EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENATATION USING MODIFIED REDUCED WIDTH TWISTED TAPES (RWTT) AS INSERTS FOR TUBE SIDE FLOW OF LIQUIDS

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Under the Guidance of
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CERTIFICATE

This is to certify that the thesis entitled, **"EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION USING MODIFIED REDUCED WIDTH TWISTED TAPES (RWTT) AS INSERTS FOR TUBE SIDE FLOW OF LIQUIDS"** submitted by **Gaurav Johar & Virendra Hasda** in partial fulfilments 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 this thesis has not been submitted to any other University / Institute for the award of any Degree or Diploma.

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ABSTRACT

This project report deals with the use of modified twisted tape inserts as Passive Heat transfer augmentation device. Effect of Reduced width twisted tape (RWTT), Baffled Reduced width twisted tape ($BRWTT_1$) & Baffled Reduced width twisted tape with holes ($BRWTT_2$) on heat transfer and friction factor for heating of water for Reynolds number range 2500-30000, was studied experimentally in a double pipe heat exchanger. Three tapes of different twist ratio (y_w=3.69, y_w=4.39, y_w=5.25) for RWTT, $BRWTT_1$ & $BRWTT_2$ were used. Based on constant flow rate, the heat transfer coefficient were found to be 1.18-3.66, 2.61-7.07 & 3.58-8.08 times the smooth tube values for RWTT, $BRWTT_1$ & $BRWTT_2$ respectively. The friction factor values were found to be 3.23-5.96, 7.79-11.23 & 8.86-14.44 times the smooth tube values for RWTT, $BRWTT_1$ & $BRWTT_2$ respectively. Based on constant pumping power, the heat transfer coefficient values were found to be 0.88-1.62, 1.59-3.70 & 2.12-4.49 times the smooth tube values for RWTT, $BRWTT_1$ & $BRWTT_2$ respectively. Based on the increase in Heat transfer coefficient, Performance evaluation criteria R_1 & R_3, it was concluded that Baffled Reduced width twisted tape & Baffled Reduced width twisted tape with holes performs much better than the Reduced width twisted tapes(RWTT) of the same twist ratio.

CONTENTS

Chapter		Topic	Page No.
		Abstract	iv
		List of Figures	vii
		List of Tables	viii
		Nomenclature	ix
Chapter 1		**Introduction**	1
Chapter 2		**Literature Review**	3
	2.1	Classification of enhancement techniques	4
	2.2	Performance Evaluation Criteria	6
	2.3	Treated Surfaces	8
	2.4	Rough Surfaces	9
	2.5	Extended Surfaces	10
	2.6	Displacement Enhanced Devices	11
	2.7	Swirl Flow Devices	12
	2.8	Coiled Tubes	13
	2.9	Additives for liquids	14
	2.10	Twisted tape in laminar flow	14
	2.11	Twisted tape in turbulent flow	19
Chapter 3		**Present experimental work**	24

	3.1	Specifications of Heat exchanger used	25
	3.2	Types of inserts used	25
	3.3	Fabrication of twisted tapes	27
	3.4	Experimental Setup	28
	3.5	Experimental Procedure	31
	3.6	Standard equations used	33
	3.7	Precautions	34
Chapter 4		**Sample Calculations**	35
	4.1	Rotameter Calibration	36
	4.2	Pressure drop & Friction factor calculations	36
	4.3	Heat transfer coefficient calculation	37
Chapter 5		**Results & Discussion**	41
	5.1	Friction Factor Results	42
	5.2	Heat Transfer Coefficient Results	47
	5.3	Testing of experimental data for repeatability	51
Chapter 6		**Conclusion**	52
Chapter 7		**Scope for future work**	55
		References	57
		Appendix	62

List of figures

Fig. No	Figure Name	Page No.
2.1	Corrugated tubes, Two-Dimensional Roughness	10
2.2	Segmented fin heat sink	10
2.3	Conical Ring inserts in circular tubes	11
2.4	Heatex wire matrix tube insert	12
2.5	Twisted Tape	12
2.6	Coiled Tubes	13
3.1	Reduced Width Twisted Tape (RWTT)	26
3.2	Baffled Reduced Width Twisted Tape ($BRWTT_1$)	26
3.3	Baffles & RWTT to make $BRTWTT_1$	27
3.4	Baffled Reduced Width Twisted Tape with holes ($BRWTT_2$)	27
3.5	Schematic Diagram for the experimental setup	29
3.6	Photograph of the experimental setup	30
3.7	Wilson chart	32
4.1	Viscosity vs. Temperature	37
4.2	Temperature in different RTDs	38
4.3	Prandtl Number vs. Temperature	39
5.1	Friction Factor vs. Reynolds number for Smooth Tube	42
5.2	Friction factor vs. Reynolds number for Smooth tube, RWTT, $BRWTT_1$ & $BRWTT_2$	43
5.3	f_a/f_o vs. Reynolds Number for RRWTT, $BRWTT_1$ & $BRWTT_2$	44
5.4	Correlations for variation of Friction factor with Reynolds Number	45
5.5	Heat transfer coefficient vs. Reynolds Number for smooth tube	47
5.6	Heat transfer coefficient vs. Reynolds Number for Smooth tube, RWTT, $BRWTT_1$ & $BRWTT_2$	48
5.7	Performance evaluation criteria, R_1 vs. Reynolds Number for RWTT, $BRWTT_1$, $BRWTT_2$	49
5.8	Performance evaluation criteria, R_3 vs. Reynolds Number for RWTT, $BRWTT_1$, $BRWTT_2$	50
5.9	Heat transfer coefficient vs Reynolds number for $BRWTT_2$ (Repeatability)	51

List of tables

Table. No	Table Name	Page No.
2.1	Performance Evaluation Criteria	7
2.2	Performance Evaluation Criteria of Bergles et.al	8
2.3	Summaries of important investigations of twisted tape in laminar flow	15
2.4	Summaries of important investigations of twisted tape in turbulent flow	21
5.1	Correlations for Friction Factor for different twisted tapes	46
6.1	Range of R_1, R_3, f_a/f_o for different twisted tapes.	53
A.1.1	Small Rotameter Calibration	63
A.1.2	Large Rotameter Calibration	63
A.1.3	RTD Calibration	63
A.2.1	STANDARDISATION OF SMOOTH TUBE (f vs. Re)	64
A.2.2	FRICTION FACTOR vs. Re FOR RWTT HAVING y_w =3.69	64
A.2.3	FRICTION FACTOR vs. Re FOR RWTT HAVING y_w =4.39	65
A.2.4	FRICTION FACTOR vs. Re FOR RWTT HAVING y_w =5.25	65
A.2.5	FRICTION FACTOR vs. Re FOR $BRWTT_1$ HAVING y_w =3.69	66
A.2.6	FRICTION FACTOR vs. Re FOR $BRWTT_1$ HAVING y_w =4.39	66
A.2.7	FRICTION FACTOR vs. Re FOR $BRWTT_1$ HAVING y_w =5.25	67
A.2.8	FRICTION FACTOR vs. Re FOR $BRWTT_2$ HAVING y_w =3.69	67
A.2.9	FRICTION FACTOR vs. Re FOR $BRWTT_2$ HAVING y_w =4.39	68
A.2.10	FRICTION FACTOR vs. Re FOR $BRWTT_2$ HAVING y_w =5.25	68
A.3.1	STANDARDISATION OF SMOOTH TUBE (h_i vs. Re)	69
A.3.2	HEAT TRANSFER COEFFICIENT vs. Re FOR RWTT HAVING y_w=3.69	70
A.3.3	HEAT TRANSFER COEFFICIENT vs. Re FOR RWTT HAVING y_w=4.39	71
A.3.4	HEAT TRANSFER COEFFICIENT vs. Re FOR RWTT HAVING y_w=5.25	72
A.3.5	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_1$ HAVING y_w=3.69	73
A.3.6	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_1$ HAVING y_w=4.39	74
A.3.7	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_1$ HAVING y_w=5.25	75
A.3.8	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_2$ HAVING y_w=3.69	76
A.3.9	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_2$ HAVING y_w=4.39	77
A.3.10	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_2$ HAVING y_w=5.25	78
A.4	EXPERIMENTAL DATA FOR REPEATABLITY	79
A.4.1	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_2$ HAVING y_w=3.69	79
A.4.2	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_2$ HAVING y_w=4.39	79
A.4.3	HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_2$ HAVING y_w=5.25	79

NOMENCLATURE

A_i	Heat transfer area, m^2
A_{xa}	Cross- section area of tube with twisted tape, m^2
A_{xo}	Cross-section area of tube, m^2
C_p	Specific heat of fluid, J/Kg.K
d_i	ID of inside tube, m
d_o	OD of inside tube, m
f	Fanning friction factor, Dimensionless
f_a	Friction factor for the tube with inserts, Dimensionless
f_o	Theoretical friction factor for smooth tube, Dimensionless
g	acceleration due to gravity, m/s^2
Gz	Graetz Number, Dimensionless
h	Heat transfer coefficient, $W/m^2°C$
h_a	Heat transfer coefficient for tube with inserts, $W/m^2°C$
h_o	Heat transfer coefficient for smooth tube, $W/m^2°C$
h_i(exp)	Experimental Heat transfer coefficient, $W/m^2°C$
h_i(theo)	Theoretical Heat transfer coefficient, $W/m^2°C$
H	Pitch of twisted tape for 180°rotation,
L	heat exchanger length, m
LMTD	Log mean temperature difference, °C
m	Mass flow rate, kg/sec
Nu	Nusselt Number, Dimensionless
Pr	Prandtl number, dimensionless
Q	Heat transfer rate, W

Re	Reynolds Number, Dimensionless
R_1	Performance evaluation criteria based on constant flow rate, Dimensionless
R_3	Performance evaluation criteria based on constant pumping power, Dimensionless
U_i	Overall heat transfer coefficient based on inside surface area, $W/m^2°C$
v	flow velocity, m/s^2
W	Width of twisted tape, m
w	Width ratio (W/d_i), Dimensionless
y_w	Twist ratio of twisted tape (H/W), Dimensionless

Greek letters

Δh	Height difference in manometer, m
ΔP	Pressure difference across heat exchanger, N/m^2
μ	Viscosity of the fluid, $N\ s/m^2$
μ_b	Viscosity of fluid at bulk temperature, $N\ s/m^2$
μ_w	Viscosity of fluid at wall temperature, $N\ s/m^2$
ρ	Density of the fluid, kg/m^3

CHAPTER 1

INTRODUCTION

INTRODUCTION:

Heat exchangers are used in different processes ranging from conversion, utilisation & recovery of thermal energy in various industrial, commercial & domestic applications. Some common examples include steam generation & condensation in power & cogeneration plants; sensible heating & cooling in thermal processing of chemical, pharmaceutical & agricultural products; fluid heating in manufacturing & waste heat recovery etc. Increase in Heat exchanger's performance can lead to more economical design of heat exchanger which can help to make energy, material & cost savings related to a heat exchange process.

The need to increase the thermal performance of heat exchangers, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as "*Heat transfer Augmentation*". These techniques are also referred as "*Heat transfer Enhancement*" or "*Intensification*". Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger.

Use of Heat transfer enhancement techniques lead to increase in heat transfer coefficient but at the cost of increase in pressure drop. So, while designing a heat exchanger using any of these techniques, analysis of heat transfer rate & pressure drop has to be done. Apart from this, issues like long term performance & detailed economic analysis of heat exchanger has to be studied. To achieve high heat transfer rate in an existing or new heat exchanger while taking care of the increased pumping power, several techniques have been proposed in recent years and are discussed in the following sections.

Twisted tapes-a type of passive heat transfer augmentation techniques have shown significantly good results in past studies. For experimental work, Reduced width twisted tapes, having width less than ID of inside tube (W/d_i=0.727)) are used. Different designs of twisted tapes used are-Reduced width twisted tapes (RWTT), Baffled reduced width twisted tapes ($BRWTT_1$) & Baffled reduced width twisted tape with holes ($BRWTT_2$). All these tapes have been studied with three different twist ratios (y_w=3.69, y_w=4.39, y_w= 5.25)

CHAPTER 2

LITERATURE REVIEW

2.1 CLASIFICATION OF ENHANCEMENT TECHNIQUES: [1, 2]

Heat transfer enhancement or augmentation techniques refer to the improvement of thermo-hydraulic performance of heat exchangers. Existing enhancement techniques can be broadly classified into three different categories:

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

1. PASSIVE TECHNIQUES: These techniques 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) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power. Heat transfer augmentation by these techniques can be achieved by using:

- **Treated Surfaces**: This technique involves using pits, cavities or scratches like alteration in the surfaces of the heat transfer area which may be continuous or discontinuous. They are primarily used for boiling and condensing duties.
- **Rough surfaces**: These surface modifications particularly create the disturbance in the viscous sub-layer region. These techniques are applicable primarily in single phase turbulent flows.
- **Extended surfaces**: Plain fins are one of the earliest types of extended surfaces used extensively in many heat exchangers. Finned surfaces have become very popular now a days owing to their ability to disturb the flow field apart from increasing heat transfer area.
- **Displaced enhancement devices**: These inserts are used primarily in confined forced convection. They improve heat transfer 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**: They produce swirl flow or secondary circulation on the axial flow in a channel. Helical twisted tape, twisted ducts & various forms of altered

(tangential to axial direction) are common examples of swirl flow devices. They can be used for both single phase and two-phase flows.

- **Coiled tubes**: In these devices secondary flows or vortices are generated due to curvature of the coils which promotes higher heat transfer coefficient in single phase flows and in most regions of boiling. This leads to relatively more compact heat exchangers.
- **Surface tension devices:** These devices direct and improve the flow of liquid to boiling surfaces and from condensing surfaces. Examples include wicking or grooved surfaces.
- **Additives for liquids**: This technique involves addition of solid particles, soluble trace additives and gas bubbles added to the liquids to reduce the drag resistance in case of single phase flows. In case of boiling systems, trace additives are added to reduce the surface tension of the liquids.

2. ACTIVE TECHNIQUES: These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. Various active techniques are as follows:

- **Mechanical Aids**: Examples of the mechanical aids include rotating tube exchangers and scrapped surface heat and mass exchangers. These devices stir the fluid by mechanical means or by rotating the surface.
- **Surface vibration**: They have been used primarily in single phase flows. A low or high frequency is applied to facilitate the surface vibrations which results in higher convective heat transfer coefficients.
- **Fluid vibration**: Instead of applying vibrations to the surface, pulsations are created in the fluid itself. This kind of vibration enhancement technique is employed for single phase flows.
- **Electrostatic fields**: Electrostatic field like electric or magnetic fields or a combination of the two from DC or AC sources is applied in heat exchanger systems which induces greater bulk mixing, force convection or electromagnetic pumping to enhance heat transfer. This technique is applicable in heat transfer process involving dielectric fluids.

- **Injection**: In this technique, same or other fluid is injected into the main bulk fluid through a porous heat transfer interface or upstream of the heat transfer section. This technique is used for single phase heat transfer process.
- **Suction**: This technique is used for both two phase heat transfer and single phase heat transfer process. Two phase nucleate boiling involves the vapour removal through a porous heated surface whereas in single phase flows fluid is withdrawn through the porous heated surface.
- **Jet impingement:** This technique is applicable for both two phase and single phase heat transfer processes. In this method, fluid is heated or cooled perpendicularly or obliquely to the heat transfer surface.

3. COMPOUND TECHNIQUES: A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

2.2 PERFORMANCE EVALUATION CRITERIA: [1]

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

It may be noted that objective 1 accounts for increase in heat transfer rate, objective 2 and 4 yield savings in operating (or energy) costs, and objective 3 leads to material savings and reduced capital costs.

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

- <u>Fixed Geometry (FG) Criteria:</u> The area of flow cross-section (N and d_i) and tube length L are kept constant. This criterion is typically 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 cross sectional area (N and d_i) 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.
- ❖ Variable Geometry (VN) Criteria - The flow frontal area (N and L) is 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.

Table 2.1 Performance Evaluation Criteria [1]

Case	Geometry	M	P	Q	ΔTi	Objective
FG-1a	N, L, D_i	X			X	Q↑
FG-1b	N, L, D_i	X		X		ΔTi↓
FG-2a	N, L, D_i		X		X	Q↑
FG-2b	N, L, D_i		X	X		Δ Ti↓
FG-3	N, L, D_i			X	X	P↓
FN-1	N, D_i		X	X	X	L↓
FN-2	N, D_i	X		X	X	L↓
FN-3	N, D_i	X		X	X	P↓
VG-1	—	X	X	X	X	(NL) ↓
VG-2a	(NL)	X	X		X	Q↑
VG-2b	(NL)	X	X	X		Δ Ti↓
VG-3	(NL)	X		X	X	P↓

Bergles et al [3] suggested a set of eight (R1-R_8) number of performance evaluation criteria as shown in Table 2.2.

Table 2.2 Performance Evaluation Criteria of Bergles et al [3]

		Criterion number							
		R_1	R_2	R_3	R_4	R_5	R_6	R_7	R_8
Fixed	Basic Geometry	×	×	×	×				
	Flow Rate	×						×	×
	Pressure Drop		×				×		×
	Pumping Power			×					
	Heat Duty				×	×	×	×	×
Objective	Increase Heat Transfer	×	×	×					
	Reduce pumping power				×				
	Reduce Exchange Size					×	×	×	×

It may be noted that FG-1a & FG-2a are similar to R_1 & R_3 respectively. Performance evaluation criteria R_1 & R_3 have been used for present experimental work to determine heat transfer enhancement for different types of tapes.

2.3 TREATED SURFACES: [1, 2]

It consists of a variety of structured surfaces (continuous or discontinuous integral surface roughness or alterations) and coatings. The roughness created by this treatment do not causes any significant effect in the single phase heat transfer. These are applicable in cases of two phase heat transfer only.

2.3.1 Boiling: Some of the treated surfaces are as follows:

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

In enhanced boiling treated surfaces provide a large number of stable vapour traps or nucleation sites on the surface for bubble formation. In case of highly wetting fluids like refrigerants, organic liquids, cryogens and alkali liquid metals the normal cavities present on the heated surfaces tend to experience sub-cooled liquid flooding. For high surface tension fluids, coatings of non-wetting material (e.g. Teflon) on either the heated surface or its pits and cavities were found to be effective in nucleate boiling. Stainless steel surface along with Teflon can be spread to create spots of the no-wetting material on the heated surface which results in three to four times higher heat transfer coefficients.

2.3.2 Condensing: In condensation of vapours, treated surfaces promote drop wise condensation which is ideal for preventing surface wetting and break up the condensate film into droplets. This process provides better drainage and more effective vapour removal at cold heat transfer interface. This technique increases heat transfer by a factor of 10 to 100 in drop wise condensation when compared with that in film wise condensation as proposed by Bergles. Non- wetting inorganic compound or a noble metals or an organic polymer can be used effectively for coating the heat transfer surfaces. Among these, organic coatings have been used considerably in steam systems.

2.4 ROUGH SURFACES: [1, 2]

Small scale roughness or surface modification promotes turbulence in the flow field near the wall region by disturbing the viscous laminar sub layer. This disturbance causes higher momentum and heart transfer. This small scale roughness has little effect in laminar flows, but is very effective in turbulent single phase flows. Nowadays instead of natural roughness, artificial and structured roughness is used in most applications. Structured roughness can be integral to the surface. Wire coil type inserts can be inserted inside the tube to provide protuberances in the surface. In case of structured roughness almost an infinite number of geometric variations can be produced by machining, casting, or welding. Corrugated tubes,a type of 2-D roughness is shown in Fig 2.1 .Rough surfaces have been employed to enhance heat transfer in single phase flows both inside tubes and outside tubes.

External rough surface can created by grooving the heat transfer surface and can be used in double pipe and shell and tube bundles to enhance annulus or shell side heat transfer. Bergles and Champagne [4] proposed the idea of variable roughness which 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.

Fig. 2.1: Corrugated tubes, Two-Dimensional Roughness

2.5 EXTENDED SURFACES: [1, 2]

Extended or finned surfaces increase the heat transfer area which could be very effective in case of fluids with low heat transfer coefficients. This technique includes finned tube for shell & tube exchangers, plate fins for compact heat exchanger and finned heat sinks for electronic cooling.

Finned surfaces enhance heat transfer in natural or forced convection which can be used for cooling of electrical and electronic devices. 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. Segmented or interrupted longitudinal fins, as shown in Fig2.2, promote boundary layer separation of the fluids and disturb the whole bulk flow field inside circular tubes. Separation and restarting of the boundary layers increases the heat transfer rate. 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.

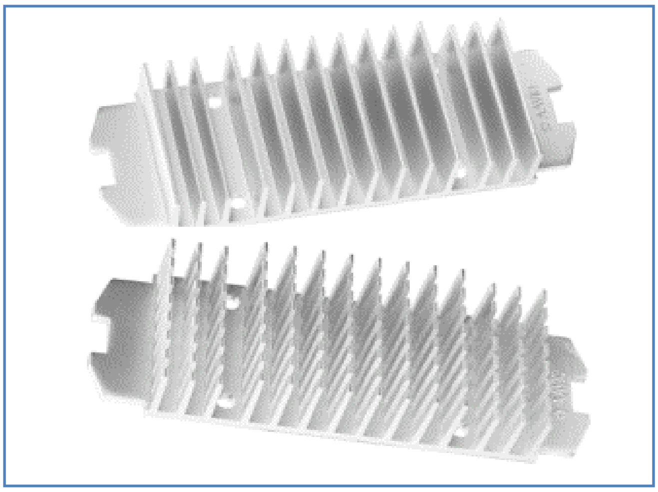
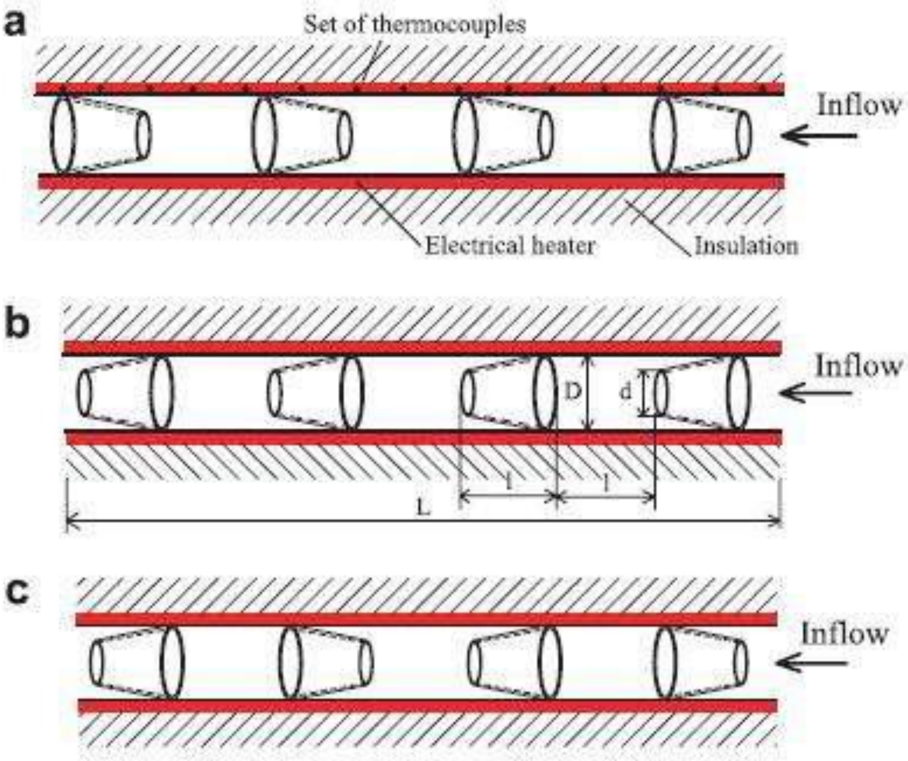


Fig. 2.2 Segmented fin heat sink

2.6 DISPLACED ENHANCEMENT DEVICES: [1, 2]

Displaced enhancement devices displace the fluid elements from the core of the channel to heated or cooled surfaces and vice versa .Displaced enhancement devices include inserts like static mixer elements (e.g. Kenics, Sulzer), metallic mesh, and discs, wire matrix inserts, rings or balls. Different types of conical ring inserts used in circular tubes are shown in Fig 2.3. These inserts do not alter heat transfer surface and provide a lot of scope for inter-mixing of the fluid particles. Disks promote higher heat transfer with moderate increase in friction factor whereas friction factor is very high for rings and round balls. Bergles found that pressure drop in the turbulent flows are extremely high. Most of the devices are suitable for laminar flow only. The main objective behind the use of static mixers is to increase the fluid mixing, so its application is limited to chemical processes with heat transfer only.



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

Fig. 2.3: Conical Ring inserts in circular tubes

Heatex wire matrix tube insert is one of the commercially available new displaced enhancement devices as shown in Fig.2.4.Degree of the disturbance and mixing depends on the coil matrix densities attached to core rod. Megerlin et al. [5] carried out the experiments with spiral brush inserts for turbulent flows 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 very high.

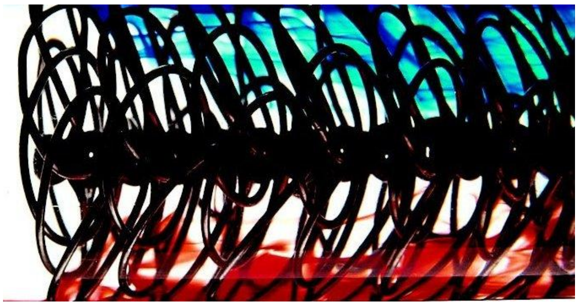


Fig. 2.4 Heatex wire matrix tube insert

2.7 SWIRL FLOW DEVICES: [1, 2]

Swirl flow devices causes swirl flow or secondary flow in the fluid .A variety of devices can be employed to cause this effect which includes tube inserts, altered tube flow arrangements, and duct geometry modifications. Dimples, ribs, helically twisted tubes are examples of duct geometry modifications. Tube inserts include twisted-tape inserts, helical strip or cored screw–type inserts and wire coils. Periodic tangential fluid injection is type of altered tube flow arrangement. Among the swirl flow devices, twisted- tape inserts had been very popular owing to their better thermal hydraulic performance in single phase, boiling and condensation forced convection, as well as design and application issues. Fig 2.5 shows a typical configuration of twisted tape which is used commonly.

Fig.2.5 Twisted Tape

Twisted tape inserts increases the heat transfer coefficients with relatively small increase in the pressure drop. They are known to be one of the earliest swirl flow devices employed in the single phase heat transfer processes. Because of the design and application convenience they have been widely used over decades to generate the swirl flow in the fluid. Size of the new heat exchanger can be reduced significantly by using twisted tapes in the new heat exchanger for a specified heat load. Thus it provides an economic advantage over the fixed cost of the equipment. Twisted tapes can be also used for retrofitting purpose. It can increase the heat duties of the existing shell and tube heat exchangers. Twisted tapes with multitube bundles are easy to fit and remove, thus enables tube side cleaning in fouling situations.

Inserts such as twisted tape, wire coils, ribs and dimples mainly obstruct the flow and separate the primary flow from the secondary flows. This causes the enhancement of the heat transfer in the tube flow. Inserts reduce the effective flow area thereby increasing the flow velocity. This also leads to increase in the pressure drop and in some cases causes' significant secondary flow. Secondary flow creates swirl and the mixing of the fluid elements and hence enhances the temperature gradient, which ultimately leads to a high heat transfer coefficient.

2.8 COILED TUBES: [1, 2]

A coiled or curved tube,as shown in Fig 2.6, causes secondary flows due to continuous change in the bulk velocity vector at the curve surface of the duct. Coiled tubes are used in domestic water heaters, chemical process reactors, solar heating system, industrial & marine boilers, kidney dialysis devices and blood oxygenators.

Secondary flows are generated due the centrifugal force on the fluid motion, induced because of the curvature of the coils. This curvature induced flow characteristics of the coiled tubes depends on the geometrical attributes like radius of curvature, helical number etc.

Fig. 2.6 Coiled Tubes

2.9 ADDITIVES FOR LIQUIDS: [1, 2]

Pressure drop in the tube flow is a consequence of the frictional loses with the solid surface. These frictional loses occurs because of the drag force of the fluid. This technique is basically concerned with reducing the drag coefficient using some additives to the fluid in single phase flows. Additives when added to the fluids are found to have operational benefits by lowering the frictional losses. These operational benefits could be fixed pressure or pumping costs. Polymeric additives induce a viscoelastic character to the solution which promotes secondary circulation in the bulk flow. These secondary flows have significant effect on the heat transfer coefficient. Some soluble polymeric additives in water have shear thinning effect on the solutions, which lead to a significant reduction in frictional loss as well as a modest increase in the heat transfer coefficient. Some of the additives used are polystyrene spheres suspension in oil and injection of gas bubbles.

2.10 TWISTED TAPE IN LAMINAR FLOW: [6]

A summary of important investigations of twisted tape in a laminar flow is represented in Table 2.3. Twisted tape increases the heat transfer coefficient with an increase in the pressure drop. Different configurations of twisted tapes, like full-length twisted tape, short length twisted tape, full length twisted tape with varying pitch, reduced width twisted tape and regularly spaced twisted tape have been studied widely by many researchers.

Use of twisted tapes for augmentation can be dated back to as early as up to the end of nineteenth century. One of the early researches on heat transfer enhancement by means of twisted tapes was carried out by Whitman, [7]. Saha et al. [8] concluded that the short length twisted tapes perform better than the full length twisted tapes because the swirl generated by the short length twisted tape decays slowly downstream which increases the heat transfer coefficient with minimum pressure drop. Regularly spaced twisted tape decreases the friction factor and reduces the heat transfer coefficient but the reduction in heat transfer coefficient is not much because the spacing of twisted tape disturbs the swirl flow. Date and Singham [9] studied the heat transfer and friction factor characteristics of fully developed laminar flows in tube containing twisted tape inserts. Laminar viscous liquid flow with uniform heat flow boundary condition for high prandtl number (appro.730) was investigated by Hong and Bergles [10. Tariq et al [11] found that twisted tape in a laminar flow was more efficient than internally threaded tube. Manglik and Bergles [12] developed the correlation between friction factor and Nusselt number for laminar flows including the swirl parameter.

Table 2.3 SUMMARIES OF IMPORTANT INVESTIGATIONS OF TWISTED TAPE IN LAMINAR FLOW [6]

SI No	Authors	Fluid	Configuration of twisted tape	Type of investigation	Observations	Comments
1	Saha and Dutta[8]	Water with (205< Pr < 518)	(a) Short length (b) Full length (c)Smoothly varying pitch (d)Regularly Spaced	Experiment 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 thermo hydraulic performance (4) Uniform pitch performs better then gradually decreasing pitch	It was observed that twisted tape is effective in laminar flow. Short length twisted tape perform better than full length tape.
2	Bergles and Hong [10]	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	Twisted tape can been used as full-length twisted tape, half-length twisted tape and varying pitch twisted tape
4	Manglik and Bergles [12]	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

5	Saha et al. [13]	Fluids with 205 < Pr<518	Twisted tape (regularly spaced)	Experiment in circular tube	(1) Pinching of twisted tape gives better results than connecting thin rod for thermo hydraulic performance (2) Reducing tape width gives poor results; larger than zero phase angle not effective	
6	Lokanath and Misal [14]	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	
7	Lokanath [15]	Water (240 < Re < 2300) (2.6 < Pr < 5.4)	Full-length and half-length twisted tapes	Experimental in horizontal tube	(1) On unit pressure drop basis and on unit pumping power basis, half-length twisted tape is more effective than full-length twisted tape	
8	Liao and Xin[17]	(1) Water (2) Ethylene glycol (3) Turbine oil 5.5 < Pr < 590, 80 < Re< 50000	Segmented twisted tape and three-dimensional extended surfaces	Experiment in tube flow	(1) In a tube with three-dimensional Extended surfaces and twisted tape increases average Stanton number up to 5.8 times compared with empty smooth tube	

9	Ujhidy [18]	Water	Twisted tape	Experiment in channel	(1) Explained flow structure (2) Proved existence of secondary flow in tubes with helical static elements.	
10	Suresh Kumar [19]	Water	Twisted tape	Experiment in large diameter annulus	(1) Observed relatively large values of friction factor (2) Measured heat transfer in annulus with different configurations of twisted tapes	
11	Saha and Chakraborty [20]	Water (145 < Re<1480) (4.5 < Pr < 5.5)	Twisted tape (regularly spaced) (1.92<y<5.0)	Experiment in circular tube flow	(1) Larger number of turns may yield improved thermo hydraulic performance compared with single turn	
12	Saha and Bhunia [22]	Servotherm medium oil (205 < Pr < 512,45< Re < 840)	Twisted tape (twist ratio 2.5<y <10)	Experiment in circular tube	(1) Heat transfer characteristics depend on twist ratio, Re and Pr	Uniform pitch twisted tape performs better than gradually varying pitch twisted tape
13	Agarwal and Raja Rao[23]	Servotherm oil	Twisted tape	Experiment in circular tube	Nusselt number for augmented tube is more than plain tube	

Saha et al. [13] found that placing twisted tape concentric to the inside tube gives better heat transfer performance than a twisted tape inserted by a loose fit. Lokanath and Misal [14] studied twisted tapes in shell and tube heat exchanger for different fluids. Their study revealed that twisted tapes of tighter twists are expected to give higher overall heat transfer coefficients. Lokanath [15] investigated the laminar flow experimentally using the tube fitted with half length tapes. He concluded that half length twisted tapes gives better performance than full length twisted tapes on the basis of unit pumping power

AI-Fahed et al. [16] investigated that , for high pressure drop and low twist ratio (y = 5.4) and, a loose fit twisted tape is a better option for the heat exchanger owing to it's easy installation and removal for cleaning purposes. For other twist ratios tight fit gives better performance that the loose- fit twisted tapes. Liao and Xin [17] carried out experimental work on compound heat transfer enhancement technique with three dimensional internal extended surfaces by using segmented twisted tape inserts. Results revealed the reduction in the friction factor with small decrease in Stanton number. The Stanton number is the ratio of heat transfer rate to the enthalpy difference and gives a measure of the heat transfer coefficient. Ujhidy et al. [18] proposed a modified dean number for the laminar flow in coils and tubes containing twisted tapes and helical elements. Dean number compensates for the curvature of the coiled tubes or helical elements and gives the measure of the magnitude of the secondary flows. Thermo-hydraulic performance of twisted tape inserts in a large hydraulic diameter annulus was reported by Suresh Kumar et al., [19].

In laminar flow, the dominant thermal resistance is distributed entirely over the cross section of the tube. Thus, a twisted tape insert is more effective than other technique as it mixes the bulk flow.

Saha and Chakraborty [20] observed the drastic reduction in the pressure drop compared to the reduction in the heat transfer in their experiment carried out with regularly spaced twisted tapes for laminar flow conditions. It was concluded that for a constant pumping power a large number of turns gives a better thermo-hydraulic performance than the single turn in the twisted tapes.

P.Sivashanmugam and S.Suresh [21] investigated heat transfer and friction factor characteristics of circular tube fitted with full length helical screw elements of different twist ratio and helical screw inserts with spacer length 100,200,300,400 mm with uniform heat flux under laminar flow conditions. They found that regularly spaced helical screw elements can safely be used for heat transfer augmentation without much increase in pressure drop than full length helical screw inserts. S.K.Agarwal and M.Raja Rao [23] experimentally determined the isothermal and non-isothermal friction factors and mean Nusselt Numbers for uniform wall temperature heating and cooling of Servotherm oil for flow in a circular tube with twisted tape insert.

2.11 TWISTED TAPE IN TURBULENT FLOW: [6]

Unlike laminar flows where thermal resistance exist entirely over the cross section, it is limited to the thin viscous sub layer. So the main objective of the twisted tape in the turbulent region is to reduce that resistance near the wall to promote better heat transfer. Besides, a tube inserted with a twisted tape produces swirl and cause intermixing of the fluid which leads to better performance than a plain tube. Heat transfer rate is improved effectively with the increase in the frictional losses. A summary of important investigations of twisted tape in turbulent flow is represented in Table 2.4.

Ventisislav D.Zimparov, Plamen J.Penchev and Joshua P. Meyer [24] evaluated the performance of angled spiralling tape inserts, a round tube inside a twisted square tube and spiralled tube inside the annulus for enhancement in the annulus side of tube-in-tube Heat exchanger. The results showed that for most of the cases, angled spiralling tube inserts technique is the most efficient. Watcharin Noothong, Smith Eiamsa-ard and Pongjet Promvonge [25] studied experimentally the effect of twisted tape insert on heat transfer and friction factor characteristics in concentric tube heat exchanger for Reynolds number 2000 to 12000. They found that enhancement efficiency and Nusselt number increases with decreasing the twist ratio and friction factor also increase with decreasing the twist ratio. Smith et. al [26] carried out experimental study on the mean Nusselt number; friction factor and enhancement efficiency characteristics in a round tube with short-length twisted tape insert under uniform wall heat flux boundary conditions for Re 4000 to 20000. Pongjet Promvonge [27] examined the thermal augmentation in a circular tube with twisted tape and wire coil turbulators for Reynolds Number 3000 to 18000.The report indicate that presence of wire coils together with the twisted tape lead to double increase in the heat transfer over the use of wire coil or twisted tape alone. Smith et. al. [28] investigated the heat transfer enhancement and pressure loss by insertion of single twisted tape, full length dual and regularly spaced dual twisted tapes as swirl generators in round tube under axially uniform wall heat flux conditions. Chinaruk Thianpong et.al. [29] experimentally investigated the friction and compound heat transfer behavior in dimpled tube fitted with twisted tape swirl generator for a fully developed flow for Reynolds number in the range of 12000 to 44000.

Al-Fahed and Chakroun [30] studied the effect of tube -tape clearance for a fully developed turbulent flow in an isothermal condition. Their results revealed that for low twist ratio loose fit tape shows nearly the same enhancement as the tight-fit of the twist ratio. Such behavior indicates the presence of an optimum tape width. Smith Eiamsa-ard and Pongjet Promvonge [31] carried out the experiments with full length helical tape with or without centered rod and regularly spaced helical tube.

T.S.Ravigururajan and A.E.Bergles [32] developed the correlations for pressure drop and heat transfer in single phase turbulent flow in enhanced tubes. Results were found to be applicable to a wide range of Prandtl number.Klepper [33] and Kidd Jr [34] studied the short-length twisted tape and compared the results with full length tape. They observed better performance of the short length twisted tapes over the full length in gas cooled nuclear reactors. Saha [35] revealed that swirl generated by the tapes breaks down in between the spacing of regularly spaces twisted tapes. So for a constant heat flux boundary conditions full length twisted tapes are found to give better performance than the regularly spaced tapes.

Blackwelder and Kreith [36] gave the concept of continuous and decaying swirl flow generated by the twisted tapes in the turbulent flow. Zozulya and Shkuratov [37] studied the effect of the pitch of the twisted tapes in the heat transfer process and reported the increase in the heat transfer coefficient for small reduction in pitch.

Shyy Woei Chang & Ker Wei Yu [38] investigated the effect of increase in number of inserts. They found that the heat transfer rate increases for turbulent flow (based on same pumping power consumption) but the same is not true for the laminar flow where negative effect has been observed. Rao and Sastri [39], studied the rotating tube with a twisted tape insert, observed that increase in heat transfer rate compensates for the increase in frictional losses.

Table 2.4 SUMMARIES OF IMPORTANT INVESTIGATIONS OF TWISTED TAPE IN TURBULENT FLOW [6]

Author	Configuration of tape	Observations	Comments
Zimparov et. al.[24]	Angled spiralling tape inserts, Spiralled tube	Enhancement in the annulus side	Angled spiralling tape inserts more efficient than spiralled tube.
Watcharin et.al[25]	Twisted tape insert	Effect on heat transfer and friction factor	Nusselt number and friction factor increases with decrease in twist ratio.
Smith et .al [26]	Short length twisted tape	Nusselt number, friction factor, enhancement efficiency characteristics.	
Promvonge [27]	Twisted tape and wire coil turbulators	Double increase in heat transfer than wire coil and tape alone	
Smith et. al. [28]	Single twisted tapes, full length dual and regularly spaced dual twisted tapes.	Full length dual twisted tapes yield higher heat transfer enhancement than regularly spaced twisted tapes.	
C.Thianpong et.al.[29]	Dimple tube fitted with twisted tape	Both heat transfer and friction factor are higher than dimpled tube and plain tube alone.	Heat transfer coefficient and friction factor increases as pitch ratio and twist ratio decreases.
Al-Fahed and Chakroun [30]	Loose fit tapes and tight –fit tape	For same twist ratio effect of loose fit and tight fit twisted tape are same.	
S.Eiamsa-ard , P. Promvonge [31]	Full length helical tape with or without centred rod and regularly spaced helical tape	Full length helical tape with rod provide highest heat transfer rate	

Klepper[33]	Short-length twisted tape	Usefulness of tape in gas-cooled nuclear reactor	
Kidds Jr[34]	Short-length twisted tape	Effectiveness of twisted tape in gas cooled nuclear reactor	
Blackwelder and Kreith[36]	Twisted tape	Recommended that optimum design of heat exchanger with tape-induced swirl flow must consider combination of continuous and decaying swirl flow	
Zozulya and Shkuratov[37]	Twisted tape	Smooth decrease in pitch of twisted tape has significant influence on heat transfer	
S.W.Chang & K.W.Yu[38]	Single, twin & triple twisted tape	With increase in number of inserts the heat transfer enhancement increases for turbulent flow	
Cresswell[40]	Full length twisted tape	Ratio of maximum velocity to mean velocity is smaller in swirl flow compared with straight flow	Heat transfer coefficient is enhanced by twisted tape
Kreith and Margolis[41]	Full length twisted tape	Centrifugal force aids convection when fluid is heated up	All configurations of twisted tape lead to high friction factor, which is due to fact that twisted tape disturbs entire flow field
Thorsen and Landis[42]	Full length twisted tape	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

Colburn and King[43]	Inserts like baffled tube and short-length twisted tape	Short-length twisted tapes more effective than full-length twisted tapes	
Seymour[44]	Short-length twisted tape	Short-length twisted tapes more effective than full-length twisted tapes	
Kreith and Sonju[45]	Short-length twisted tape	Short-length (25–45 per cent of tube length) tapes perform better than full-length tapes	
Huang and Tsou[46]	Twisted tape	Studied free swirl flow	

CHAPTER 3

PRESENT EXPERIMENTAL WORK

3.1 SPECIFICATIONS OF HEAT EXCHANGER USED

The experimental study is done in a double pipe heat exchanger having the specifications as listed below:-

Specifications of Heat Exchanger:

Inner pipe ID = 22mm

Inner pipe OD=25mm

Outer pipe ID =53mm

Outer pipe OD =61mm

Material of construction= Copper

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 60°C) flowed through the annulus side in the counter current direction.

3.2 TYPES OF INSERTS USED:

For experimentation, three types of twisted tape inserts made from stainless steel strips of thickness 1.80 mm were used.

1. Reduced Width Twisted Tape(RWTT): Twisted tapes of width 16mm, thickness 1.80 mm were used in the inner pipe of ID 22mm as shown in Fig 3.1.
2. Baffled Reduced Width Twisted Tape ($BRWTT_1$): Baffles in the shape of rectangular strips of size 16mm×10mm×1.80mm were attached in such a way that they were projecting at right angles on each side to the surface of twisted tape as shown in Fig. 3.2 & Fig 3.3. A constant distance of 20 cm was kept in between two consecutive strips.
3. Baffled Reduced Width Twisted Tape with holes ($BRWTT_2$): In these twisted tapes, holes of diameter 6 mm were drilled at midpoint of two consecutive strips of $BRWTT_1$ as shown in Fig 3.4.

The insert used for the experiment were made of twisted stainless steel strip of thickness 1.80mm.

The present work deals with finding the friction factor and the heat transfer coefficient for the various types of twisted tapes with twist ratios (y_w=3.69, 4.39, 5.25) and comparing those results with that of smooth tube and finally finding the heat transfer enhancement in comparison to a smooth tube on constant flow rate basis (R_1) as well as constant pumping power basis (R_3).

Twist ratio y=3.69

Twist ratio y=4.39

Twist ratio y=5.25

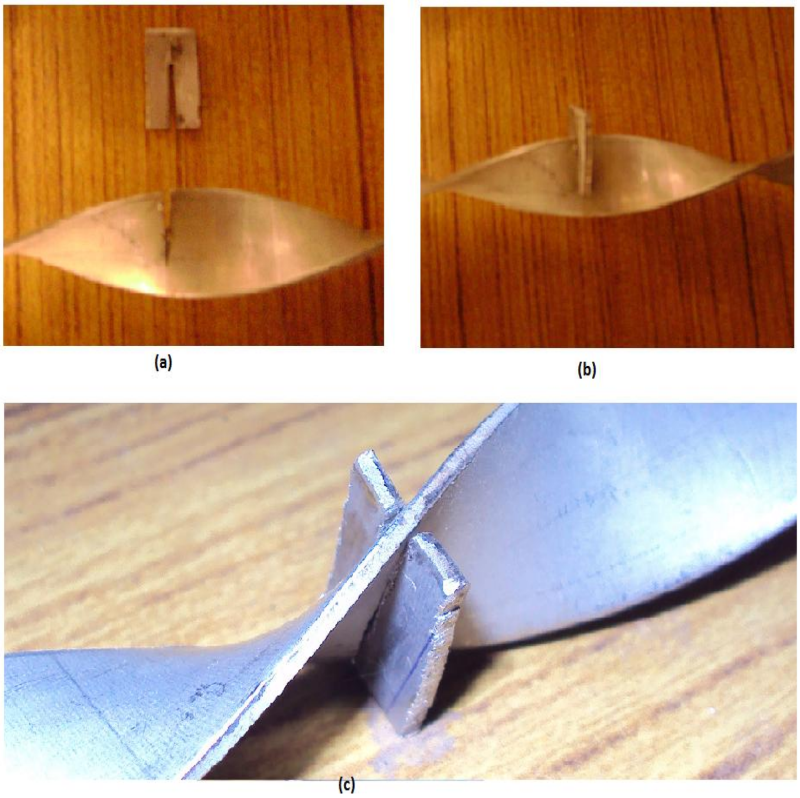
Fig 3.1 Reduced Width Twisted Tape (RWTT)

Twist ratio y=3.69

Twist ratio y=4.39

Twist ratio y=5.25

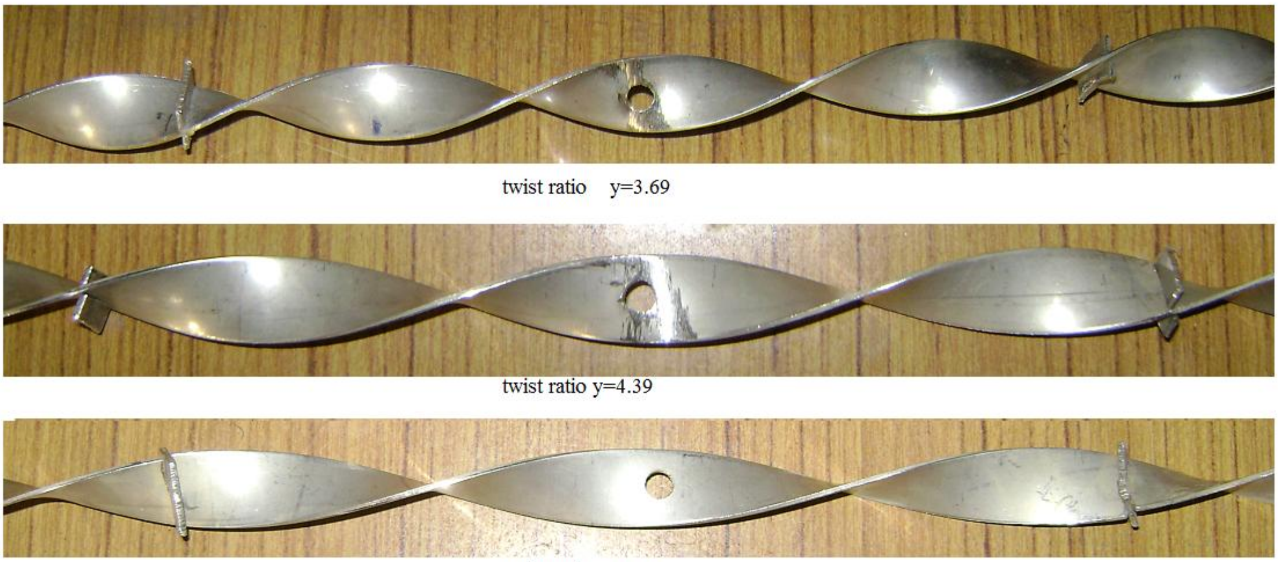
Fig 3.2 Baffled Reduced Width Twisted Tape ($BRWTT_1$)



(a) (b)

(c)

Fig 3.3 Baffles & RWTT to make $BRWTT_1$



twist ratio y=3.69

twist ratio y=4.39

twist ratio y=5.25

Fig 3.4 Baffled Reduced Width Twisted Tape with holes ($BRWTT_2$)

3.3 FABRICATION OF TWISTED TAPES:

The stainless steel strip of length 125cm, width 16mm and thickness 1.80mm were taken. Holes were drilled at both ends of every tape so that the two ends could be fixed to the metallic clamps. Desired twist was obtained using a Lathe machine. One end was kept fixed on the tool post of the lathe while the other end was given a slow rotatory motion by rotating the chuck side. During the whole operation the tape was kept under tension by applying a mild pressure on the tool post side to avoid its distortion. Three tapes with varying twist ratios were fabricated (y_w=5.25, y_w =4.39, y_w =3.69) as shown in fig 3.1. The end portions of the fabricated tapes were cut and holes of 3mm size were drilled for joining the two tapes. Three tapes with the same twist ratio and twist in the same direction were joined by using small screws with nuts, thus giving a total length of 3.0m, which is sufficient enough for the double pipe heat exchanger, used for the experiment.

Since the tape width is less than inside diameter of inner tube, so holes of 2.5mm size were drilled into the tape & cycle spokes of length 20mm were welded into the tape at an interval of 40cm. This allows the tape to be concentric with the tube.

For making baffled twisted tape ($BRWTT_1$), small cuts were made on original twisted tape & on the strips of size 16mm×10mm which were then fitted together. The distance between two consecutive baffles was kept 20cm. Baffle strips were projecting at right angles to the surface of twisted tape on each side as shown in fig 3.2 & fig 3.3

For making $BRWTT_2$, holes of 6 mm diameter were drilled at midpoint of two consecutive baffles.

3.4 EXPERIMENTAL SETUP

Fig 3.5 shows the schematic diagram of the experimental setup. It is a double pipe heat exchanger consisting of a calming section, test section, rotameters, overhead water tank for supplying cold water & a constant temperature bath (500 litre capacity) for supplying hot water with in-built heater, pump & the control system. The test section is a smooth copper tube with dimensions of 2430mm length, Inner tube-22mm ID, and 25mm OD; Outer GI pipe-53mm ID, and 61 mm OD. The outer pipe is well insulated using 15mm dia asbestos rope to reduce heat losses to the atmosphere. Two calibrated rotameters, with the flow ranges 1 to 5 LPM and 300 to 1250 LPH, are used to measure the flow of cold water. The water, at room temperature is drawn from an overhead tank using gravity flow. Similarly a rotameter is provided to control the flow rate of hot water from the inlet hot water tank. Hot water flow rate is kept constant at 1000LPH. Two pressure tapings- One just before the test section and the other just after the test section are attached to the U-tube manometer for pressure drop measurement. Carbon tetrachloride is used as the manometric fluid. Bromine crystals were dissolved in it to impart pink colour to it for easy identification. Four RTDs measure the inlet & outlet temperature of hot water & cold water (T_1 –T_4) through a multipoint digital temperature indicator. Fig 3.6 shows photograph of the setup.

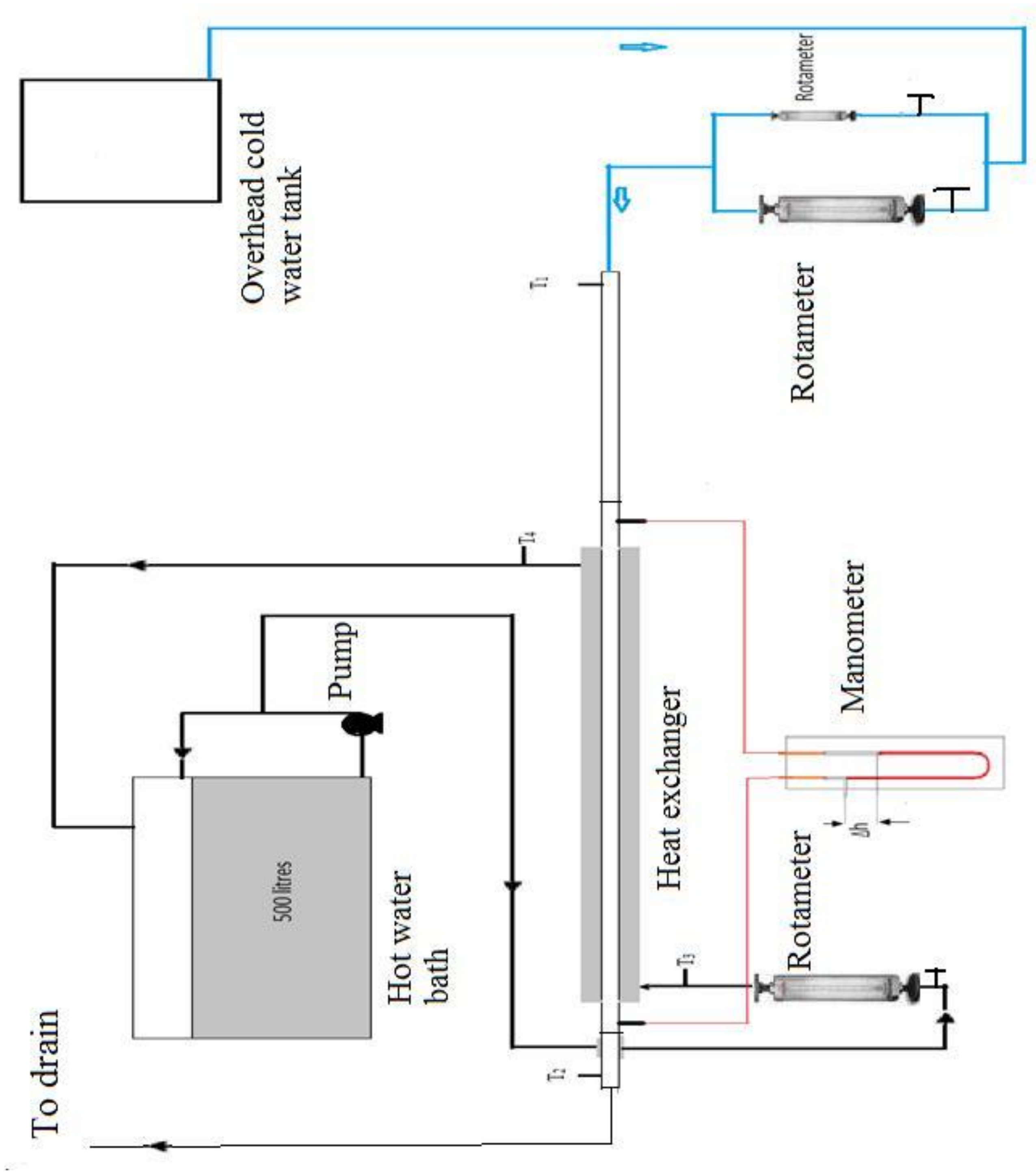


Fig 3.5 Schematic Diagram for the experimental setup

Fig 3.6 Photograph of the experimental setup

3.5 EXPERIMENTAL PROCEDURE:

1. All the rotameters & RTD are calibrated first.
   i. For rotameter calibration, water is collected in a bucket. Weight of water collected & time of collection is noted to calculate mass flow rate of water.
   ii. A minimum of 3 readings are taken for each flow rate & average flow rate is used for calculations. The readings are given in A.1.1 & A.1.2
   iii. For RTD calibration, all the RTDs are dipped in a constant water bath & readings shown by each RTD are noted. Temperature shown by one of the RTD (T_1) was taken as reference & corrections were made to other RTDs values (i.e. T_2-T_4) accordingly.
2. Twist Ratio(y) of the twisted tapes were calculated.

   Twist Ratio, $y_w = H/W$

   Where H = Linear distance of the tape for 180° rotation

   W = Width of twisted tape
3. Standardization of the set-up:

   Before starting the experimental study on friction & heat transfer in heat exchanger using inserts, standardization of the experimental setup is done by obtaining the friction factor & heat transfer results for the smooth tube & comparing them with the standard equations available.
4. For friction factor determination: Pressure drop is measured for each flow rate with the help of manometer at room temperature.
   a. The U-tube manometer used carbon tetrachloride as the manometric liquid. A little of bromine crystals were added to it to impart a colour to the CCl_4.
   b. Air bubbles are removed from the manometer so that the liquid levels in both the limbs were equal when the flow is stopped.
   c. Water at room temperature is allowed to flow through the inner pipe of the heat exchanger.
   d. The manometer reading is noted.
5. For heat transfer coefficient calculation:
   a) Then, heater is put on to heat the water to 60°C in a constant temperature water tank of capacity 500 litres. The tank is provided with a centrifugal pump & a bypass valve for recirculation of hot water to the tank & to the experimental setup.

b) Hot water at about 60°C is allowed to pass through the annulus side of heat exchanger at 1000LPH ($\mathbf{m}_h$=0.2715 Kg/sec).

c) Cold water is now allowed to pass through the tube side of heat exchanger in counter current direction at a desired flow rate.

d) The water inlet and outlet temperatures for both hot water & cold water (T_1-T_4) are recorded only after temperature of both the fluids attains a constant value.

e) The procedure was repeated for different cold water flow rates ranging from 0.0331-0.3492 Kg/sec.

6. Preparation of Wilson chart:

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{h_i}{d_o \times h_o} + \frac{x_w \times d_i}{k_w \times d_l} + R_d \tag{3.1}$$

where R_d is the dirt resistance

All the resistances, except the first term on the RHS of equation (1), are constant for this set of experiments.

For Re>10000, Seider Tate equation for smooth tube is of the form:

$h_i = A \times Re^{0.8}$

Therefore Eq. (3.1) can be written as

$$\frac{1}{U_i} = \frac{1}{A \times Re^{0.8}} + K \quad \text{, where K is a constant.} \tag{3.2}$$

K is to be found from the Wilson chart ($1/U_i$ vs. $1/Re^{0.8}$) as the intercept on the y-axis.

$K=6.613\times10^{-4}$ (Refer Fig 3.7)

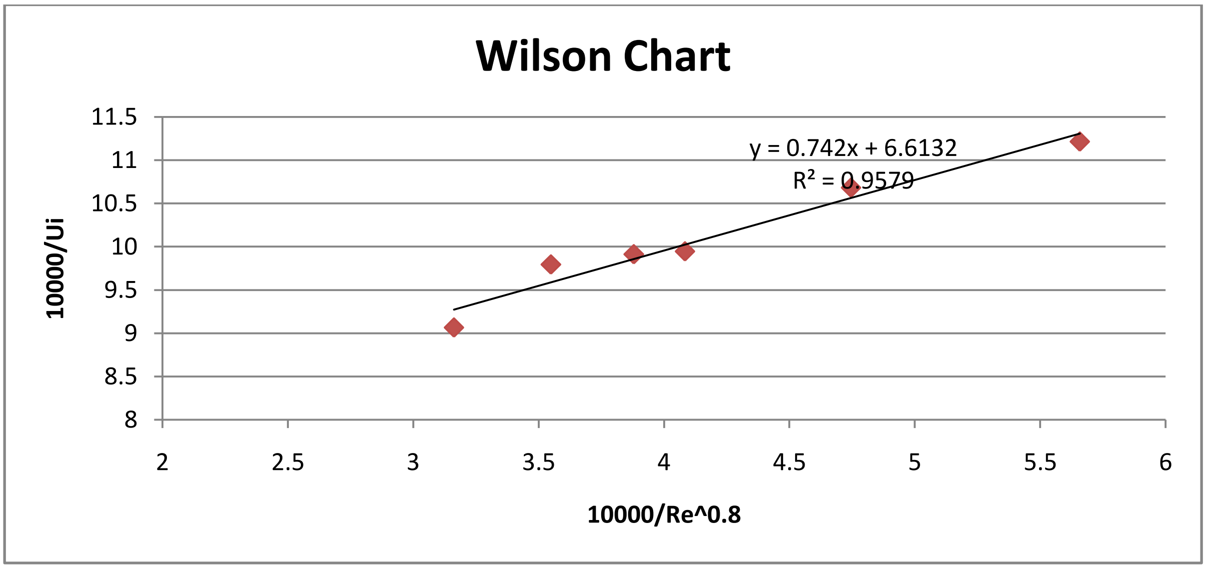


Fig 3.7 Wilson chart

7. After confirmation of validity of experimental values of friction factor & heat transfer coefficient in smooth tube with standard equations, friction factor & heat transfer studies with inserts were conducted.
8. The friction factor & heat transfer observations & results for all the cases are presented in Tables A.2.1-A.2.10 & A.3.1-A.3.10 respectively.

3.6 STANDARD EQUATIONS USED:

I. *Friction factor (f_0) calculations:*

a. For Re< 2100

$$f = \frac{16}{Re} \tag{3.3}$$

b. For Re>2100

Colburn's Equation:

$$f = \frac{0.046}{Re^{0.2}} \tag{3.4}$$

II. *Heat transfer calculations*

i. Laminar Flow:

For Re<2100

Nu= f(Gz)

Where $Gz=\frac{Re\times Pr\times d_i}{L}$ (3.5)

a. For Gz<100, Hausen Equation is used.

$$Nu = 3.66 + \frac{0.085Gz}{1+0.045Gz^{0.67}}\left(\frac{\mu b}{\mu w}\right)^{0.14} \tag{3.6}$$

b. For Gz>100, Seider Tate equation is used.

$$Nu = 1.86Gz^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3.7}$$

ii. Transition Zone:

For 2100<Re<10000, Hausen equation is used

$$Nu = 0.116\left(Re^{2/3} - 125\right) \times Pr^{1/3} \times (1 + (\frac{D}{l})^{2/3}(\frac{\mu_b}{\mu_w})^{0.14} \qquad (3.8)$$

iii. Turbulent Zone:

For Re>10000, Seider-Tate equation is used.

$$Nu = 0.023 \times Re^{0.8} \times Pr^{1/3} \times (\frac{\mu_b}{\mu_w})^{0.14} \qquad (3.9)$$

Viscosity correction Factor$(\frac{\mu_b}{\mu_w})^{0.14}$ is assumed to be equal to 1 for all calculations as this value for water in present case will be very close to 1 & the data for wall temperatures is not measured.

3.7 PRECAUTIONS:

1. While fabricating twisted tapes, exact number of rotations should be measured for a given twist so that other tapes could be made of exact twist ratio.
2. Rotameters should be calibrated properly to measure exact flow rate of water for a given rotameter reading.
3. RTDs should be calibrated properly. This is done by measuring temperature the temperature of water bath by all RTDs at the same time & then taking one of them as reference.
4. Air bubbles are removed from manometer so that liquid levels in both the limbs are equal when the flow is stopped. The presence of air bubbles in manometer can lead to inaccurate readings because of density difference.
5. Temperature readings should be taken only when the inlet & outlet temperature of both the liquids reach a constant value.

CHAPTER 4

SAMPLE CALCULATIONS

4.1 ROTAMETER CALIBRATION:

SMALL ROTAMETER

For 4 lpm (Table No. A1.1)

Observation No.1

Weight of water collected=10.2 kg

Time=156 sec

$\mathbf{m}_1$=0.06538 kg/sec

Observation No.2

Weight of water collected=10.4 kg

Time=159 sec

$\mathbf{m}_2$=0.06541 kg/sec

Observation No.3

Weight of water collected=10.8 kg

Time=165 sec

$\mathbf{m}_3$=0.06545 kg/sec

$\mathbf{m}=\frac{m1+m2+m3}{3}$=0.06542kg/sec

% difference=$\frac{\frac{4}{60}-0.06542}{4/60} \times 100$= 1.87%

4.2 PRESSURE DROP & FRICTION FACTOR CALULATIONS:

For $BRWTT_1$ having y_w=3.69 (Table No.A2.5)

m=0.2090 Kg/sec

Experimental friction factor

Area A=$\frac{\pi}{4} \times d_i^2 = \frac{\pi}{4} \times 0.022^2 = 3.8\times10^{-4}$ m^2

$v = \frac{m}{A\times\rho_w} = \frac{0.209}{3.8\times10^{-4}\times1000}$ =0.55m/sec

$\Delta P = (\rho_{ccl_4} - \rho_w) \times g \times \Delta h$ = (1603-1000) ×9.81×0.831=4916 N/m^2

$f_a = \frac{\Delta P\times d_i}{2\times\rho\times L\times v^2} = \frac{4915.72\times0.022}{2\times1000\times2.825\times0.55^2} = 63.29\times10^{-3}$

For viscosity calculation:

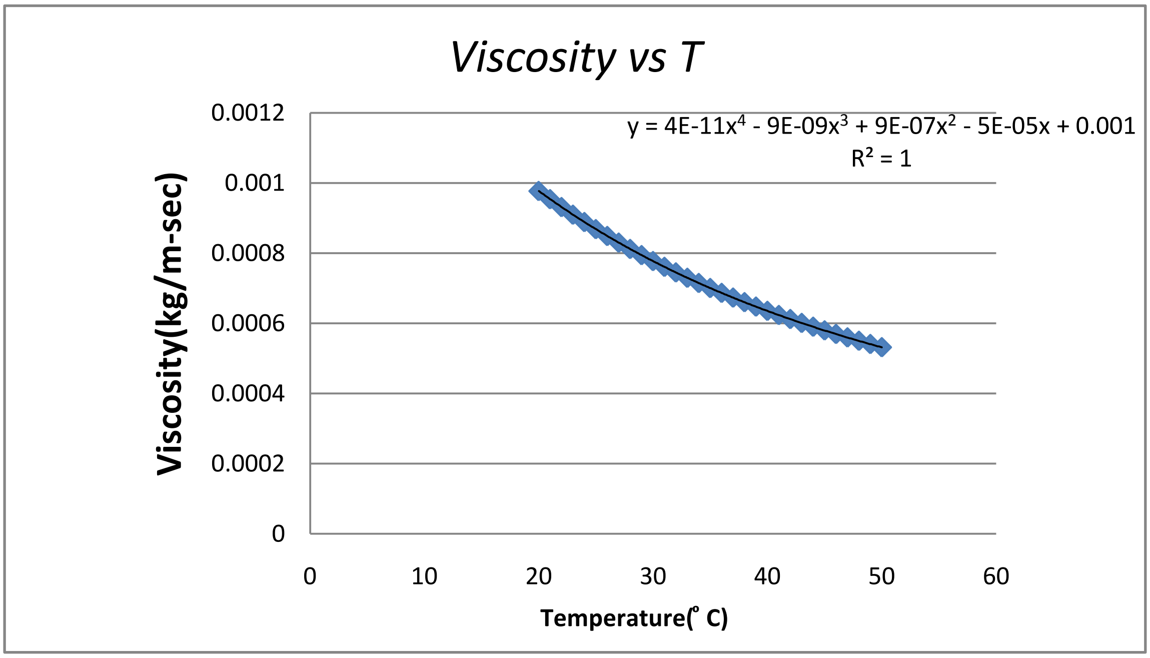


Fig 4.1 Viscosity vs. Temperature

μ =4×10^{-11}T^{4}-9×10^{-09}T^{3}+9×10^{-07}T^{2}-5×10^{-05}T+0.0017 (4.1)

Theoretical friction factor calculation for smooth tube:

$$Re=\frac{4\times m}{\pi\times d_i\times\mu}=\frac{4\times 0.2090}{\pi\times 0.022\times 7.02\times 10^4}=17218$$

f_o =0 .046 ×Re$^{-0.2}$ =0 .046 × 17218$^{-0.2}$=6.54×10^{-3}

$$\frac{f_a}{f_o}=\frac{63.29}{6.54}=9.68$$

4.3 HEAT TRANSFER COEFFICIENT CALCULATION:

For BRWTT$_1$ having y_w=3.69 (Table No.A3.5)

m$_c$= 0.209 kg/sec (750lph) & **m**$_h$=0.2715kg/sec

NOTE: Temperature correction has already been taken into account while giving data in Appendix.

T_1 = 38.5°C

T_2 =41.1°C

T_3 =51°C

T_4 =49.1°C

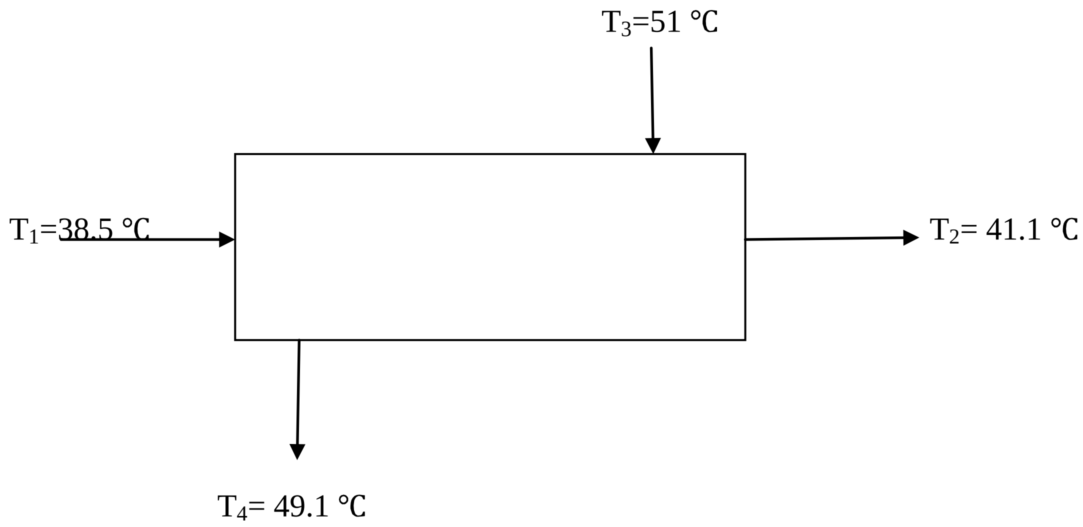


Fig. 4.2 Temperature in different RTDs

$\Delta T_1 = T_4 - T_1 = (49.1\text{-}38.5)\ °C = 10.5°C$

$\Delta T_2 = T_3 - T_2 = (51.0\text{-}41.1)\ °C = 9.9°C$

$LMTD = \frac{\Delta T1 - \Delta T2}{\ln\frac{\Delta T1}{\Delta T2}} = \frac{10.5-9.9}{\ln\frac{10.5}{9.9}} °C = 10.25°C$

$Q_1 = m_c \times C_{pc} \times (T_2 - T_1) = 0.2090 \times 4187 \times (41.1\text{-}38.5) = 2275W$

$Q_2 = m_h \times C_{ph} \times (T_3 - T_4) = 0.2715 \times 4187 \times (51.0\text{-}49.1) = 2160W$

Heat balance error = $\frac{2275-2160}{2160} \times 100 = 5.32\ \%$

$Q_{avg} = (Q_1+Q_2)/2 = (2275+2160)/2 = 2218\ W$

Heat transfer Area, $A_i = \pi \times d_i \times l = \pi \times 0.022 \times 2.43 = 0.1680\ m^2$

$U_i = \frac{Q}{Ai \times LMTD} = \frac{2218}{0.16795 \times 10.25} = 1289\ \ W/m^2°C$

$Re_a = \frac{4 \times m}{\pi \times di \times \mu} = \frac{4 \times 0.209}{\pi \times .022 \times 0.000669} = 18089$

h_i can be calculated using Eq. (3.1)

$$\frac{1}{U_i} = \frac{1}{h_i} + K \tag{4.2}$$

K is found from the Wilson chart ($1/U_i$ vs. $1/Re^{0.8}$) as the intercept on the y-axis.

$K=6.613\times10^{-4}$ (Refer Fig 3.7)

$\Rightarrow \frac{1}{h_i} = \frac{1}{U_i} - K = \frac{1}{1289} - 6.613\times10^{-4}$

$\Rightarrow h_a = 8721$ W/m^2 °C

Theoretical Calculation for smooth tube

$$Nu = 0.023 \times Re^{0.8} \times Pr^{1/3}$$

$$\Rightarrow \frac{h_i \times d_i}{k} = 0.023 \times Re^{0.8} \times Pr^{1/3}$$

$$\Rightarrow h_i = \frac{0.023 \times k}{d_i} \times Re^{0.8} \times Pr^{1/3} \quad (4.3)$$

For Prandtl Number calculation:

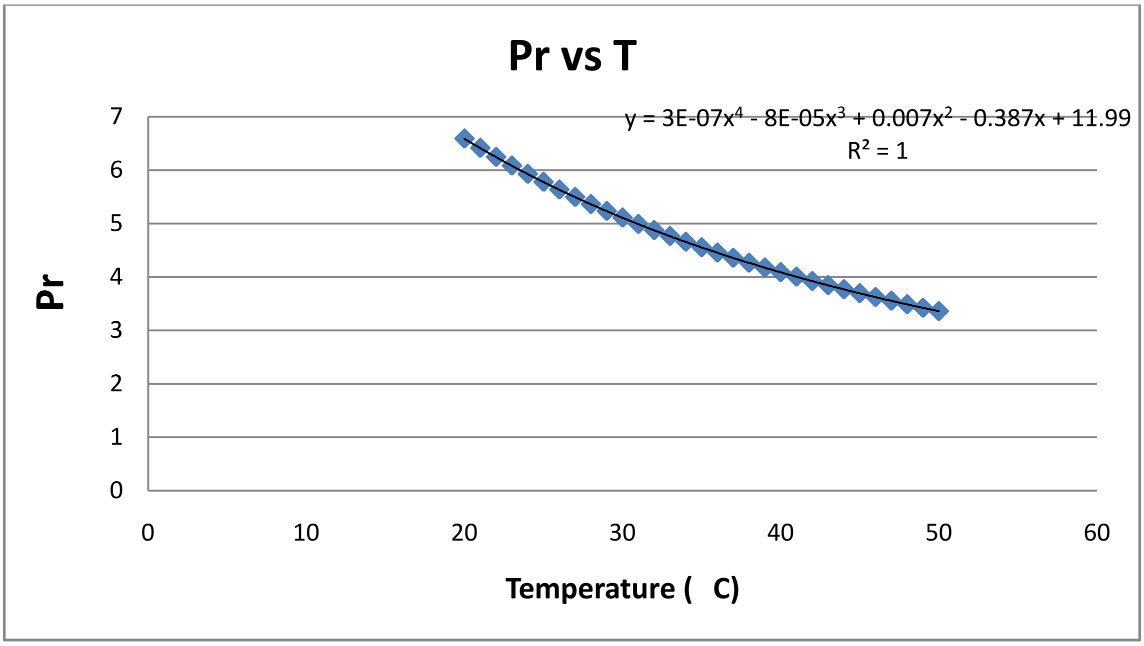


Fig 4.3 Prandtl Number vs. Temperature

$$Pr=3\times10^{-07}T^4-8\times10^{-05}T^3+0.0072\times T^2-0.3873\times T+11.995 \quad (4.4)$$

$T_{avg}=\frac{38.5+41.1}{2} = 39.8°C$

Pr (at $T=T_{avg}$) =3.695

$$h_o(h_i \text{ for smooth tube}) = \frac{0.023\times 0.6322}{0.022} \times 18089^{0.8} \times 3.695^{1/3}$$

$h_0 = 2601$W/m^2 °C

$$R_1 = \frac{h_a}{h0} = \frac{8721}{2601} = 3.35$$

To calculate f_a at Re=18089, use correlation for f_a vs. Re for $BRWTT_1$ having y_w=3.69

$$f_a= 0.9409\times Re^{(-0.277)} =0.06226$$

For equal pumping power,

$(f\times Re^3\times A_x)_a = (f\times Re^3\times A_x)_o$ (4.5)

$A_{xo}=\frac{\pi}{4}\times d_i^2=\frac{\pi}{4}\times 0.022^2 = 3.80\times 10^{-4}$

$A_{xa}= \frac{\pi}{4}\times d_i^2 - t\times w = \frac{\pi}{4}\times 0.022^2 - 0.002\times 0.016 = 3.48\times 10^{-4}$

$f_o\times Re_o^3 = (f_a\times Re_a^3\times A_{xa}/A_{xo})$

$Re_o = 39341$

h_o at Re_o(Equivalent Reynolds number in the smooth tube for same pumping power)

$$h_o = \frac{0.023\times k}{d_i}\times Re^{0.8}\times Pr^{\frac{1}{3}}$$

$$h_o = \frac{0.023\times 0.6322}{.022}\times 39341^{0.8}\times 3.695^{1/3} = 4843\ W/m^{2\circ}C$$

$R_3 = \frac{h_a}{h_o} = \frac{8721}{4843} = 1.80$

CHAPTER 5

RESULTS & DISCUSSION

5.1 FRICTION FACTOR RESULTS:

Tables A.2.1-A.2.10 represents the friction factor results & f_a/f_o values for all the cases. As shown in fig.5.1, except at low Re, the difference between f_{exp} & f_{theo} is limted to ±10%, so we can easily assume that the friction factor equations hold true for our experimental setup. Higher deviation between f_{exp} & f_{theo} for low Re is due to limitations of experimental setup. As the ΔH values were very small (0.1-0.8cm) for low Re & the manometer's least count was 0.1cm, so we cannot measure those low pressure drops with higher accuracy.

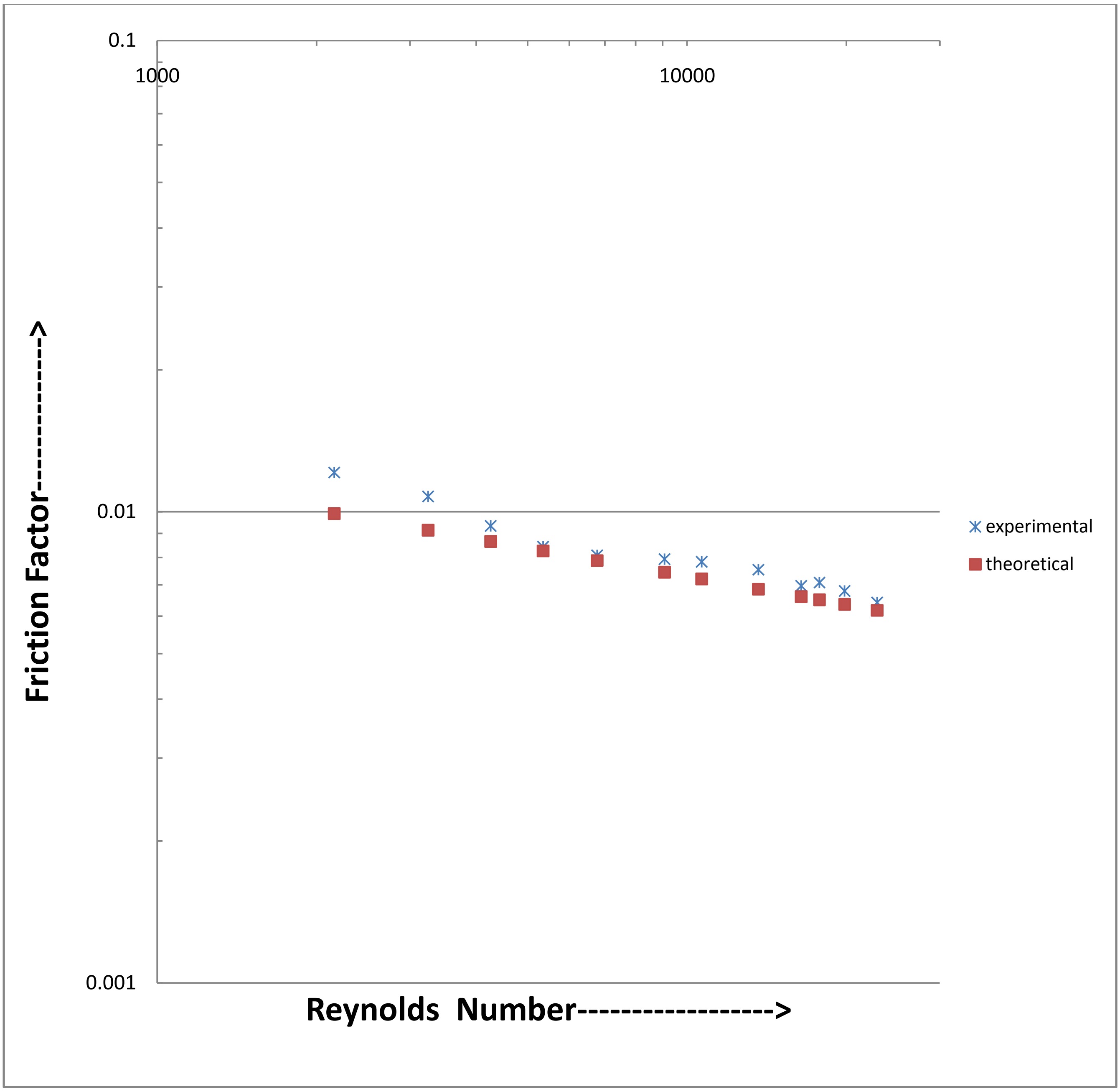


Fig 5.1 Friction Factor vs. Reynolds number for Smooth Tube

Fig 5.2 shows the variation of friction factor (f_a) with Reynolds Number for Smooth tube, Reduced width twisted tape (RWTT), Baffled Reduced width twisted tape ($BRWTT_1$), Baffled Reduced width twisted tape with holes ($BRWTT_2$) for different twist ratios (y_w=3.69, y_w=4.39, y_w=5.25). As the twist ratio decreases, a higher degree of swirl is created which leads to higher pressure drop & hence higher friction factor. In case of $BRWTT_1$ & $BRWTT_2$, a much higher friction factor is observed because of increase in degree of turbulence created by the respective tapes.

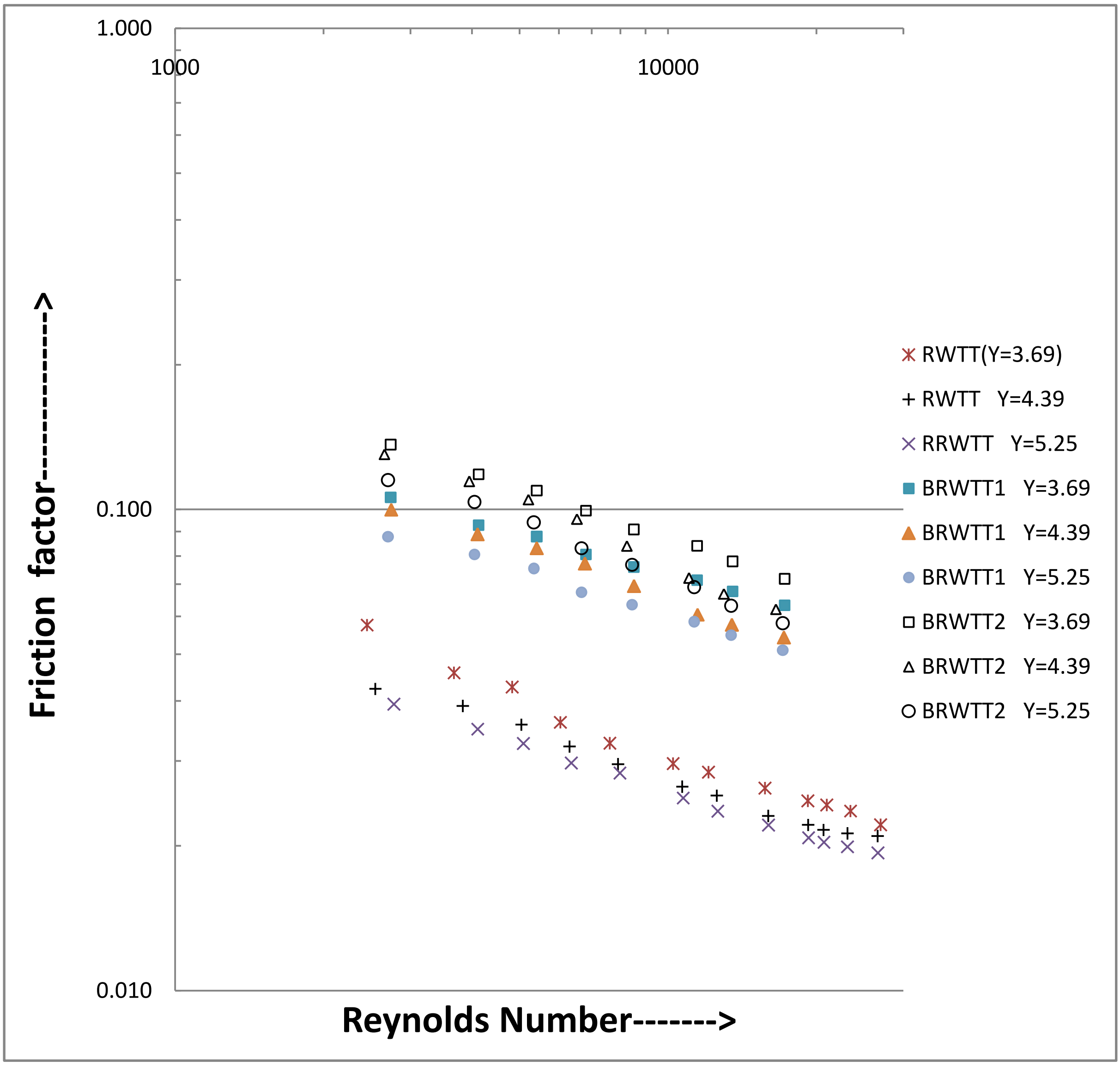


Fig 5.2 Friction factor vs. Reynolds number for Smooth tube, RWTT, $BRWTT_1$ & $BRWTT_2$

Fig 5.3 shows the variation of f_a/f_o with Reynolds number for RRWTT, $BRWTT_1$ & $BRWTT_2$.

1. f_a/f_o increases with decrease in twist ratio due to increase in swirl flow created with decreasing twist ratio.
2. f_a/f_o is found to be maximum for $BRWTT_2$ (8.86-14.44) followed by $BRWTT_1$ (7.79-11.23) followed by RWTT (3.23-5.96) because degree of turbulence decreases in the same order.

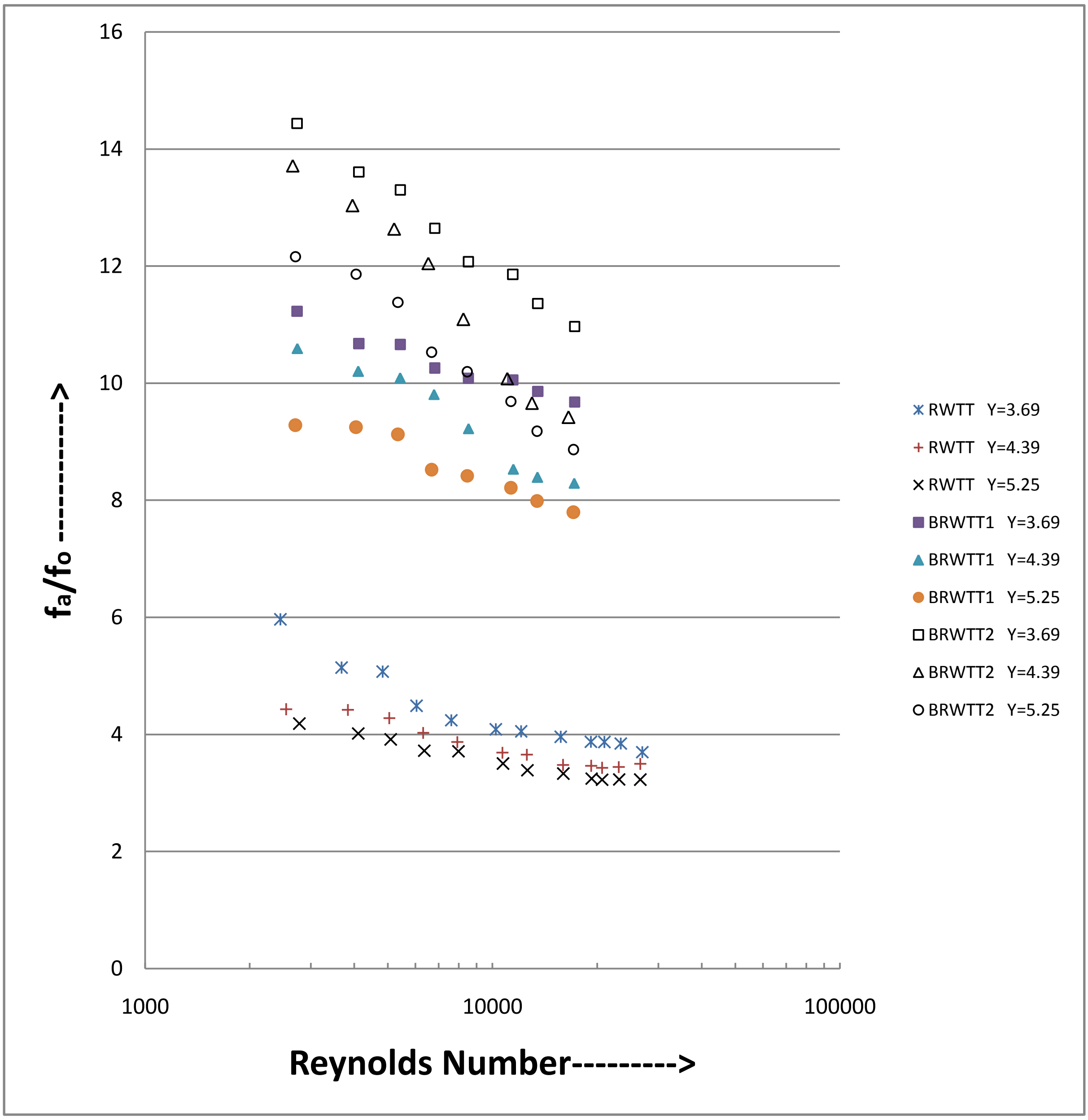


Fig 5.3 f_a/f_o vs. Reynolds Number for RRWTT, $BRWTT_1$ & $BRWTT_2$

Fig 5.4 shows the correlations for friction factor for different twisted tapes for RRWTT, $BRWTT_1$ & $BRWTT_2$. These correlations were used while calculating performance evaluation criteria R_3.

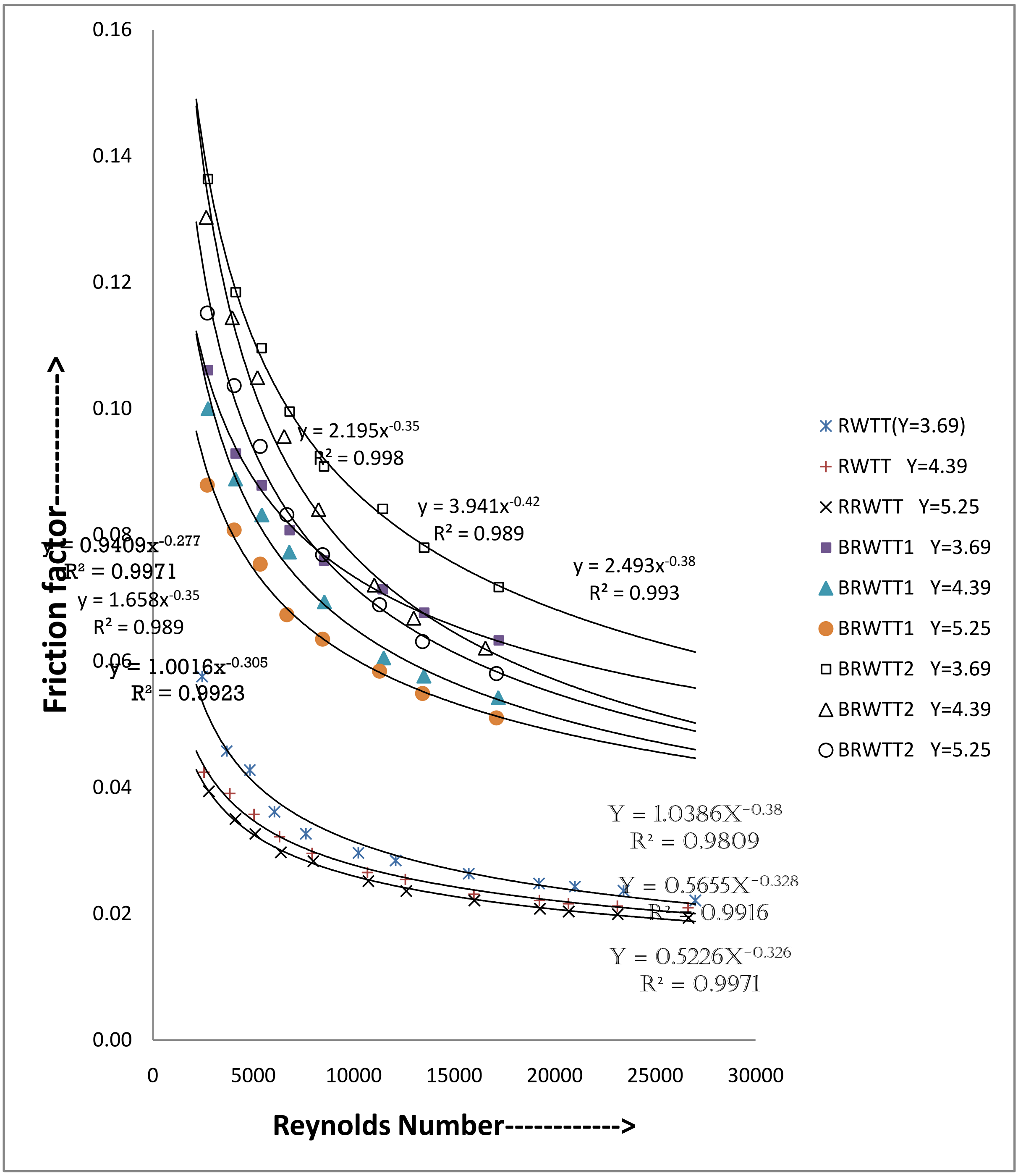


Fig 5.4 Correlations for variation of Friction factor with Reynolds Number

Table 5.1 gives correlations for variation of friction factor with Reynolds number for different twisted tapes alongwith the correlation coefficient, R^2 based on regression analysis.

As we can see form the correlations it is quite clear that friction factor is increasing with decrease in twist ratio. Also, for a given twist ratio friction factor is increasing in the sequence RWTT<$BRWTT_1$<$BRWTT_2$.

As the R^2value is very close to 1, so we can easily make out that the correlation holds true for respective twisted tapes in the given range of Reynolds Number.

Table 5.1 Correlations for Friction Factor for different twisted tapes

SI No.	y_w	Correlation, f_a=	R^2
	RWTT		
1	3.69	$1.0386\times Re^{-0.380}$	0.9809
2	4.39	$0.5655\times Re^{-0.328}$	0.9916
3	5.25	$0.5226\times Re^{-0.326}$	0.9971
	$BRWTT_1$		
4	3.69	$0.9409\times Re^{-0.277}$	0.9971
5	4.39	$1.6585\times Re^{-0.351}$	0.9890
6	5.25	$1.0016\times Re^{-0.305}$	0.9923
	$BRWTT_2$		
7	3.69	$2.1954\times Re^{-0.35}$	0.9981
8	4.39	$3.941\times Re^{-0.428}$	0.9895
9	5.25	$2.4939\times Re^{-0.385}$	0.9938

5.2 HEAT TRANSFER COEFFICIENT RESULTS:

Table A.3.1-A.3.10 gives the heat transfer results for RWTT, $BRWTT_1$ & $BRWTT_2$ along with the corresponding performance evaluation criteria R_1 & R_3 for each of the readings. As depicted in fig.5.5, there is a very small difference between h_{exp} & h_{theo} , so we can easily assume that the heat transfer equations hold true for our experimental setup. Higher deviation (-108.82%) between h_{exp} & h_{theo} for low Reynolds number can be attributed to the phenomenon of natural convection taking place along with forced convection. The phenomenon of natural convection is negligible in comparison to forced convection for higher Re but is significant at low Re. The percentage difference for Re>5000 were found to be well within ±8% for most of the readings. This can be taken as an indication of heat transfer results in the case of smooth tube, to be reasonably accurate.

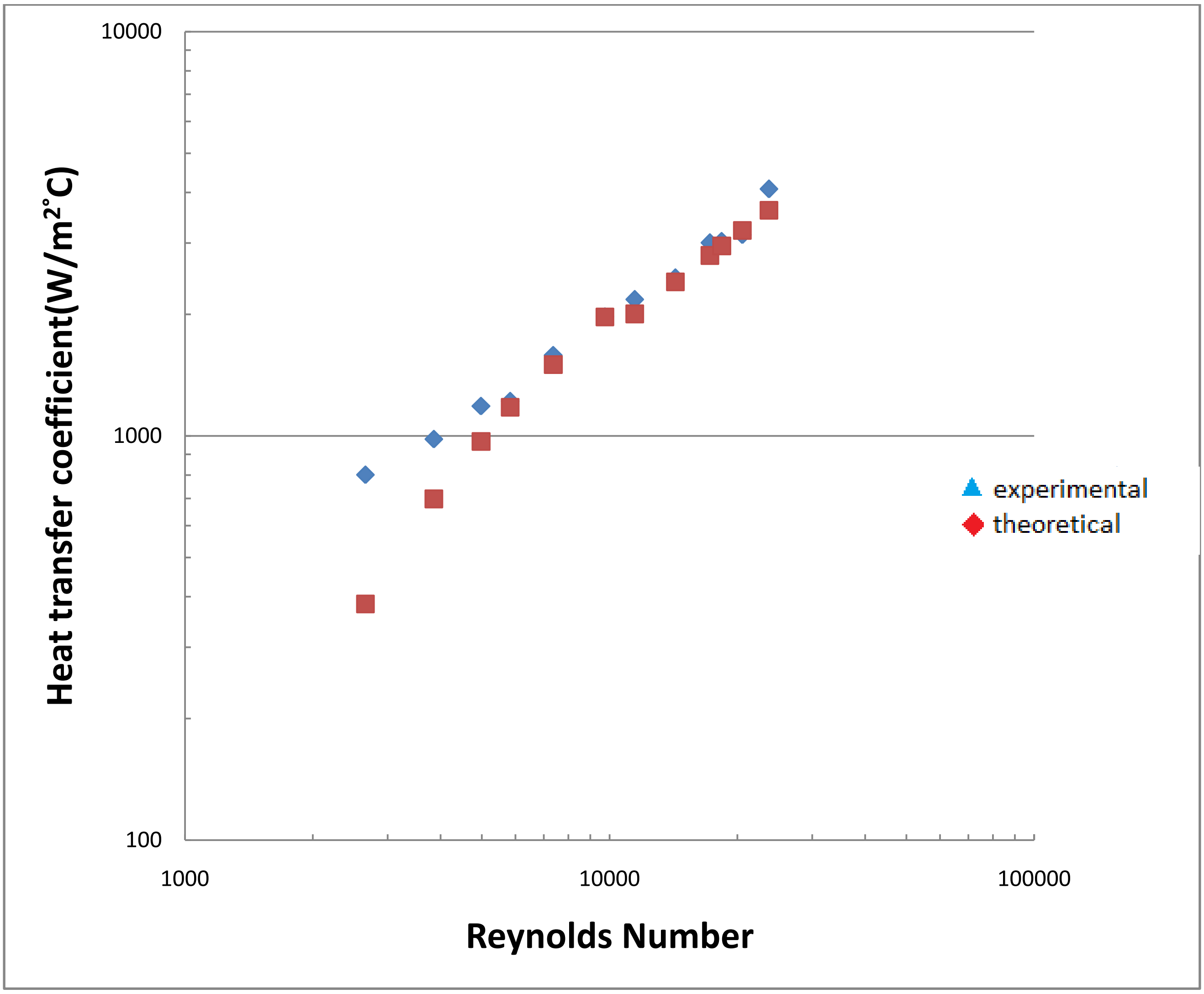


Fig 5.5Heat transfer coefficient vs. Reynolds Number for smooth tube

Fig 5.6 shows the variation of heat transfer coefficient (h_a) with Reynolds Number for Smooth tube, Reduced width twisted tape (RWTT), Baffled Reduced width twisted tape ($BRWTT_1$), Baffled Reduced width twisted tape with holes ($BRWTT_2$) for different twist ratios (y_w=3.69, y_w=4.39, y_w=5.25). As the twist ratio decreases, a higher degree of swirl is created which increases turbulence & hence the heat transfer coefficient increases as the twist ratio decreases. In case of $BRWTT_1$ & $BRWTT_{2,}$ a much higher heat transfer coefficient is observed because of increase in degree of secondary flow created which disturbs the entire thermal boundary layer & hence the heat transfer coefficient increases as the twist ratio decreases.

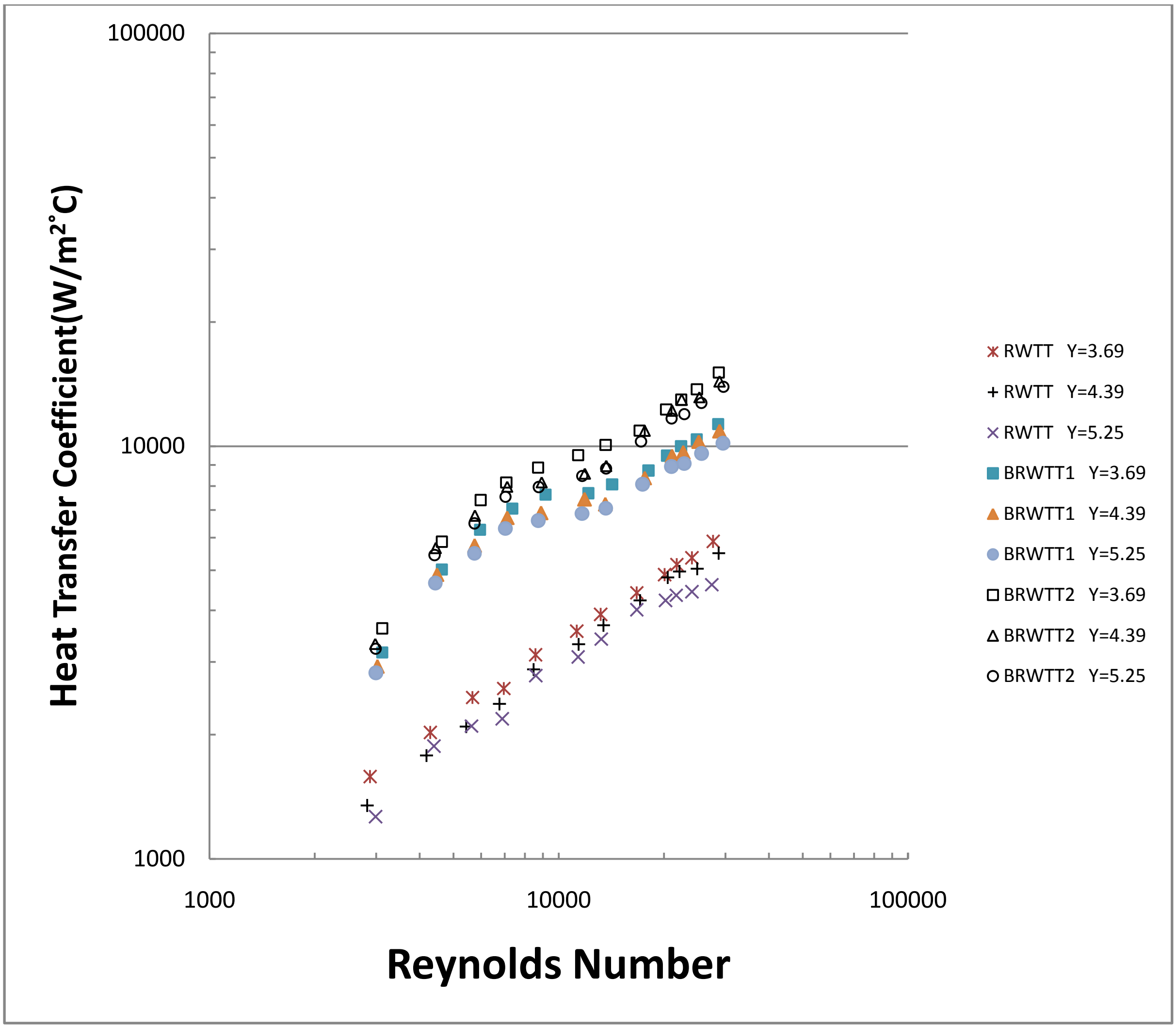


Fig 5.6 Heat transfer coefficient vs. Reynolds Number for Smooth tube, RWTT, $BRWTT_1$ & $BRWTT_2$

Plot for Performance evaluation criteria, R_1 (based on constant flow rate) vs. Reynolds number for different tapes is shown in Fig 5.7.

1. Maximum R_1 at a given condition is observed for $BRWTT_2$ & then decreases for $BRWTT_1$ & RWTT. From this we can infer that $BRWTT_2$ is the best design & is giving better than previously used designs like RWTT.
2. Maximum Value of R_1 is observed for lowest Reynolds Number & then decreases with increasing Reynolds Number. This is because as the Reynolds number increases, degree of turbulence increases in the smooth tube itself as the Reynolds number exponent in Nusselt number increases from 0.33 to 0.67 to 0.80 from laminar to transition to turbulent flow. So when we use an enhancement technique, the relative effect of enhancement technique (in this case twisted tape) to enhance secondary flow is not that high. This indicates that such tapes are much more effective at lower Reynolds number than that higher values.

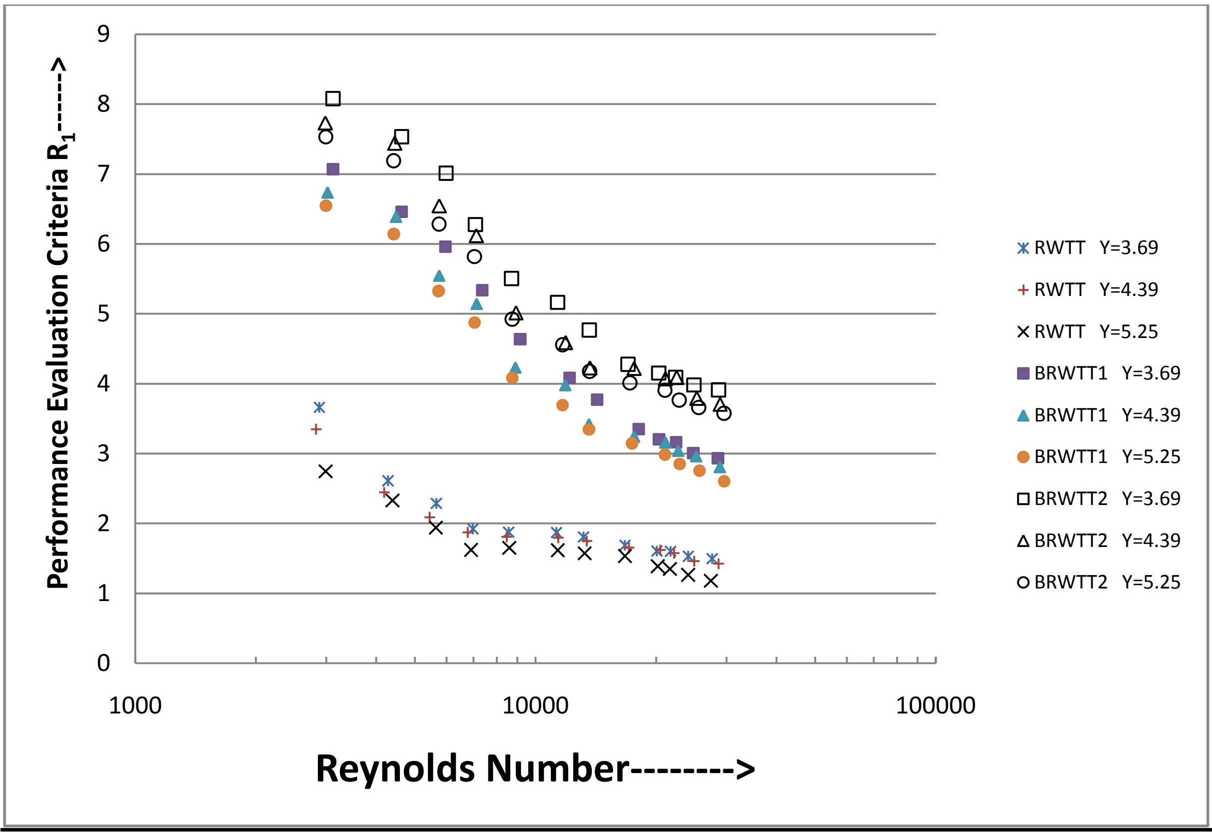


Fig 5.7 Performance evaluation criteria, R_1 vs. Reynolds Number for RWTT, $BRWTT_1$, $BRWTT_2$

Plot for Performance evaluation criteria, R_3 (based on constant pressure drop) vs. Reynolds number for different tapes is shown in Fig 5.8. For obtaining R_3, correlation between friction factor & Reynolds Number for respective tape is used.

1. Maximum R_3 at a given condition is observed for $BRWTT_2$ & then decreases for $BRWTT_1$ & RWTT. From this we can infer that $BRWTT_2$ is the best design & is working better than previously used designs like RWTT.
2. For RWTT, R_3 is found to increase with decrease in twist ratio y_w. But for $BRWTT_1$ & $BRWTT_2$, the values of R_3 were found to be higher for tapes with twist ratio y_w=4.39 & minimum for the twist ratio of y_w=3.69. This is possibly because of a higher increase of frictional losses in the latter case without corresponding increase in the heat transfer coefficient.
3. $BRWTT_1$ & $BRRWTT_2$ with y_w=4.39 were found to be the best on the basis of performance evaluation criterion R_3.
4. Maximum R_3 is observed for Re_0≈10000 as at this value of Re, the exponent of Re in Nusselt number changes from 0.67 to 0.8.

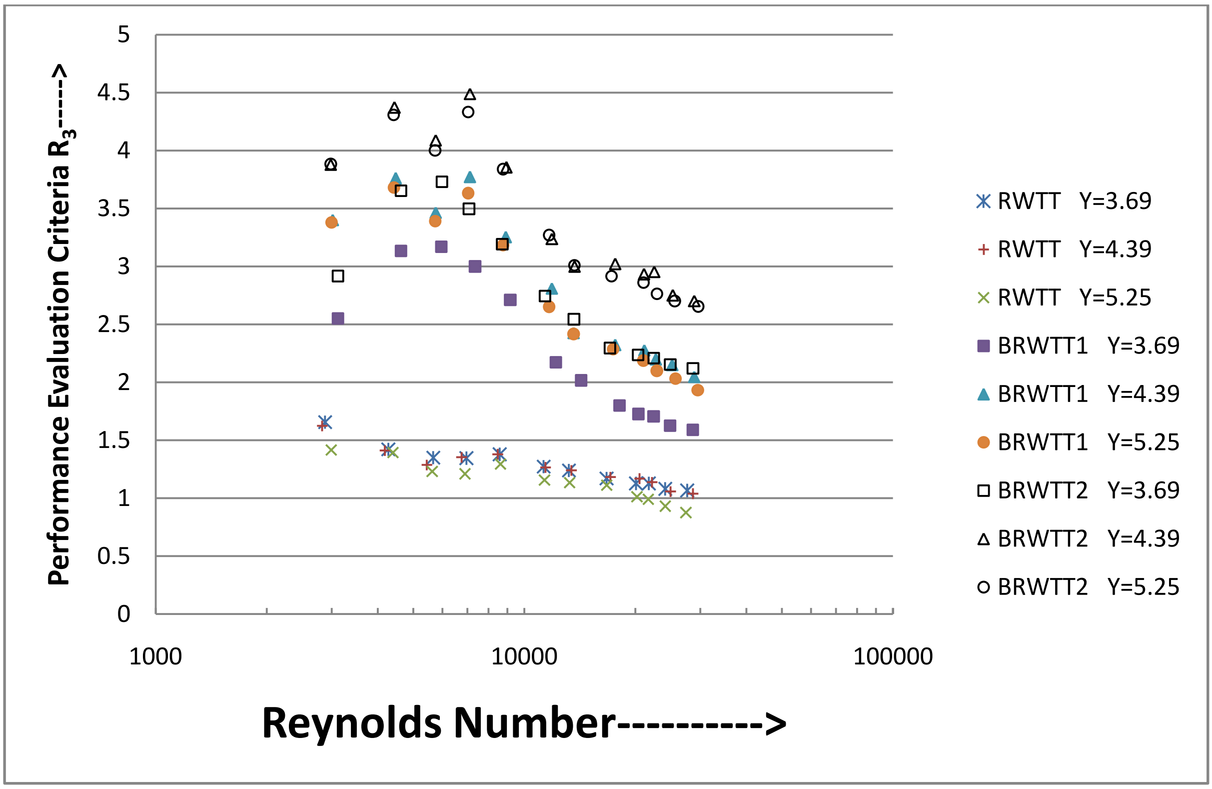


Fig 5.8 Performance evaluation criteria, R_3 vs. Reynolds Number for RWTT, $BRWTT_1$, $BRWTT_2$

5.3 TESTING OF EXPERIMENTAL DATA FOR REPEATABILITY:

Heat transfer experiments were done again for $BRWTT_2$ in the Reynolds number range 2500-7500 (Table Nos.A.4.1-A.4.3) to verify the previously obtained results. While repeating the experiment, the values for heat transfer coefficient were found to be well within ±5% error when compared to previous values. This can be taken as a degree of the accuracy of experimental work. Fig 5.9 shows the two sets of heat transfer coefficient values. Data points for the repeated section are shown as hollow points.

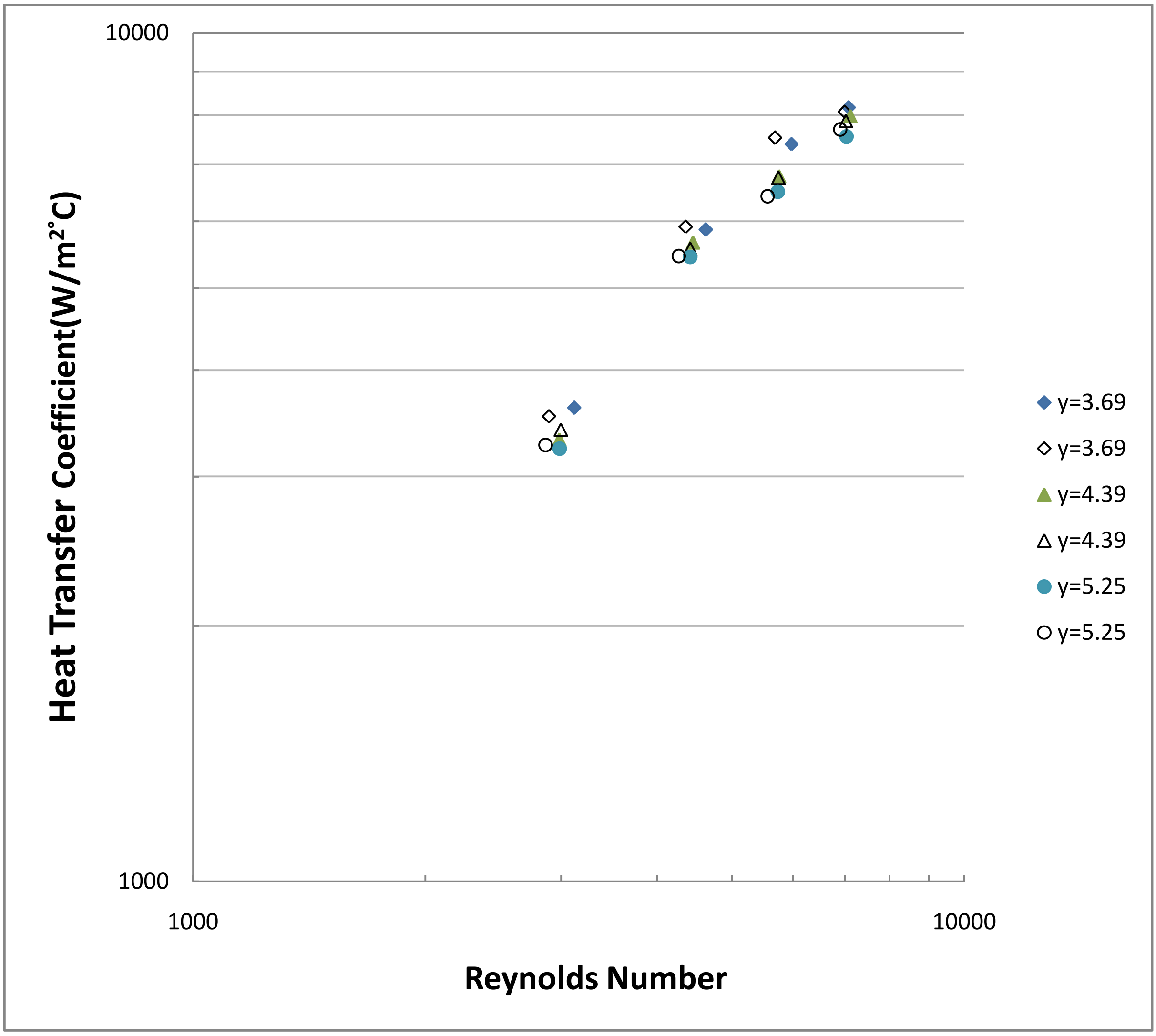


Fig 5.9 Heat transfer coefficient vs. Reynolds number for $BRWTT_2$ for repeatability

CHAPTER 6

CONCLUSION

The range of Performance evaluation criteria R_1 (based on constant mass flow rate) & R_3 (based on constant pumping power), & fa/fo for different tapes used is given below:

Table 6.1 Range of R_1, R_3, f_a/f_o for different twisted tapes.

SI No.	Y	Range of R_1	Range of R_3	Range of f_a/f_o
	RWTT			
1	3.69	1.50-3.66	1.07-1.66	3.70-5.96
2	4.39	1.43-3.35	1.04-1.62	3.43-4.43
3	5.25	1.18-2.75	0.88-1.42	3.23-4.18
	$BRWTT_1$			
4	3.69	2.94-7.07	1.59-3.17	9.68-11.23
5	4.39	2.81-6.73	2.05-3.77	8.29-10.59
6	5.25	2.61-6.55	1.93-3.68	7.79-9.28
	$BRWTT_2$			
7	3.69	3.91-8.08	2.12-3.73	10.97-14.44
8	4.39	3.71-7.73	2.70-4.49	9.42-13.71
9	5.25	3.58-7.53	2.65-4.33	8.86-12.16

1. For same twist ratio, Baffled reduced width twisted tape with holes & Baffled reduced width twisted tape shows higher heat transfer coefficient & friction factor increase because of higher degree of turbulence created.
2. On the basis of performance evaluation criteria R_1 & R_3, we can say that Baffled reduced width twisted tape with holes ($BRWTT_2$) & Baffled reduced width twisted tape ($BRWTT_1$) shows better performance than previously studied twisted tapes like reduced width twisted tapes (RWTT).
3. For same twist ratio, Baffled reduced width twisted tape with holes & Baffled reduced width twisted tape gives higher heat transfer coefficient than the reduced width twisted tapes.

4. The correlations derived from friction factor values have R^2(Correlation coefficient) values very close to 1. So, the correlations can be used for finding friction factor values for respective designs in the given range of Reynolds number.
5. With decrease in twist ratio, heat transfer coefficient increases but at the same time pressure drop also increases.
6. For RWTT, R_3 is found to increase with decrease in twist ratio y_w. But for $BRWTT_1$ & $BRWTT_2$, the values of R_3 were found to be higher for tapes with twist ratio y_w=4.39 & minimum for the twist ratio of y_w=3.69. This is possibly because of a higher increase of frictional losses in the case of y_w=3.69 without a corresponding increase in heat transfer coefficient.
7. On the basis of performance evaluation criterion $R_{3,}$ twisted tapes-$BRWTT_1$ & $BRRWTT_2$ with y_w=4.39 were found to be the best tape for heat transfer augmentation.

CHAPTER 7

SCOPE FOR FUTURE WORK

Further studies can be done using this study as base. Some of the possibilities are mentioned below:

1. Experimental work can be done at **low Reynolds number** using viscous liquids, as the tapes have shown comparatively better results at low Reynolds number.
2. **Distance between two consecutive baffles & holes** can be varied to see their effect on heat transfer & friction factor.
3. **Geometry of baffles**:
   a. Circular baffles instead of rectangular baffles can be used.
   b. Baffles can be kept at an angle to flow of liquid instead of putting them perpendicular to flow of liquid.
   c. Size of baffles can be varied.
4. The proposed designs can be used for **cooling** of liquids.

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APPENDIX

A.1. CALIBRATION

A.1.1 SMALL ROTAMETER CALIBRATION

	Observation 1			Observation 2			Observation 3				
Rotameter Reading (lpm)	**Wt (kg)**	**Time (sec)**	**m (Kg/sec)**	**Wt (kg)**	**Time (sec)**	**m (Kg/sec)**	**Wt (kg)**	**Time (sec)**	**m (Kg/sec)**	**Average m (Kg/sec)**	**% diff**
1	10.6	650	0.0163	10.8	652	0.0166	10.4	630	0.0165	0.0165	1.24
2	10.3	313	0.0329	10.5	320	0.0328	10.1	300	0.0337	0.0331	0.61
3	11	220	0.0500	11.4	230	0.0496	11.2	226	0.0496	0.0497	0.58
4	10.2	156	0.0653	10.4	159	0.0654	10.8	165	0.0655	0.0654	1.88
5	10.3	125	0.0824	10.9	133	0.0820	11	135	0.0815	0.0819	1.67

A.1.2 LARGE ROTAMETER CALIBRATION

	Observation 1			Observation 2			Observation 3				
Rotameter Reading (Kg/hr)	**Wt (kg)**	**Time (sec)**	**m (Kg/sec)**	**Wt (kg)**	**Time (sec)**	**m (Kg/sec)**	**Wt (kg)**	**Time (sec)**	**m (Kg/sec)**	**Average m (Kg/sec)**	**% diff**
400	11.35	109	0.1041	10.89	105	0.1037	10.51	102	0.1030	0.1036	6.74
500	11.90	85	0.1400	11.50	84	0.1369	11.80	85	0.1388	0.1386	0.23
600	12.80	78	0.1641	12.30	75	0.1640	11.40	70	0.1629	0.1637	1.81
750	12.23	59	0.2073	12.01	57	0.2107	11.70	56	0.2089	0.2090	-0.31
900	11.40	45	0.2533	10.58	42	0.2519	10.30	41	0.2512	0.2523	-0.86
1000	10.96	40	0.2740	9.70	36	0.2694	10.10	37	0.2723	0.2721	2.03
1100	10.92	35	0.3120	11.15	37	0.3014	10.70	36	0.2972	0.3035	0.66
1250	11.53	33	0.3494	10.53	30	0.3510	8.67	25	0.3468	0.3492	-0.53

A.1.3 RTD CALIBRATION:

SI No.	Temperature Readings			
	T_1	**T_2**	**T_3**	**T_4**
1	20.0	20.2	19.9	20.1
2	20.0	20.2	19.9	20.1
3	19.9	20.1	19.8	20.0
4	20.0	20.2	19.9	20.1
5	19.9	20.1	19.8	20.0
Correction	**0.0**	**-0.2**	**+0.1**	**-0.1**

A.2. FRICTION FACTOR RESULTS:

A.2.1 STANDARDISATION OF SMOOTH TUBE (f vs. Re)

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_{exp}*1000	f_{theo}*1000	% diff
0.0165	23.9	1062	0.1	5.92	12.28	15.06	18.51
0.0331	24.0	2158	0.4	23.66	12.12	9.91	-22.35
0.0497	24.1	3245	0.8	47.32	10.77	9.13	-17.94
0.0654	24.0	4260	1.2	70.99	9.33	8.65	-7.86
0.0819	24.1	5349	1.7	100.6	8.42	8.26	-1.91
0.1034	24.2	6764	2.6	153.8	8.09	7.88	-2.64
0.1389	24.1	9068	4.6	272.1	7.93	7.43	-6.66
0.1637	24.0	10658	6.3	372.7	7.82	7.20	-8.69
0.2090	24.1	13642	9.9	585.6	7.54	6.85	-10.05
0.2522	24.0	16422	13.3	786.8	6.96	6.60	-5.39
0.2715	24.2	17768	15.7	928.7	7.08	6.50	-8.97
0.3035	24.1	19814	18.8	1112	6.79	6.36	-6.74
0.3491	24.2	22840	23.5	1390	6.41	6.18	-3.79

A.2.2 FRICTION FACTOR vs. Re FOR RWTT HAVING y_w =3.69

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	fa/fo
0.0331	29.70	2451	1.9	112.4	57.58	9.66	5.96
0.0497	29.70	3677	3.4	201.1	45.77	8.91	5.14
0.0654	29.60	4829	5.5	325.4	42.75	8.43	5.07
0.0819	29.60	6049	7.3	431.8	36.16	8.06	4.49
0.1034	29.50	7615	10.50	621.1	32.68	7.70	4.24
0.1389	29.50	10231	17.2	1018	29.65	7.26	4.09
0.1637	29.60	12080	22.9	1355	28.44	7.02	4.05
0.2090	30.50	15726	34.6	2047	26.35	6.66	3.96
0.2522	31.10	19219	47.4	2804	24.80	6.40	3.88
0.2715	31.80	21005	53.9	3188	24.31	6.28	3.87
0.3035	31.70	23430	65.5	3875	23.65	6.15	3.85
0.3491	31.80	27002	81.0	4792	22.11	5.98	3.70

A.2.3 FRICTION FACTOR vs. Re FOR RWTT HAVING y_w =4.39

m (kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	33.10	2547	1.4	82.82	42.43	9.58	4.43
0.0497	33.30	3837	2.9	171.6	39.04	8.83	4.42
0.0654	33.20	5040	4.6	272.1	35.75	8.36	4.28
0.0819	33.20	6313	6.5	384.5	32.20	7.99	4.03
0.1034	32.90	7919	9.5	562.0	29.56	7.64	3.87
0.1389	33.10	10681	15.4	911.0	26.55	7.20	3.69
0.1637	33.00	12560	20.5	1213	25.46	6.97	3.65
0.2090	32.80	15978	30.3	1792	23.08	6.64	3.48
0.2522	32.70	19243	42.3	2502	22.13	6.40	3.46
0.2715	32.60	20683	47.9	2834	21.61	6.30	3.43
0.3035	32.60	23119	58.8	3478	21.23	6.17	3.44
0.3491	32.70	26638	76.8	4543	20.96	5.99	3.50

A.2.4 FRICTION FACTOR vs. Re FOR RWTT HAVING y_w =5.25

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	35.80	2781	1.3	76.90	39.40	9.42	4.18
0.0497	35.00	4106	2.6	153.8	35.00	8.71	4.02
0.0654	32.10	5092	4.2	248.5	32.64	8.34	3.91
0.0819	32.00	6365	6.0	354.9	29.72	7.98	3.72
0.1034	31.70	7980	9.1	538.3	28.32	7.63	3.71
0.1389	31.70	10722	14.6	863.7	25.17	7.19	3.50
0.1637	31.60	12606	19.0	1124	23.60	6.96	3.39
0.2090	31.30	15996	29.0	1716	22.09	6.64	3.33
0.2522	31.20	19260	39.7	2348	20.77	6.39	3.25
0.2715	31.10	20698	45.1	2668	20.34	6.30	3.23
0.3035	31.10	23135	55.2	3265	19.93	6.16	3.23
0.3491	31.20	26663	70.8	4188	19.33	5.99	3.23

A.2.5 FRICTION FACTOR vs. Re FOR BRWTT$_1$ HAVING y_w =3.69

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	37.10	2739	3.5	207.0	106.07	9.45	11.23
0.0497	37.30	4124	6.9	408.2	92.88	8.70	10.67
0.0654	37.20	5418	11.3	668.4	87.83	8.24	10.66
0.0819	37.40	6811	16.3	964.2	80.74	7.87	10.26
0.1034	36.90	8518	24.4	1443	75.93	7.53	10.09
0.1389	36.90	11445	41.4	2449	71.37	7.10	10.06
0.1637	37.00	13508	54.5	3224	67.68	6.86	9.86
0.2090	36.90	17218	83.1	4916	63.29	6.54	9.68

A.2.6 FRICTION FACTOR vs. Re FOR BRWTT$_1$ HAVING y_w =4.39

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	37.20	2744	3.3	195.2	100.01	9.44	10.59
0.0497	37.10	4110	6.6	390.4	88.84	8.71	10.20
0.0654	37.20	5418	10.7	633.0	83.17	8.24	10.09
0.0819	37.20	6787	15.6	922.8	77.27	7.88	9.81
0.1034	37.00	8533	22.3	1319	69.40	7.53	9.22
0.1389	37.10	11485	35.1	2076	60.51	7.09	8.53
0.1637	36.90	13484	46.4	2745	57.62	6.87	8.39
0.2090	36.80	17188	71.2	4212	54.23	6.54	8.29

A.2.7 FRICTION FACTOR vs. Re FOR BRWTT$_1$ HAVING y_w =5.25

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	36.40	2706	2.9	171.6	87.88	9.47	9.28
0.0497	36.20	4045	6.0	354.9	80.77	8.74	9.24
0.0654	36.40	5343	9.7	573.8	75.39	8.26	9.12
0.0819	36.20	6669	13.6	804.5	67.36	7.91	8.52
0.1034	36.40	8444	20.4	1206.8	63.49	7.54	8.42
0.1389	36.10	11285	33.9	2005.3	58.44	7.12	8.21
0.1637	36.60	13413	44.2	2614.6	54.89	6.87	7.98
0.2090	36.50	17098	67.0	3963.3	51.03	6.55	7.79

A.2.8 FRICTION FACTOR vs. Re FOR BRWTT$_2$ HAVING y_w =3.69

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	37.10	2739	4.50	266.2	136.37	9.45	14.44
0.0497	37.30	4124	8.80	520.6	118.46	8.70	13.61
0.0654	37.20	5418	14.10	834.1	109.59	8.24	13.30
0.0819	37.40	6811	20.10	1189	99.56	7.87	12.65
0.1034	36.90	8518	29.20	1727	90.87	7.53	12.07
0.1389	36.90	11445	48.80	2887	84.12	7.10	11.85
0.1637	37.00	13508	62.80	3715	77.99	6.86	11.36
0.2090	36.90	17218	94.20	5572	71.75	6.54	10.97

A.2.9 FRICTION FACTOR vs. Re FOR BRWTT$_2$ HAVING y_w =4.39

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	35.40	2658	4.3	254.4	130.31	9.50	13.71
0.0497	34.90	3952	8.5	502.8	114.42	8.78	13.03
0.0654	35.00	5210	13.5	798.6	104.93	8.31	12.63
0.0819	35.10	6539	19.3	1142	95.60	7.94	12.04
0.1034	35.10	8249	27.0	1597	84.02	7.58	11.09
0.1389	34.80	11023	41.8	2473	72.06	7.15	10.08
0.1637	34.80	12987	53.8	3183	66.81	6.92	9.66
0.2090	34.70	16553	81.5	4821	62.07	6.59	9.42

A.2.10 FRICTION FACTOR vs. Re FOR BRWTT$_2$ HAVING y_w =5.25

m(kg/sec)	T(°C)	Re	ΔH(cm)	ΔP(N/m^2)	f_a*1000	f_o*1000	f_a/f_o
0.0331	36.40	2706	3.8	224.8	115.16	9.47	12.16
0.0497	36.20	4045	7.7	455.5	103.65	8.74	11.86
0.0654	36.40	5343	12.1	715.8	94.05	8.26	11.38
0.0819	36.20	6669	16.8	993.8	83.21	7.91	10.53
0.1034	36.40	8444	24.7	1461	76.87	7.54	10.19
0.1389	36.10	11285	40.0	2366	68.95	7.12	9.69
0.1637	36.60	13413	50.8	3005	63.09	6.87	9.18
0.2090	36.50	17098	76.2	4508	58.04	6.55	8.86

A.3. HEAT TRANSFER RESULTS:

A.3.1 STANDARDISATION OF SMOOTH TUBE (h_i vs. Re)

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_i(expt)	h_i(theo)	% diff
0.0331	24.1	43.1	65.0	62.3	29.30	2660	523.8	801.4	383.8	-108.82
0.0497	24.2	39.6	66.9	63.3	32.85	3853	595.0	981.1	698.5	-40.45
0.0654	24.0	38.1	66.0	62.9	33.10	4981	664.3	1185	968.4	-22.35
0.0819	24.1	31.7	51.1	49.0	22.04	5829	674.7	1218	1177	-3.52
0.1034	23.7	32.3	58.5	55.2	28.77	7370	773.3	1583	1500	-5.50
0.1389	23.8	30.8	56.2	52.9	27.21	9750	855.8	1972	1968	-0.21
0.1637	24.2	30.2	56.4	52.8	27.38	11462	891.8	2174	2004	-8.50
0.2090	23.8	28.4	53.3	49.8	25.45	14279	936.2	2458	2405	-2.23
0.2522	24.0	28.2	54.3	50.4	26.25	17230	1006	3002	2794	-7.42
0.2716	23.9	27.4	50.5	47.1	23.15	18367	1009	3030	2949	-2.75
0.3035	24.1	27.8	55.3	51.2	27.30	20530	1021	3142	3224	2.51
0.3491	24.2	27.5	53.5	49.4	25.60	23718	1103	4077	3614	-12.83

A.3.2 HEAT TRANSFER COEFFICIENT vs. Re FOR RWTT HAVING y_w=3.69

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	31.1	44.0	53.0	51.4	13.89	2879	773.3	1583	432	3.66	5089	1.66
0.0497	31.5	42.7	54.8	52.7	16.23	4281	865.7	2025	776	2.61	7378	1.42
0.0654	32.0	42.6	57.5	54.9	18.61	5657	937.1	2464	1076	2.29	9575	1.35
0.0819	31.8	41.1	57.9	55.1	19.87	6969	954.9	2590	1347	1.92	11639	1.35
0.1034	31.4	39.0	56.0	53.1	19.25	8575	1018	3120	1662	1.88	14132	1.38
0.1389	31.4	36.8	54.0	51.0	18.37	11268	1062	3562	1900	1.87	18247	1.28
0.1637	31.7	35.9	51.4	48.7	16.24	13195	1090	3910	2159	1.81	21151	1.24
0.2090	31.9	34.9	48.4	46.1	13.85	16711	1127	4418	2614	1.69	26384	1.17
0.2522	31.7	34.8	51.3	48.5	16.65	20102	1154	4881	3033	1.61	31363	1.13
0.2715	32.0	34.9	51.0	48.3	16.20	21760	1170	5173	3227	1.60	33777	1.13
0.3035	31.8	34.2	49.8	47.1	15.45	24073	1179	5358	3508	1.53	37126	1.08
0.3491	31.9	34.0	49.5	46.8	15.20	27656	1203	5874	3921	1.50	42273	1.07

A.3.3 HEAT TRANSFER COEFFICIENT vs. Re FOR RWTT HAVING y_w =4.39

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	33.8	44.1	52.3	51.0	12.15	2827	712.3	1347	402.6	3.35	4668	1.62
0.0497	34.0	42.3	51.7	50.2	12.49	4185	818.1	1782	728.4	2.45	6787	1.41
0.0654	34.2	40.6	50.9	49.2	12.50	5437	877.6	2091	1002	2.09	8713	1.29
0.0819	34.1	39.9	50.6	48.9	12.64	6764	924.0	2376	1270	1.87	10731	1.35
0.1034	34.2	39.1	50.9	48.9	13.20	8481	991.5	2879	1592	1.81	13316	1.38
0.1389	34.6	38.8	52.0	49.8	14.18	11405	1038	3313	1846	1.80	17667	1.26
0.1637	34.7	38.6	52.5	50.2	14.69	13425	1072	3682	2104	1.75	20642	1.24
0.2090	35.0	38.1	53.0	50.4	15.15	17113	1114	4231	2557	1.66	26021	1.18
0.2522	34.8	37.6	52.8	50.2	15.30	20521	1150	4810	2965	1.62	30946	1.17
0.2715	35.1	37.8	52.2	49.9	14.60	22198	1159	4968	3151	1.58	33355	1.14
0.3035	35.7	37.8	50.4	48.4	12.65	24943	1163	5042	3450	1.46	37281	1.06
0.3491	35.8	37.7	51.1	48.9	13.25	28686	1186	5499	3859	1.43	42601	1.04

A. 3.4 HEAT TRANSFER COEFFICIENT vs. Re FOR RWTT HAVING y_w =5.25

m (kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	29.1	49.8	67	64.4	25.17	2992	689.1	1266	460.7	2.75	4817	1.42
0.0497	29.5	47.2	67.1	63.8	26.45	4395	836.9	1874	803.6	2.33	6956	1.40
0.0654	29.8	44.1	65.8	62.3	26.74	5629	879.2	2100	1081	1.94	8810	1.23
0.0819	30.1	41.7	64.8	61.2	26.90	6893	893.4	2183	1348	1.62	10690	1.21
0.1034	30.6	40.1	62.6	58.9	25.29	8601	979.3	2779	1680	1.65	13206	1.30
0.1389	30.8	37.9	60.7	57	24.46	11348	1015	3083	1908	1.62	17209	1.16
0.1637	30.9	36.9	59.4	55.7	23.63	13249	1048	3413	2165	1.58	19951	1.14
0.2090	31	35.7	57.4	53.8	22.25	16711	1098	4011	2615	1.53	24903	1.11
0.2522	31.5	35.4	57.2	53.6	21.95	20226	1114	4226	3044	1.39	29884	1.02
0.2715	31.6	35	54.8	51.6	19.90	21670	1123	4358	3222	1.35	31919	0.99
0.3035	31.6	34.4	53.6	50.4	19.00	24073	1128	4436	3510	1.26	35290	0.93
0.3491	31.4	33.7	51.4	48.5	17.40	27456	1139	4623	3908	1.18	40012	0.88

A. 3.5 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_1$ HAVING y_w =3.69

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	41.4	49.0	52.3	51.4	6.04	3120	1023	3166.55	447.8	7.07	7121	2.55
0.0497	41.2	47.5	51.8	50.7	6.56	4622	1163	5030.12	778.4	6.46	10437	3.14
0.0654	39.9	45.8	51.6	50.2	7.84	5946	1219	6281.19	1053	5.96	13333	3.17
0.0819	39.5	44.6	51.5	49.9	8.53	7356	1246	7064.60	1323	5.34	16398	3.00
0.1034	39.1	43.4	50.8	49.2	8.68	9163	1262	7640.70	1646	4.64	20303	2.71
0.1389	38.8	42.1	50.4	48.7	9.08	12153	1263	7683.95	1881	4.08	26721	2.17
0.1637	38.6	41.6	50.6	48.8	9.59	14237	1274	8082.19	2142	3.77	31166	2.02
0.2090	38.5	41.1	51.0	49.1	10.25	18089	1289	8721.02	2601	3.35	39341	1.80
0.2522	34.2	37.4	52.8	49.8	15.50	20375	1304	9476.75	2957	3.20	44168	1.73
0.2715	35.8	38.3	51.8	49.1	13.40	22433	1314	10002.82	3162	3.16	48500	1.71
0.3035	35.3	37.6	50.4	48.0	12.75	24812	1320	10368.13	3444	3.01	53494	1.63
0.3491	35.5	37.6	51.3	48.7	13.45	28585	1334	11312.31	3854	2.94	61389	1.59

A. 3.6 HEAT TRANSFER COEFFICIENT vs. Re FOR $BRWTT_1$ HAVING y_w =4.39

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	35.4	50.8	57.7	55.9	12.49	3023	997.0	2927	434.6	6.73	4975	3.40
0.0497	35.7	48.9	58.3	55.9	14.12	4479	1155	4885	764.6	6.39	7242	3.76
0.0654	35.8	45.6	55.7	53.3	13.46	5747	1197	5743	1036	5.55	9185	3.46
0.0819	35.7	44.4	55.9	53.3	14.33	7123	1234	6706	1304	5.14	11273	3.77
0.1034	36.0	43.0	55.6	52.9	14.64	8905	1240	6890	1628	4.23	13951	3.25
0.1389	36.2	41.8	55.7	52.9	15.26	11865	1257	7440	1869	3.98	18346	2.81
0.1637	35.1	39.7	53.9	51.1	15.08	13602	1251	7230	2113	3.42	20902	2.43
0.2090	36.7	39.8	52.1	49.7	12.65	17625	1281	8375	2581	3.24	26764	2.32
0.2522	36.6	39.2	51.6	49.2	12.50	21140	1304	9458	2994	3.16	31835	2.28
0.2715	36.7	38.9	50.6	48.3	11.65	22727	1307	9660	3176	3.04	34112	2.20
0.3035	36.3	38.3	50.1	47.8	11.65	25184	1318	10257	3462	2.96	37624	2.15
0.3491	36.2	38.1	50.7	48.3	12.35	28887	1327	10864	3868	2.81	42886	2.05

A. 3.7 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_1$ HAVING y_w =5.25

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	36.3	48.8	55.0	53.4	10.74	2997	984.5	2822	430.8	6.55	4826	3.38
0.0497	36.2	46.9	54.5	52.6	11.44	4427	1142	4658	758.8	6.14	7003	3.68
0.0654	36.1	44.9	53.9	51.8	12.04	5728	1186	5505	1034	5.32	8957	3.39
0.0819	35.7	42.9	52.7	50.5	12.13	7035	1220	6324	1297	4.88	10900	3.63
0.1034	35.6	41.2	51.5	49.3	11.92	8742	1230	6598	1615	4.09	13413	3.18
0.1389	36.0	40.0	50.4	48.3	11.32	11665	1239	6871	1859	3.70	17668	2.65
0.1637	35.6	39.3	50.6	48.4	12.03	13614	1246	7079	2114	3.35	20476	2.42
0.2090	36.0	39.0	50.9	48.6	12.25	17399	1274	8082	2571	3.14	25882	2.29
0.2522	36.5	38.8	50.1	47.9	11.35	21049	1293	8924	2990	2.98	31044	2.19
0.2715	37.1	39.3	51.1	48.8	11.75	22883	1296	9084	3182	2.85	33623	2.10
0.3035	37.3	39.5	52.5	50.0	12.85	25665	1306	9599	3483	2.76	37516	2.03
0.3491	37.5	39.5	53.1	50.5	13.30	29566	1316	10160	3897	2.61	42943	1.93

A. 3.8 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_2$ HAVING y_w =3.69

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	41.4	49.0	52.4	51.4	6.12	3120	1066	3616	447.8	8.08	7121	2.92
0.0497	41.2	47.5	51.9	50.7	6.63	4622	1202	5864	778.5	7.53	10437	3.66
0.0654	40.1	46.1	51.7	50.3	7.67	5969	1255	7397	1055	7.01	13383	3.73
0.0819	36.2	43.2	51.8	49.8	10.91	7082	1276	8168	1301	6.28	15804	3.50
0.1034	35.3	41.2	51.4	49.1	11.91	8719	1292	8880	1613	5.50	19347	3.19
0.1389	33.9	39.0	51.3	48.7	13.51	11355	1305	9521	1843	5.17	25013	2.75
0.1637	35.6	39.5	51.3	48.8	12.49	13638	1315	10076	2115	4.76	29891	2.54
0.2090	34.5	38.1	51.9	49.1	14.20	17037	1328	10918	2553	4.28	37114	2.29
0.2522	34.0	37.3	53.0	49.8	15.75	20321	1346	12275	2954	4.16	44052	2.24
0.2715	35.7	38.3	51.6	48.9	13.25	22414	1354	12944	3161	4.09	48459	2.21
0.3035	35.5	37.8	50.5	48.0	12.60	24900	1362	13745	3448	3.99	53678	2.16
0.3491	35.7	37.8	51.4	48.7	13.30	28686	1374	15090	3859	3.91	61600	2.12

A. 3.9 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_2$ HAVING y_w =4.39

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	34.9	49.7	55.8	54.1	11.42	2985	1039	3317	429.0	7.73	4917	3.88
0.0497	35.9	47.8	56.3	54.0	12.70	4448	1194	5664	761.2	7.44	7193	4.37
0.0654	35.6	45.9	55.7	53.3	13.36	5751	1236	6781	1036	6.54	9192	4.09
0.0819	35.5	44.5	56.0	53.3	14.42	7117	1271	7978	1304	6.12	11265	4.49
0.1034	36.1	43.3	55.6	52.9	14.43	8934	1276	8173	1631	5.01	13995	3.85
0.1389	36.5	41.7	54.6	51.9	14.11	11885	1286	8578	1870	4.59	18375	3.24
0.1637	35.4	40.0	53.7	50.9	14.58	13673	1294	8947	2117	4.23	21005	3.00
0.2090	36.7	39.9	51.9	49.5	12.40	17640	1328	10894	2582	4.22	26786	3.02
0.2522	36.5	39.2	51.7	49.2	12.60	21122	1345	12181	2994	4.07	31809	2.93
0.2715	36.1	38.4	50.2	47.8	11.75	22512	1354	12944	3166	4.09	33804	2.95
0.3035	36.5	38.6	50.6	48.2	11.85	25294	1356	13143	3467	3.79	37780	2.75
0.3491	36.2	38.2	51.0	48.5	12.55	28913	1368	14337	3869	3.71	42922	2.70

A. 3.10 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_2$ HAVING y_w =5.25

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	h_o	R_1	Re_o	R_3
0.0331	35.6	49.1	54.9	53.3	10.67	2988	1030	3235	429.4	7.53	4811	3.88
0.0497	35.7	47.0	54.6	52.6	11.64	4413	1183	5443	757.2	7.19	6982	4.31
0.0654	36.2	44.9	53.8	51.6	11.85	5733	1227	6496	1034	6.28	8964	4.00
0.0819	35.6	43.0	52.5	50.3	11.91	7035	1260	7548	1297	5.82	10900	4.33
0.1034	35.5	41.4	51.8	49.5	12.11	8749	1271	7955	1616	4.92	13423	3.84
0.1389	35.9	40.1	50.6	48.4	11.47	11665	1283	8473	1860	4.56	17668	3.27
0.1637	35.7	39.5	50.8	48.5	12.03	13650	1291	8830	2115	4.17	20527	3.01
0.2090	35.2	38.6	51.6	49.0	13.40	17218	1318	10272	2562	4.01	25625	2.92
0.2522	36.4	38.8	50.2	47.9	11.45	21031	1339	11676	2989	3.91	31019	2.86
0.2715	37.0	39.3	51.2	48.8	11.85	22864	1343	11970	3182	3.76	33595	2.77
0.3035	37.1	39.4	52.5	49.9	12.95	25599	1352	12733	3480	3.66	37424	2.70
0.3491	37.5	39.6	53.3	50.6	13.40	29591	1364	13939	3898	3.58	42978	2.65

A.4EXPERIMENTAL DATA FOR REPEATABILITY:

A.4.1 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_2$ HAVING y_w=3.69

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	From table A.3.8		% diff
									Re	h_a	
0.0331	35.3	45.4	50.2	48.8	8.41	2894	1059	3534	3120	3616	2.33
0.0497	36.1	44.9	50.8	49.2	9.03	4353	1204	5910	4622	5864	-0.78
0.0654	36.3	43.7	50.5	48.8	9.36	5681	1259	7524	5969	7397	-1.70
0.0819	35.9	42	50.6	48.5	10.47	6994	1274	8077	7082	8168	1.13

A.4.2 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_2$ HAVING y_w=4.39

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	From Table A.3.9		% diff
									Re	h_a	
0.0331	35.4	49.8	55.8	54.1	11.17	2999	1047	3407	2985	3317	-2.73
0.0497	35.1	47.6	55.3	53.3	12.21	4413	1189	5567	4448	5664	1.72
0.0654	35.5	45.8	55	52.8	12.83	5742	1235	6750	5751	6781	0.47
0.0819	35	43.4	54.1	51.6	13.43	7023	1269	7874	7117	7978	1.30

A.4.3 HEAT TRANSFER COEFFICIENT vs. Re FOR BRWTT$_2$ HAVING y_w=5.25

m(kg/sec)	T_1	T_2	T_3	T_4	LMTD	Re	U_i	h_a	From Table A.3.10		% diff
									Re	h_a	
0.0331	34.1	45.4	50.1	48.8	8.77	2865	1034	3269	2988	3235	1.05
0.0497	34.2	44.2	50.4	48.8	9.81	4260	1184	5455	4413	5443	0.22
0.0654	34.5	42.8	50.7	48.8	10.79	5555	1224	6418	5733	6496	-1.20
0.0819	34.4	41.9	51.4	49.2	11.95	6900	1264	7693	7035	7548	1.92