SOME EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION FOR FLOW OF LIQUID THROUGH CIRCULAR TUBES USING TWISTED ANGLES AND TAPES

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

Bachelor of Technology
in
Chemical Engineering

By

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&
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Department of Chemical Engineering
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Under the Guidance of
Prof. S.K.AGARWAL

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CERTIFICATE

This is to certify that the thesis entitled, "SOME EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION FOR FLOW OF LIQUID THROUGH CIRCULAR TUBES USING TWISTED ANGLES AND TAPES" submitted by Ranjit Gouda and Amit Bikram Das 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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Date: Ranjit Gouda

Amit Bikram Das

CONTENTS

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. Therefore any augmentation device should optimize between the benefits due to the increased heat transfer coefficient and the higher cost involved because of the increased frictional losses.

The present paper includes various heat transfer augmentation techniques. A literature review of heat transfer augmentation using passive techniques has been included. Experimental work on heat transfer augmentation using twisted tape and a new kind of insert called twisted angles 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 twisted tapes and twisted angles having different twist ratios. The results of twisted tapes and twisted angles having different twist ratios have been compared with the values for the smooth tube. Five twisted angles($y=\infty$, $y=2.915$, $y=3.612$, $y=4.105$ & $y=5.07$) and three twisted tapes($y=2.149$, $y=3.127$ & $y=4.705$) having different twist ratios are used in the study.

For twisted angles it was observed that the heat transfer coefficient could vary from 1.16 to 2.87 times the smooth tube value but the corresponding friction factor increases by 4 to 9.6 times the smooth tube values. Similarly for twisted tapes it was observed that the heat transfer coefficient varied from 1.28 to 2.48 times and the friction factor increased by 3.19 to 9.1 times the smooth tube value. It was also observed that with an increase in Reynolds number (Re), the heat transfer coefficient increases where as the friction factor decreases.

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION:-

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 10000 m^2/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 passive techniques.

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

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 Classification of Augmentation Techniques:

They are broadly classified into three different categories:

1. Passive Techniques
2. Active Techniques
3. Compound Techniques.

1) **Passive Techniques:** These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop. They generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour except for extended surfaces. Heat transfer augmentation by these techniques can be achieved by using;

(i) *Treated Surfaces*: Such surfaces have a fine scale alteration to their finish or coating which may be continuous or discontinuous. They are primarily used for boiling and condensing duties.

(ii) *Rough surfaces*: These are the surface modifications that promote turbulence in the flow field in the wall region, primarily in single phase flows, without increase in heat transfer surface area.

(iii) *Extended surfaces*: They provide effective heat transfer enlargement. The newer developments 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.

(iv) *Displaced enhancement devices*: These are the 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.

(v) *Swirl flow devices*: They produce and superimpose swirl flow or secondary recirculation on the axial flow in a channel. These include helical strip or cored screw type tube inserts, twisted tapes. They can be used for single phase and two-phase flows.

(vi) *Coiled tubes*: These lead to relatively more compact heat exchangers. It produces secondary flows and vortices which promote higher heat transfer coefficients in single phase flows as well as in most regions of boiling.

(vii) *Surface tension devices*: These consist of wicking or grooved surfaces, which direct and improve the flow of liquid to boiling surfaces and from condensing surfaces.

(viii) *Additives for liquids*: These include the addition of solid particles, soluble trace additives and gas bubbles in single phase flows and trace additives which usually depress the surface tension of the liquid for boiling systems.

(ix) *Additives for gases*: These include liquid droplets or solid particles, which are introduced in single-phase gas flows either as dilute phase (gas-solid suspensions) or as dense phase (fluidized beds).

2) **Active Techniques:** In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer. Augmentation of heat transfer by this method can be achieved by

(i) *Mechanical Aids*: Such instruments stir the fluid by mechanical means or by rotating the surface. These include rotating tube heat exchangers and scrapped surface heat and mass exchangers.

(ii) *Surface vibration*: They have been applied in single phase flows to obtain higher heat transfer coefficients.

(iii) *Fluid vibration*: These are primarily used in single phase flows and are considered to be perhaps the most practical type of vibration enhancement technique.

(iv) *Electrostatic fields*: It can be in the form of electric or magnetic fields or a combination of the two from dc or ac sources, which can be applied in heat exchange systems involving dielectric fluids. Depending on the application, it can also produce greater bulk mixing and induce forced convection or electromagnetic pumping to enhance heat transfer.

(v) *Injection*: Such a technique is used in single phase flow and 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.

(vi) *Suction*: It 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.

(vii) *Jet impingement*: It involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface.

3) Compound Techniques: When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement. This technique involves complex design and hence has limited applications.

2.2 Performance Evaluation Criteria:

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

- Increase in the heat duty of an existing heat exchanger without altering the pumping power or flow rate requirements.
- Reduction in the approach temperature difference between the two heat exchanging fluid streams for a specified heat load and size of exchanger.
- Reduction in the size or heat transfer surface area requirements for a specified heat duty and pressure drop.
- Reduction in the process stream's pumping power requirements for a given heat load and exchanger surface area.

Different Criteria used for evaluating the performance of a single phase flow are:

- ➢ *Fixed Geometry (FG) Criteria*: The area of flow cross-section (N and d_i) and tube length L are kept constant. They would typically be applicable for retrofitting the smooth tubes of an existing exchanger with enhanced tubes, thereby maintaining the same basic geometry and size (N, d_i, L). The objectives then could be to increase the heat load Q for the same approach temperature ΔT_i and mass flow rate m or pumping power P; or decrease ΔT_i or P for fixed Q and m or P; or reduce P for fixed Q.

- *Fixed Number (FN) Criteria*: The flow frontal area or cross-section (N and d_i) is kept constant and the heat exchanger length is allowed to vary. Here the objectives are to reduce either the heat transfer surface area (A→ L) or the pumping power P for a fixed heat load.
- *Variable Geometry (VN) Criteria*: The number of tubes and their length (N & 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 frictional 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 grater number of parallel flow circuits.

		Fixed				
Case	**Geometry**	**m**	**P**	**Q**	**ΔT_i**	**Objective**
FG-1a	N, L	X			X	Q↑
FG-1b	N, L	X		X		ΔT_i↓
FG-2a	N, L		X		X	Q↑
FG-2b	N, L		X	X		ΔT_i↓
FG-3	N, L			X	X	P↓
FN-1	N		X	X	X	L↓
FN-2	N	X		X	X	L↓
FN-3	N	X		X	X	P↓
VG-1	—	X	X	X	X	$(NL)^a$↓
VG-2a	$(NL)^a$	X	X		X	Q↑
VG-2b	$(NL)^a$	X	X	X		ΔT_i↓
VG-3	$(NL)^a$	X		X	X	P↓

Table 2.1: Performance Evaluation Criteria for Single Phase Forced Convection in Enhanced Tubes of Same Envelope Diameter (d_i) as the smooth Tube

2.3 TREATED SURFACES:

These are primarily applicable in two phase heat transfer and they consist of a variety of structured surfaces (continuous or discontinuous integral surface roughness or alterations) and coatings. Though the treatment provides a roughness to the surface, it is not large enough to influence single phase heat transfer.

2.3.1 Boiling: Different types of treated surfaces used are

- Machined or grooved surfaces
- Formed or modified low-fin surfaces
- Multilayered surfaces
- Coated surfaces

The principle of providing treated surfaces for enhanced boiling is to produce a large number of stable vapor traps or nucleation sites on the surface. This is applicable for highly wetting fluids like refrigerants, organic liquids, cryogens and alkali liquid metals where the normal cavities present on the heated surfaces tend to experience sub-cooled liquid flooding. For less wetting or relatively higher surface tension fluids, coatings of non-wetting material (eg.teflon) on either the heated surface or its pits and cavities have been found to improve stable nucleation and reduce the required wall super heat were proposed by Griffith and Wallis,(1960);Young and Hummel,(1965);Gaertner,(1967);Vachon et al.,(1969). When the stainless steel surface along with Teflon is spread to create spots of the no-wetting material on the heated surface it was found to promote nucleate boiling in water with relatively low wall super heat and three to four times higher heat transfer coefficients, was proposed by Young and Hummel(1965). In a more recent study of boiling of alcohols (methanol, ethanol and isopropanol) at atmospheric and sub-atmospheric pressures on a horizontal brass tube coated with poly-tetra-fluoro-ethylene (PTFE), Vijaya Vittala et al. (2001), found a significant enhancement in heat transfer.

2.3.2 Condensing: Vapor space condensation heat transfer coefficients can be enhanced primarily by treated surfaces that promote drop wise condensation. The intent here is to prevent surface wetting and break up the condensate film into droplets which lends to better drainage and more effective vapor renewal at the cold heat transfer interface. This technique had been found to enhance the heat transfer by a factor of 10 to 100 in comparison with that in film wise condensation proposed by Bergles, (1998). Non-wetting coatings of an inorganic compound or a noble metals or an organic polymer have been used effectively. Among these, organic coatings have been used considerably in steam systems.

Glicksman et al. (1973) have been found out that, by placing strips of Teflon or other non-wetting material in a helical or axial arrangement around the circumference of horizontal tubes, the average condensation heat transfer coefficients of steam on horizontal tubes can be improved by 20 to 50%. The application of hydrophobic coatings of self-assembled monolayers, formed by chemisorption of alkylthiols on metallic surfaces, to promote dropwise condensation has been proposed by Das et al. (2000). It was found that steam condensation on coated corrugated tubes with gold and copper-nickel alloy surfaces under atmospheric and sub-atmospheric pressure conditions with wall sub-cooling of about 16°C and 6°C respectively showed that condensation heat transfer coefficients increased by factors of 2.3 to 3.6 compared to those for un-coated tubes.

2.4 ROUGH SURFACES:

2.4.1 Single Phase Flow: The use of surface roughness in turbulent single phase flow is one of the simplest and highly effective techniques; small scale roughness has little effect in laminar flows. It essentially disturbs the viscous laminar sub-layer near the wall to promote higher momentum and heat transport. Surface roughness can be introduced in the form of wire-coiled type inserts or it may be integral to the surface.

Rough surfaces have been employed to enhance heat transfer in single phase flows both inside tubes and outside tubes. Dong et al. (2001) developed a new set of analogy based friction factor and Nusselt number correlations for turbulent flows of water and oil in spirally corrugated tubes. Adopting an empirical approach, combined with a statistical analysis of a fairly large database for heat transfer coefficients and friction factors for various roughness shown above, Ravigururajan and Bergles (1996) proposed correlations for Nusselt number and fanning factor as:

$$Nu=Nu_o\{1+ [2.64Re^{0.036}(e/d)^{0.212}(p/d)^{-0.21}(\alpha/90)^{0.29}. (Pr)^{-0.024}]^7\}^{1/7}$$

$$f=f_o\{1+ [29.1Re^{a1}(e/d)^{a2} (p/d)^{a3}(\alpha/90)^{a4} (1+2.94\sin(\beta/n))]^{15/16}\}^{16/15}$$

Where $a1=0.67 - 0.06(p/d) - 0.49(\alpha/90)$; $a2=1.37 - 0.157(p/d)$

$a3=-1.66 X 10^{-6}Re-0.33(\alpha/90)$; $a4=4.59 + 4.11 X 10^{-6}Re-0.15(p/d)$

and the respective smooth tube Nu_o and f_o performances are given by

$$Nu_o=Re. Pr (f_{o/2})/ [1 + 12.7(f_o/2)^{0.5}(Pr^{2/3} - 1)]$$

$$f_o=(1.58 \ln Re-3.28)^{-2}$$

These above correlations have been shown very good compared with more than 1800 experimental data points.

Tubes with grooves provide an external rough surface and have been used in double pipe and shell and tube bundles to enhance annulus or shell side heat transfer. Variable roughness can be obtained by using a wire-coil insert made of a shape memory alloy (SMA) that alters its geometry in response to change in temperature proposed by Bergles and Champagne, 1999. With a fixed roughness height (e/d), the wire coil inserts change from a compressed shape, which occupies a smaller fraction of the tube length, to an expanded shape that has the desired roughness pitch (p/d) and helix pitch (α/90) upon being heated. Champagne and Bergles (2001) have also shown that by using SMA (NiTi) wire coil inserts, heat transfer coefficients can be increased from 30 to 64% in single phase turbulent flow.

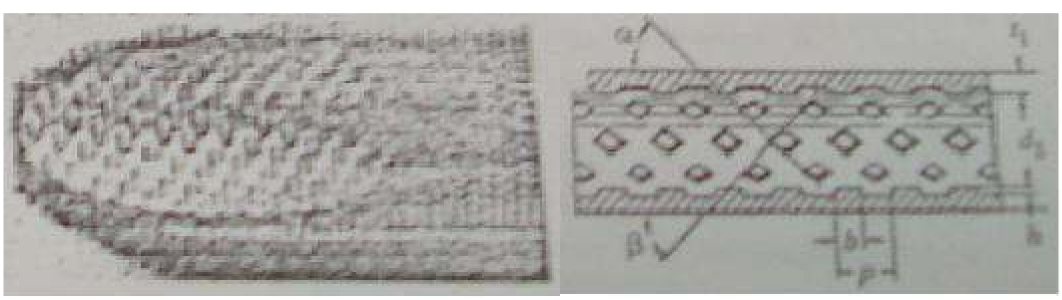


Fig. 2.1: (Three-Dimensional Roughness)

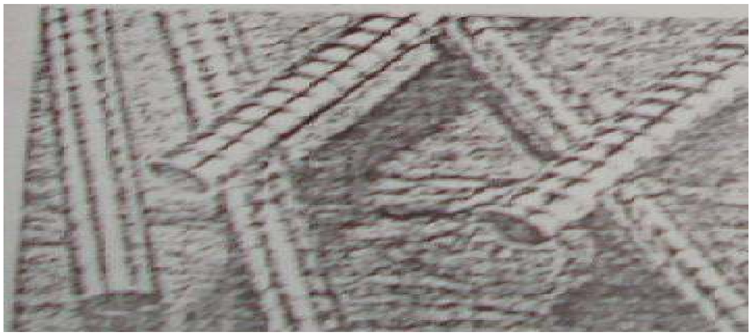


Fig. 2.2: (Corrugated tubes, Two-Dimensional Roughness)

Some of the recent work with water and propylene glycol flows in annuli with corrugated tubes indicate up to four times higher Nusselt number, with up to ten times higher friction factor in turbulent flow regime proposed by Garimella and Christensen,1995; Salim et al.,1999; Kang and Christensen,2000. Durant et al. (1965) have reported that by using three-dimensional diamond- knurls type of roughness on the inner heated tube's outer surface, the heat transfer coefficient can be increased up to 75% higher than those for the equivalent smooth annuli. In case of pyramid shaped roughness element considered by Achenbach (1977) and Zhukauskas et al. (1978), with air and water flows, 150% increase in Nusselt number is reported.

2.4.2 Boiling:

The use of helical ribbed tubes in high-pressure power boilers, as considered by Bergles, (1998) found to increase the heat transfer coefficient and critical heat flux (CHF) in once through boiling of water. Commercially structured rough surfaces in the form of corrugated tubing have extensively employed in refrigerant evaporators. Withers and Habdas (1974) have reported up to 100% increase in the heat transfer coefficient and up to 200% enhancement in critical heat flux (CHF) in bulk boiling, in helically corrugated tubes. Artificial roughness in the form of longitudinal ribs or grooves has been applied in gravity driven, horizontal tube evaporators. Though these types of surfaces promote turbulence, it tends to impede film drainage proposed by Bergles (1998).Cox et al.(1969) found that three-dimensional rough surfaces tend to promote turbulence as well as the liquid spreading thereby increasing the heat transfer coefficient as much as 100%.

2.4.3 Condensing:

Corrugated tubes have been extensively used for enhancement of vapor space condensation. Rough surfaces also improve in-tube forced convective condensation. The overall heat transfer coefficient with forced convection condensation inside and spray film evaporation outside could be improved by using spirally indented and V-grooved tubes. Thomas (1967) attached axial wires around the periphery of vertical smooth tubes, which facilitated better surface tension driven condensate film drainage to produce three to four fold enhancement in heat transfer. He further proposed that square profiled wires were more effective than circular ones of the same roughness height. In steam condensation on horizontal helically corrugated tubes, Mehta and Raja Rao (1979) and Zimparov et. al. (1991), Webb (1994), Das et al. (2000) have reported about 1.1 to 1.4 times increase in heat transfer coefficient. Dreitser et al. (1988) have reported 1.8 to 2.65

times higher steam condensation heat transfer coefficients on horizontal tubes with transverse grooves or corrugations.

2.5 EXTENDED SURFACES:

Extended or finned surfaces are most widely used techniques which include finned tube for shell & tube exchangers, plate fins for compact heat exchanger and finned heat sinks for electronic cooling.

2.5.1 Single-Phase Flow: Enhanced heat transfer from finned surfaces for buoyancy driven natural or free convection has been considered primarily for cooling of electrical and electronic devices and for hot water base-board room heaters. The use of extended surfaces for cooling electronic devices is not restricted to the natural convection heat transfer regime but also can be used for forced convective heat transfer. By using segmented or interrupted longitudinal fins inside circular tubes, heat transfer can be increased by periodically disrupting and restarting the boundary layer on the finned surface and perturbing the bulk flow field. Plate fin or tube and plate fin type of compact heat exchangers , where the finned surfaces provide a very large surface area density, are used increasingly in many automotive, waste heat recovery, refrigeration and air conditioning , cryogenic, propulsion system and other heat recuperative applications.

A variety of finned surfaces typically used, include offset strip fins, louvered fins, perforated fins and wavy fins.

Fig. 2.3: (Tubes with Circumferential and strip fins on their outer surface)

Watkinson et al. (1975) gave the expression for hydraulic diameter based isothermal fanning friction factor and Nusselt number for internally finned tubes with straight o r spiral fins and laminar flows.

$$f_h = (16.4/Re_h) \times (d_h/d)^{1.4}$$

For straight fin tubes,

$$Nu_h = [(1.08 \log Re_h)/ \{n^{0.5} (1 + 0.01\ Gr_h)\}].Re_h^{0.46}.\ Pr^{(1/3).} (L/d_h)^{(1/3)}(\mu_w/\mu_b)^{0.14}$$

For spiral fin tubes,

$$Nu_h = [(8.533 \log Re_h)/ (1 + 0.01\ Gr_h)].Re_h^{0.26}.\ Pr^{(1/3).}\ (t/p)^{0.5}(L/d_h)^{(1/3)}(\mu_w/\mu_b)^{0.14}$$

Carnavo (1979) recommended following expressions for f_h and Nu_h in turbulent flows in tubes with straight and spiral fins.

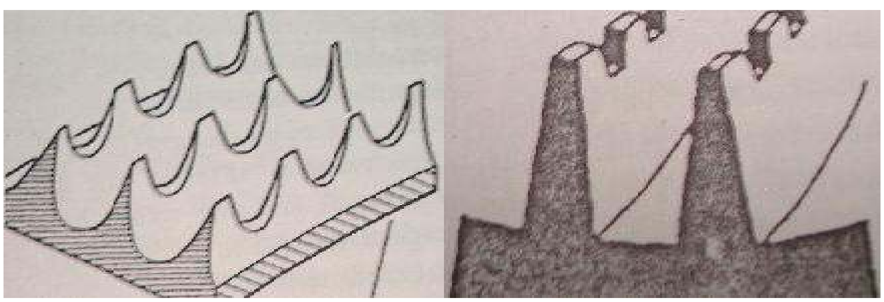
$$f_h = 0.046\ Re_h^{-0.2}\ (A_f / A_{fi})^{0.5}\ (\sec \alpha)^{0.75}$$

$$Nu_h = 0.023\ Re_h^{0.8}\ Pr^{0.4}\ (A_f / A_{fi})^{0.1}\ (A_i / A)^{0.5}\ (\sec \alpha)^3$$

Kelkar and Patankar (1990) considered in-line segmented fins which had half the fin surface area of staggered or continuous fins, were found to perform better with 6% higher Nusselt number and 22% lower friction factor.

2.5.2 Boiling: Internal finned tubes are widely used in refrigerant evaporators and for flow boiling. In pool-boiling, finned tubes have higher heat transfer coefficients compared with the performance of equivalent smooth tubes. In most heat exchangers for refrigeration and air-conditioning systems, micro finned tubes are extensively used.
Bergeles (2000) pointed out that by using fin structures, the heat transfer coefficients can be increased up to 200%.

2.5.3 Condensing: Extended surfaces that include a variety of large, medium and micro sized fins are used extensively for condensation heat transfer enhancement in power, process, and air conditioning and refrigeration applications. The heat exchangers in these duties involve both horizontal and vertical tube condensers with fins on the inside or outside surfaces of tubes. For integral fin tubes, besides the increased surface area, high heat transfer coefficients are obtained because a relatively thin condensate film tends to be formed near the fin tubes and surface tension forces pull the condensate into the inter fin grooved spaces, there by promoting better drainage and reduction of liquid film resistance.



(Notched fins) (Serrated-tip microfins)

Fig. 2.4: Three-dimensional finned surfaces for enhanced condensation:

Chandran and Watson (1976) found that by using circular pin fins, average heat transfer coefficient can be increased to 20% more than those for a smooth tube. Itoh et al. (1997) had shown that micro fins with serrated tips (as shown above) provide 30 to 60% improvements in the heat transfer coefficients over same sized conventional micro fin tubes.

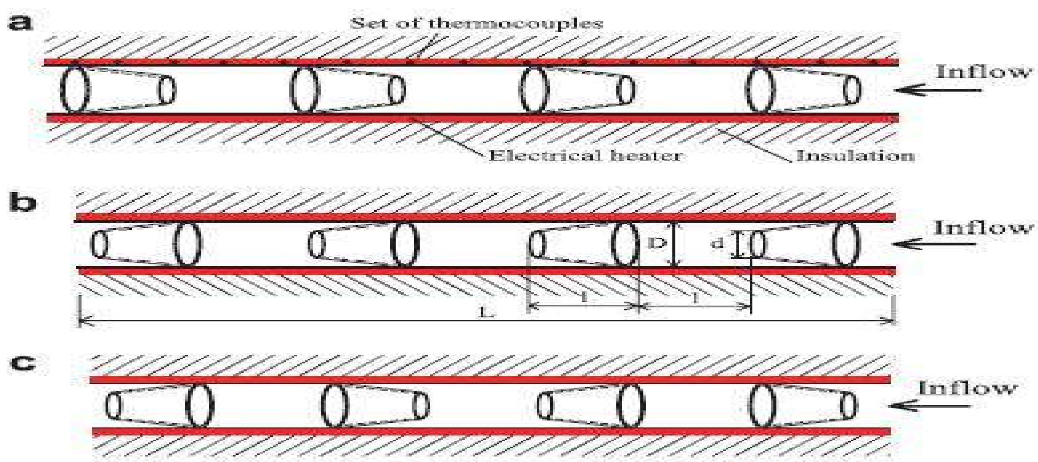
2.6 DISPLACED ENHANCEMENT DEVICES:

2.6.1 Single-Phase Flow: Several types of inserts which are categorized as displaced enhancement devices include static mixer elements (e.g. Kenics, Sulzer), metallic mesh, discs, wire matrix inserts, rings or balls which tend to displace the fluid from the core of the channel to its heated or cooled wall and vice versa, keeping the heat transfer surface unaltered. Rings and round balls have comparable heat transfer improvements, but the friction factors are exorbitantly high. Most of the devices are effective only in laminar flows, as in turbulent flows, the pressure drop penalties are extremely high as reported by

Bergles (1998). The applications of static mixers are generally restricted to chemical processing with heat transfer, where fluid mixing is the primary need.

Spiral brush inserts in short channels with turbulent flows and high wall heat flux have been shown by Megerlin et al. (1974) and found out that heat transfer coefficient can be improved as much as 8.5 times that in a smooth tube, but pressure drop was exorbitantly high; which restricted its use in practical applications.

P. Promvonge (2007), conducted experiments by inserting several conical rings as turbulators over a test tube. Conical rings with three different diameter ratios of the ring to the diameter (d/D = 0.5, 0.6, 0.7) were introduced in the tests and for each ratio, the rings were placed with three different arrangements (Converging conical Ring-CR, Diverging conical Ring-DR, Converging Diverging conical Ring-CDR). Cold air at ambient air temperature was passed through the tube. He found out that such inserts lead to a higher heat transfer rates than plane tubes and DR yielded better heat transfer than the others. The Nusselt number was found to increase by 197%, 333%, 237% in case of CR, DR and CDR array respectively. It leads to a substantial increase in friction factor.



a:- Diverging Ring
b:- Converging Ring
c:- Converging and Diverging Rings

Fig. 2.5: Conical Ring inserts in circular tubes

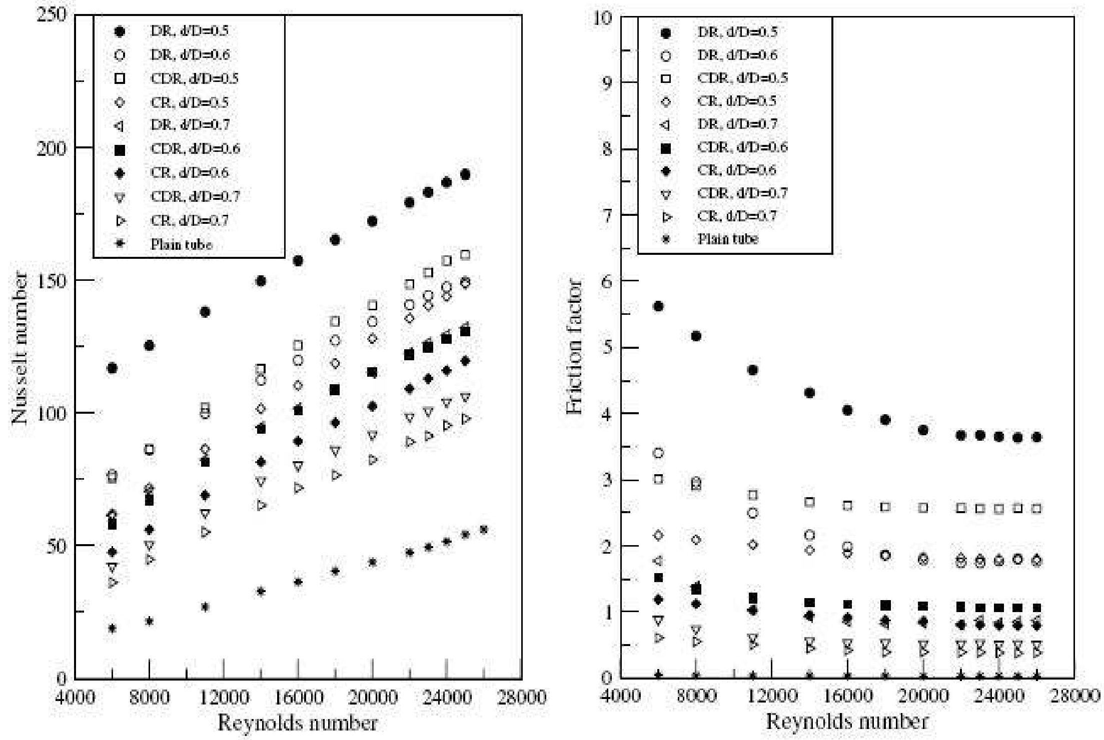
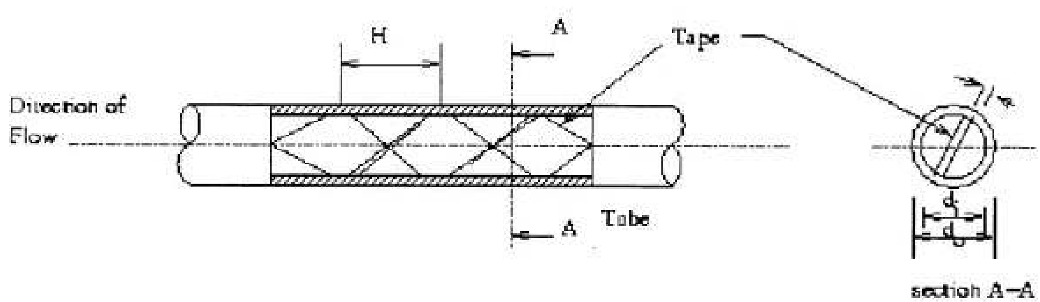


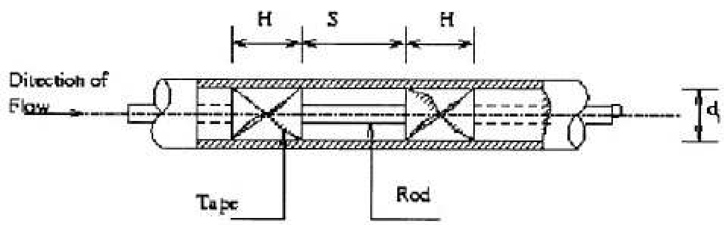
Fig. 2.6: Variation of Nu and friction factor w.r.t. Re for conical ring inserts

2.7 SWIRL FLOW DEVICES:

Swirl flow devices generally consist of a variety of tube inserts, geometrically varied flow arrangements and duct geometry modifications that produce flows. These techniques include twisted tape inserts, periodic tangential fluid injection and helically twisted tubes.



(a)



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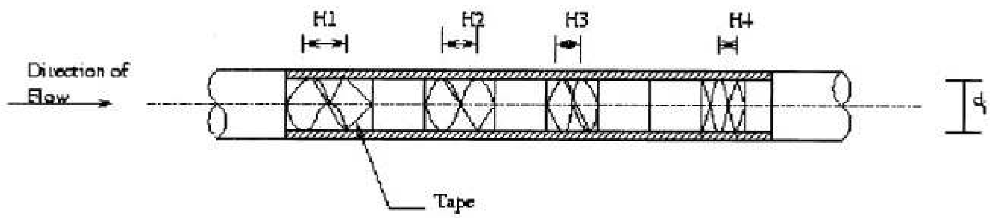

(c)

Fig 2.7 Example of (a) full-length twisted tape, (b) regularly spaced twisted tape and (c) smoothly varying (gradually decreasing) pitch full-length twisted tape

2.7.1 Single-Phase flows: Twisted tape inserts are the most widely used swirl flow device for single-phase flows. These inserts increase the heat transfer coefficient significantly with a relatively small pressure drop penalty as reported by Smithberg and Landis (1964); Lopina and Bergles (1969); Date and Singham (1972); Manglik and Bergles (1992); Manglik and Yera (2002). Twisted tapes can be used in the existing shell and tube heat exchangers to upgrade their heat duties or when employed in a new exchanger for a specified heat duty, significant reduction in size can be achieved. The ease of fitting multiple bundles with tape inserts and their removal makes them useful in fouling situations, where frequent tube-side cleaning may be required. When swirl flow devices are placed inside a circular tube, the flow field gets altered in several ways like an increase in axial velocity and wetted perimeter due to the blockage and partitioning of the flow cross-section, longer effective flow length in the helically twisting partitioned duct and tape's helical curvature induces secondary fluid circulation or swirl. Swirl generation is the most dominant mechanism which effects transverse fluid transport across the tape partitioned duct, thereby promoting greater fluid mixing and higher heat transfer coefficients.

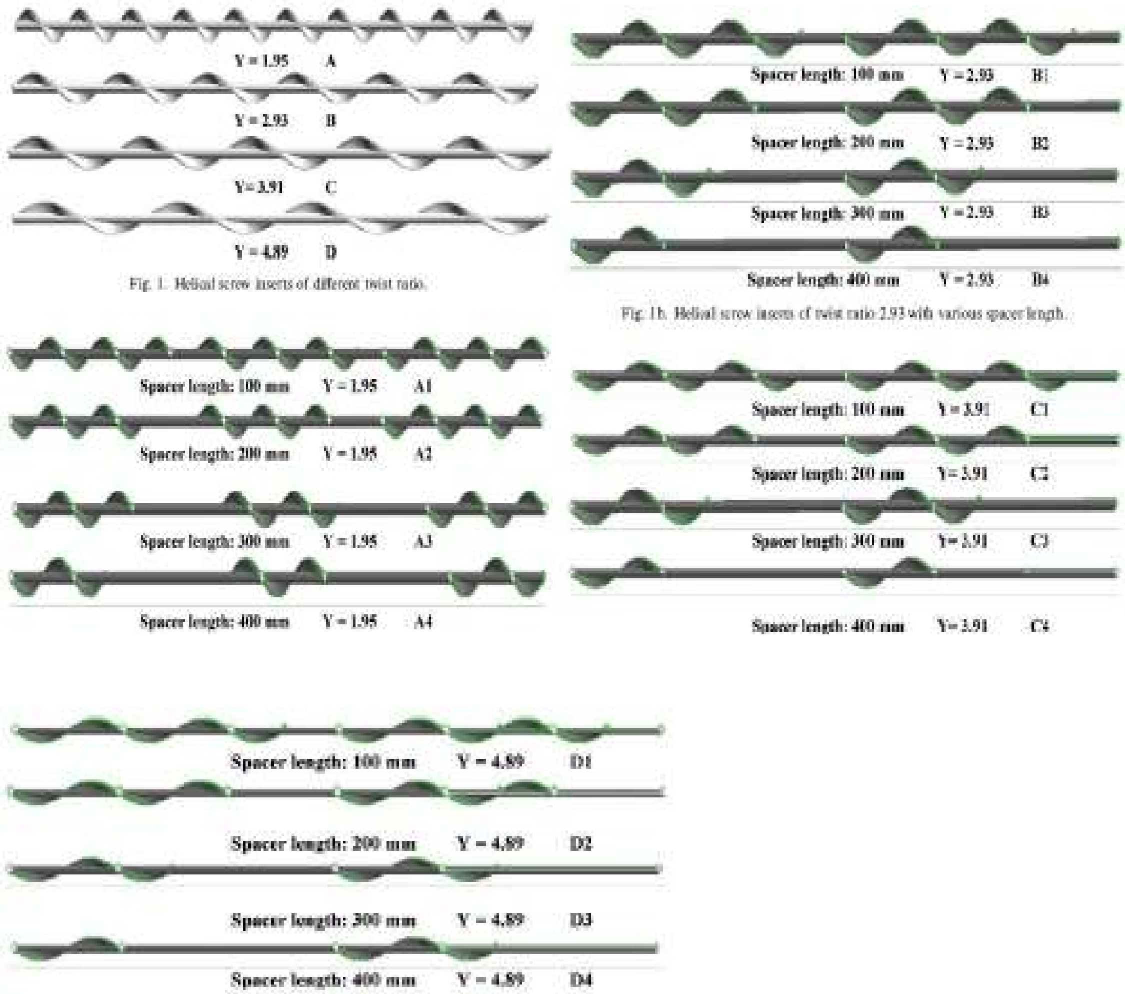

Fig. 2.8: Twisted tape inserts with different spacer lengths and twist ratios

P.Sivashanmugam and S.Suresh (2006) conducted experimental investigation of heat transfer and friction factor characteristics of circular tube fitted with full length helical screw element of different twist ratio (1.95, 2.93, 3.91, 4.89), and helical screw inserts with spacer length 100, 200, 300 and 400 mm as shown above with uniform heat flux under turbulent flow condition. The friction factor for helical twist insert with spacer length 100 mm was found to be very close to the value of that of full length helical twist for all Reynolds number and decreases by 5% for each 100 mm increment space length indicating that there is no much reduction in pumping power. The increase in Nusselt number from twist ratio 4.89 to 1.95 is nearly 30 to 40% for all Reynolds number for full length helical twist where as the decrease in friction factor is about 40 to 45% for various spacer lengths.

They developed empirical equations for Nusselt number and friction factor:

$$Nu = 0.258\,(Re)^{0.254}\,(Pr)\,(Y)^{-0.242}(1+S/D_h)^{-0.042}$$

$$f = (Re)^{-0.384}(Y)^{-0.852}(1+S/D_h)^{-0.047}$$

Various graphical representations obtained by them are as follows:

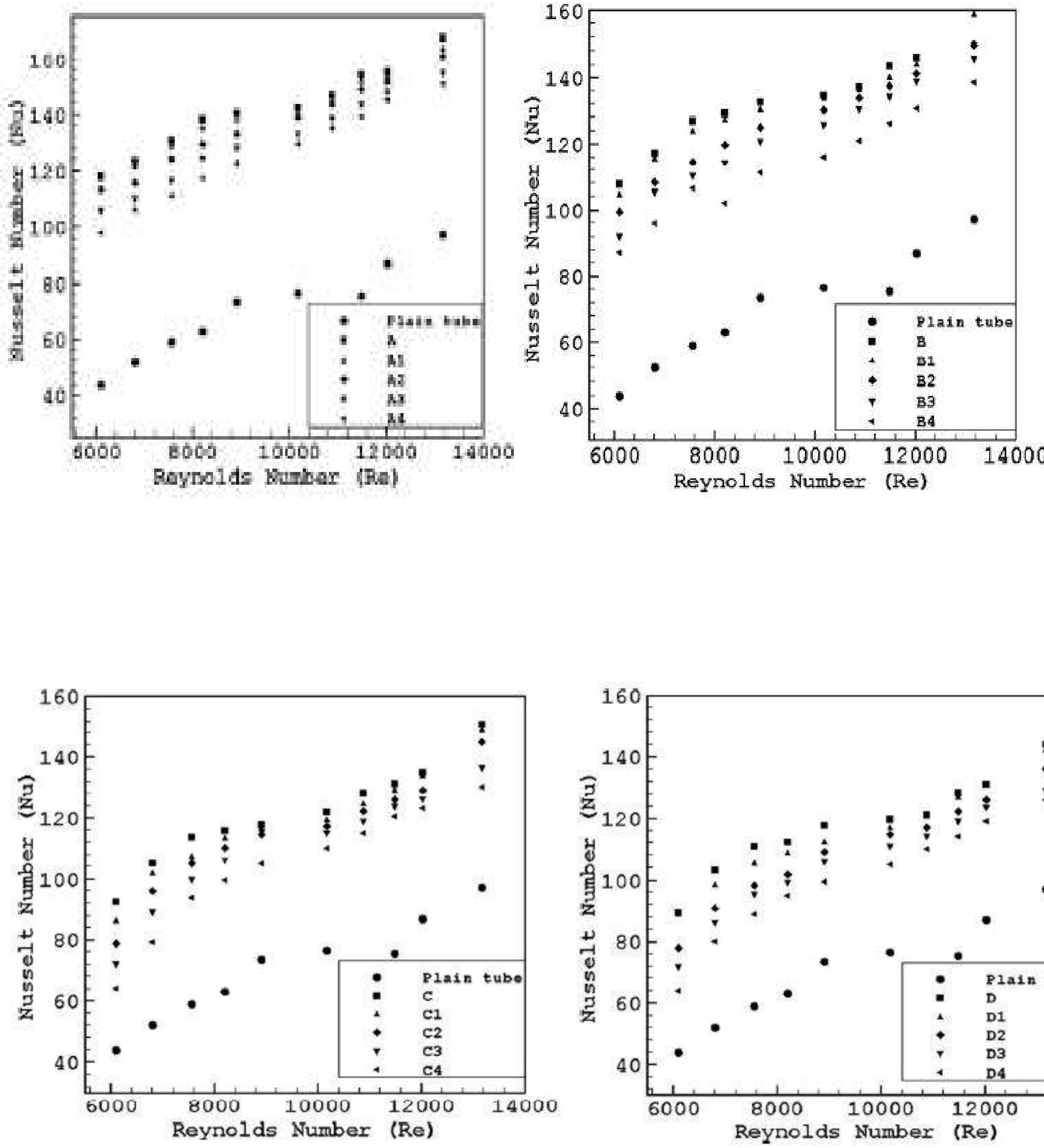

Fig. 2.9: Variation of Nu w.r.t. Re for twisted tapes with different spacer lengths.

2.7.2 Boiling: The heat transfer enhancement due to the tape inserts is reflected in the reduced wall temperature along the tube length in a single phase liquid, sub-cooled boiling, bulk boiling and dispersed film boiling. The primary enhancement mechanism is the tape induced swirl, which tend to increase vapor removal and wetting of the heated surface.

2.7.3 Condensing: Steam condensation in tubes fitted with twisted tape inserts enhanced the heat transfer coefficients by 30% over the empty tube values.

2.8 COILED TUBES:

A coiled or curved tube is a swirl producing geometry where the secondary fluid motion is generated by the continuous change in direction of the tangential vector to the bounding curve surface of the duct, which results in the local deflection of the bulk flow velocity vector. Applications of coiled tubes in domestic water heaters, chemical process reactors, industrial & marine boilers, kidney dialysis devices and blood oxygenators were found by Bergles et al. (1991); Nandakumar and masliyah, (1986).

2.8.1 Single-Phase Flow: The single phase flow behavior, thermal-hydraulic performance and applications of curved and coiled tubes of circular as well as noncircular cross section have been proposed by Nandakumar and masliyah, 1986; Shah and Joshi, 1987; Bergles et al., 1991; Ebadian and Dong, 1998.The curvature induced swirl flow characteristics of curved or helically coiled tubes are strongly dependent on their geometrical attributes.

The tube curvature acts to impose a centrifugal force on the fluid motion, thereby generating a secondary circulation in laminar flows which consist of two symmetrical counter-rotating helical vortices was proposed by Mori and Nakayama,1965;Collin and Dennis,1975;. Nandakumar and masliyah, 1982;Prusa and Yao,1982;Cheng and Yuen,1987.The thermal entrance region for curved tubes is significantly smaller than that for straight tubes for the same flow conditions.

2.8.2 Boiling: Coiled tubes are commonly employed in commercial vapor generators, as they provide a substantial improvement in the evaporation heat transfer coefficient with a significantly smaller surface area to volume ratio. In forced convective evaporation of refrigerants, heat transfer coefficients were found to increase by 60% (Barskii and Chukhman, 1971).

2.9 ADDITIVES FOR LIQUIDS:

2.9.1 Single-Phase flow: This technique for single-phase liquid flows has focused primarily on drag reducing consequences on the additives. The lowering of frictional losses has the indirect effect of providing heat transfer enhancement when evaluated on a fixed pressure drop or pumping power basis. In the case of soluble polymeric additives in water, where the solution has a shear thinning rheology, the non-Newtonian effects lead to a significant reduction in frictional loss as well as a modest increase in the heat transfer coefficient as reported by Joshi and Bergles (1982), Prusa and Manglk (1995), Hartnett and Cho (1998), Chhabra and Richardson (!999), Manglik and Fang (2002). With polymeric additives that imparts a viscoelastic character to the solution, the heat transfer has been found to be further enhanced in rectangular ducts due to a visoelasticity driven secondary circulation that is imposed over the bulk flow (Hartnett and Kostic, 1985; Hartnett, 1992; Hartnett and Cho, 1998). Some of the additives used are polystyrene spheres suspension in oil and injection of gas bubbles. By injecting air bubbles at the base of a heated vertical wall, Tamari and Nishikawa (1976) found up to 400% higher free convection heat transfer coefficient in water and ethylene glycol. In a turbulent flow of water, Kenning and Kao (1972) obtained upto 50% increase in heat transfer by injecting nitrogen bubbles.

2.9.2 Boiling: The use of various additives like surfactants, polymers, etc. that lower the surface tension of the solution and binary mixtures of liquid (wetting agents, alcohols) have been found to enhance pool boiling substantially. Nucleate boiling heat transfer coefficient increases up to 20 to 160% in surfactant solutions depending on their concentrations (Tzan and Yang, 1990; Ammerman and You, 1996; Wu et al.,1998; Manglik, 1998; Hetsroni et al., 2000; Wasekar and Manglik, 2002) and 20 to 40% in binary liquid mixtures with wetting agents or alcohols.The improved thermal performance is strongly depended on the type and concentration of the surfactant additive, its chemistry (ionic nature, molecular and chemical composition and structure) and the diffusion kinetics at the dynamic liquid interface. The lowering of the solution's surface tension promotes nucleation of smaller bubbles, with a clustered activation of nucleation sites which depart at much higher frequencies than seen in pure water.

2.10 ACTIVE TECHNIQUES:

Rotating surfaces substantially enhance heat transfer coefficient up to 350% for laminar flows in straight tubes rotating around their own axis or a parallel axis (Mori and Nakayama, 1967; McElhiney and Preckshot, 1977; Bidyanidhi et al., 1977).Tang and McDonald, 1971 reported that, with high speed rotation of heated cylinders in saturated pools, the convective coefficients are so high that boiling can be suppressed. Depending on oscillation amplitude –to –tube diameter ratios and vibration Reynolds number, the heat transfer coefficients increase up to 20 times compared with those of stationary tubes (Bergles, 1998).

In case of fluid vibrations, improvements of 100 to 200% over natural convection heat transfer coefficients in air were obtained by Sprott et al.,1960;Fand and Kaye,1961;and Lee and Richardson,1965;by generating intense sound fields and directing them transversely to a horizontal heated cylinder. Robinson et al.,(1958);Zhukauskas et al.,1961;Larson and London,1962;Fand ,1965;and Li and Parker,1967;reported 30 to 45% increase in free convection heat transfer by means of sonic and ultrasonic vibrations. As per Wong and Chon, 1969, ultrasonic vibrations do not promote any improvements in nucleate pool boiling; but they enhance vapor removal and tend to increase critical heat flux (CHF) by 50% as reported by Ornatskii and Shcherbakov, 1959. Mathewson and Smith (1963) investigated the effects of up to 176 dB acoustic field with frequencies in the range 50 to 330 Hz and found laminar film condensation coefficients for isopropanol to be enhanced by about 60% at low vapor flow rates.

2.11 COMPOUND ENHANCEMENT:

Some examples of compound enhancement techniques are:

- Corrugated (rough) tube with a hydrophobic coating (treated surface) to promote dropwise condensation of steam.
- Single phase mass transfer enhancement in grooved (finned) channel with flow pulsations and heat transfer in an acoustically excited flow field over a rough cylinder.
- Gas-solid suspension flows in an electric field.
- Surfactant additives for sea water evaporation in spirally corrugated or doubly fluted (rough surface) tubes.
- Application of Electro Hydro Dynamic (EHD) fields in pool boiling of refrigerants from micro finned and treated tubes.
- Use of rough surface along with swirl flow devices.

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CHAPTER 3

Present experimental work

3.1 PRESENT EXPERIMENTAL WORK

The experimental study on passive heat transfer augmentation using *twisted aluminium angles* and *twisted tapes* 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

3.2 About the inserts:

The insert used for the experiment are twisted aluminium angles and stainless steel twisted tapes. While much literature can be found about passive heat transfer augmentation using twisted tapes as mentioned earlier, twisted aluminium angles are a new kind of insert where no such experiments have been done thus giving us ample room for experimental studies.

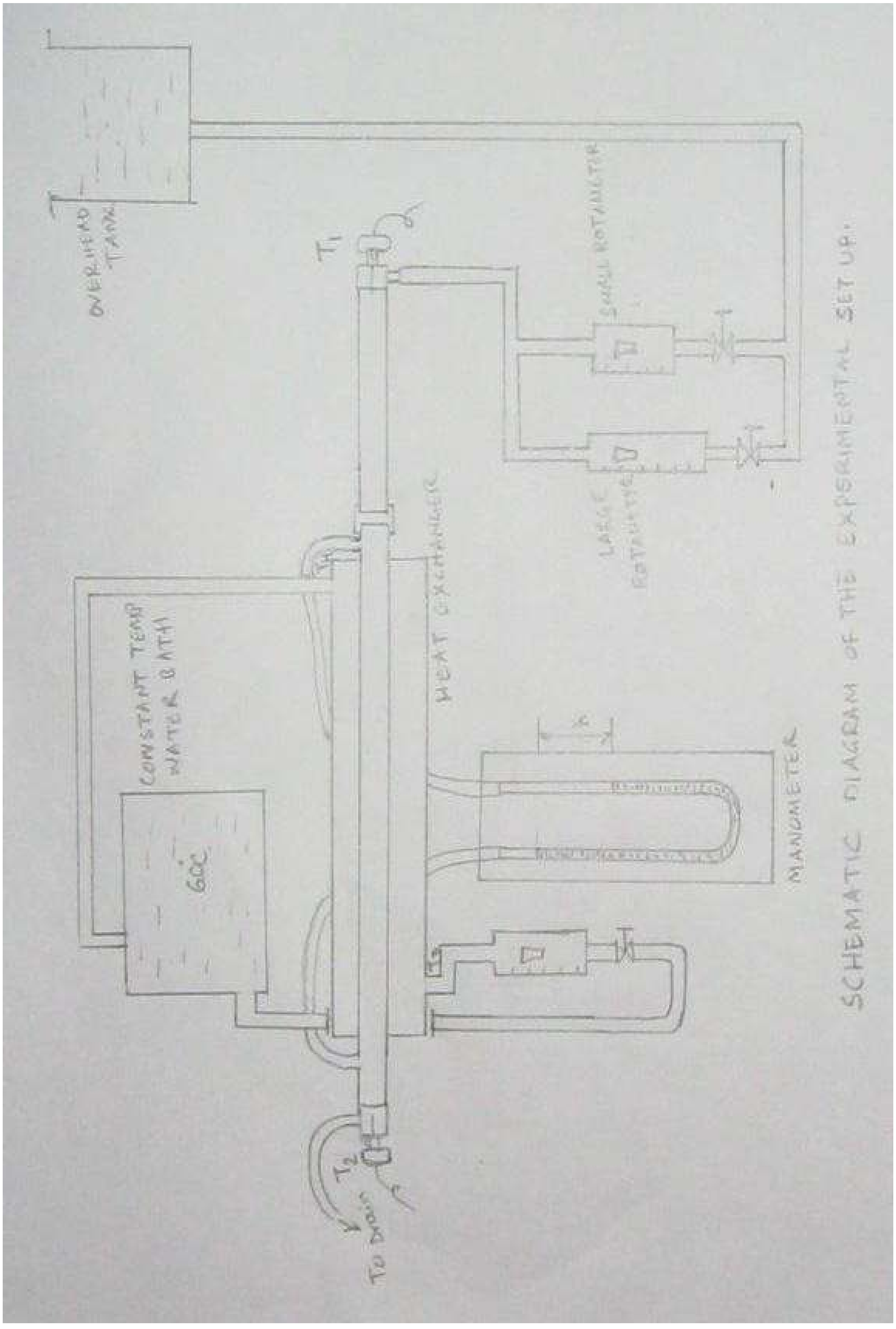
The present work deals with finding the friction factor and the heat transfer coefficient for the twisted angles with twist ratios ($y=\infty$, $y=2.915$, $y=3.612$, $y=4.105$ & $y=5.07$) and twisted tapes with twist ratios ($y=2.149$, $y=3.127$ & $y=4.705$) and comparing those results with that of smooth tube and among themselves.

3.3 Fabrication:

The aluminium angles of size 19mm, thickness about 1mm and length 366cm were first cut into 3 equal length (122cm) .About 2.5 cm on each side was clipped into a flat tape and a hole of about 5mm was drilled in it. Then the ends were tightened in the clamps and fixed on the lathe – one end being fixed on the tool part side and the other on the chuck side. The chuck was then rotated slowly by hand, while the angle was being held in tension, to give it a desired twist. Four angle tapes with varying twist ratios were fabricated as shown in fig 3.2 to fig 3.5. The end portions of the fabricated tapes were cut and drilled to join the tapes by thin high tension wires. Three tapes with the same twist ratio and twist in the same direction were joined to give a length of around 2.95 m, which was sufficient for the heat exchanger used for the experimental study.

The twisted tapes are shown in fig 3.6 to fig 3.8

The cross sectional view of the Al angle is shown in fig 3.9.


OVERHEAD TANK
SMALL ROTAMETER
LARGE ROTAMETER
T_1
HEAT EXCHANGER
CONSTANT TEMP WATER BATH
60°C
x
MANOMETER
T_2
To Drain
SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET UP.

Experimental Set Up

Fig. 3.1: Plain angle, $y = \infty$

Fig. 3.2: Twisted Angle, y = 5.07

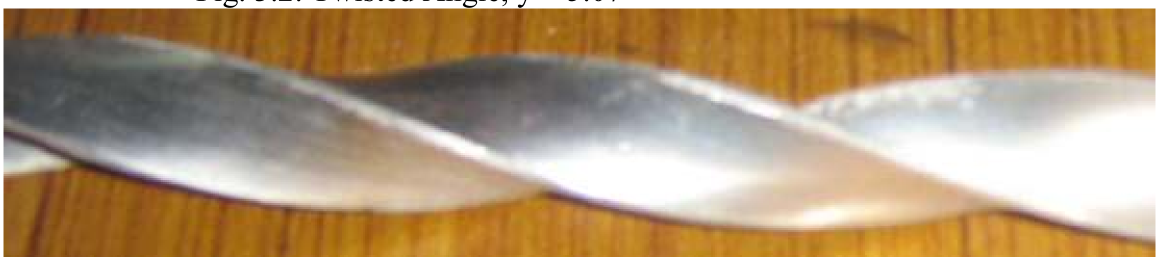


Fig. 3.3: Twisted Angle, y = 4.105

Fig. 3.4: Twisted Angle, y = 3.612

Fig. 3.5: Twisted Angle, y = 2.915

Fig. 3.6: Twisted Tape, y = 4.705

Fig. 3.7: Twisted Tape, y = 3.127

Fig. 3.8: Twisted Tape, y = 2.149

Fig. 3.9: Cross-Sectional View of Plain Angle

3.4 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. Twist Ratio(y) of the inserts were calculated.

   Twist Ratio, $y = H/d_i$

   d_i = Inside diameter of the tube, m

   H = Linear distance of the tape for 180° rotation, m.
4. For friction factor determination:
   a. The manometer used contained CCl_4 as the manometric liquid and a little of bromine crystals were 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:-

(i) Re< 2100

$$f = 16/Re$$

(ii) Re>2100

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

5. 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:-

(i) For Re<2100

Nu = f(Gz)

For Gz < 100

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

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

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

(iii) Re>10000(DITTUS BOILTER EQUATION)

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

b. Water to be flown throw the annulus side was first heated to a set temperature of 60°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 (1000kg/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

4.1ROTAMETER CALIBRATION

4.1.1 LARGE ROTAMETER (Table No.A1.1)

For SL.NO-4

Observation1: Wt= 10.4(kg)
Time= 75 (secs)
m1 = 0.1387(kg/s)

Observation2: Wt = 10.7(kg)
Time = 78(secs)
m2 = 0.1372(kg/s)

Observation3: Wt = 10.4(kg)
Time = 76(secs)
m3 = 0.1368(kg/s)

m_{avg} = (m1 + m2 + m3)/3 = 0.1376 kg/s

Twist Ratio Calculation: (for y = 3.127)

Length of Twisted Tape, L = 289cm
Number of 180° turns ,N = 42
H = L/N = 289/42 = 6.88
d_i = 2.2cm
y = H/d_i = 6.88/2.2 = 3.127

4.2 PRESSURE DROP AND FRICTION FACTOR CALCULATIONS:-

For Y=3.127 (Twisted tape) (Table No A2.3)
SL.NO. -9
m = 0.1376 kg/s
h= 0.194 m

$A_c = \pi/4 \times d_i^2 = 3.8 \times 10^{-4}\ m^2$

$V = m/(A_c \times \rho_w) = 0.3620$ m/s

$\Delta P = (\rho_{ccl4} - \rho_w) \times g \times h = (1603 - 1000) \times 9.81 \times 0.194 = 1147.59$

$f_a = \Delta P \times d_i / (2 \times \rho \times L_p \times V^2) = 1147.59 \times 0.022 / (2 \times 1000 \times 2.825 \times 0.3620^2)$
$= 0.0341$

μ = 0.85 cP

$Re = 4 \times m / (\pi \times d_i \times \mu) = 9368$

$f_o = 0.046 \times Re^{-0.2} = 0.0074$

$f_a/f_o = 0.0341 / 0.0074 = 4.62$

4.3 Heat Transfer Coefficient Calculation

For Y=3.127 (Twisted tape) (Table No A2.3)

SL.NO. – 4

m_1 = 0.2806 kg/s (hot water)

m_2 = 0.1376 kg/s (cold water)

	Actual Temp	Corrected Temp
T1 = 30.6°C	30.1	30.6
T2 = 35.2°C	35.2	35.2
T3 = 49.4°C	49.4	49.4
T4 = 47.1°C	47.0	47.1

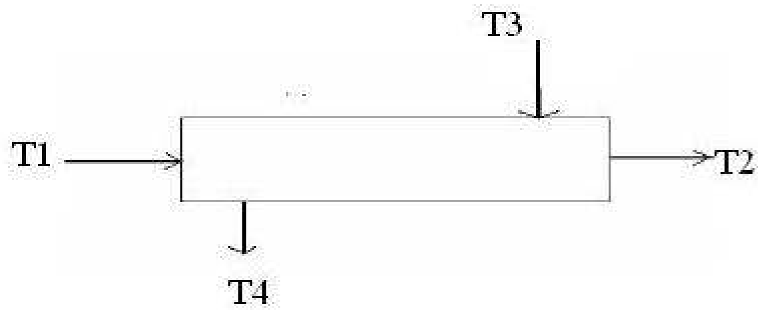


Fig. 4.1: Basic Representation of Counter Current Flow

$Q_1 = m_1 \times C_p \times (T3 – T4) = 0.2806 \times 4187 \times (49.4 – 47.1) = 2702.21$ Watt

$Q_2 = m_2 \times C_p \times (T2 – T1) = 0.1376 \times 4187 \times (35.2 – 30.6) = 2650.20$ Watt

$Q_{avg} = (Q_1 + Q_2)/2 = 2676.20$ Watt

$\%diff = (Q_1 - Q_2) \times 100/ Q_{avg} = 1.94$

T4 – T1 = 47.1 – 30.6 = 16.5°C

T3 – T2 = 49.4 – 35.2 = 14.2°C

L.M.T.D. = {(T4 – T1) – (T3 – T2)}/ ln {(T4 – T1) / (T3 – T2)} = 15.32°C

$A_i = \pi \times d_i \times L_h = \pi \times 0.022 \times 2.43 = 0.168$ m^2

$U_i = Q/(A_i \times LMTD) = 2676.20/(0.168 \times 15.32) = 1040$ W/ m^2/°C

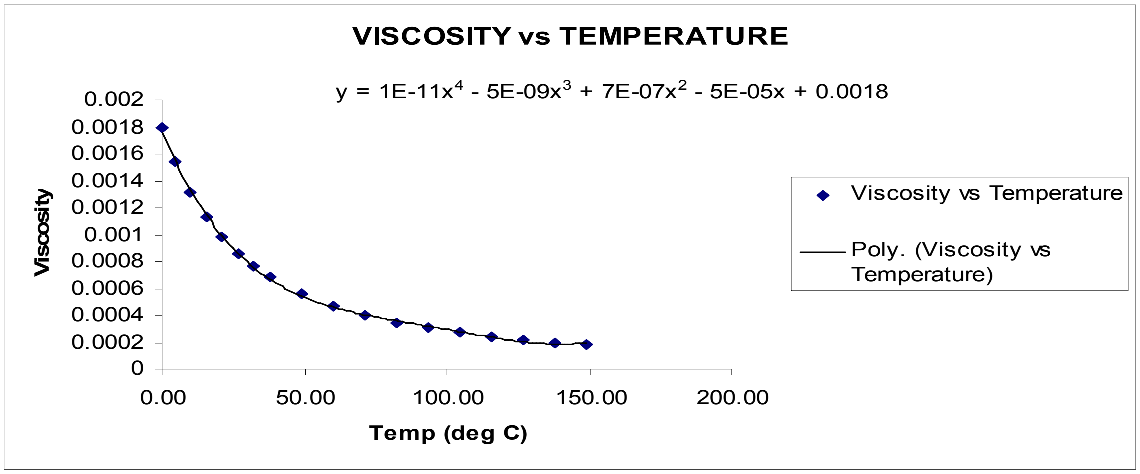


Fig. 4.2: Relationship between Viscosity & Temperature

$\mu = 10^{-11} \times T^4 - 5 \times 10^{-09} \times T^3 + 7 \times 10^{-07} \times T^2 - 5 \times 10^{-05} \times T + 0.0018 = 0.000746$

where, $T = (T1 + T2)/2 = 32.90°C$

$Re = 4 \times m / (\pi \times d_i \times \mu) = 10672$

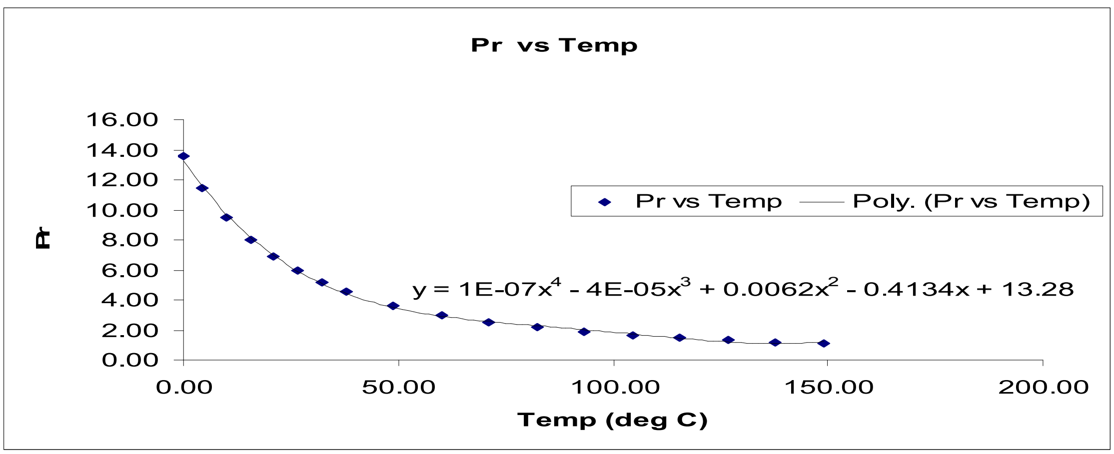


Fig. 4.3: Relationship between Pr & Temperature

$Pr = 10^{-07} \times T^4 - 4 \times 10^{-05} \times T^3 + 0.0062 \times T^2 - 0.4134T + 13.28 = 5.0828$

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

Which reduces to

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

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

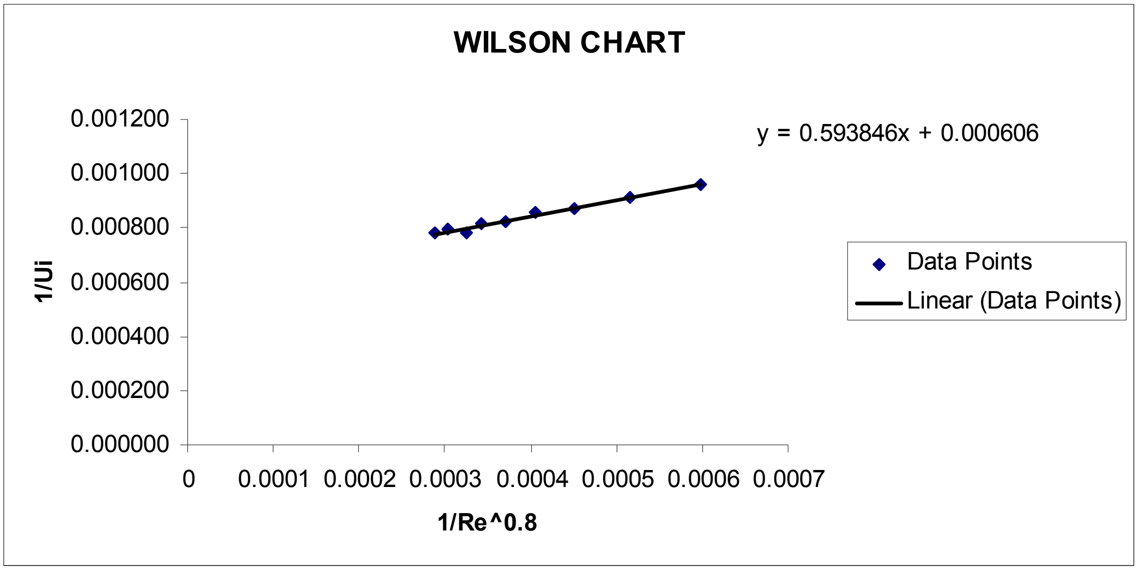


Fig. 4.4: Wilson chart for twisted tape insert (y=3.127, Table No:3.3)

K=0.000606 m °C/W

$1/h_a = (1/U_i) - K = (1/1040) - 0.000606 = 0.0003556$

$h_a = 2811$ W/m^2/°C

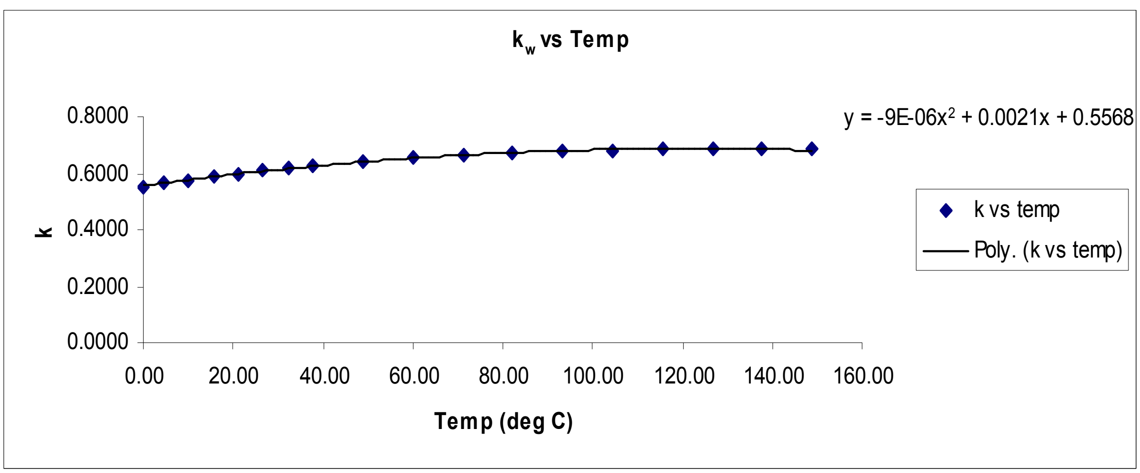


Fig. 4.5: Relationship between k_w & Temperature

$k_w = -9 \times 10^{-6} \times T^2 + .0021\ T + .5568 = 0.6161$ W/m/°C (at T = 32.9°C)

$h_o = 0.023 \times Re^{0.8} \times Pr^{0.4} \times (k/d_i) = 2061$ W/m^2/°C

$R1 = h_a / h_o = 2811/2061 = 1.36$

CHAPTER 5

Results and discussion

5.1 friction factor results:

In Fig no.5.1.1 it was observed that for Re> 3400 i.e., in turbulent region f is nearly equal to f_o. For very low Re (Re < 3400) the observed height difference in the manometer was very small, resulting in a larger deviation. The friction factor relation used was

$$f = 16/Re \text{ (For Re<2100)}$$

$$f = 0.046 \times Re^{-0.2} \text{ (For Re>2100)}$$

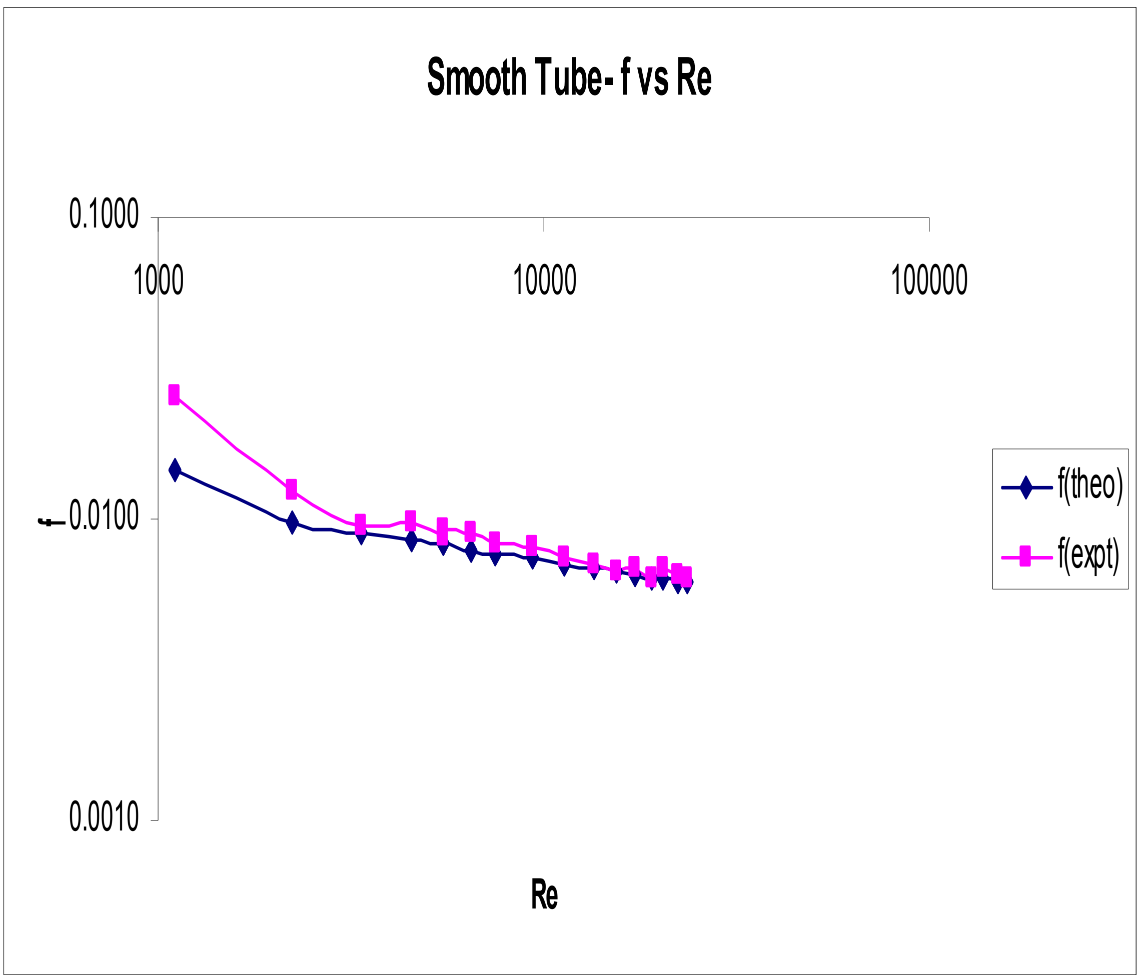


Fig 5.1.1

In Fig 5.1.2 & Fig. 5.1.3 it is observed that friction factor increases as the degree of twist in the twisted tapes increases which is indicated by a lower value of twist ratio. Hence the pressure drop increases accordingly i.e,

f(smooth) < f(y = 4.705) < f(y = 3.127) < f(y = 2.149).

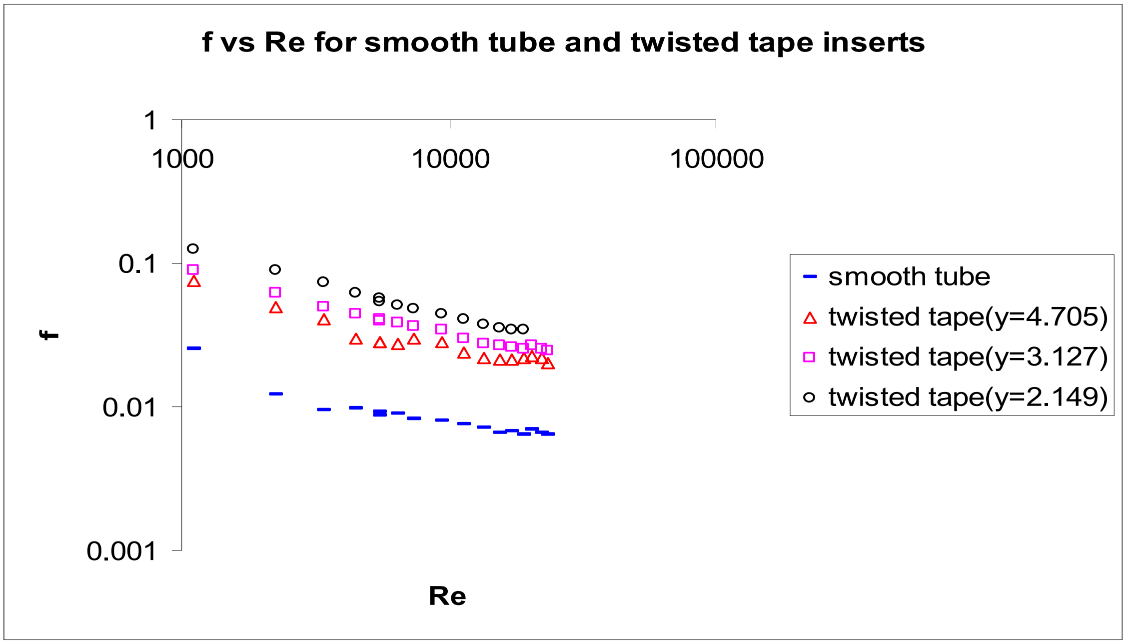


Fig 5.1.2

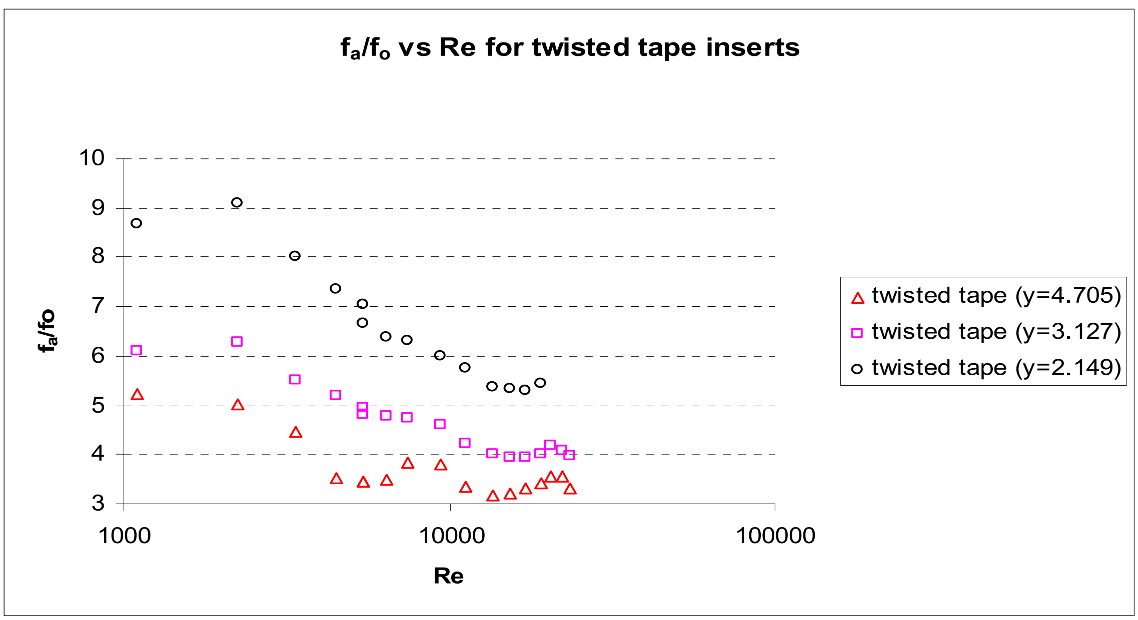


Fig 5.1.3

In Fig 5.1.4 & Fig. 5.1.5 it is also observed that friction factor increases as the degree of twist in the twisted angles increases which is indicated by a lower value of twist ratio. Hence the pressure drop increases accordingly i.e.,

f(smooth) < f(y = 0) < f(y = 5.07) < f(y = 4.105) < f(y = 3.612) < f(y = 2.915).

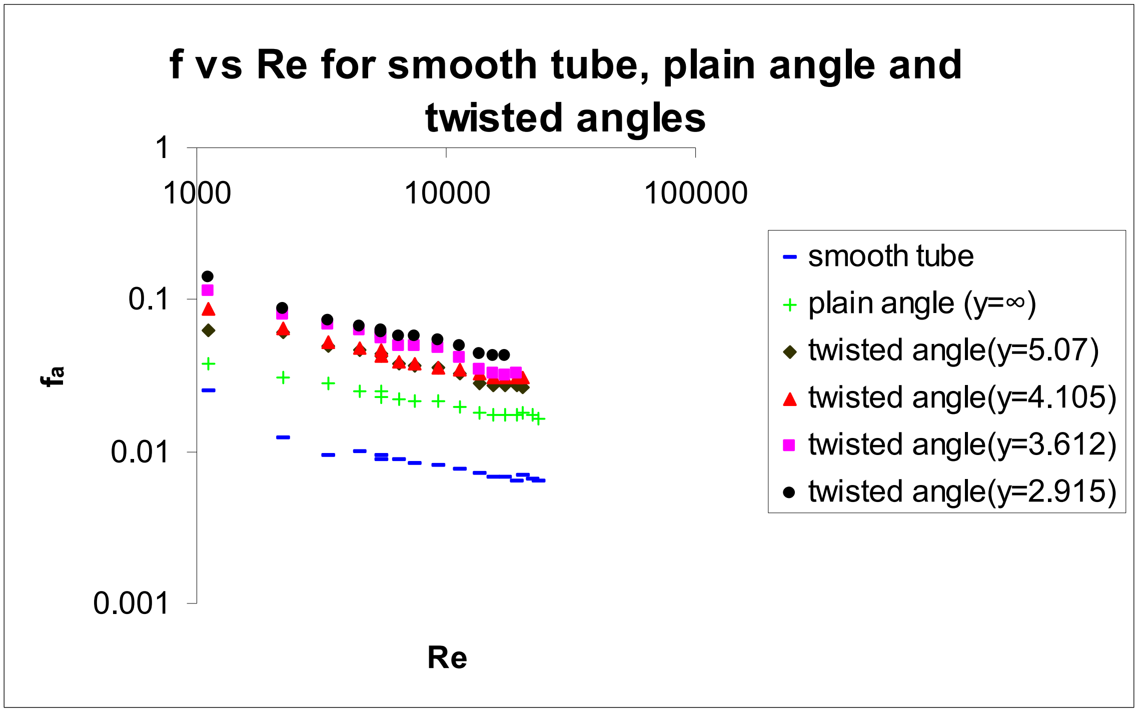


Fig 5.1.4

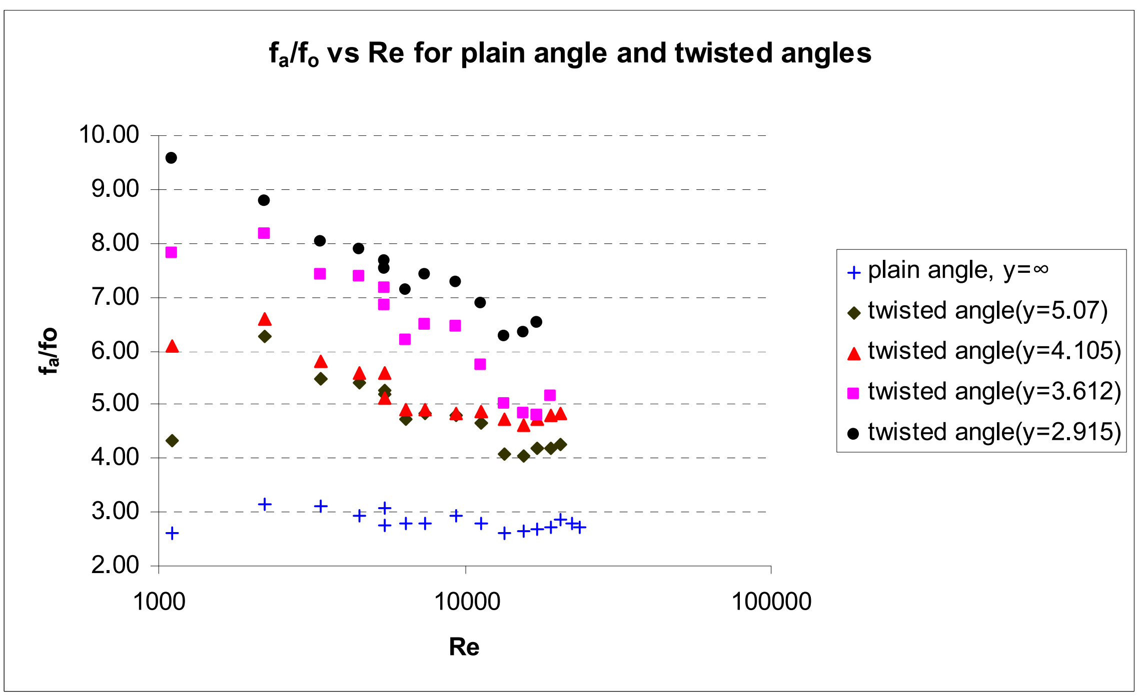


Fig 5.1.5

Fig. 5.1.6 & Fig. 5.1.7 shows the plot of f vs Re and f_a/f_o for all the inserts used in the experiment satisfying the above discussed trend. It is noticed that for twisted angles and twisted tapes having almost the same twist ratio, the twisted angles show a greater increase in the friction factor for the same Re. f_a/f_o goes on increasing as the degree of twist increases.

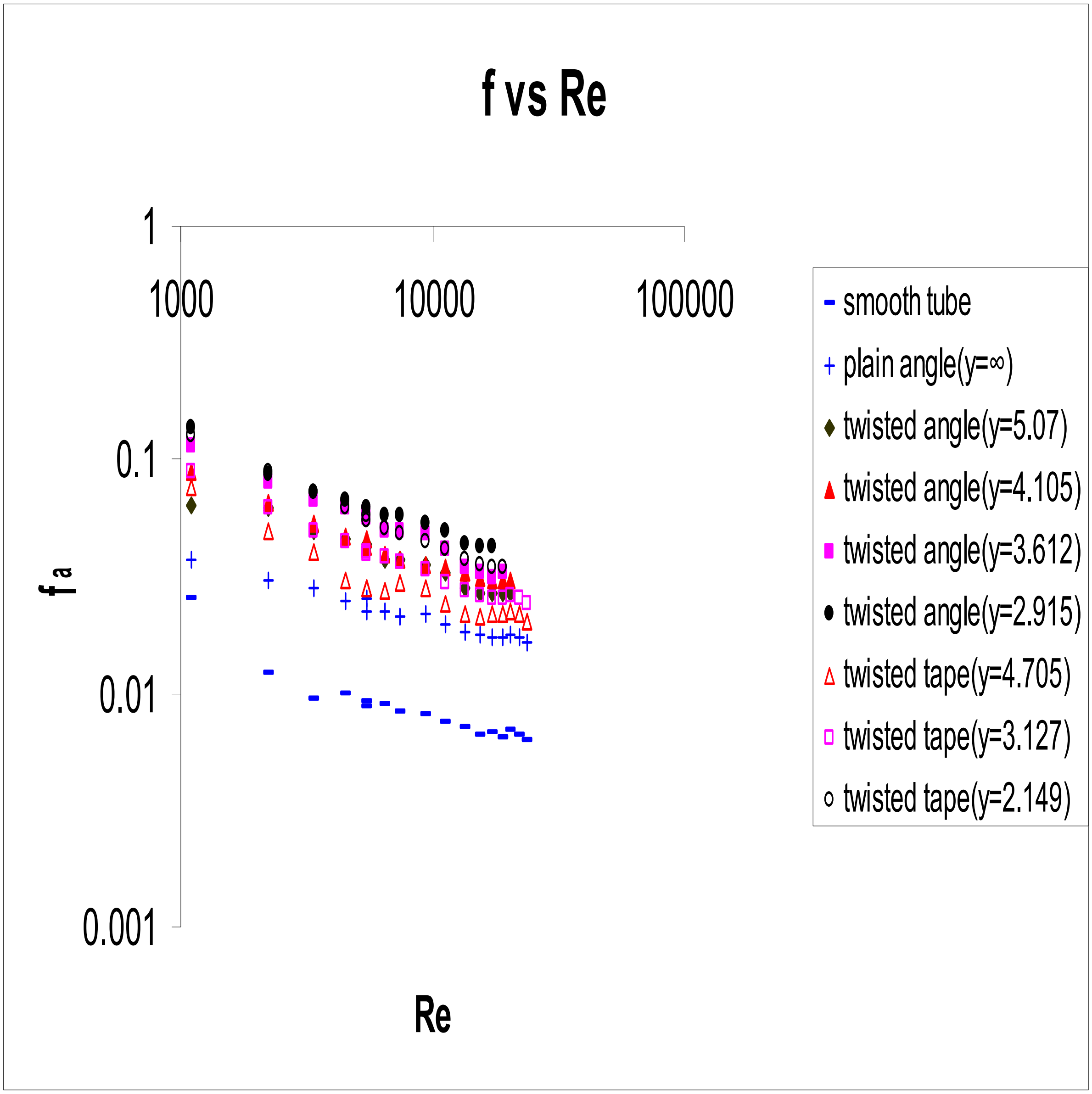
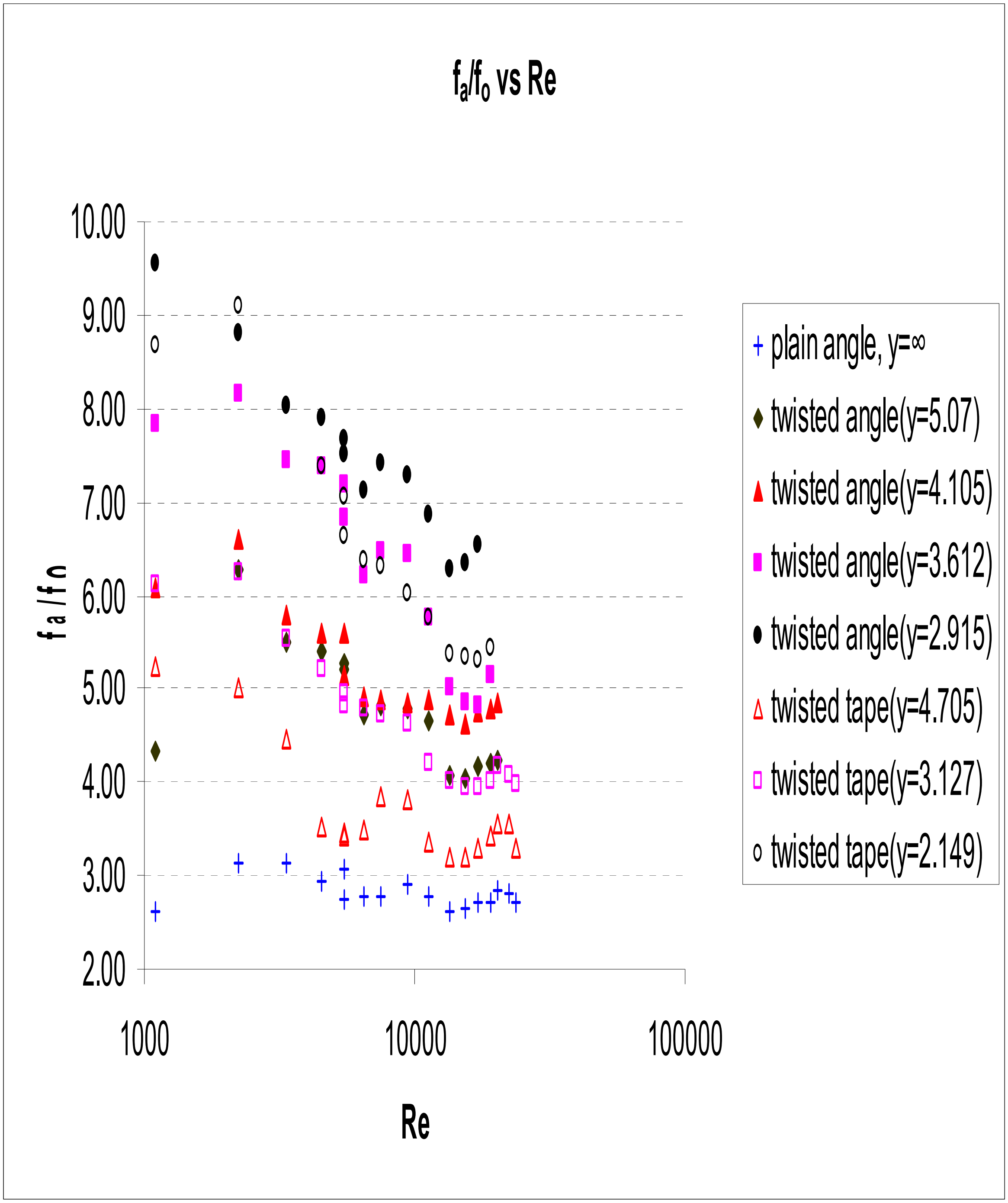


Fig 5.1.6



fa/fo vs Re
10.00
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
f a / f o
1000
10000
100000
Re
plain angle, y=∞
twisted angle(y=5.07)
twisted angle(y=4.105)
twisted angle(y=3.612)
twisted angle(y=2.915)
twisted tape(y=4.705)
twisted tape(y=3.127)
twisted tape(y=2.149)

Fig 5.1.7

5.2 Heat transfer coefficient results

From Fig. 5.2.1 it is observed that h_i (expt) and hi (theo) are almost the same in turbulent region. It is also found that from Fig. 5.2.2, U_i increases with an increase in Re.

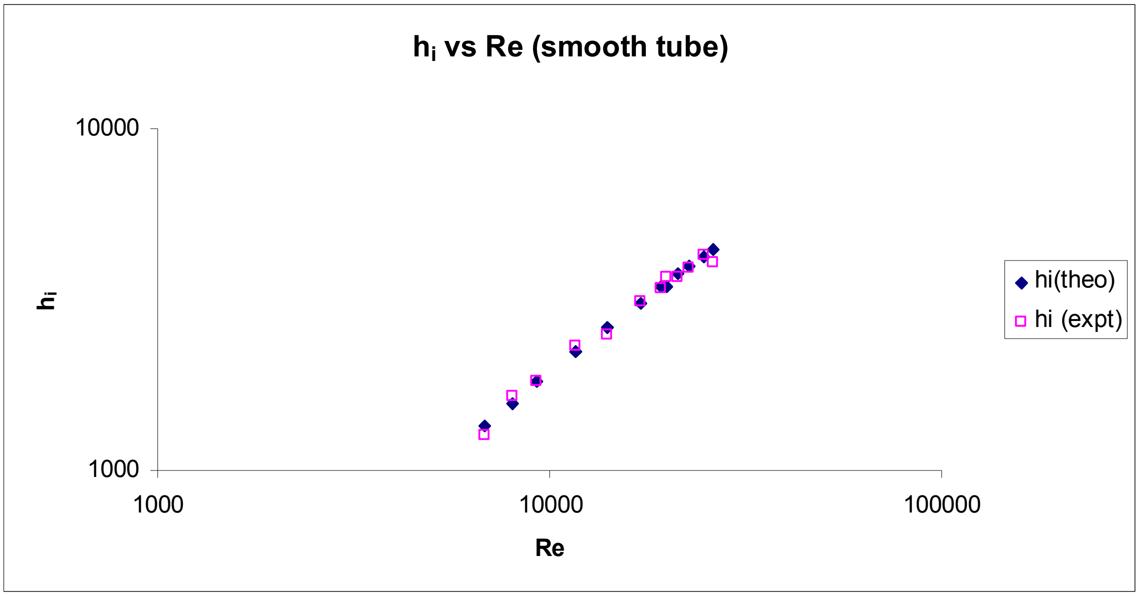


Fig 5.2.1

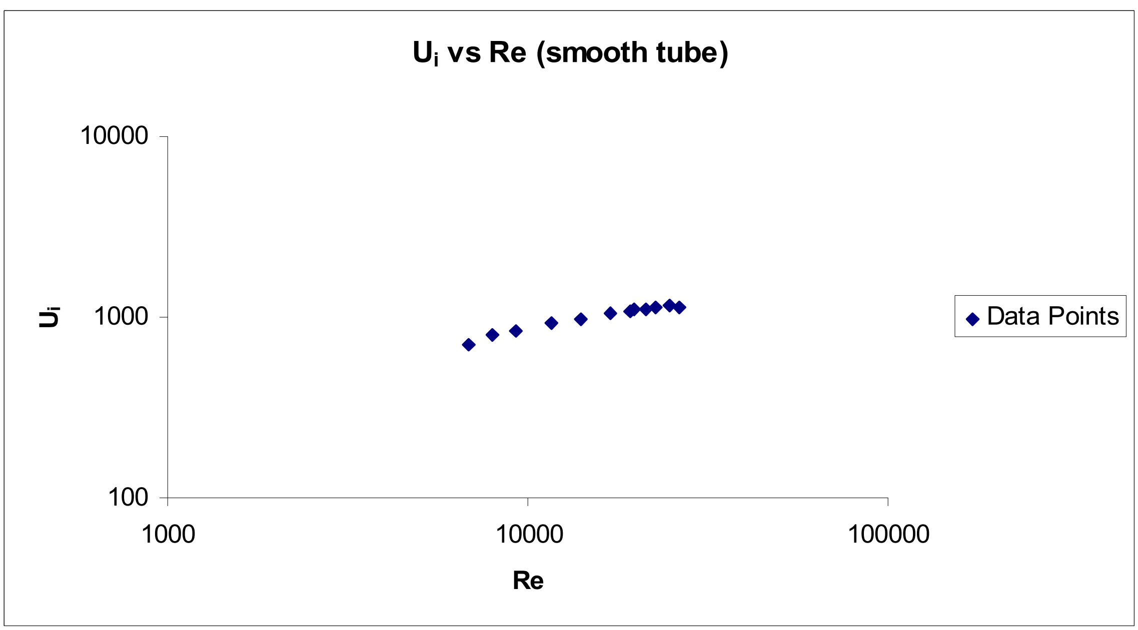


Fig 5.2.2

In Fig no. 5.2.3 & Fig no 5.2.5 the heat transfer coefficient(film and overall) increases as the twist in the inserts increases (i.e. with decreasing twist ratio).

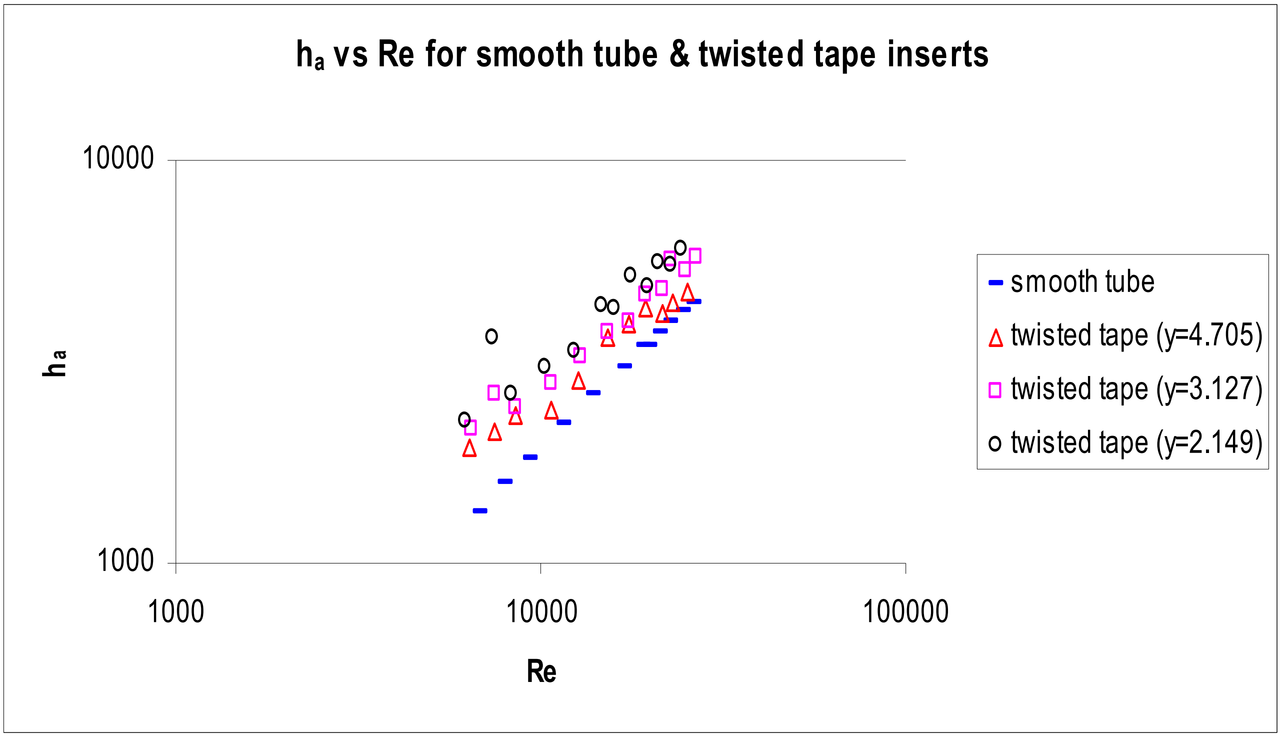


Fig 5.2.3

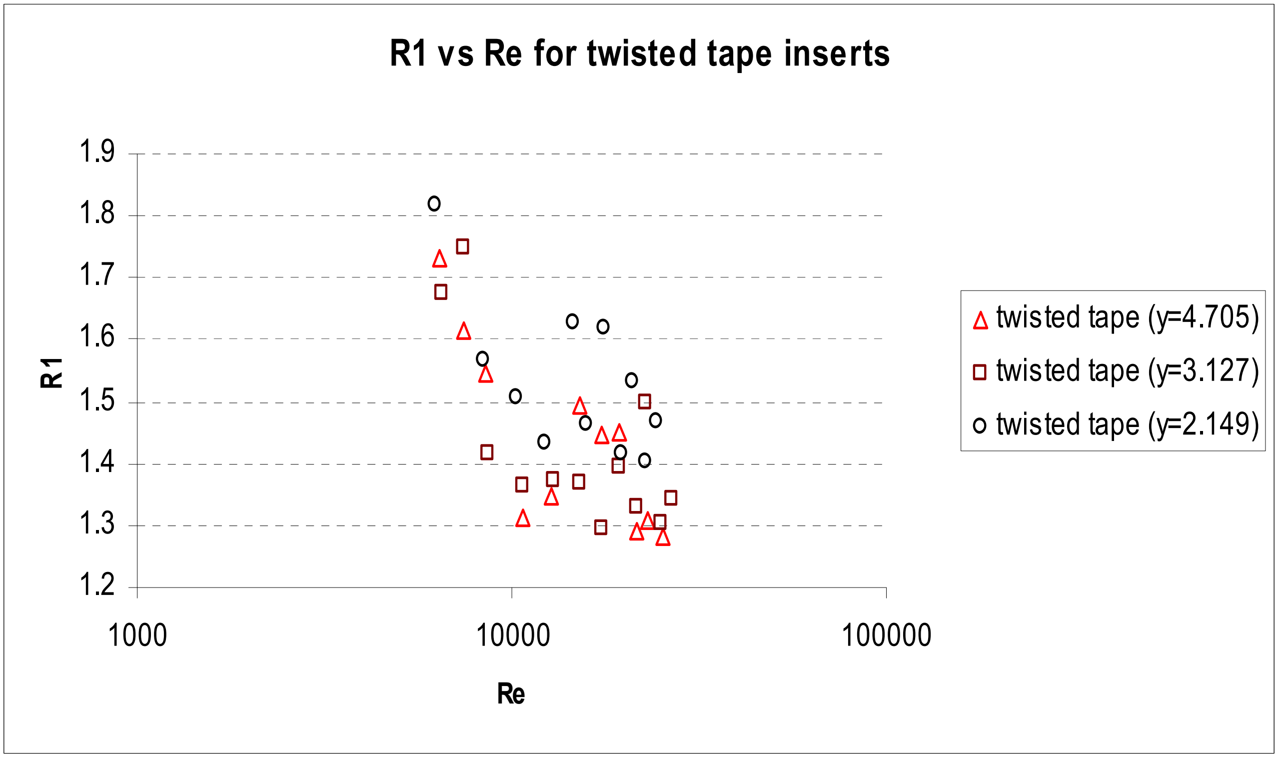
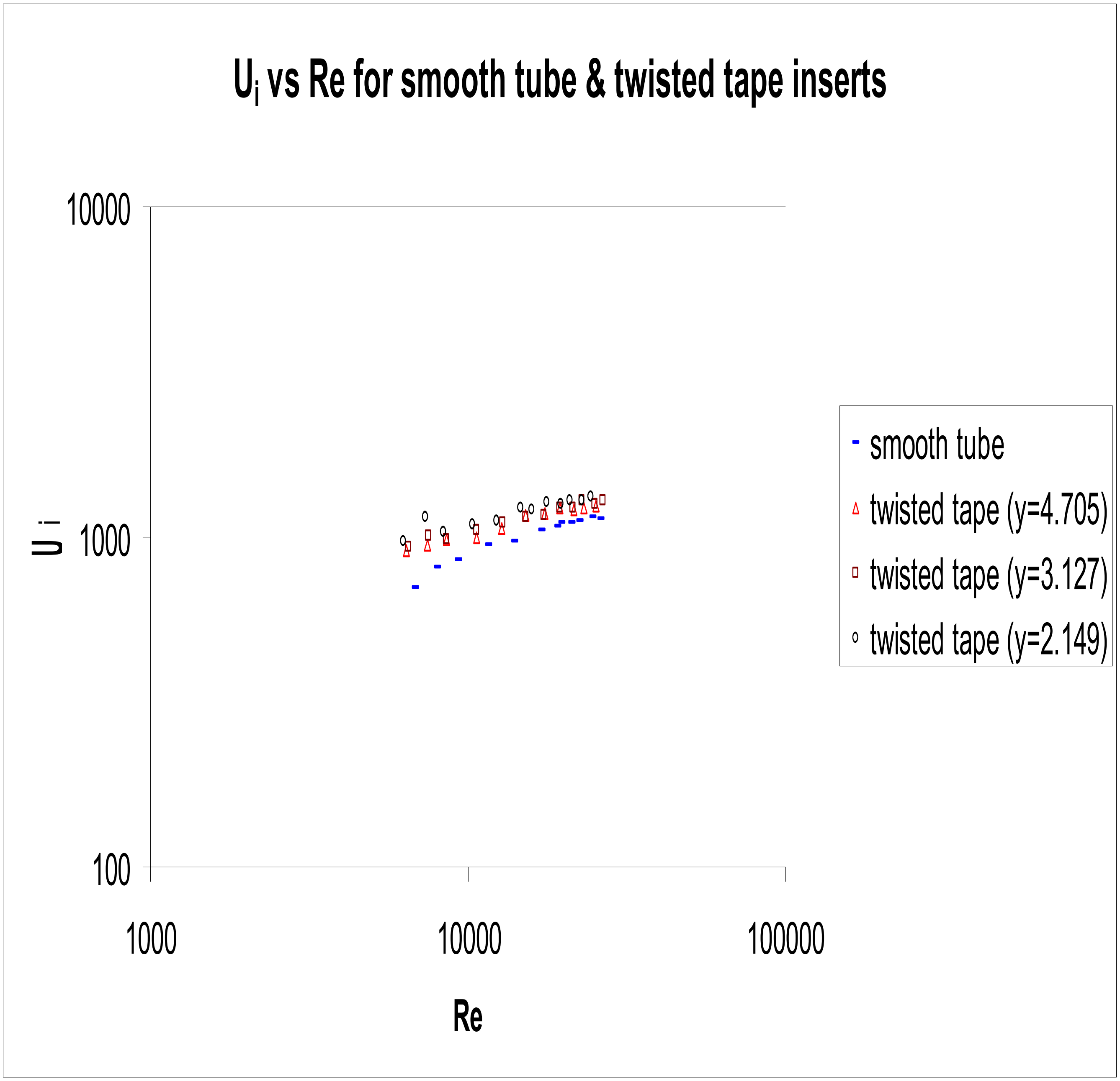


Fig 5.2.4



U_i vs Re for smooth tube & twisted tape inserts
10000
1000
100
U i
1000
10000
100000
Re
smooth tube
twisted tape (y=4.705)
twisted tape (y=3.127)
twisted tape (y=2.149)

Fig 5.2.5

In Fig no. 5.2.6 & Fig no 5.2.7 the heat transfer coefficient increases as the twist in the inserts increases (i.e. with decreasing twist ratio) as found in the case of twisted tapes

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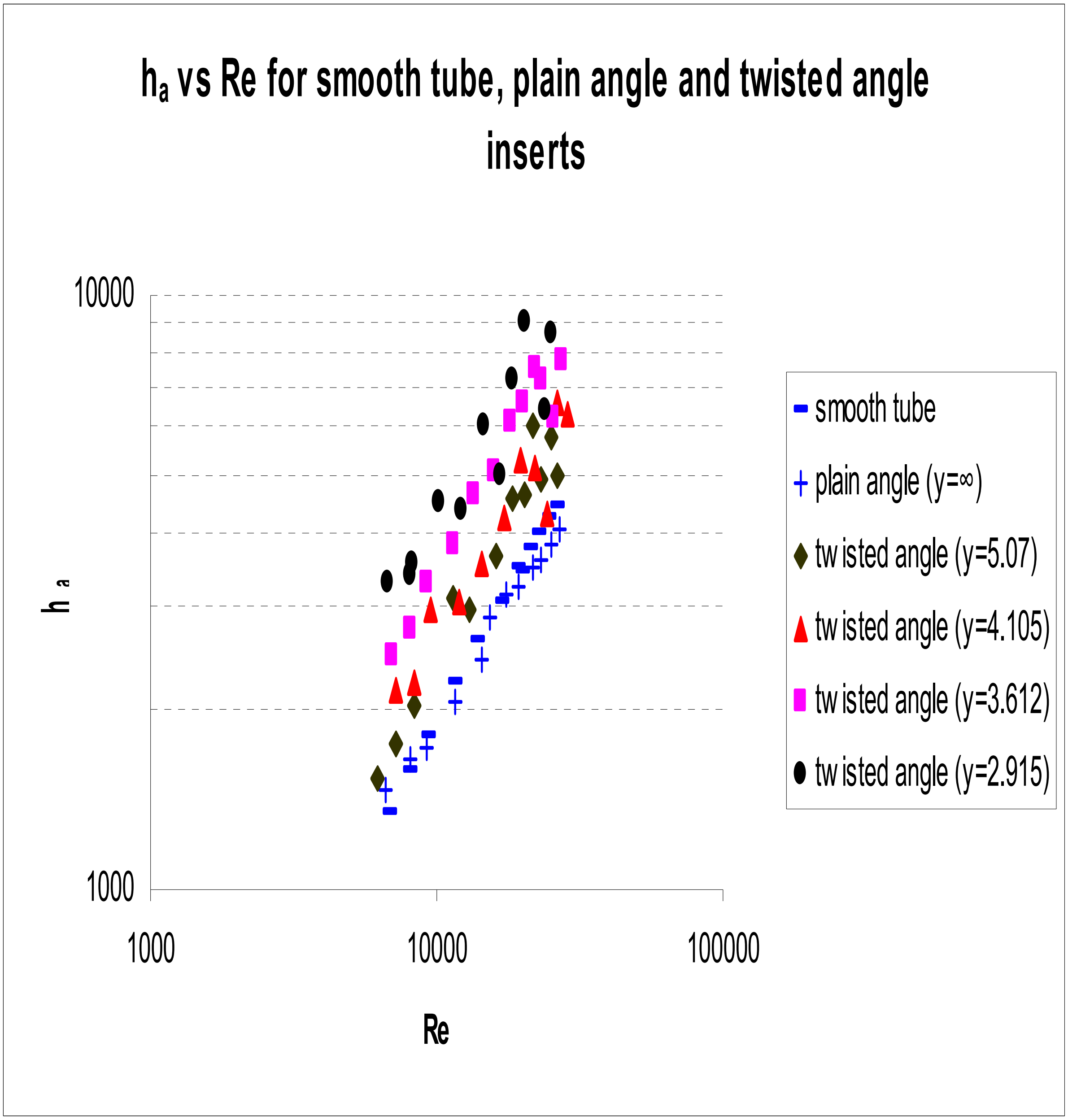

Fig 5.2.6

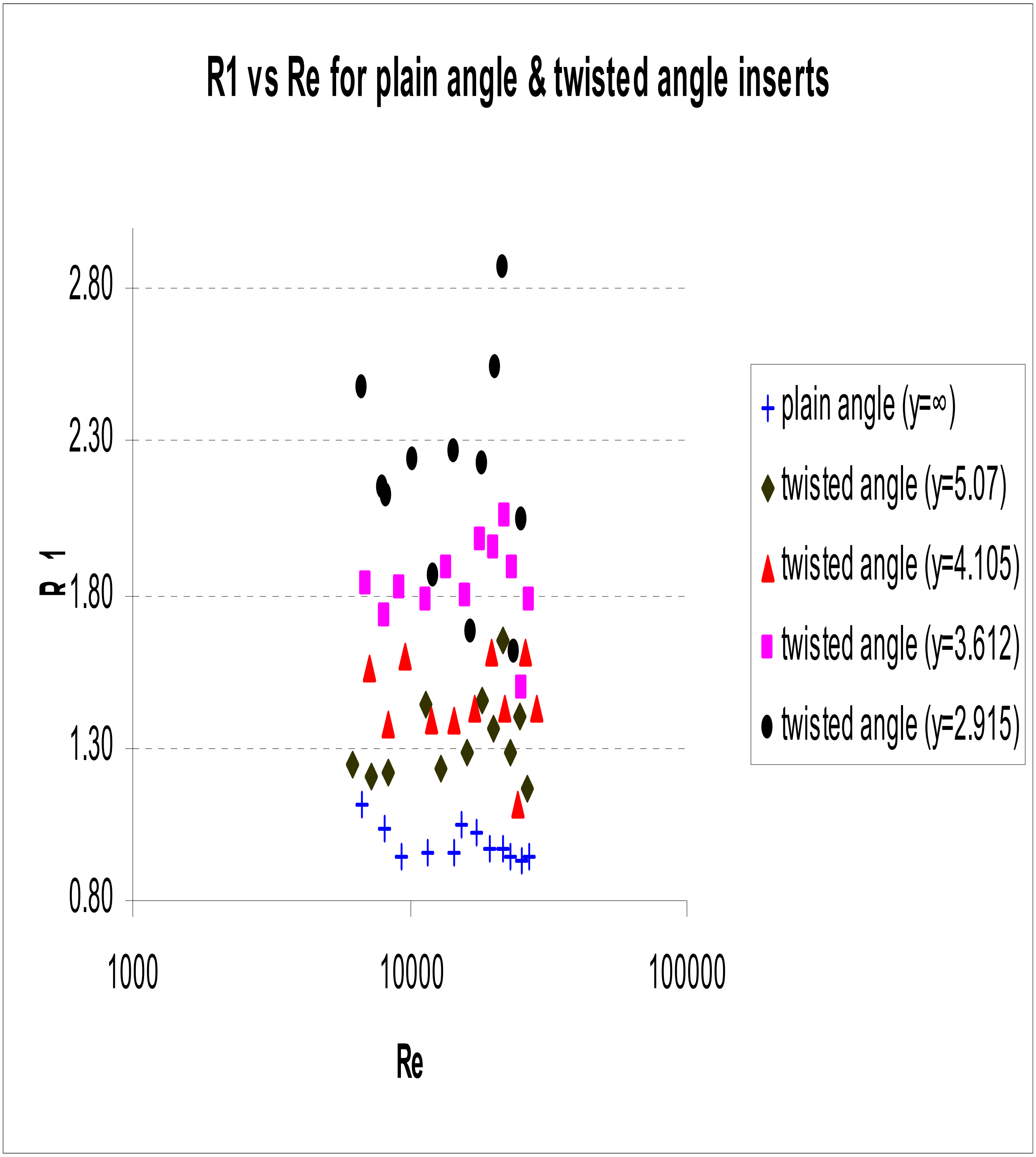
R1 vs Re for plain angle & twisted angle inserts
R 1
2.80
2.30
1.80
1.30
0.80
1000
10000
100000
Re
plain angle (y=∞)
twisted angle (y=5.07)
twisted angle (y=4.105)
twisted angle (y=3.612)
twisted angle (y=2.915)

Fig 5.2.7

From Fig. 5.2.8 it is observed that for all the inserts the heat transfer coefficient increases with an increase in twist ratio and for twisted angles the increase in heat transfer coefficient is grater than that of twisted tapes with similar twist ratio.

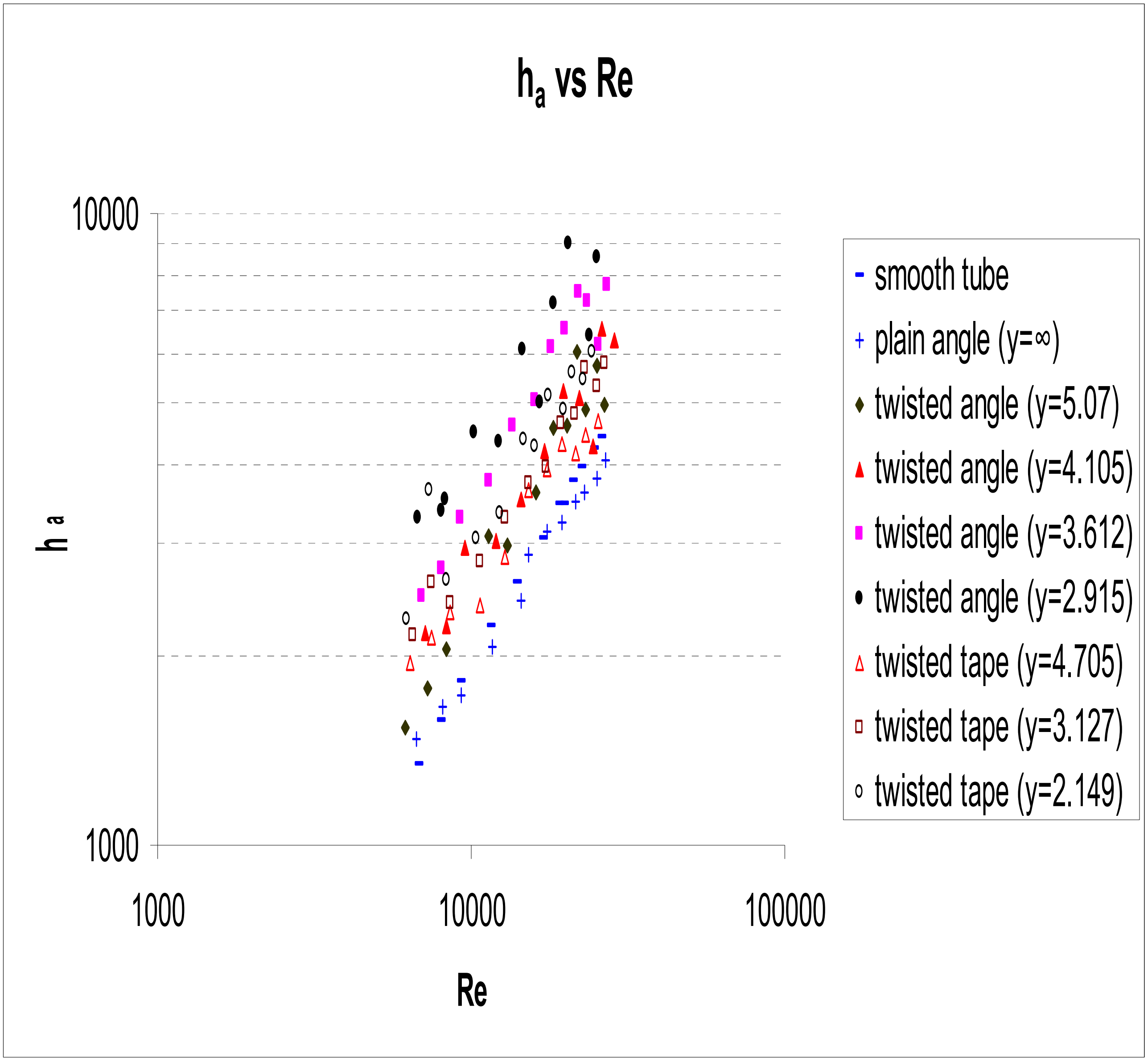

Fig 5.2.8

In Fig 5.2.9 it is observed that for same twist ratios of twisted angles and twisted tape, twisted angle tape gave better heat transfer enhancement than twisted tape.

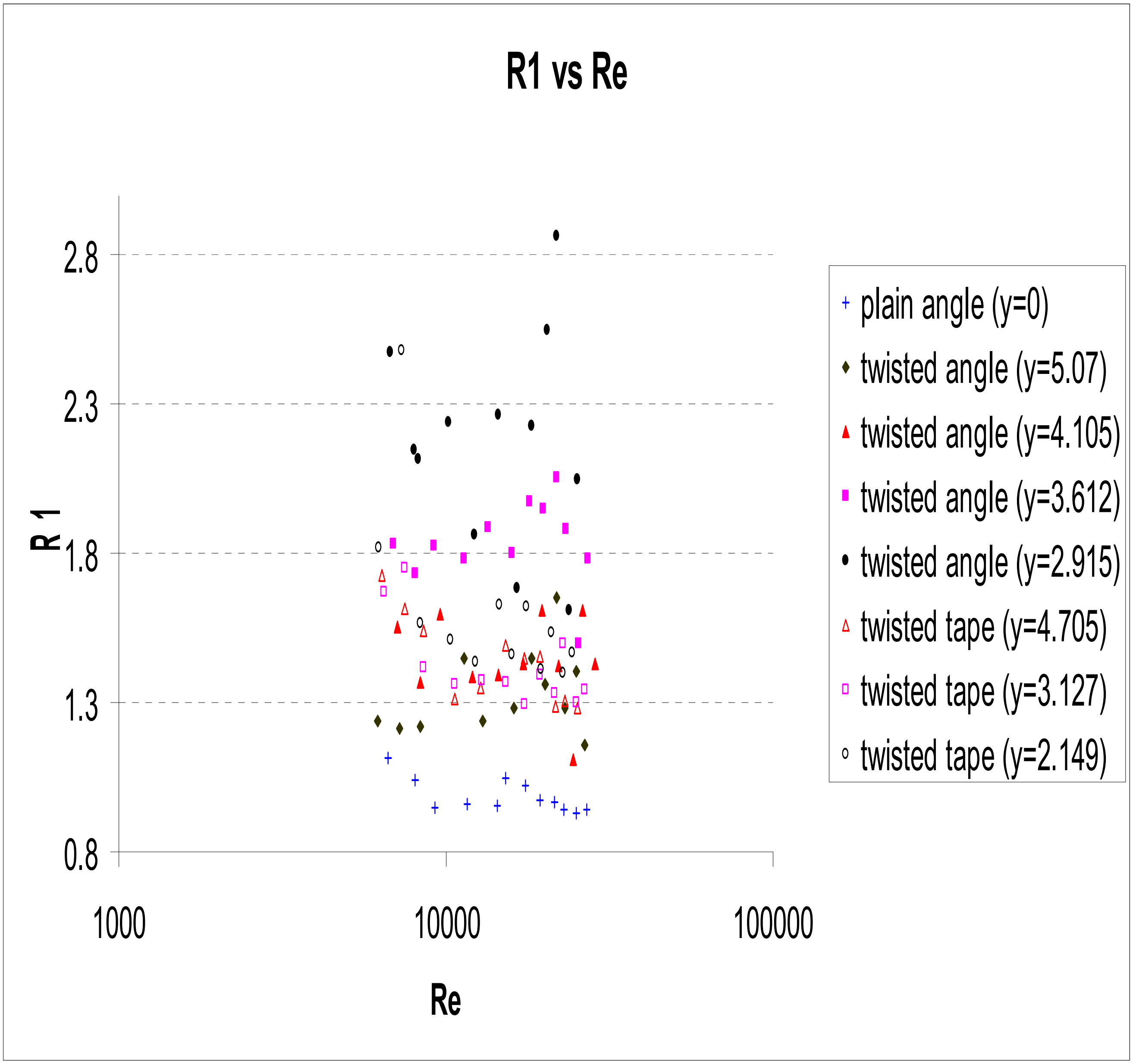

Fig. 5.2.9

CHAPTER 6

Conclusion

Conclusion:-

1. The pressure drop and the heat transfer coefficient increase as the degree of twist in the tapes and angles goes on increasing.
2. For almost the same twist ratio, twisted angles 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 twisted angle and twisted tapes are shown in the table 6.1 and table 6.2 respectively.

Table 6.1: For Twisted Angles

Sl.No.	**Y**	**Range of f_a/f_o**	**Range of R1**
1	∞ (plain angle)	2.61 to 3.14	0.93 to 1.11
2	5.07	4.04 to 6.28	1.16 to 1.65
3	4.105	4.62 to 6.59	1.11 to 1.61
4	3.612	4.80 to 8.16	1.50 to 2.05
5	2.915	6.27 to 9.56	1.61 to 2.87

For twisted angles it was observed that the heat transfer coefficients varied from 1.16 to 2.87 times the smooth tube value. But this increase is at the cost of increased frictional losses. The corresponding friction factor varied from 4 to 9.56 times the smooth tube values.

Table 6.2: For Twisted Tapes

Sl.No.	**Y**	**Range of f_a/f_o**	**Range of R1**
1	4.705	3.19 to 5.23	1.28 to 1.73
2	3.127	3.93 to 6.26	1.30 to 1.75
3	2.149	5.29 to 9.10	1.40 to 2.48

In the case of twisted tapes the heat transfer coefficients varied from 1.28 to 2.48 times the smooth tube value and the corresponding friction factor varied from 3.19 to 9.10 times.
We observe that with an increase in the Reynolds number (Re) the heat transfer coefficients increases for both twisted angles and twisted tapes while the friction factor decreases.
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.
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 twisted angles gives better performance as compared to the twisted tape having the same twist ratio.

NOMENCLATURE:--

A_i	Inside heat transfer surface area, m^2
A_c	Cross sectional area, m^2
C_p	Specific heat of fluid, J/kg.k
d_i	Inside diameter of the tube, m
f	Fanning friction factor, dimensionless
f_a	Fanning friction factor for augmented tube, dimensionless
f_o	Theoretical Fanning friction factor for smooth tube, dimensionless
Gz	Graetz number, dimensionless
h	Difference in level of CCl_4 in the manometer, m
h_i	Inside htc, W / m^2. ºC
h_i (expt)	Experimental inside htc, W / m^2. ºC
h_i (theo)	Theoretical inside htc, W / m^2. ºC
h_o	Heat transfer coefficient for smooth tube, W / m^2. ºC
ha	Augmented value of heat transfer coefficient, W / m^2. ºC
H	Linear distance of the tape for 180° rotation, m.
k_w	Thermal conductivity of the tube wall, w/ m .°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 (ratio of augmented value of heat transfer coefficient to smooth tube heat transfer coefficient i.e, h_a/h_o), dimensionless
T	Temperature in °C
U_i	Overall htc based on the inside surface area, W / m^2. ºC
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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Appendix

1. CALIBRATION
1.1 SMALL ROTAMETER

	Observation1			Observation2			Observation3			
Rotameter Reading (lpm)	**Wt (kg)**	**Time (sec)**	**m (kg/s)**	**Wt (kg)**	**Time (sec)**	**m (kg/s)**	**Wt. (kg)**	**Time (sec)**	**m (kg/s)**	**Average m**
1	12	749	0.0160	10.7	655	0.0163	10.4	635	0.0164	0.0162
2	10.4	317	0.0328	10.6	322	0.0329	10.3	313	0.0329	0.0329
3	10.2	205	0.0498	10.2	205	0.0498	10.4	211	0.0493	0.0496
4	10.2	154	0.0662	10.2	155	0.0658	10.5	156	0.0673	0.0664
5	11.6	145	0.0800	10.0	124	0.0806	10.6	132	0.0803	0.0803

1.2 LARGE ROTAMETER

	Observation1			Observation2			Observation3			
Rotameter Reading (kg/hr)	**Wt (kg)**	**Time (sec)**	**m (kg/s)**	**Wt (kg)**	**Time (sec)**	**m (kg/s)**	**Wt. (kg)**	**Time (sec)**	**m (kg/s)**	**Average m**
300	10.5	130	0.0808	10.2	127	0.0803	10.2	127	0.0803	0.0805
350	10.4	111	0.0937	10.4	109	0.0954	10.2	107	0.0953	0.0948
400	10.3	94	0.1096	10.3	93	0.1107	10.5	97	0.1082	0.1095
500	10.4	75	0.1387	10.7	78	0.1372	10.4	76	0.1368	0.1376
600	10.4	63	0.1651	10.6	64	0.1656	10.7	64	0.1672	0.1660
700	10.6	53	0.2000	10.4	53	0.1962	10.6	53	0.2000	0.1987
800	10.6	47	0.2255	10.8	46	0.2348	10.4	47	0.2213	0.2272
900	10.5	41	0.2561	10.6	42	0.2524	10.8	43	0.2512	0.2532
1000	10.7	39	0.2744	10.5	37	0.2838	10.5	37	0.2838	0.2806
1100	10.8	36	0.3000	10.7	37	0.2892	10.5	34	0.3088	0.2993
1200	10.6	33	0.3212	10.3	31	0.3323	10.8	33	0.3273	0.3269
1250	10.5	30	0.3500	10.7	31	0.3452	10.4	30	0.3467	0.3473

1.3 RTD CALIBRATION

TEMPERATURE	ACTUAL TEMP	CORRECTION	CORRECTED TEMP
T1	19.4	+0.5	19.9
T2	19.9	0.0	19.9
T3	19.9	0.0	19.9
T4	19.8	+0.1	19.9

2. FRICTION FACTOR RESULTS

2.1 STANDARDISATION – SMOOTH TUBE (f vs Re)

Sl. No.	m(kg/s)	h(cm)	$\Delta P(N/m^2)$	f(expt)	Re	f(theo)	f(expt)/f(theo)	%diff
1	0.0163	0.2	11.83	0.0252	1107	0.0145	1.74	74.23
2	0.0329	0.4	23.66	0.0123	2239	0.0098	1.25	25.14
3	0.0496	0.7	41.41	0.0095	3378	0.0091	1.04	4.48
4	0.0664	1.3	76.90	0.0098	4522	0.0085	1.15	14.77
5	0.0803	1.7	100.56	0.0088	5470	0.0082	1.07	6.55
6	0.0805	1.8	106.48	0.0092	5479	0.0082	1.12	12.47
7	0.0948	2.4	141.97	0.0089	6457	0.0080	1.12	11.60
8	0.1095	3.0	177.46	0.0083	7458	0.0077	1.08	7.61
9	0.1376	4.6	272.11	0.0081	9368	0.0074	1.09	9.46
10	0.1660	6.2	366.76	0.0075	11303	0.0071	1.05	5.23
11	0.1987	8.4	496.90	0.0071	13534	0.0069	1.03	3.08
12	0.2272	10.3	609.29	0.0066	15472	0.0067	0.99	-0.67
13	0.2532	13.1	774.92	0.0068	17244	0.0065	1.04	3.95
14	0.2806	15.2	899.15	0.0064	19112	0.0064	1.00	0.22
15	0.2993	18.6	1100.27	0.0069	20385	0.0063	1.09	9.20
16	0.3269	21.0	1242.24	0.0065	22263	0.0062	1.05	5.21
17	0.3473	23.0	1360.55	0.0063	23650	0.0061	1.03	3.35

2.2 (f vs Re) FOR TWISTED TAPE HAVING y=4.705

Sl. No.	m(kg/s)	h(cm)	$\Delta P(N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	0.6	35.49	1107	0.0756	0.0145	5.23
2	0.0329	1.6	94.65	2239	0.0492	0.0098	5.01
3	0.0496	3	177.46	3378	0.0406	0.0091	4.48
4	0.0664	4	236.62	4522	0.0302	0.0085	3.53
5	0.0803	5.5	325.35	5470	0.0284	0.0082	3.45
6	0.0805	5.5	325.35	5479	0.0283	0.0082	3.44
7	0.0948	7.5	443.66	6457	0.0278	0.0080	3.49
8	0.1095	10.7	632.95	7458	0.0297	0.0077	3.84
9	0.1376	16	946.47	9368	0.0281	0.0074	3.81
10	0.1660	19.8	1171.26	11303	0.0239	0.0071	3.36
11	0.1987	26	1538.01	13534	0.0219	0.0069	3.19
12	0.2272	33.2	1963.92	15472	0.0214	0.0067	3.20
13	0.2532	41.6	2460.82	17244	0.0216	0.0065	3.30
14	0.2806	51.9	3070.11	19112	0.0219	0.0064	3.42
15	0.2993	60.3	3567.00	20385	0.0224	0.0063	3.54
16	0.3269	70.7	4182.21	22263	0.0220	0.0062	3.54
17	0.3473	73.7	4359.67	23650	0.0203	0.0061	3.31

2.3 (f vs Re) FOR TWISTED TAPE HAVING y=3.127

Sl. No.	m(kg/s)	h(cm)	$\Delta P(N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	0.7	41.41	1107	0.0882	0.0145	6.10
2	0.0329	2	118.31	2239	0.0615	0.0098	6.26
3	0.0496	3.7	218.87	3378	0.0500	0.0091	5.52
4	0.0664	5.9	349.01	4522	0.0445	0.0085	5.21
5	0.0803	7.9	467.32	5470	0.0407	0.0082	4.95
6	0.0805	7.7	455.49	5479	0.0396	0.0082	4.81
7	0.0948	10.3	609.29	6457	0.0381	0.0080	4.79
8	0.1095	13.2	780.84	7458	0.0366	0.0077	4.73
9	0.1376	19.4	1147.59	9368	0.0341	0.0074	4.62
10	0.1660	24.8	1467.03	11303	0.0299	0.0071	4.21
11	0.1987	32.6	1928.43	13534	0.0275	0.0069	4.00
12	0.2272	40.9	2419.41	15472	0.0264	0.0067	3.94
13	0.2532	49.5	2928.14	17244	0.0257	0.0065	3.93
14	0.2806	60.6	3584.75	19112	0.0256	0.0064	4.00
15	0.2993	71	4199.96	20385	0.0264	0.0063	4.17
16	0.3269	81.6	4826.99	22263	0.0254	0.0062	4.09
17	0.3473	88.3	5223.32	23650	0.0244	0.0061	3.97

2.4 (f vs Re) FOR TWISTED TAPE HAVING y=2.149

Sl.No.	m(kg/s)	h(cm)	$\Delta P(N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	1	59.15	1107	0.1260	0.0145	8.69
2	0.0329	2.9	171.55	2239	0.0892	0.0098	9.10
3	0.0496	5.4	319.43	3378	0.0730	0.0091	8.02
4	0.0664	8.3	490.98	4522	0.0626	0.0085	7.37
5	0.0803	11.2	662.53	5470	0.0577	0.0082	7.04
6	0.0805	10.6	627.04	5479	0.0545	0.0082	6.64
7	0.0948	13.8	816.33	6457	0.0511	0.008	6.38
8	0.1095	17.5	1035.20	7458	0.0485	0.0077	6.30
9	0.1376	25.3	1496.60	9368	0.0445	0.0074	6.01
10	0.1660	33.9	2005.33	11303	0.0409	0.0071	5.77
11	0.1987	44	2602.79	13534	0.0371	0.0069	5.37
12	0.2272	55.4	3277.15	15472	0.0357	0.0067	5.33
13	0.2532	66.3	3921.93	17244	0.0344	0.0065	5.29
14	0.2806	82.2	4862.48	19112	0.0347	0.0064	5.42

2.5 (f vs Re) FOR PLAIN ANGLE INSERT HAVING y=∞

Sl.No.	m(kg/s)	h(cm)	$\Delta P (N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	0.3	17.75	1107	0.0378	0.0145	2.61
2	0.0329	1	59.15	2239	0.0308	0.0098	3.14
3	0.0496	2.1	124.22	3378	0.0284	0.0091	3.12
4	0.0664	3.3	195.21	4522	0.0249	0.0085	2.93
5	0.0803	4.9	289.86	5470	0.0253	0.0082	3.08
6	0.0805	4.4	260.28	5479	0.0226	0.0082	2.76
7	0.0948	6	354.93	6457	0.0222	0.008	2.78
8	0.1095	7.7	455.49	7458	0.0214	0.0077	2.77
9	0.1376	12.3	727.60	9368	0.0216	0.0074	2.92
10	0.1660	16.4	970.13	11303	0.0198	0.0071	2.79
11	0.1987	21.5	1271.82	13534	0.0181	0.0069	2.62
12	0.2272	27.4	1620.83	15472	0.0177	0.0067	2.64
13	0.2532	33.8	1999.42	17244	0.0175	0.0065	2.70
14	0.2806	41	2425.33	19112	0.0173	0.0064	2.71
15	0.2993	48.4	2863.07	20385	0.0180	0.0063	2.85
16	0.3269	55.9	3306.73	22263	0.0174	0.0062	2.81
17	0.3473	60.2	3561.09	23650	0.0166	0.0061	2.72

2.6 (f vs Re) FOR TWISTED ANGLE INSERT HAVING y=5.07

Sl.No.	m(kg/s)	h(cm)	$\Delta P (N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	0.5	29.58	1107	0.0630	0.0145	4.34
2	0.0329	2.0	118.31	2239	0.0615	0.0098	6.28
3	0.0496	3.7	218.87	3378	0.0500	0.0091	5.50
4	0.0664	6.1	360.84	4522	0.0460	0.0085	5.41
5	0.0803	8.4	496.90	5470	0.0433	0.0082	5.28
6	0.0805	8.3	490.98	5479	0.0426	0.0082	5.20
7	0.0948	10.2	603.37	6457	0.0377	0.008	4.72
8	0.1095	13.4	792.67	7458	0.0372	0.0077	4.83
9	0.1376	20.2	1194.92	9368	0.0355	0.0074	4.80
10	0.1660	27.3	1614.91	11303	0.0330	0.0071	4.64
11	0.1987	33.5	1981.67	13534	0.0282	0.0069	4.09
12	0.2272	42.0	2484.48	15472	0.0271	0.0067	4.04
13	0.2532	52.3	3093.77	17244	0.0271	0.0065	4.17
14	0.2806	63.6	3762.21	19112	0.0269	0.0064	4.20
15	0.2993	72.1	4265.03	20385	0.0268	0.0063	4.25

2.7 (f vs Re) FOR TWISTED ANGLE INSERT HAVING y=4.105

Sl.No.	m(kg/s)	h(cm)	$\Delta P (N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	0.7	41.41	1107	0.0882	0.0145	6.08
2	0.0329	2.1	124.22	2239	0.0646	0.0098	6.59
3	0.0496	3.9	230.70	3378	0.0527	0.0091	5.79
4	0.0664	6.3	372.67	4522	0.0475	0.0085	5.59
5	0.0803	8.9	526.47	5470	0.0459	0.0082	5.60
6	0.0805	8.2	485.07	5479	0.0421	0.0082	5.14
7	0.0948	10.6	627.04	6457	0.0392	0.008	4.90
8	0.1095	13.6	804.50	7458	0.0377	0.0077	4.90
9	0.1376	20.4	1206.75	9368	0.0359	0.0074	4.85
10	0.1660	28.7	1697.73	11303	0.0347	0.0071	4.88
11	0.1987	38.8	2295.19	13534	0.0327	0.0069	4.74
12	0.2272	48.0	2839.41	15472	0.0309	0.0067	4.62
13	0.2532	59.4	3513.77	17244	0.0308	0.0065	4.74
14	0.2806	72.6	4294.60	19112	0.0307	0.0064	4.79
15	0.2993	82.2	4862.48	20385	0.0305	0.0063	4.84

2.8 (f vs Re) FOR TWISTED ANGLE INSERT HAVING y=3.612

Sl.No.	m(kg/s)	h(cm)	$\Delta P (N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	0.9	53.24	1107	0.1134	0.0145	7.82
2	0.0329	2.6	153.80	2239	0.0800	0.0098	8.16
3	0.0496	5.0	295.77	3378	0.0676	0.0091	7.43
4	0.0664	8.3	490.98	4522	0.0626	0.0085	7.37
5	0.0803	11.4	674.36	5470	0.0588	0.0082	7.17
6	0.0805	10.9	644.78	5479	0.0560	0.0082	6.83
7	0.0948	13.4	792.67	6457	0.0496	0.008	6.20
8	0.1095	18.0	1064.78	7458	0.0499	0.0077	6.48
9	0.1376	27.1	1603.08	9368	0.0476	0.0074	6.44
10	0.1660	33.8	1999.42	11303	0.0408	0.0071	5.75
11	0.1987	41.1	2431.24	13534	0.0346	0.0069	5.02
12	0.2272	50.4	2981.38	15472	0.0325	0.0067	4.85
13	0.2532	60.2	3561.09	17244	0.0312	0.0065	4.80
14	0.2806	78.0	4614.04	19112	0.0329	0.0064	5.15

2.9 (f vs Re) FOR TWISTED ANGLE INSERT HAVING y=2.915

Sl.No.	m(kg/s)	h(cm)	$\Delta P(N/m^2)$	Re	f_a	f_o	f_a/f_o
1	0.0163	1.1	65.07	1107	0.1386	0.0145	9.56
2	0.0329	2.8	165.63	2239	0.0861	0.0098	8.79
3	0.0496	5.4	319.43	3378	0.0730	0.0091	8.02
4	0.0664	8.9	526.47	4522	0.0671	0.0085	7.90
5	0.0803	12.2	721.68	5470	0.0629	0.0082	7.67
6	0.0805	12.0	709.85	5479	0.0617	0.0082	7.52
7	0.0948	15.4	910.98	6457	0.0570	0.008	7.12
8	0.1095	20.6	1218.58	7458	0.0571	0.0077	7.42
9	0.1376	30.6	1810.12	9368	0.0538	0.0074	7.27
10	0.1660	40.4	2389.83	11303	0.0488	0.0071	6.87
11	0.1987	51.4	3040.53	13534	0.0433	0.0069	6.27
12	0.2272	65.8	3892.35	15472	0.0424	0.0067	6.33
13	0.2532	81.7	4832.91	17244	0.0424	0.0065	6.52

3. HEAT TRANSFER RESULTS

3.1 STANDARDISATION – SMOOTH TUBE (hi vs Re)

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i(W/m^2/°C)	Re	h_i(theo)	h_i (expt)	h_i (expt)/h_i (theo)
1	0.0805	34.1	39	50.2	49	12.96	703	6836	1345	1269	0.94
2	0.0948	34.3	38.4	49.2	47.8	12.10	805	8014	1577	1646	1.04
3	0.1095	34.5	38.6	50.6	49	13.21	847	9303	1815	1833	1.01
4	0.1376	34.5	38.1	50.1	48.4	12.93	937	11618	2221	2316	1.04
5	0.1660	34.8	37.7	49.5	47.8	12.39	964	13998	2611	2485	0.95
6	0.1987	35.7	38.2	49.8	48	11.95	1045	17050	3068	3106	1.01
7	0.2272	36.3	38.9	49.8	48.3	11.44	1102	19812	3455	3669	1.06
8	0.2532	30.6	33.4	49.2	46.8	16.00	1077	19198	3468	3404	0.98
9	0.2806	30.6	33.1	48.6	46.2	15.55	1102	21195	3757	3669	0.98
10	0.2993	30.6	33	48.8	46.3	15.75	1123	22579	3952	3919	0.99
11	0.3269	30.7	33	49.2	46.6	16.05	1150	24692	4238	4266	1.01
12	0.3473	30.8	32.8	48.4	45.9	15.35	1133	26200	4440	4045	0.91

3.2 (h_i vs Re) FOR TWISTED TAPE HAVING y=4.705

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_0	h_a	h_a/h_o
1	0.0805	30.4	37.1	49.6	47.7	14.77	905	6375	1124	1944	1.73
2	0.0948	30.5	36.5	49.5	47.5	14.91	944	7464	1322	2137	1.62
3	0.1095	30.5	35.9	49.3	47.2	14.99	981	8557	1514	2337	1.54
4	0.1376	30.6	35.1	48.7	46.7	14.81	993	10659	1829	2403	1.31
5	0.1660	30.6	34.5	48.5	46.3	14.83	1062	12762	2119	2855	1.35
6	0.1987	30.7	34.1	47.8	45.5	14.24	1156	15218	2443	3646	1.49
7	0.2272	30.8	34.0	48.7	46.2	15.05	1183	17401	2720	3933	1.45
8	0.2532	31.0	33.9	49.1	46.4	15.30	1215	19417	2968	4311	1.45
9	0.2806	31.2	34.0	49.6	47.0	15.70	1203	21599	3226	4158	1.29
10	0.2993	31.4	34.0	50.0	47.2	15.90	1226	23097	3401	4447	1.31
11	0.3269	31.7	34.0	49.0	46.4	14.85	1243	25322	3654	4686	1.28

3.3 (h_i vs Re) FOR TWISTED TAPE HAVING y=3.127

Sl No.	m(kg/s)	T_1	T_2	T_3	T_4	LMTD	U_i	Re	h_o	h_a	h_a/h_0
1	0.0805	30.8	38.1	50.5	48.6	14.94	935	6487	1288	2156	1.67
2	0.0948	30.6	36.6	49	46.9	14.26	1012	7483	1496	2616	1.75
3	0.1095	30.5	36	49.5	47.4	15.14	981	8568	1709	2419	1.42
4	0.1376	30.6	35.2	49.4	47.1	15.32	1040	10672	2061	2811	1.36
5	0.1660	30.7	34.9	49.2	46.9	15.23	1098	12842	2391	3286	1.37
6	0.1987	30.8	34	48.9	46.2	15.15	1146	15218	2747	3754	1.37
7	0.2272	30.8	33.8	48.7	46.1	15.10	1165	17357	3054	3958	1.30
8	0.2532	30.9	33.9	48.6	46.1	14.95	1218	19392	3335	4651	1.39
9	0.2806	31	33.6	48.4	45.8	14.80	1229	21437	3616	4808	1.33
10	0.2993	31	33.6	48.6	45.9	14.95	1280	22866	3807	5709	1.50
11	0.3269	31.3	33.6	48.4	45.8	14.65	1260	25069	4094	5332	1.30
12	0.3473	31.4	33.6	48.3	45.7	14.50	1284	26666	4300	5780	1.34

3.4 (h_i vs Re) FOR TWISTED TAPE HAVING y=2.149

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_o	h_a	h_a/h_o
1	0.0805	29	36.8	49.8	47.7	15.68	967	6240	1247	2270	1.82
2	0.0948	29	36.6	49.9	47.3	15.67	1153	7334	1472	3653	2.48
3	0.1095	29.1	35.3	49.5	47.2	16.07	1027	8344	1676	2628	1.57
4	0.1376	28.9	34.3	50.2	47.5	17.21	1086	10328	2026	3056	1.51
5	0.1660	28.8	33.4	49	46.4	16.58	1122	12303	2339	3358	1.44
6	0.1987	28.7	32.9	49.8	46.7	17.44	1218	14616	2691	4385	1.63
7	0.2272	26.7	31	50.8	47.3	20.20	1209	15903	2922	4273	1.46
8	0.2532	26.9	30.6	49	45.6	18.55	1270	17678	3183	5156	1.62
9	0.2806	27	30.5	49.7	46.3	19.25	1253	19591	3455	4888	1.41
10	0.2993	27.2	30.5	50.2	46.5	19.50	1295	20950	3643	5584	1.53
11	0.3269	27.2	30.2	50.4	46.6	19.80	1288	22794	3902	5470	1.40
12	0.3473	27.4	30.4	49.8	46.3	19.15	1317	24340	4106	6027	1.47

3.5 (h_i vs Re) FOR PLAIN ANGLE INSERT HAVING $y=\infty$

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_o	h_a	h_a/h_o
1	0.0805	32.3	38.9	54	52.1	17.34	765	6676	1319	1469	1.11
2	0.0948	34.5	38.8	49.7	48.3	12.29	811	8074	1586	1652	1.04
3	0.1095	34.6	38.3	49.2	47.8	12.01	828	9280	1812	1721	0.95
4	0.1376	34.9	37.8	48.9	47.3	11.74	900	11632	2157	2067	0.96
5	0.1660	35.9	38.8	50.2	48.6	12.04	963	14386	2541	2431	0.96
6	0.1987	30.9	34.1	50	47.5	16.25	1026	15257	2751	2874	1.04
7	0.2272	30.9	34.2	51.3	48.7	17.45	1057	17467	3064	3130	1.02
8	0.2532	31.1	33.9	49.8	47.4	16.10	1070	19441	3339	3251	0.97
9	0.2806	31.2	33.6	49.3	46.8	15.65	1095	21491	3621	3492	0.96
10	0.2993	31.3	33.6	50.1	47.4	16.30	1106	22952	3815	3603	0.94
11	0.3269	31.4	33.7	51	48.2	17.05	1124	25132	4099	3805	0.93
12	0.3473	31.7	34	52.2	49.2	17.85	1145	26902	4319	4065	0.94

3.6 (h_i vs Re) FOR TWISTED ANGLE INSERT HAVING $y=5.07$

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_o	h_a	h_a/h_o
1	0.0805	29.3	35.8	50.2	48.4	16.64	770	6186	1238	1534	1.24
2	0.0948	29.5	35.1	49.4	47.5	16.08	825	7242	1458	1768	1.21
3	0.1095	29.7	34.6	49.2	47.1	15.96	879	8334	1674	2038	1.22
4	0.1376	32.5	38.3	56.4	53.5	19.51	1029	11361	2130	3081	1.45
5	0.1660	30.6	36.0	56.9	53.7	21.98	1017	13005	2407	2975	1.24
6	0.1987	32.2	37.1	57.4	54.3	21.19	1084	16101	2829	3632	1.28
7	0.2272	32.3	36.7	58.0	54.4	21.70	1154	18342	3143	4559	1.45
8	0.2532	31.9	36.1	58.1	54.5	22.30	1159	20187	3405	4631	1.36
9	0.2806	30.9	35.1	58.3	54.3	23.30	1231	21818	3649	6038	1.65
10	0.2993	30.9	34.6	58.4	54.4	23.65	1175	23126	3830	4899	1.28
11	0.3269	30.8	34.2	58.7	54.3	24.00	1218	25100	4096	5754	1.40
12	0.3473	30.7	33.9	58.6	54.4	24.20	1179	26533	4289	4976	1.16

3.7 (h_i vs Re) FOR TWISTED ANGLE INSERT HAVING y=4.105

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_o	h_a	h_a/h_o
1	0.0805	35.8	40.8	50.1	48.7	11.00	901	7139	1394	2167	1.55
2	0.0948	35.9	40.0	49.4	48.0	10.69	911	8338	1626	2227	1.37
3	0.1095	36.0	39.6	49.0	47.4	10.37	1013	9596	1857	2961	1.59
4	0.1376	35.9	39.4	50.2	48.5	11.68	1023	12013	2196	3044	1.39
5	0.1660	35.9	39.0	50.2	48.4	11.84	1073	14421	2544	3538	1.39
6	0.1987	36.0	38.6	49.5	47.7	11.30	1127	17198	2932	4199	1.43
7	0.2272	36.1	38.4	48.9	47.1	10.75	1191	19641	3262	5254	1.61
8	0.2532	36.4	38.7	50.6	48.6	12.05	1183	22051	3572	5088	1.42
9	0.2806	36.7	38.4	49.6	47.7	11.10	1134	24438	3878	4296	1.11
10	0.2993	36.9	38.6	49.1	47.2	10.40	1249	26196	4094	6581	1.61
11	0.3269	37.0	38.7	50.1	48.1	11.25	1237	28682	4400	6280	1.43

3.8 (h_i vs Re) FOR TWISTED ANGLE INSERT HAVING y=3.612

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_o	h_a	h_a/h_o
1	0.0805	30.4	43.6	67.7	63.9	28.54	929	6912	1357	2486	1.83
2	0.0948	30.4	42.6	68.2	64.2	29.51	962	8044	1582	2739	1.73
3	0.1095	30.4	41.8	68.7	64.3	30.27	1022	9199	1800	3288	1.83
4	0.1376	30.4	40.2	68.9	64.2	31.18	1066	11332	2127	3786	1.78
5	0.1660	30.4	39	69	63.9	31.72	1123	13468	2452	4621	1.88
6	0.1987	30.5	37.8	67.3	62.4	30.68	1147	15901	2811	5063	1.80
7	0.2272	30.5	36.9	66	61	29.79	1195	17978	3110	6140	1.97
8	0.2532	30.6	36.2	64.6	59.7	28.75	1211	19886	3379	6577	1.95
9	0.2806	30.5	35.6	63.4	58.6	27.95	1239	21845	3651	7496	2.05
10	0.2993	30.9	35.2	61.1	56.6	25.80	1231	23301	3845	7245	1.88
11	0.3269	31.2	34.7	58.9	54.8	23.90	1196	25386	4120	6180	1.50
12	0.3473	31.5	34.8	57.2	53.4	22.15	1245	27106	4336	7728	1.78

3.9 (h_i vs Re) FOR TWISTED ANGLE INSERT HAVING y=2.915

Sl No.	m(kg/s)	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)	LMTD	U_i	Re	h_o	h_a	h_a/h_o
1	0.0805	32.9	39.1	49.8	48	12.77	979	6743	1330	3289	2.47
2	0.0948	33.8	38.9	49.4	47.7	12.12	987	8014	1577	3382	2.14
3	0.1095	28.4	35	50.8	48.3	17.77	999	8240	1660	3517	2.12
4	0.1376	28.4	33.9	49.5	47	17.06	1065	10211	2015	4513	2.24
5	0.1660	28.6	33.2	50.6	47.8	18.29	1056	12241	2333	4345	1.86
6	0.1987	28.5	32.6	49.9	47	17.89	1134	14523	2682	6069	2.26
7	0.2272	28.6	32.1	48.7	46.2	17.10	1091	16522	2978	5010	1.68
8	0.2532	28.5	31.9	49.5	46.6	17.85	1169	18343	3242	7229	2.23
9	0.2806	28.7	31.8	50.1	46.9	18.25	1207	20354	3522	8972	2.55
10	0.2993	28.8	31.9	50.4	47.2	18.45	1233	21765	3713	10645	2.87
11	0.3269	29.1	31.5	49.5	46.5	17.70	1145	23742	3982	6399	1.61
12	0.3473	29.2	31.5	48.9	45.9	17.05	1199	25256	4182	8556	2.05

4.1 Pr, k & μ vs Temperature

Temp °F	Temp °C	μ	k (Btu/ft/hr/°F)	k (W/m/°C)	Pr
32	0.00	0.001794	0.32	0.5538	13.56
40	4.44	0.001546	0.326	0.5642	11.47
50	10.00	0.00131	0.333	0.5763	9.51
60	15.56	0.001129	0.34	0.5884	8.03
70	21.11	0.000982	0.346	0.5988	6.86
80	26.67	0.000862	0.352	0.6092	5.92
90	32.22	0.000764	0.358	0.6196	5.16
100	37.78	0.000683	0.362	0.6265	4.56
120	48.89	0.000559	0.371	0.6421	3.64
140	60.00	0.00047	0.378	0.6542	3.01
160	71.11	0.000401	0.384	0.6646	2.53
180	82.22	0.000347	0.388	0.6715	2.16
200	93.33	0.000305	0.392	0.6784	1.88
220	104.44	0.00027	0.394	0.6819	1.66
240	115.56	0.000242	0.396	0.6854	1.48
260	126.67	0.000218	0.396	0.6854	1.33
280	137.78	0.000199	0.396	0.6854	1.22
300	148.89	0.000185	0.396	0.6854	1.13