0.0369	0.0074	4.9865	9244
10	0.1622	26.4	1561.67	0.0330	0.0071	4.6479	11049
11	0.1901	36.2	2141.39	0.0329	0.0069	4.7681	12950
12	0.2205	46.4	2744.76	0.0313	0.0067	4.6716	15021
13	0.2471	57.4	3395.46	0.0309	0.0066	4.6818	16833
14	0.2752	69.2	4093.48	0.0300	0.0064	4.6875	18747
15	0.302	81.1	4797.41	0.0292	0.0063	4.6349	20573
16	0.3319				0.0062		22610
17	0.3431				0.0062		23373

2.4 (f vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=5.23

SL.NO.	m	h	▲P	fa	fo	fa/fo	Re
1	0.0163	1	59.15	0.1236	0.0144	8.5833	1110
2	0.031	2.5	147.89	0.0854	0.0076	11.2368	2112
3	0.048	4.8	283.94	0.0684	0.0091	7.5165	3270
4	0.063	8	473.23	0.0662	0.0086	7.6977	4292
5	0.077	11	650.70	0.0609	0.0083	7.3373	5245
6	0.0789	11	650.70	0.0580	0.0083	6.9880	5375
7	0.0927	12.9	763.09	0.0493	0.0080	6.1625	6315
8	0.1081	15.2	899.15	0.0427	0.0078	5.4744	7364
9	0.1357	21.5	1271.82	0.0383	0.0074	5.1757	9244
10	0.1622	30.7	1816.04	0.0383	0.0071	5.3944	11049
11	0.1901	40.7	2407.58	0.0370	0.0069	5.3623	12950
12	0.2205	50.3	2975.46	0.0340	0.0067	5.0746	15021
13	0.2471	60.4	3572.92	0.0325	0.0066	4.9242	16833
14	0.2752	75.6	4472.065	0.0328	0.0064	5.1250	18747
15	0.302	90	5323.887	0.0324	0.0063	5.1429	20573
16	0.3319				0.0062		22610
17	0.3431				0.0062		23373

2.5 (f vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=3.27

SL.NO.	m	h	▲P	fa	fo	fa/fo	Re
1	0.0163	1.2	70.99	0.1483	0.0144	10.2986	1110
2	0.031	3.2	189.29	0.1094	0.0076	14.3947	2112
3	0.048	5.4	319.43	0.0770	0.0091	8.4615	3270
4	0.063	9.4	556.05	0.0778	0.0086	9.0465	4292
5	0.077	13.1	774.92	0.0726	0.0083	8.7470	5245
6	0.0789	12.6	745.34	0.0665	0.0083	8.0120	5375
7	0.0927	16.6	981.96	0.0634	0.0080	7.9250	6315
8	0.1081	21.6	1277.73	0.0607	0.0078	7.7821	7364
9	0.1357	29.5	1745.05	0.0526	0.0074	7.1081	9244
10	0.1622	39.4	2330.68	0.0492	0.0071	6.9296	11049
11	0.1901	54.8	3241.66	0.0498	0.0069	7.2174	12950
12	0.2205	70.5	4170.38	0.0476	0.0067	7.1045	15021
13	0.2471	85	5028.12	0.0457	0.0066	6.9242	16833
14	0.2752				0.0064		18747
15	0.302				0.0063		20573
16	0.3319				0.0062		22610
17	0.3431				0.0062		23373

2.6 (f vs Re) FOR TWISTED TAPE HAVING y=5.05

SL.NO.	m	h	▲P	fa	fo	fa/fo	Re
1	0.0163	0.4	23.66	0.0494	0.0144	3.4306	1110
2	0.031	1.5	88.73	0.0513	0.0076	6.7500	2112
3	0.048	3	177.46	0.0428	0.0091	4.7033	3270
4	0.063	4.7	278.03	0.0389	0.0086	4.5233	4292
5	0.077	6.4	378.59	0.0355	0.0083	4.2771	5245
6	0.0789	6.1	360.84	0.0322	0.0083	3.8795	5375
7	0.0927	8.5	502.81	0.0325	0.0080	4.0625	6315
8	0.1081	9.3	550.13	0.0261	0.0078	3.3462	7364
9	0.1357	14.4	851.82	0.0257	0.0074	3.4730	9244
10	0.1622	20	1183.09	0.0250	0.0071	3.5211	11049
11	0.1901	27.2	1609.00	0.0247	0.0069	3.5797	12950
12	0.2205	34.3	2028.99	0.0232	0.0067	3.4627	15021
13	0.2471	42.9	2537.72	0.0231	0.0066	3.5000	16833
14	0.2752	51.9	3070.11	0.0225	0.0064		18747
15	0.302	60	3549.26	0.0216	0.0063		20573
16	0.3319	70.7	4182.21	0.0211	0.0062		22610
17	0.3431	76	4495.73	0.0212	0.0062		23373

2.7 (f vs Re) FOR TWISTED TAPE HAVING y=3.25

SL.NO.	m	h	▲P	fa	fo	fa/fo	Re
1	0.0163	0.7	41.41	0.0865	0.0144	6.0069	1110
2	0.031	2.1	124.22	0.0718	0.0076	9.4474	2112
3	0.048	3.8	224.79	0.0542	0.0091	5.9560	3270
4	0.063	4.9	289.86	0.0405	0.0086	4.7093	4292
5	0.077	7.6	449.57	0.0421	0.0083	5.0723	5245
6	0.0789	7.6	449.57	0.0401	0.0083	4.8313	5375
7	0.0927	9.6	567.88	0.0367	0.0080	4.5875	6315
8	0.1081	12.3	727.60	0.0346	0.0078	4.4359	7364
9	0.1357	18.4	1088.44	0.0328	0.0074	4.4324	9244
10	0.1622	25.4	1502.52	0.0317	0.0071	4.4648	11049
11	0.1901	33.7	1993.50	0.0306	0.0069	4.4348	12950
12	0.2205	42.5	2514.06	0.0287	0.0067	4.2836	15021
13	0.2471	50.2	2969.55	0.0270	0.0066	4.0909	16833
14	0.2752	61.3	3626.16	0.0266	0.0064		18747
15	0.302	74	4377.42	0.0266	0.0063		20573
16	0.3319	84.5	4998.54	0.0252	0.0062		22610
17	0.3431				0.0062		23373

3.1 STANDARDISATION – SMOOTH TUBE (h_i vs Re)

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	hi(thero)	hi(exp)/ hi(thero)
1	0.0163	27.5	45.9	53.8	52.8	14.94	539	1110	739	176	4.20
2	0.0313	27.5	39.3	54.4	53.1	19.89	514	2132	692	218	3.18
3	0.048	27.6	37.5	53.9	52.5	20.35	589	3270	834	563	1.48
4	0.063	27.6	36.4	53.6	51.9	20.54	694	4292	1063	826	1.29
5	0.077	27.6	35.9	54.3	52.2	21.35	799	5245	1330	1052	1.26
6	0.0789	27.7	36.3	54.4	52.5	21.27	784	5375	1289	1079	1.19
7	0.0927	27.7	35.4	53	51.8	20.68	681	6315	1033	1291	0.80
8	0.1081	27.7	34.8	53.5	51.4	21.10	884	7364	1584	1515	1.05
9	0.1357	27.8	34.1	53.9	51.4	21.64	992	9244	1968	1891	1.04
10	0.1622	27.8	33.6	54.2	51.5	22.11	1058	11049	2248	2051	1.10
11	0.1901	27.8	33.2	54.5	51.6	22.52	1125	12950	2571	2334	1.10
12	0.2205	27.8	32.7	54	51.2	22.33	1145	15021	2680	2635	1.02
13	0.2471	27.9	32.3	53.5	50.5	21.89	1212	16833	3075	2891	1.06
14	0.2752	27.9	32	53.7	50.6	22.19	1238	18747	3248	3156	1.03
15	0.302	28.1	31.9	53.9	50.7	22.29	1262	20573	3422	3397	1.01
16	0.3319	28.2	31.8	54.2	50.7	22.45	1338	22610	4041	3664	1.10
17	0.3431	28.2	31.8	54.3	50.9	22.60	1332	23373	3988	3763	1.06

3.2 (h_i vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=0.0

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	R1
1	0.0163	32.6	45.1	53.1	52.3	12.98	462	1110	601	3.42
2	0.0313	32.6	43.3	54.1	53	15.09	592	2132	841	3.86
3	0.048	32.6	41.6	53.9	52.6	15.84	695	3270	1065	1.94
4	0.063	32.6	40.5	53.7	52.2	16.19	784	4292	1289	1.61
5	0.077	32.6	39.9	53.4	51.8	16.18	861	5245	1511	1.48
6	0.0789	32.6	40.2	54.3	52.6	16.88	879	5375	1567	1.49
7	0.0927	32.6	39.5	53.6	51.8	16.52	954	6315	1824	1.46
8	0.1081	32.6	39.3	54.5	52.5	17.44	1013	7364	2055	1.40
9	0.1357	32.6	38.3	53.6	51.5	17.04	1099	9244	2440	1.33
10	0.1622	32.6	37.9	54.4	52.1	17.96	1151	11049	2710	1.37
11	0.1901	32.6	37.3	53.6	51.3	17.47	1207	12950	3043	1.35
12	0.2205	32.7	37	54.3	51.7	18.14	1272	15021	3493	1.38
13	0.2471	32.7	36.6	53.5	51	17.59	1298	16833	3695	1.33
14	0.2752	32.8	36.4	53.6	51	17.70	1333	18747	4001	1.32
15	0.302	32.9	36.4	54.3	51.4	18.20	1413	20573	4818	1.47
16	0.3319	33	36.3	53.8	51.1	17.80	1423	22610	4936	1.40
17	0.3431	33.1	36.3	54.3	51.2	18.05	1501	23373	6018	1.66

3.3 (h_i vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=8.73

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	R1
1	0.0163	31.6	47	54.5	53.7	13.51	488	1110	645	3.67
2	0.0313	31.5	44	54.4	53.2	15.36	655	2132	975	4.47
3	0.048	31.6	41.9	54.3	52.8	16.40	771	3270	1255	2.28
4	0.063	31.7	40.6	54	52.4	16.78	829	4292	1415	1.76
5	0.077	31.7	39.6	53.8	52	17.06	900	5245	1638	1.60
6	0.0789	31.5	39.7	53.5	51.8	16.84	915	5375	1688	1.60
7	0.0927	31.5	39.1	54.2	52.2	17.75	982	6315	1929	1.53
8	0.1081	31.5	38.2	53.6	51.5	17.60	1029	7364	2120	1.44
9	0.1357	31.4	37.4	54.3	51.9	18.64	1101	9244	2451	1.33
10	0.1622	31.4	36.8	54.2	51.6	18.76	1181	11049	2885	1.45
11	0.1901	31.4	36.3	54.6	51.8	19.33	1227	12950	3176	1.40
12	0.2205	31.3	35.8	53.9	51.2	18.98	1267	15021	3454	1.35
13	0.2471	31.3	35.3	53.7	50.9	18.99	1286	16833	3605	1.28
14	0.2752	31.3	35	53.6	50.6	18.94	1355	18747	4199	1.37
15	0.302	31.4	34.9	53.9	51	19.29	1333	20573	3995	1.21
16	0.3319	31.4	34.7	54.1	50.9	19.45	1414	22610	4822	1.35
17	0.3431	31.4	34.5	53.4	50.4	18.95	1384	23373	4498	1.23

3.4 (h_i vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=5.23

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	R1
1	0.0163	32.5	48.8	55.8	54.8	13.20	578	1110	814	4.62
2	0.0313	32.4	45.9	56	54.5	15.32	767	2132	1244	5.71
3	0.048	32.6	44	55.6	54	16.00	859	3270	1505	2.74
4	0.063	32.7	42.6	55.5	53.6	16.58	965	4292	1863	2.33
5	0.077	32.8	41.5	55.2	53.2	16.83	1010	5245	2042	2.00
6	0.0789	32.8	41.6	55	53	16.57	1045	5375	2187	2.09
7	0.0927	32.9	40.8	54.8	52.7	16.73	1088	6315	2388	1.91
8	0.1081	32.9	40.1	54.5	52.4	16.82	1117	7364	2529	1.73
9	0.1357	32.9	38.9	54.4	52.1	17.28	1163	9244	2778	1.52
10	0.1622	33	38.5	54.1	52.5	17.48	1032	11049	2133	1.08
11	0.1901	33.1	38	54.2	51.6	17.32	1319	12950	3877	1.73
12	0.2205	33.1	37.6	55.1	52.2	18.29	1362	15021	4272	1.69
13	0.2471	33.1	37.2	54.6	51.8	18.04	1371	16833	4362	1.57
14	0.2752	33.1	36.9	54.2	51.2	17.70	1470	18747	5546	1.83
15	0.302	33.2	36.7	54.7	51.7	18.25	1433	20573	5056	1.55
16	0.3319	33.2	36.3	54.3	51.3	18.05	1430	22610	5011	1.42
17	0.3431	33.2	36.3	53.9	50.9	17.65	1486	23373	5788	1.60

3.5 (h_i vs Re) FOR TWISTED ALUMINIUM TAPER CLIP HAVING y=3.27

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	R1
1	0.0163	31.5	48.4	53.5	52.4	11.20	731	1110	1153	6.55
2	0.0313	31.5	45.4	53.5	52.1	13.39	857	2132	1500	6.88
3	0.048	30.5	43.2	53.9	52.1	15.51	991	3270	1966	3.58
4	0.063	30.1	41.3	54.3	52.2	17.19	1043	4292	2178	2.70
5	0.077	30.1	40.2	54.3	52	17.71	1109	5245	2489	2.42
6	0.0789	30	40.2	54	51.8	17.49	1117	5375	2531	2.40
7	0.0927	30	39	53.1	51.4	17.49	1015	6315	2059	1.63
8	0.1081	30.1	38	53.4	51	18.01	1167	7364	2805	1.89
9	0.1357	30.1	36.9	53.3	50.7	18.42	1235	9244	3229	1.75
10	0.1622	30.1	36.2	53.4	50.6	18.80	1300	11049	3716	1.85
11	0.1901	30.1	35.6	53.8	50.8	19.42	1339	12950	4053	1.77
12	0.2205	30.1	34.9	53.9	50.8	19.83	1341	15021	4070	1.58
13	0.2471	30.1	34.7	53.4	50.3	19.44	1419	16833	4880	1.73
14	0.2752	30.1	34.3	53.2	50	19.39	1457	18747	5360	1.74
15	0.302	30.2	34.1	53.3	50	19.49	1485	20573	5768	1.73
16	0.3319	30.3	33.9	53.4	50.1	19.65	1484	22610	5759	1.60
17	0.3431	30.3	33.8	53.5	50.2	19.80	1477	23373	5648	1.53

3.6 (h_i vs Re) FOR TWISTED TAPE HAVING y=5.05

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	R1
1	0.0163	30.8	48.3	56.1	54.8	14.41	637	1110	934	5.31
2	0.0313	30.7	45.4	56	54.5	16.32	749	2132	1198	5.49
3	0.048	30.8	43	55.9	54.1	17.59	858	3270	1501	2.73
4	0.063	30.9	41.4	55.7	53.7	18.22	927	4292	1729	2.15
5	0.077	31.1	40.5	55.6	53.5	18.51	978	5245	1914	1.87
6	0.0789	31.3	40.7	55.4	53.4	18.14	986	5375	1945	1.85
7	0.0927	31.4	39.9	55.3	53.1	18.37	1053	6315	2223	1.77
8	0.1081	31.4	39	55.1	52.8	18.62	1084	7364	2367	1.61
9	0.1357	31.4	38	54.9	52.4	18.87	1164	9244	2787	1.52
10	0.1622	31.3	37.2	54.6	52	19.00	1220	11049	3126	1.57
11	0.1901	31.3	36.5	54.4	51.6	19.07	1281	12950	3563	1.57
12	0.2205	31.2	35.9	54	51.3	19.08	1289	15021	3626	1.42
13	0.2471	31.2	35.6	54.5	51.5	19.59	1354	16833	4193	1.49
14	0.2752	31.2	35.2	54.7	51.5	19.89	1385	18747	4507	1.47
15	0.302	31.2	35.1	54.9	51.6	20.09	1441	20573	5152	1.56
16	0.3319	31.2	34.9	55.1	51.7	20.35	1475	22610	5619	1.58
17	0.3431	31.2	34.9	55.3	51.9	20.55	1486	23373	5778	1.58

3.7 (h_i vs Re) FOR TWISTED TAPE HAVING y=3.25

SL.NO.	m	T1	T2	T3	T4	LMTD	Ui	Rec	hi(exp)	R1
1	0.0163	32.8	49.2	54.7	53.7	11.536	664	1110	994	5.65
2	0.0313	32.6	46.7	55.7	54.3	14.430	801	2132	1336	6.13
3	0.048	32.6	44.6	56	54.2	15.960	938	3270	1765	3.22
4	0.063	32.6	43	56.1	54.1	16.955	992	4292	1968	2.46
5	0.077	32.5	42	56.5	54.3	17.903	1041	5245	2171	2.13
6	0.0789	32.5	42.3	56.8	54.6	18.034	1062	5375	2265	2.16
7	0.0927	32.4	41.6	57	54.6	18.593	1130	6315	2598	2.08
8	0.1081	32.4	40.7	57.2	54.7	19.255	1142	7364	2665	1.82
9	0.1357	32.4	39.8	57.7	54.8	20.066	1249	9244	3326	1.82
10	0.1622	32.4	39	57.9	54.9	20.648	1275	11049	3516	1.78
11	0.1901	32.4	38.4	58	54.8	20.969	1338	12950	4044	1.80
12	0.2205	32.4	37.9	58.1	54.8	21.281	1381	15021	4463	1.76
13	0.2471	32.3	37.4	58.2	54.8	21.639	1406	16833	4730	1.70
14	0.2752	32.4	37.1	58.4	54.8	21.845	1451	18747	5285	1.74
15	0.302	32.4	36.8	58.4	54.7	21.948	1484	20573	5750	1.76
16	0.3319	32.5	36.6	58.6	54.7	22.100	1531	22610	6527	1.85
17	0.3431	32.5	36.6	58.6	54.8	22.150	1534	23373	6579	1.82

4.1 Pr No vs Temperature

SL.NO.	T1	T2	(T1+T2)/2	Prc
1	27.5	45.9	36.7	5.019
2	27.5	39.3	33.4	5.192
3	27.6	37.5	32.55	5.405
4	27.6	36.4	32	5.479
5	27.6	35.9	31.75	5.483
6	27.7	36.3	32	5.479
7	27.7	35.4	31.55	5.553
8	27.7	34.8	31.25	5.557
9	27.8	34.1	30.95	5.561
10	27.8	33.6	30.7	5.565
11	27.8	33.2	30.5	5.57
12	27.8	32.7	30.25	5.708
13	27.9	32.3	30.1	5.776
14	27.9	32	29.95	5.846
15	28.1	31.9	30	5.777
16	28.2	31.8	30	5.777
17	28.2	31.8	30	5.777