EXPERIMENTAL STUDIES ON PRESSURE DROP FOR FLOW THROUGH TUBES USING TWISTED GALVANISED IRON WIRE INSERT WITH AND WITHOUT BAFFLES

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

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by

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Under the Guidance

of

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National Institute of Technology

Rourkela

CERTIFICATE

This is to certify that the thesis entitled, **“EXPERIMENTAL STUDIES ON PRESSURE DROP FOR FLOW THROUGH TUBES USING TWISTEDGALVANISED IRON WIRE INSERT WITH AND WITHOUT BAFFLES”** submitted by **SARTHAK SUBUDHI** (110CH0471)in partial fulfilments of the requirements for the award of Bachelor of Technology Degree in Chemical Engineering at National Institute of Technology, Rourkela 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 work, "Experimental studies on pressure drop for flow through tubes using Galvanised Iron wire insert with and without baffles" was undertaken in a view of studying the effect of turbulence on pressure drop of a heat exchanger. Most of the commercial, domestic and industrial applications where conversion or utilisation of energy is involved require a heat exchange process. This project deals with the introduction of three and four Galvanised Iron wires with and without baffles as passive augmentation device. The baffles used in the experiment were made up of thin tin sheets. By introduction of these inserts in the flow path of liquid in the inner tube of heat exchanger the effect of turbulence on pressure drop was observed. It was compared with the value of smooth tube. The effect of baffle was also taken into account and a comparative study was made on the basis of varying baffle space ($\beta = 24,12 \,\&\, 6cm$). The flow rate was varied from 350-1250 litres/hour. All the readings and results were compared with the standard data from the smooth tube. The friction factor for inserts without baffles was in range of 1.31-4.28 and with baffles was in range of 2.38-21.87. The pressure drop reading was found to increase with decreasing baffle space. The friction factor was highest for the four wire insert with 6cm baffle spacing.

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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 with insert, Dimensionless
f_{exp}	Experimental friction factor without insert, Dimensionless
f_{theo}	Theoretical friction factor without insert, Dimensionless
h	Heat transfer coefficient, $W/m^2 °C$
L	Pressure tapping to pressure tapping 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
U_i	Overall heat transfer coefficient based on inside surface area, $W/m^2 °C$
v	Flow velocity, m/s^2
P	Pumping power
R_1	Performance evaluation criteria based on constant flow rate

GREEK LETTERS

ΔT_i	Approach temperature difference
Δh	Height difference in manometer, m
ΔP	Pressure difference across heat exchanger, N/m2
μ	Viscosity of the fluid, N s/m2
μ_b	Viscosity of fluid at bulk temperature, N s/m2
μ_w	Viscosity of fluid at wall temperature, N s/m2
ρ	Density of the fluid, kg/m3
β	Baffle spacing in cm.

1. INTRODUCTION

The most of the applications in various industries involves conversion, utilization and recovery of energy. Operations like heating and cooling for thermal processing in viscous media of chemical, steam generation in various power plants, pharmaceutical and agricultural products, waste heat recovery, gas and liquid cooling of engines cooling of electronic devices are the common examples of these processes. So improved heat exchange in these industries can significantly improve the thermal efficiency of the process along with the economics of the design and applications.

The need for increasing 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 types of techniques are also referred as ―Heat transfer Enhancement or Intensification. Augmentation techniques may increase convective heat transfer by reducing the thermal resistance in a heat exchanger. In this process the pressure drop plays an important role and it should be kept in mind to reduce the pressure drop in order to reduce the pumping power requirements. On the other hand heat exchanger system in spacecraft, electronic device and medical application may rely primarily on enhanced thermal performance for their successful operations.

Use of Heat transfer augmentation techniques leads to increase in heat transfer coefficient but the pressure drop of the heat exchanger also increases. So, while designing a heat exchanger using any of the techniques, analysis ofheat transfer rate & pressure drop has to be done. Apart from that, 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 methods have been proposed in recent years.

For the present experimental work inserts are made up of twisted GI wires and baffles are made up of tin sheets and its effect on pressure drop at different baffle spacing is studied.

2. LITERATURE REVIEW

2.1 CLASSIFICATION OF HEAT TRANFER AUGMENTATION TECHNIQUES:

The heat transfer enhancement or augmentation may be broadly classified into three different categories:

1. Passive Techniques
2. Active Techniques

The difference between the two techniques is that the active techniques require external power supply to bring about the effect while the passive techniques don't need any power supply.

3. Compound Techniques

Sixteen different enhancement techniques have been identified by Bergles et al, which is shown in the given table.

Table 2.1: Different types of augmentation techniques:

PASSIVE TECHNIQUES	ACTIVE TECHNIQUES
Treated Surfaces . Extended Surfaces Rough Surfaces Displaced Enhancement Devices Swirl Flow Devices Surface Tension Devices Coiled Tubes Additives for Liquids Additives for Gases	Mechanical Aids Surface Vibration Electrostatic Fields Fluid Vibration Injection Jet impingement Suction

1.PASSIVE TECHNIQUES: These procedures for the most part utilize surface or geometrical changes to the flow channel by fusing inserts or extra devices. They advertise higher heat transfer coefficients by irritating or adjusting the current flow conduct (with the exception of extended surfaces) which additionally prompts build in the pressure drop. If there should be an occurrence of extended surfaces, successful heat transfer zone as an afterthought of the extended surface is expanded. Passive methods hold the preference over the active strategies as they don't oblige any immediate info of outer force. Heat transfer augmentation by these procedures might be attained by utilizing:

a. Treated Surfaces: These techniques include the fine-scale alteration of the surface finish or application of a coating (continuous or discontinuous). They are generally used for boiling and condensing; the roughness height is below that which affects single-phase heat transfer.

b. Rough Surfaces: The surface adjustments that advertise turbulence in the flow field, essential in single stage flows and don't expand the heat transfer surface zone. Their geometric characteristics range from arbitrary sand-grain roughness to discrete three dimensional surface bulges.

c. Extended Surfaces: These surfaces mostly in the form of fins are now regularly employed in many heat exchangers to increase the heat transfer surface area, especially on the side with the highest thermal resistances.

d. Displaced Enhancement Devices: These are the inserts primarily used in confined forced convection. These inserts are inserted into the flow channel so as to indirectly improve energy transport at the heated surface by displacing the fluid from the surface of the duct with bulk fluid from the core flow.

e. Swirl Flow Devices: These consist of a number of geometric arrangements or tube inserts for forced flow that create rotating or secondary flow. Some of the different types are Inlet Vortex Generators, Twisted Tape Inserts, Stationary Propellers and Axial-Core Inserts with a screw type winding. They can be used for both single phase flow and two-phase flows.

f. Coiled Tubes: These tubes leads to more compact heat exchangers. The secondary flows or vortices are generated due to curvature of coils that promote higher single phase heat transfer coefficients as well as improvement in most regimes of boiling.

g. Surface Tension Devices: These techniques include wicking or grooved surfaces that direct and improve the flow of fluid to boiling surfaces and from condensing surfaces. Many manifestations of devices involving capillary flow is also possible.

h. Additives for Liquids: These include solid particles, soluble trace additives and gas bubbles in single phase flows and trace additives which reduce the surface tension of the liquid for boiling systems.

i. Additives for Gases: Additives for gases are liquid droplets or solid particles, which are introduced in single phase gas flows either as dilute-phase (gas-solid suspensions) or dense-phase (fluidised beds).

2.ACTIVE TECHNIQUES: These techniques require the use of external power to facilitate the desired flow modifications and improvement in the rate of heat transfer. Thus, these techniques are more complex from the use and design point of view. It finds very limited practical applications. As compared to passive techniques, these techniques have not shown much potential as it is very difficult to provide external power input in many cases. Heat Transfer Enhancement by this technique can be achieved by incorporating one of the following methods.

a. Mechanical Aids: In this, the stirring of the fluid is done by mechanical means or by rotating the surface. Another type is surface "Scrapping", which is widely used in the chemical process industry for batch processing of viscous liquids.

b. Surface Vibration: They are applied in single phase flows to obtain higher convective heat transfer coefficients, at either low or high frequency. This is possible only in certain circumstances as the vibrations of sufficient amplitude to affect the heat transfer may destroy the heat exchanger itself.

c. Fluid Vibration: This kind of vibration augmentation technique is employed for single phase flows. Instead of applying vibrations to the surface, pulsations are created in the fluid itself. It is the practical type of vibration augmentation because of large mass of most heat exchangers.

d. Electrostatic Fields: Electrostatic fields from a AC or DC source can be applied in different ways to dielectric liquids to cause bulk mixing or disruption of flow in the vicinity of heat transfer surface to enhance heat transfer.

e. Injection: It is utilized by supplying gas to a stagnant or flowing liquid through a porous heat transfer surface or by injecting similar fluid into the liquid. The surface degassing of liquids can produce augmentation similar to gas injection.

f. Suction: This can be used for both single phase and 2-phase heat transfer process. It involves vapour removal, in nucleate or film boiling, or fluid withdrawal, in single phase flow, through a porous heated surface.

3.COMPOUND TECHNIQUES: A compound technique is the one which involves the simultaneous combination of two or more of the above techniques with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

2.2 PERFORMANCE EVALUATION CRITERIA:

In most practical applications of augmentation 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 heat exchanger surface area.

It can be seen 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:

FIXED GEOMETRY (FG) CRITERIA: The area of flow cross-section (N and di) and tube length 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, di, L). The objectives then could be to increase the heat load Q for the same approach temperature ΔTi and mass flow rate *m* or pumping power P; or decrease ΔTi 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 di) is kept constant, and the heat exchanger length is allowed to vary. Here the objectives are to seek a reduction in either the heat transfer area ($A \rightarrow L$) 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 2.2: Performance Evaluation Criteria[1]

Case	**Geometry**	**m**	**P**	**Q**	**ΔT_i**	**Objective**
FG-1a	N, L, Di	X			X	Q↑
FG-1b	N, L, Di	X		X		ΔTi↓
FG-2a	N, L, Di		X		X	Q↑
FG-1b	N, L, Di		X	X		Δ Ti↓
FG-3	N, L, Di			X	X	P↓
FN-1	N, Di		X	X	X	L↓
FN-2	N, Di	X		X	X	L↓
FN-3	N, Di	X		X	X	P↓
VG-1	---	X	X	X	X	(NL) ↓
VG-2a	N, L	X	X		X	Q↑
VG-2b	N, L	X	X	X		Δ Ti↓
VG-3	N, L	X		X	X	P↓

Bergles et al [2] suggested a set of eight (R1-R8) numberof performance evaluation criteria as shown in Table 2.3

Table 2.3: Performance Evaluation Criteria of Bergles et al [3]

	Criterion number							
	R1	R2.	R3	R4	R5	R6	R7	R8
Basic Geometry	×	×	×	×				
Flow Rate	×						×	×
Pressure Drop		×				×		×
Pumping Power			×					
Heat Duty				×	×	×	×	×
Increase Heat Transfer	×	×	×					
Reduce pumping power				×				
Reduce ExchangeSize					×	×	×	×

2.2 **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 2.1 shows a typical configuration of twisted tape which is used commonly.

Figure 2.2: 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 multitude 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.

3. EXPERIMENTAL SETUP:

3.1. SPECIFICATIONS OF THE HEAT EXCHANGER:

The experiments were carried out in a Double Pipe Heat Exchanger with the following specifications:

Table 3.1: Specification of heat exchanger:

Inner pipe Internal Diameter	22mm
Inner pipe Outer Diameter	25mm
Outer pipe Internal Diameter	53mm
Outer pipe Outer Diameter	61mm
Material of Construction of inner tube	Copper
Heat Transfer Length	2.93m
Pressure tapping to pressure tapping length	2.825m

Water at room temperature was allowed to flow through the inner pipe.

3.2: TYPES OF INSERTS USED:

For the present experimental work the insert used was Galvanised Iron wires (GI wires). The GI wires were taken in a bundle of three and four and they are twisted in order to make a suitable insert. The lengths of the inserts were about 3m in long. While much literature can be found about passive heat transfer augmentation using twisted tapes as mentioned earlier, twisted GI wires are a new kind of insert where no such experiments have been done; thus giving us ample room for experimental studies.The present work deals with finding the friction factor for the GI wire insert with various numbers of wires and comparing those results with that of the smooth tube. The GI wires used in this experiment are low in cost and widely used in different construction works. Fig 3.1 shows the picture of G.I wires used for fabrication of insert.

Figure3.1: GI wires (used for making the inserts)

3.2: TYPES OF BAFFLES USED:

The baffles used for present experimental work is made up of thin tin sheets. They were cut into circular shape with a diameter of 16mm.With the help of the drilling machines wholes were made in the centre of the baffles.. While fabricating the baffles special attention was given towards the diameter which shouldn’t increase 16mm and should be same for all he baffles. Fig 3.2 shows photo of some of the baffles used.

Figure 3.3: Baffles used

3.3: FABRICATION OF INSERTS:

Basically there were two types of inserts fabricated using GI wires. First four numbers of equal lengths of GI wires were taken and they were straightened. The one end of the four wires was tied to one point and the other end was tied to the 4 way lug wrench. The 4 way lug wrench was used in order to twist the GI wires one over another. The wrench was tightly held and it was rotated in one direction so that the wires get twisted over one another. The wrench was rotated carefully such that the pitching was even everywhere. After many rotations the back pressure continuously increased and a time came when the back pressure decreased. When the pitching was even the 4 way lug wrench was ceased to rotate and the wires were cut off from both the ends. The same process was repeated for three GI wires and another insert was fabricated. Using the Baffles at different distances the pressure drop was measured. Fig 3.3a shows the picture of 4 way lug wrench used for fabricating the inserts.

Figure 4.3a:4 way lug wrench

Fig 3.3 b to 3.3i shows the different types of GI wire inserts used with various baffle spacing.

Fig 3.3b: Insert made up of three GI wires

Fig 3.3c: Insert made up of four GI wires

Fig3.3d: 3 wire insert with β= 24cm

Fig3.3e: 3 wire insert with β= 12cm

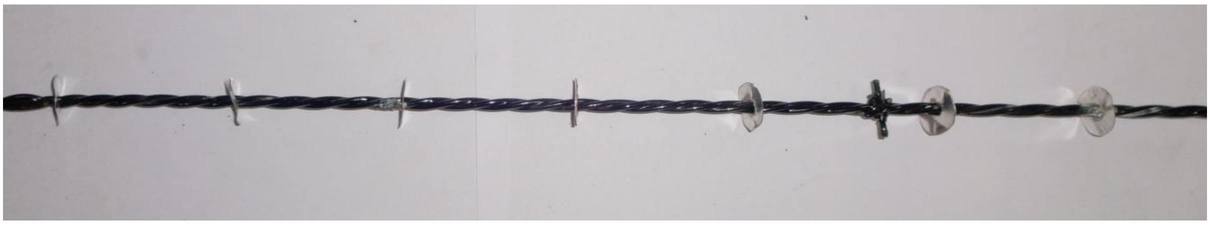

Fig3.3f: 3 wire insert with β= 6cm

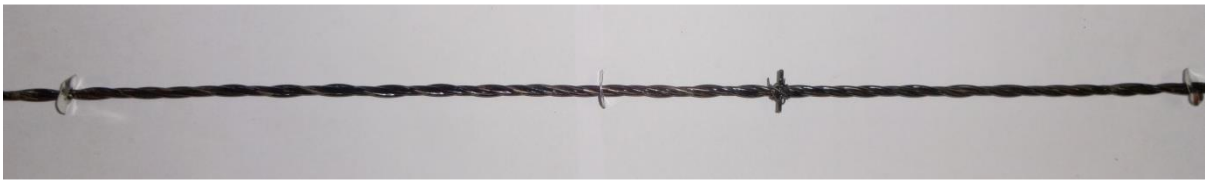

Fig3.3g: 4 wire insert with β= 24cm

Fig3.3h: 4 wire insert with β= 12cm

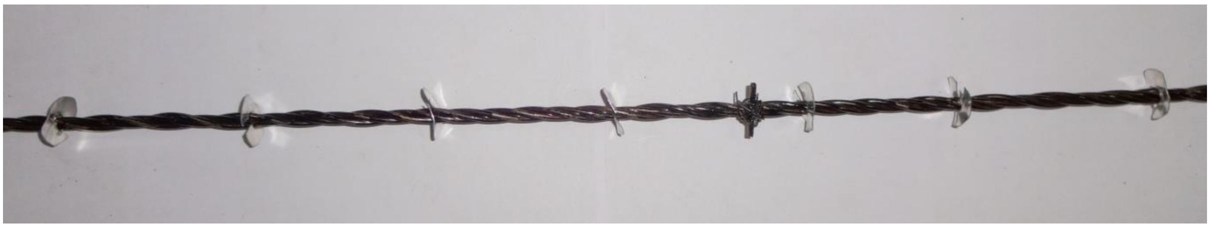

Fig3.3i: 4 wire insert with β= 6cm

3.4. EXPERIMENTAL SET UP:

The Fig.3.4a shows the schematic diagram of the experimental setup. Basically, it is a double pipe heat exchanger consisting of an inner pipe of ID 22 mm and OD 25 mm, and an outer pipe of ID 53 mm and OD 61 mm. The test section is a smooth copper tube with a length of 3meter. The apparatus is also equipped with two rota meters for continuously measuring and maintaining the particular flow rate; one for measuring hot water flow and another for measuring flow rate of cold water. The source for the cold water was from a bore-well from where water was pumped through a submersible pump. There is another tank of capacity 500 litres which has an in-built heater and pump for providing hot water at a particular desired temperature and flow rate. It is also equipped with a digital temperature indicator connected to four RTD sensors. They have four different sensors situated at different locations to give the temperatures T1-for Inner Tube Inlet, T2- for Inner Tube Outlet, T3-for Outer Tube Inlet and T4-Outer Tube Outlet. One calibrated rotameter with flow ranges from 300 to 1250 litres/hour was used to measure the flow rate of cold water. There is a U-tube manometer for measuring the pressure drop in the inner tube. Two pressure tapings- one just before the test section and one after the test section are connected to the manometer for measuring pressure drop. The fluid filled inside the manometer is Carbon Tetra-Chloride (CCl_4) with Bromine to give it a pinkish colour for easy identification.

Fig 3.4b shows the photograph of the experimental setup.

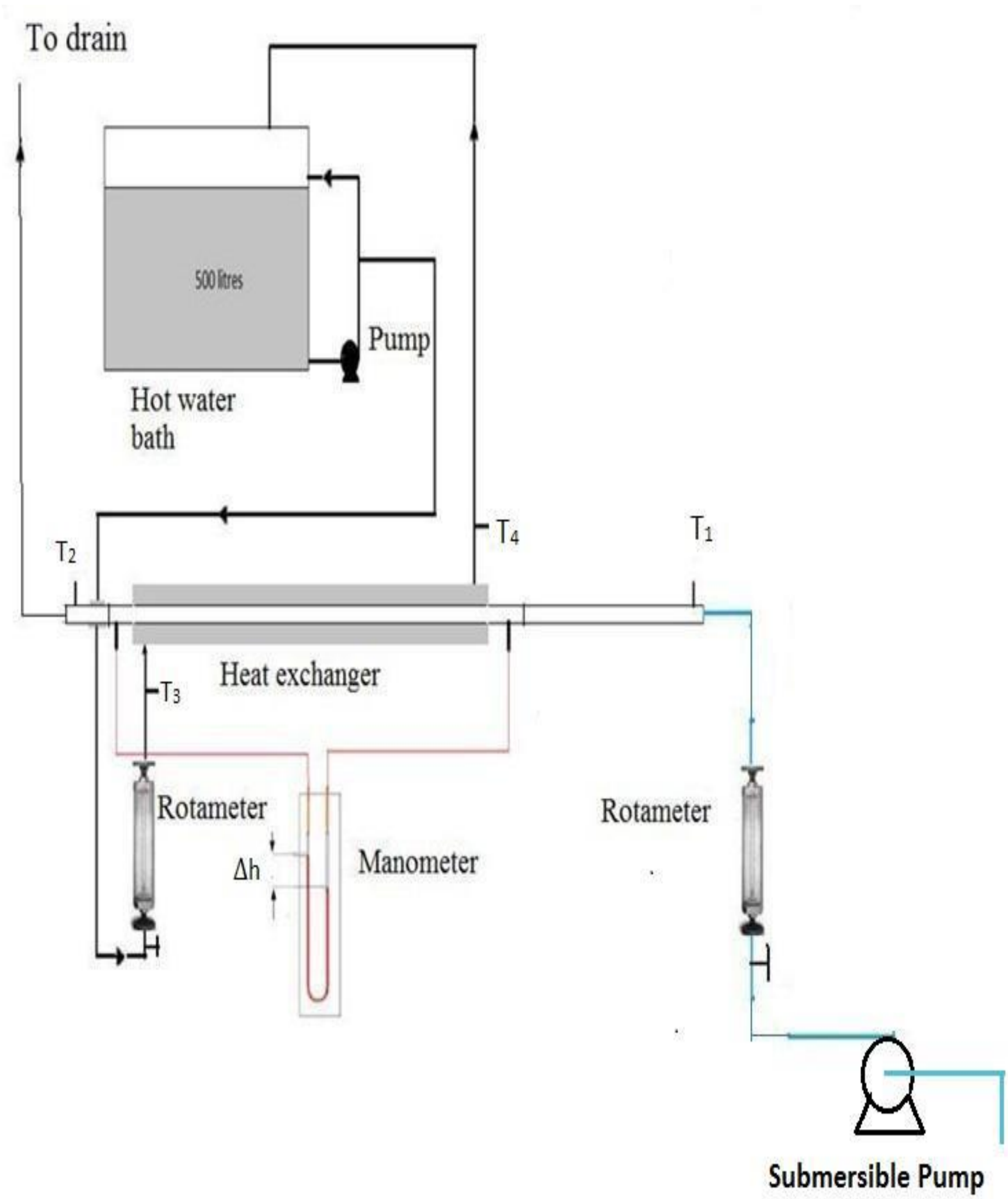

Fig. 3.4a: Schematic diagram of experimental set up

Fig.-3.4b: Photograph of the Experimental Setup

3.5 EXPERIMENTAL PROCEDURE:

1. First the Rotameter for the cold water flow rate was calibrated.

i. For the rotameter calibration, i collected water in a bucket, and simultaneously the time and weights were noted. Thus the mass flow rate was calculated.

ii. We repeated the same procedure for three times for each particular reading and then average of all the three was taken. The readings are given in A.1.1.

2.Standardization of the set-up:

Before beginning dealing with the experimental study on friction factor in heat exchanger using inserts, the standardization of the experimental setup was done by obtaining the friction factor results for the smooth tube & comparing the obtained data with the standard equations available.

3.For friction factor determination:

Pressure drop was measured for each flow rate varying from 350-1250Kg/hr with the help of manometer at room temperature.

i. The U-tube manometer used carbon tetrachloride as the manometer liquid.

ii. Air bubbles were removed from the manometer so that the liquid levels in both the limbs are same when the flow rate is made zero. The air bubbles were removed by removing the clips attached to the open ends of the pipes connected to the U-Tube limbs and then allowed the water to flow the open ends in a controlled manner by controlling the flow with the help of hand to ensure that the air bubbles in the manometer escape out. Then, the ends were closed with the help of clips. This procedure was repeated every time the experiment is done.

iii. Water at the room temperature was allowed to flow through the inner pipe of the Heat Exchanger.

iv. The manometer reading was then noted.

4. After the confirmation of validity of the experimental values of friction factor in smooth tube with standard equations, friction factor studies with inserts were conducted.

5. The friction factor results for all the cases are presented in Tables A.2.1-A.2..9.

<u>3.6 STANDARD EQUATIONS USED:</u>

For Plain Tube

Friction factor (f_{theo}) calculations:

a. For Re < 2100

$$\boldsymbol{f} = \frac{\mathbf{16}}{\boldsymbol{Re}} \text{ ....Eq. 3.2}$$

b. For Re > 2100

Colburn's Equation:

$$\boldsymbol{f} = \frac{\mathbf{.046}}{\boldsymbol{Re}^{\mathbf{0.2}}} \text{ .... Eq. 3.3}$$

4. SAMPLE CALCULATION:

4.1 ROTAMETER CALIBRATION:

For 900 kph (Table No. A.1.1)

Observation No. 1

Weight of water collected = 13.3Kg

Time = 62 sec

$\boldsymbol{m_1}$ = 0.2145 Kg/sec

Observation No. 2

Weight of water collected = 11.65 Kg

Time = 52 sec

$\boldsymbol{m_2}$ = 0.224 Kg/sec

Observation No. 3

Weight of water collected = 11.55 Kg

Time = 51 sec

$\boldsymbol{m_3}$= 0.2264 Kg/sec

$$m = \frac{\boldsymbol{m_1}+\boldsymbol{m_2}+m_3}{3} = \frac{.2145+.224+.2464}{3} = .2471 \text{ kg/sec}$$

4.2 PRESSURE DROP & FRICTION FACTOR CALCULATIONS:

For 4 wires insert without baffles (Table A.2.6)

m=0 .2205 kg/hr

Augmented friction factor is f_a

$$Area = \frac{\pi \times \boldsymbol{d_1}^2}{4} = \frac{\pi \times .022^2}{4} = 3.8 \times 10^{-4} meter^2$$

$$v = \frac{m}{A \times \rho_w} = \frac{.2205}{3.8 \times 10^{-4} \times 1000} = 0.5802\, {}^{m}/_{sec}$$

$$\Delta p = \left(\rho_{CCl_4} - \rho_{H_2O}\right) \times g \times \Delta h = (1587.6 - 1000) \times 9.81 \times 0.355 = 2043.21\, {}^{N}/_{m^2}$$

$$f_a = \frac{\Delta p \times d_i}{2 \times \rho \times L \times v^2} = \frac{2043.21 \times .022}{2 \times 1000 \times 2.83 \times .5802^2} = 0.02359$$

For viscosity calculation:

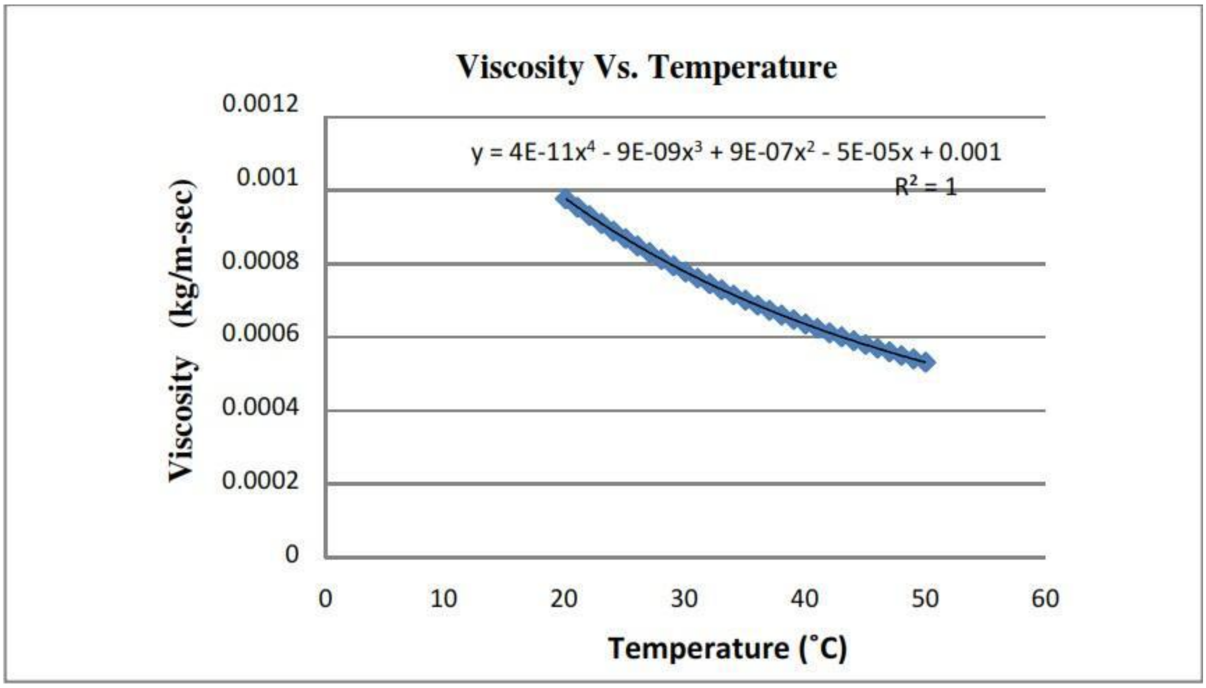

Fig. 4.1: Viscosity vs Temperature Graph

$$\boldsymbol{\mu = 4 \times 10^{-11} T^4 - 9 \times 10^{-9} T^3 + 9 \times 10^{-7} T^2 - 5 \times 10^{-5} T + 0.0017}$$

Theoretical friction factor calculation for smooth tube:

$$Re = \frac{4 \times m}{\pi \times d_i \times \mu} = \frac{4 \times .2205}{\pi \times .022 \times .00072} = 17724.07$$

$$f_0 = 0.046 \times Re^{-0.2} = .046 \times 17724.07^{-0.2} = 0.00651$$

$$\frac{f_a}{f_0} = \frac{.02359}{0.00651} = 3.6236$$

5. RESULTS AND DISCUSSION:

5.1 FRICTION FACTOR RESULTS:

Fig 5.1 shows the plot between Reynolds number and friction factor (both experimental and theoretical) for smooth tube. In almost all Reynolds number range the difference of f_{exp} and $f_{theo}$is within $\pm 20\%$. This indicates that the experimental setup can produce friction factor results with reasonable degree of accuracy. Thus it validates the standardisation of the setup.

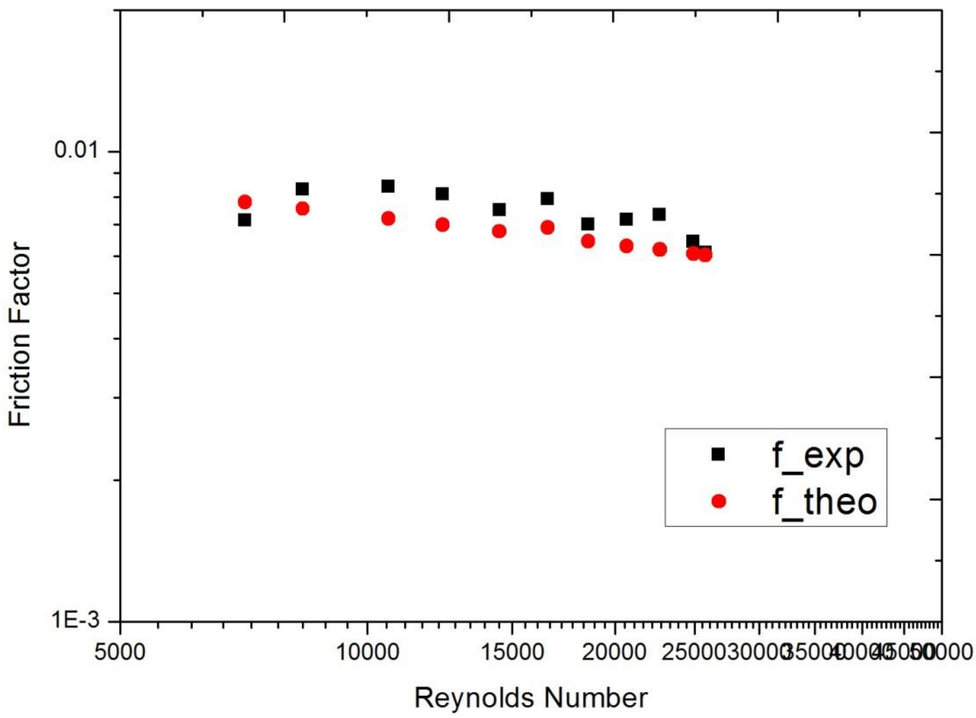

Fig 5.1 Friction Factor vs. Reynolds number for Smooth Tube

All the friction factor results and the ratio f_a/f_o for the different cases are tabled in tables A.2.2- A.2.9.

Fig 5.2 shows the variation of Friction factor with Reynolds number for the three wires insert with baffles and without baffles. The baffles spacing is varied as $(\beta = 24, 12, and\ 6\ cm)$. From the plot it can be seen that as number of baffles increase the friction factor also increases. The insert with baffles give more friction factor due to increase in degree of turbulence of the flow. So for the three wire insert the friction factor is highest for with baffle spacing ($\beta = 6cm$).

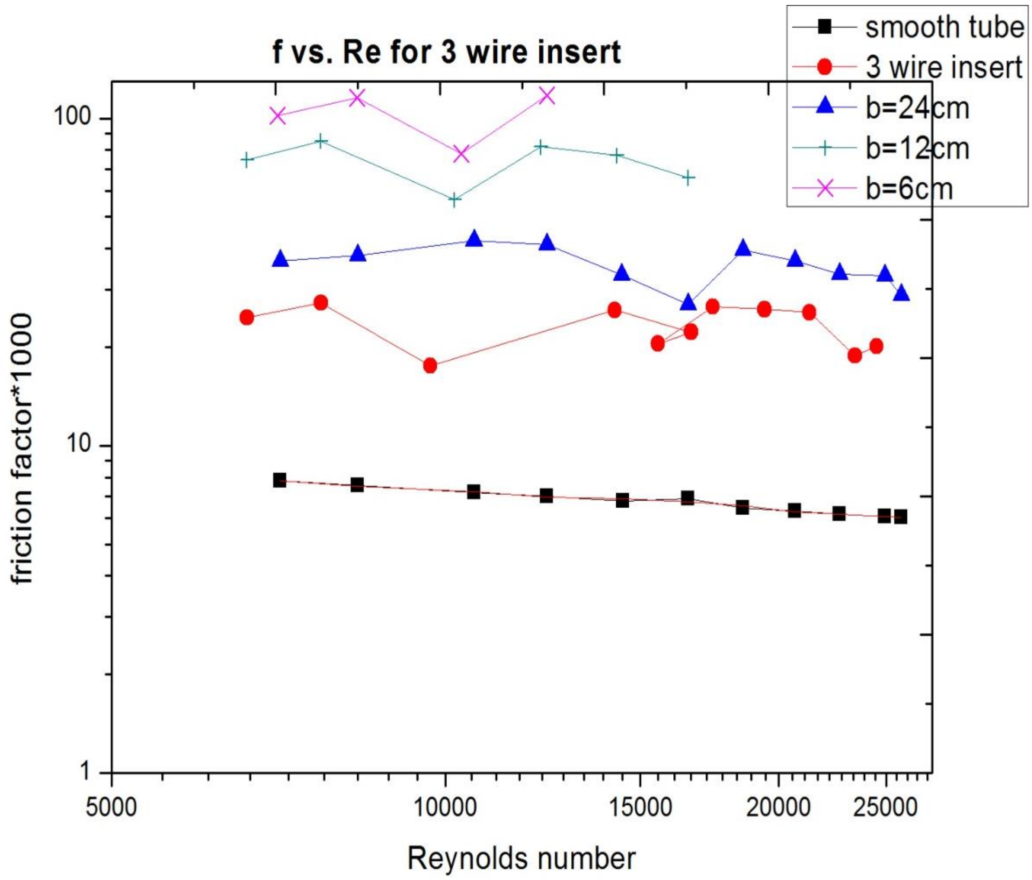

Fig 5.2: Friction Factor vs. Reynolds number for three wire insert with and without baffles

Fig 5.3 shows the variation of Friction factor with Reynolds number for the four wires insert with baffles and without baffles. The baffles spacing is varied as $(\beta = 24, 12, and\ 6\ cm)$. From the plot it can be seen that like the three wire insert for this four wires insert also with the decrease in baffle spacing the friction factor increases So for the four wire insert the friction factor is highest for with baffle spacing ($\beta = 6cm$).

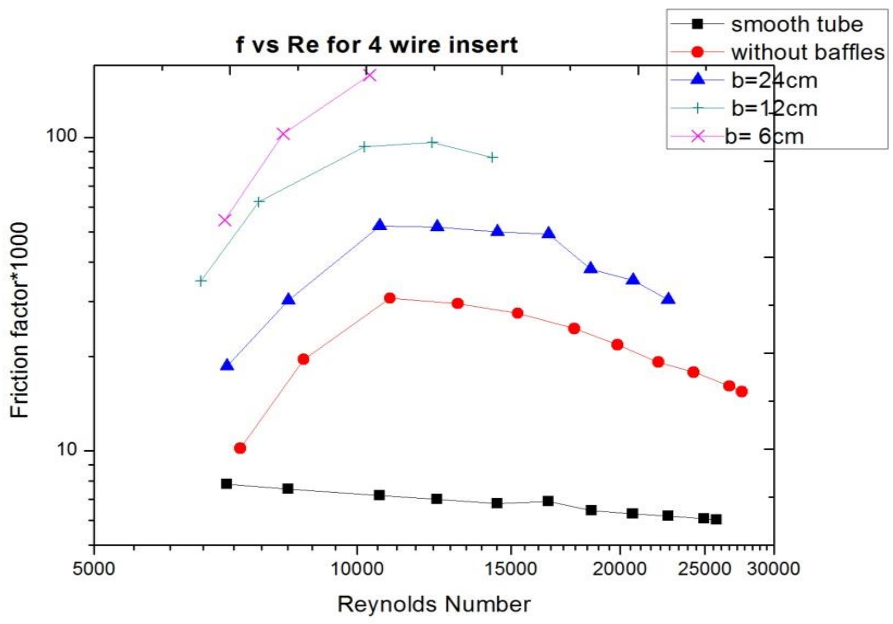

Fig 5.3: Friction Factor vs. Reynolds number four three wire insert with and without baffles

Fig 5.4-5.7 shows the variation of friction factor and Reynolds number for the three and four wire inserts with and without baffles. The different baffle spacing taken are ($\beta = 24\ ,12\ and\ 6cm$). From the plot it can be seen that the friction factor is usually more for 4 wire inserts comparing to three wire inserts. So the friction factor is highest for the four wire insert with baffle spacing ($\beta = 6cm$).

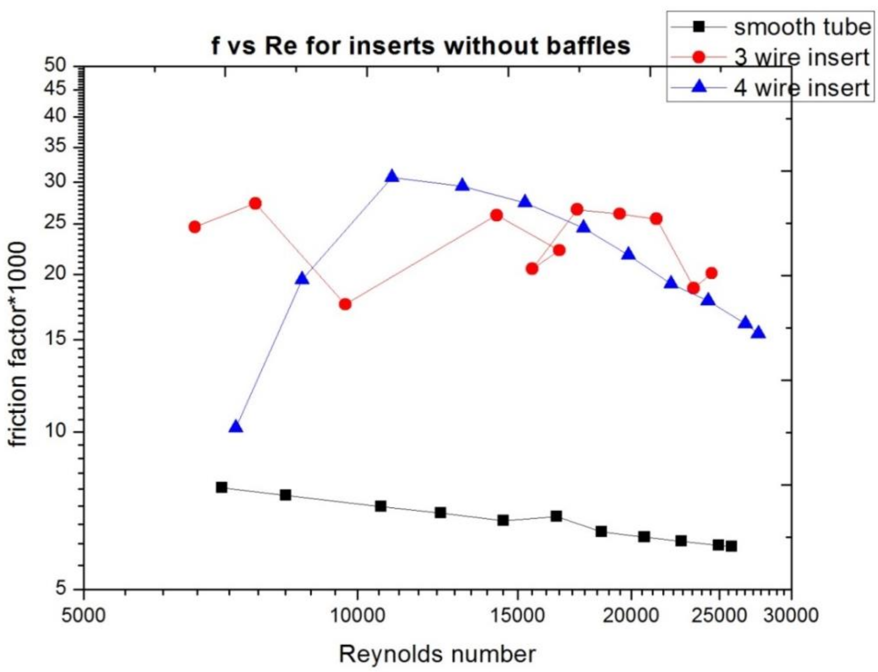

Fig 5.4: Friction Factor vs. Reynolds number for three and four wire inserts without baffles

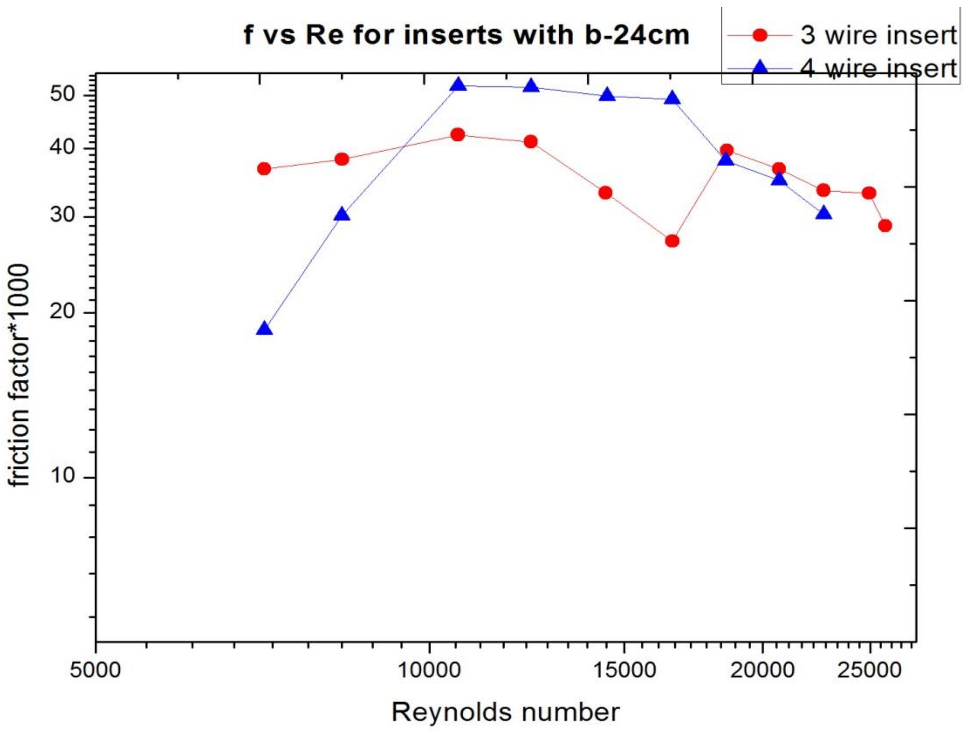

Fig 5.5: Friction Factor vs. Reynolds number for three and four wire inserts with baffle spacing$\boldsymbol{\beta = 24cm}$

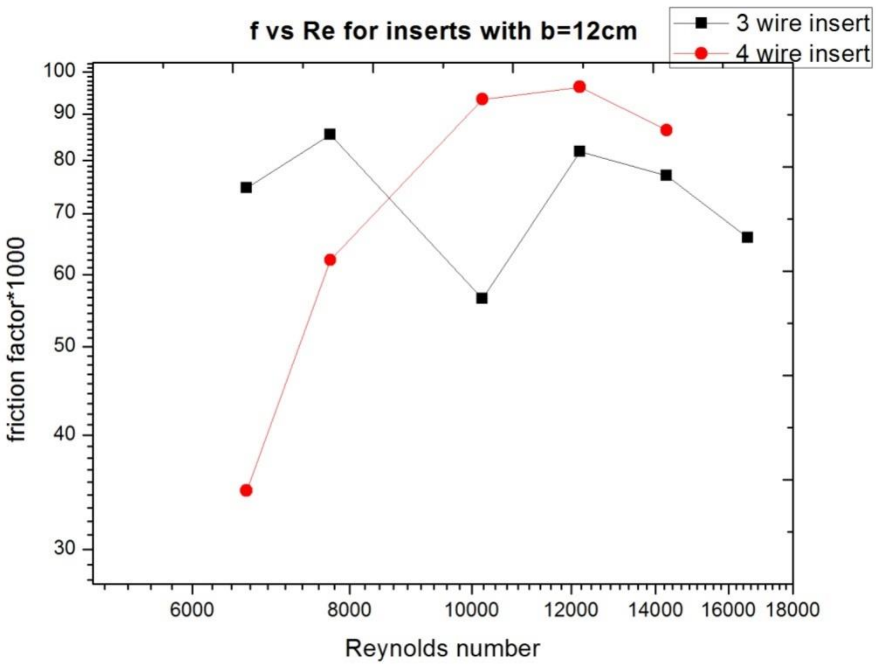

Fig 5.6: Friction Factor vs. Reynolds number for three and four wire inserts with baffle spacing$\boldsymbol{\beta = 12cm}$

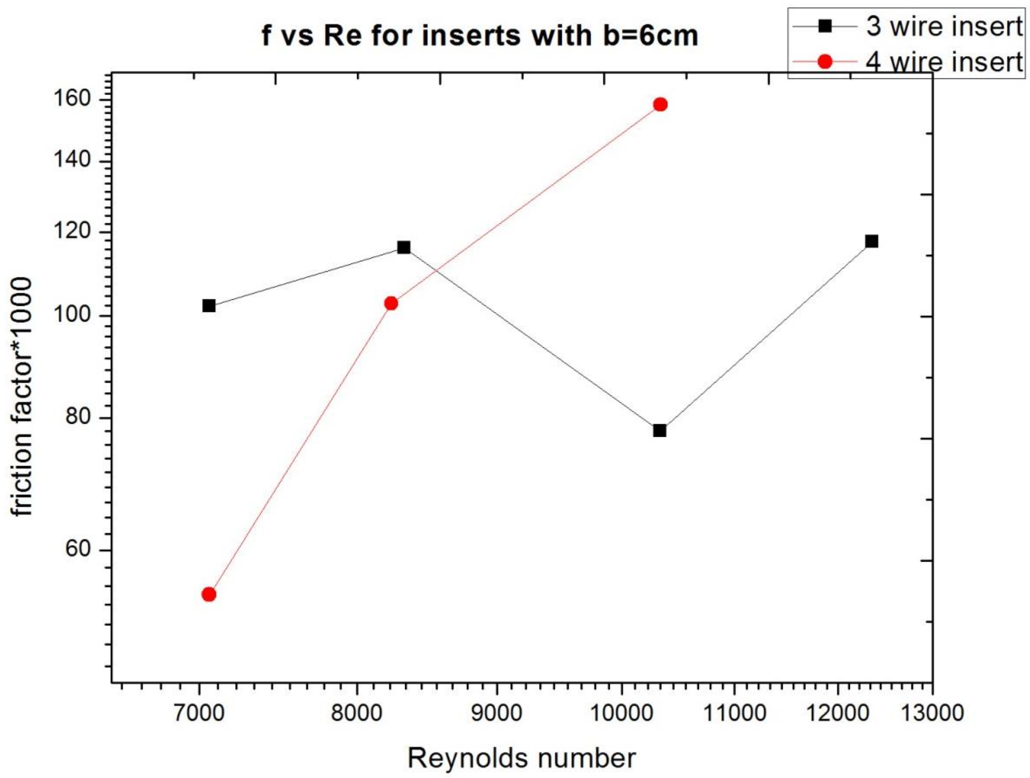

Fig 5.7: Friction Factor vs. Reynolds number for three and four wire inserts with baffle spacing$\boldsymbol{\beta = 6cm}$

Fig 5.8 shows the variation of ratio ofaugmented and theoretical friction factorsthat is$^{f_a}/_{f_o}$with Reynolds number for the three wire inserts with and without baffles. The baffles spacing is varied as ($\beta = 24, 12, and\ 6\ cm$). From the plot it can be seen that the value of ratio increases with increase in number of baffles. The ratio is highest for the three wire insert with ($\beta = 6cm$).

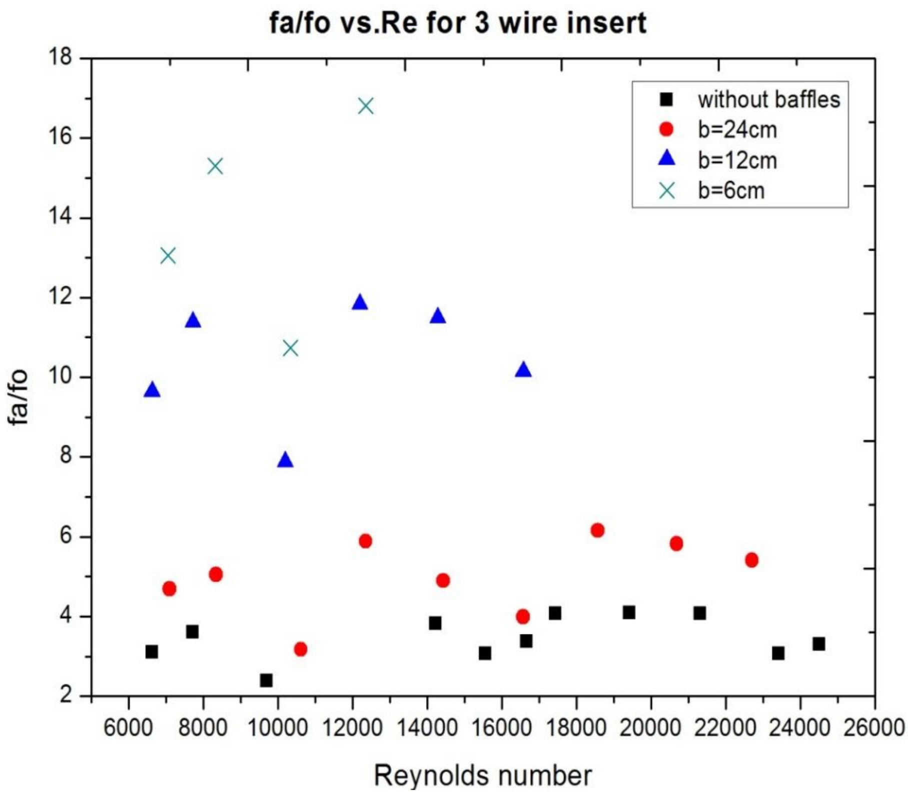

Fig 5.8: f_a/f_o vs. Reynolds number for three wire insert with and without baffles

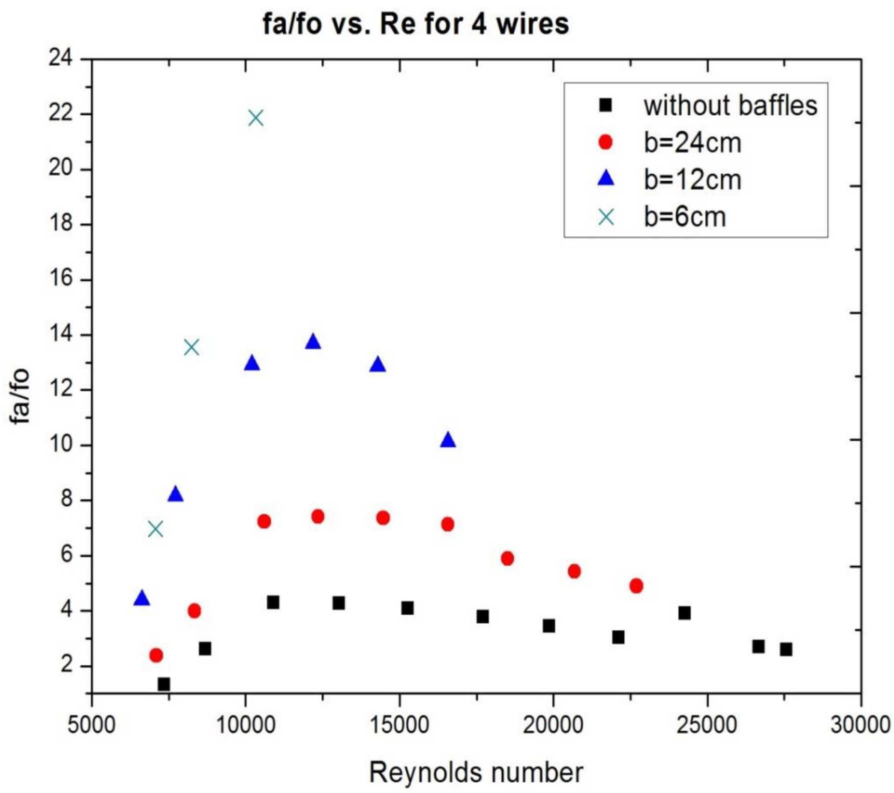

Fig 5.9: f_a/f_o vs. Reynolds number for four wire insert with and without baffles

Fig 5.10-5.13 shows the variation of ratio that is f_a/f_o with Reynolds number for both the three and four wire inserts with and without baffles. The baffles spacing is varied as $(\beta = 24, 12, and\ 6\ cm)$ for both the inserts From the plot it can be seen that the value of ratio is more for the four wire insert comparing to three wire insert. Also the ratio value increases with increase in baffles for both the wires.

a. The ratio f_a/f_o is maximum for the four wire insert with $\beta = 6cm$.

b. The ratio f_a/f_o is minimum for the three wire insert without any baffles.

c. The ratio f_a/f_o is large for the four wire insert with baffles.

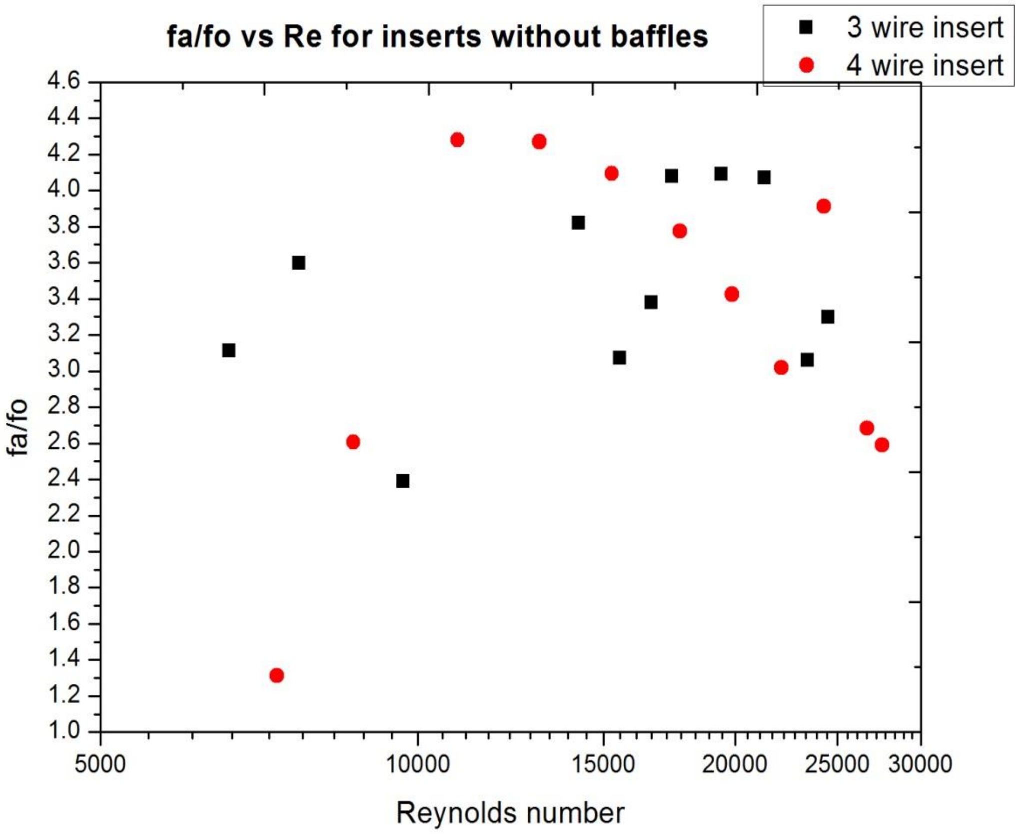

Fig 5.10: f_a/f_o vs. Reynolds number for both inserts without baffles

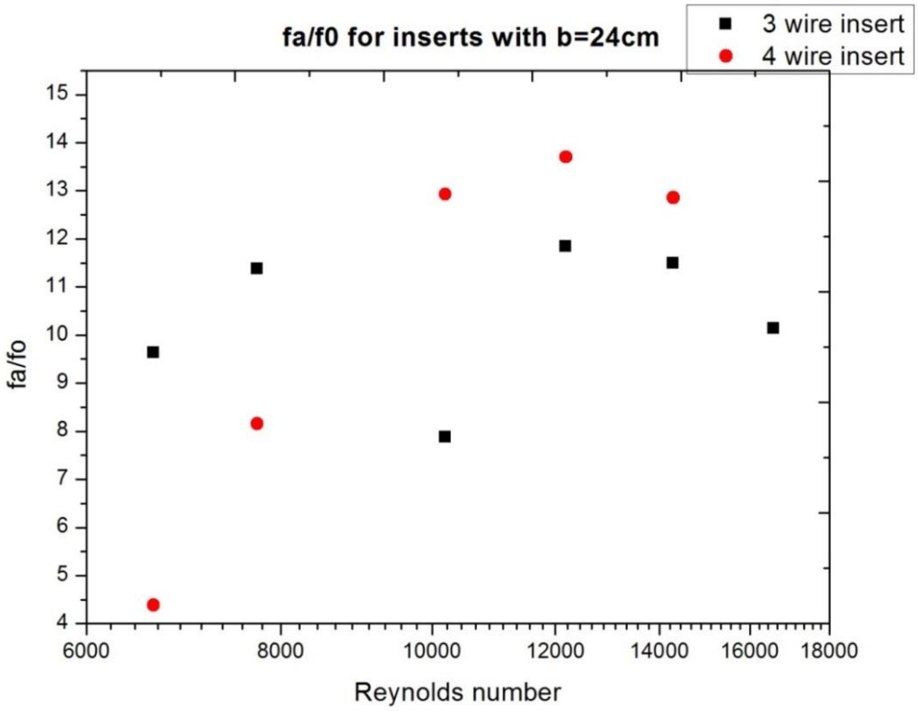

Fig 5.11

${f_a}/{f_o}$ Vs. Reynolds number for both inserts with baffle spacing $\boldsymbol{\beta = 24cm}$

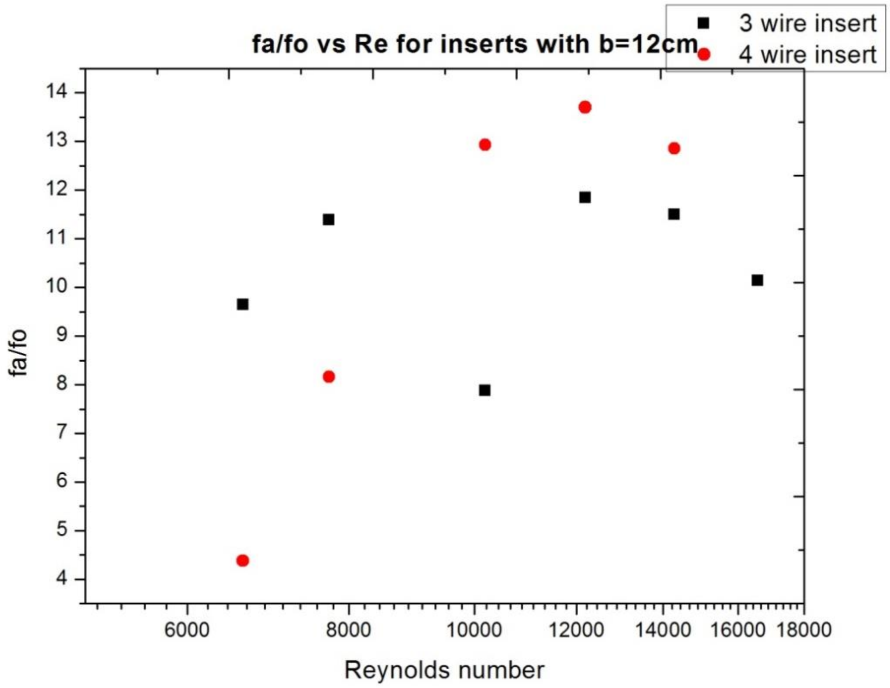

Fig 5.12: ${f_a}/{f_o}$ vs. Reynolds number for both inserts with baffle spacing $\boldsymbol{\beta = 12cm}$

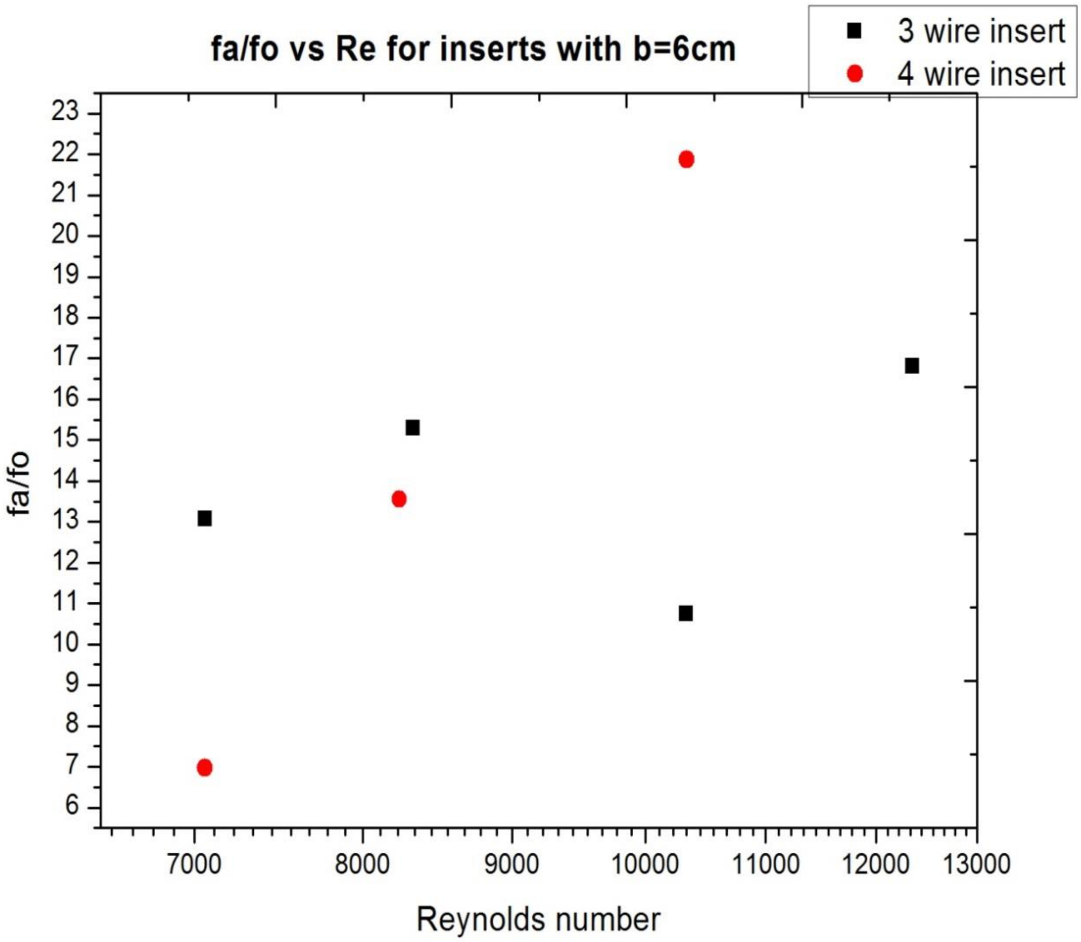

Fig 5.13: ${f_a}/{f_o}$ vs. Reynolds number for both inserts with baffle spacing $\boldsymbol{\beta = 6cm}$

6. CONCLUSION

From the different observations and plots drawn the range of ratio of friction factors that is f_a/f_o for the three wire and four wire inserts without baffles and with different baffle spacing is shown in this table.

Table 6.1: Range of ratio $\boldsymbol{f_a/f_o}$ for different types of inserts

SL No.	Insert	f_a/f_o
1	3 wire inserts without baffles	2.38 – 4.09
2	3 wire inserts with baffle spacing $\beta = 24cm$	3.17 – 6.16
3	3 wire inserts with baffle spacing $\beta = 12cm$	7.88 – 14.27
4	3 wire inserts with baffle spacing $\beta = 6cm$	13.06 – 16.8
5	4 wire inserts without baffles	1.31 - 4.28
6	4 wire inserts with baffle spacing $\beta = 24cm$	2.38 – 7.4
7	4 wire inserts with baffle spacing $\beta = 12cm$	4.38 - 14.56
8	4 wire inserts with baffle spacing $\beta = 6cm$	6.96 - 21.87

1. From the above table, we can say that with the decrease in baffle spaces the friction factor increases due to increase in degree of turbulence.
2. The ratio of friction factor is slightly more for the four wire inserts comparing to the three wire inserts.
3. From the above data it can be seen that the ratio value is highest for the four wire inserts with baffle spacing $\beta = 6cm$.
4. The maximum height difference that can be measured using the manometer is 100cm. For inserts with baffle spacing $\beta = 6cm$at higher flow rate the difference in height is more than 100cm. So the friction factor at higher flow rates cannot be determined.

5. For inserts with baffle spacing $\beta = 6cm$ the range of Reynolds number is in between 7000-11000 whereas for inserts without baffles the range varies between 7000-27000. So at higher Reynolds number the friction factor for these inserts is expected to increase. Thus the friction factor ratio range will be more for these inserts.

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APPENDIX

A.1 CALIBRATION:

A.1.1 ROTAMETER CALIBRATION:

Rotameter reading LPH	Mass flow rate kg/sec	Observation 1			Observation 2			Observation 3			Average m kg/sec
		Wt kg	Time sec	m kg/sec	Wt kg	Time sec	m kg/sec	Wt kg	Time sec	m kg/sec	
350	0.0972	12.6	150	0.1284	11.2	138	0.081159	11.96	153	0.07817	0.0927
400	0.1111	12.4	125	0.0992	12.4	132	0.093939	11.16	121	0.092231	0.1081
500	0.1381	11.2	105	0.106667	12	111	0.108108	11.14	103	0.108155	0.1357
600	0.16667	12.45	92	0.135326	13.7	99	0.138384	12.3	92	0.133696	0.1622
700	0.1945	13.3	82	0.162195	11.8	72	0.163889	12.6	78	0.161538	0.1901
800	0.2223	12.25	66	0.185606	12.55	65	0.193077	11.6	62	0.187097	0.2205
900	0.2501	13.3	62	0.214516	11.65	52	0.224038	11.55	51	0.226471	0.2471
1000	0.2778	12.75	53	0.240566	12.05	48	0.251042	12.9	45	0.286667	0.2752
1100	0.3056	13.6	51	0.266667	11.7	42	0.278571	12.3	41	0.3	0.302
1200	0.3334	12.25	42	0.291667	12.4	40	0.31	12.1	43	0.281395	0.3319
1250	0.3473	12.5	39	0.320513	13.2	39	0.338462	12.7	40	0.3175	0.3431

A.2 FRICTION FACTOR RESULTS:

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

m (kg/sec)	Δh (meter)	T(℃)	ΔP (N/m^2)	Re	$f_{exp} \times 1000$	$f_{theo} \times 1000$	f_{exp}/f_{theo}
0.0927	0.019	32.9	109.52	7096.52	7.15	7.81	0.91
0.1081	0.030	33.1	172.93	8341.65	8.31	7.56	1.09
0.1357	0.048	33.1	276.68	10612.94	8.43	7.2	1.17
0.1622	0.066	33.1	380.44	12351.64	8.11	6.99	1.16
0.1901	0.084	33.4	484.2	14476.25	7.52	6.77	1.11
0.2205	0.119	33.4	685.95	16573.16	7.92	6.89	1.14
0.2471	0.132	33.4	760.89	18572.46	6.99	6.44	1.08
0.2752	0.168	33.4	968.41	20684.51	7.17	6.3	1.13
0.302	0.207	33.4	1193.22	22698.84	7.34	6.18	1.185
0.3319	0.219	33.4	1262.39	24946.18	6.43	6.07	1.05
0.3431	0.222	33.4	1279.68	25787.99	6.1	6.03	1.01

A.2.2: Friction Factor vs. Reynolds number for three wire insert without baffles

m (kg/sec)	Δh (m)	$T(°C)$	ΔP (N/m^2)	Re	$f_a \times 1000$	$f_0 \times 1000$	f_a/f_0
0.0927	0.063	29.3	372.67	6623.42	24.63	7.92	3.1098
0.1081	0.095	29.3	561.97	7723.75	27.32	7.68	3.5973
0.1357	0.096	29.3	567.88	9695.77	17.52	7.34	2.3883
0.1622	0.097	30.4	573.79	14223.1	25.94	6.79	3.8203
0.1901	0.098	30.4	579.71	16669.62	22.24	6.58	3.3799
0.2205	0.103	28.5	609.29	15562.6	20.48	6.67	3.0719
0.2471	0.128	28.5	757.18	17439.99	26.6	6.52	4.0797
0.2752	0.157	28.5	928.73	19423.2	26.09	6.38	4.0893
0.302	0.182	28.7	1076.61	21314.76	25.52	6.27	4.0702
0.3319	0.213	28.7	1259.99	23425.07	18.82	6.15	3.0602
0.3431	0.278	28.9	1644.49	24514.51	20.09	6.09	3.2988

A.2.3: Friction Factor vs. Reynolds number for three wire insert with baffle spacing $\boldsymbol{\beta = 24cm}$

m (kg/sec)	Δh (m)	T(℃)	ΔP (N/m^2)	Re	$f_a \times 1000$	$f_0 \times 1000$	f_a/f_0
0.0927	0.094	29.3	556.05	7096.52	36.67	7.81	4.6957
0.1081	0.132	29.3	785.57	8341.65	38.19	7.56	5.0514
0.1357	0.232	29.3	1373.56	10612.94	42.36	7.20	3.1764
0.1622	0.322	30.4	1904.18	12351.64	41.12	6.99	5.8833
0.1901	0.357	30.4	2111.22	14426.25	33.18	6.77	4.9009
0.2205	0.391	28.5	2314.71	16573.16	27.04	6.89	3.9935
0.2471	0.721	28.5	4624.43	18572.46	39.67	6.44	6.1603
0.2752	0.827	28.5	4889.10	20684.51	36.67	6.30	5.821
0.302	0.909	28.7	5378.90	22698.84	33.50	6.189	5.4134
0.3319	–	28.7	–	24946.18	–	6.072	–
0.3431	–	28.9	–	25787.9	–	6.032	–

A.2.4: Friction Factor vs. Reynolds number for three wire insert with baffle spacing $\boldsymbol{\beta = 12cm}$

m (kg/sec)	Δh (m)	T(℃)	ΔP $({N}/{m^2})$	Re	$f_a \times 1000$	$f_0 \times 1000$	${f_a}/{f_0}$
0.0927	0.191	29.3	1130.44	6623.42	85.38	7.75	9.6404
0.1081	0.297	29.3	1756.29	7723.75	56.43	7.50	11.3834
0.1357	0.309	29.3	1829.64	10199.45	81.83	7.16	7.8814
0.1622	0.641	31.5	3790.12	12191.23	76.99	6.91	11.8429
0.1901	0.828	31.5	4898.57	14288.24	65.89	6.70	11.4917
0.2205	0.953	31.6	5639.77	16573.16	90.81	6.50	10.1373
0.2471	–	31.6	–	18572.46	–	6.36	–
0.2752	–	31.7	–	20684.51	–	6.22	–
0.302	–	31.7	–	22698.84	–	6.11	–
0.3319	–	31.7	–	24946.18	–	5.99	–
0.3431	–	31.7	–	25787.97	–	5.95	–

A.2.5: Friction Factor vs. Reynolds number for three wire insert with baffle spacing $\boldsymbol{\beta = 6cm}$

m (kg/sec)	Δh (m)	T(℃)	ΔP (N/m^2)	Re	$f_a \times 1000$	$f_0 \times 1000$	f_a/f_0
0.0927	0.261	32.9	1543.92	7059.17	102.14	7.82	13.0612
0.1081	0.402	33.1	2378.52	8323.18	115.95	7.58	15.2964
0.1357	0.426	32.3	2519.97	10333.65	77.81	7.24	10.7474
0.1622	0.919	32.4	5436.82	12351.64	117.49	6.99	16.8093
0.1901	–	32.4	–	14669.26	–	6.73	–
0.2205	–	32.4	–	17015.11	–	6.56	–
0.2471	–	31.7	–	19067.73	–	6.41	–
0.2752	–	31.7	–	21236.09	–	6.27	–
0.302	–	31.7	–	23304.14	–	6.16	–
0.3319	–	31.7	–	26313.09	–	6.00	–
0.3431	–	31.7	–	27201.03	–	5.97	–

A.2.6: Friction Factor vs. Reynolds number for four wire insert without baffles

m (kg/sec)	ΔH (m)	T(°C)	ΔP $({}^{N}/{}_{m^2})$	Re	$f_a \times 1000$	$f_0 \times 1000$	${}^{f_a}/{}_{f_0}$
0.0927	0.026	34.5	153.80	7349.27	10.17	7.75	1.3123
0.1081	0.068	34.5	402.25	8689.22	19.55	7.50	2.6067
0.1357	0.168	35.2	993.79	10907.74	30.65	7.16	4.2807
0.1622	0.231	35.2	1366.46	13037.84	29.50	6.91	4.2691
0.1901	0.295	35.2	1745.05	15280.48	27.43	6.70	4.0940
0.2205	0.355	35.2	2099.98	17724.07	24.53	6.50	3.7738
0.2471	0.396	35.3	2342.51	19862.22	21.79	6.36	3.4261
0.2752	0.432	35.3	2555.46	22120.93	19.17	6.22	3.0189
0.302	0.483	35.3	2857.15	24275.15	17.79	6.11	3.9116
0.3319	0.527	35.3	3117.43	26678.55	16.07	5.99	2.6828
0.3431	0.540	35.3	3194.33	27578.82	15.41	5.95	2.59

A.2.7: Friction Factor vs. Reynolds number for four wire insert with baffle spacing $\boldsymbol{\beta = 24cm}$

m (kg/sec)	Δh (m)	T(℃)	ΔP (N/m^2)	Re	$f_a \times 1000$	$f_0 \times 1000$	f_a/f_0
0.0927	0.048	32.9	283.94	7096.52	18.6	7.81	2.3814
0.1081	0.105	33.1	621.12	8341.65	30.15	7.56	3.9883
0.1357	0.286	33.4	1691.8	10612.94	52.09	7.20	7.2344
0.1622	0.406	33.4	2401.66	12351.64	51.79	6.99	7.409
0.1901	0.537.	33.4	3176.59	14476.25	49.89	6.77	7.3693
0.2205	0.711	33.4	4205.87	16573.16	49.14	6.89	7.1325
0.2471	0.690	33.5	4081.65	18512.46	37.95	6.44	5.8929
0.2752	0.788	33.6	4661.36	20684.51	34.93	6.30	5.4241
0.302	0.822	33.6	4862.48	22698.84	30.27	6.189	4.8915
0.3319	–	33.6	–	24946.18	–	6.072	–
0.3431	–	33.6	–	25787.9	–	6.032	–

A.2.8: Friction Factor vs. Reynolds number for four wire insert with baffle spacing $\boldsymbol{\beta = 12cm}$

m (kg/sec)	Δh (m)	$T(°C)$	ΔP (N/m^2)	Re	$f_a \times 1000$	$f_0 \times 1000$	f_a/f_0
0.0927	0.089	29.3	526.47	6623.42	34.8	7.94	4.3831
0.1081	0.216	29.3	1277.73	7723.75	62.25	7.63	8.159
0.1357	0.511	29.5	3022.78	10199.45	93.21	7.21	12.9277
0.1622	0.754	29.5	4460.23	12191.23	96.34	7.03	13.7038
0.1901	0.929	29.6	5495.43	14288.24	86.39	6.72	12.8552
0.2205	–	29.6	–	16573.16	–	6.51	–
0.2471	–	29.6	–	18572.46	–	6.45	–
0.2752	–	29.7	–	20684.51	–	6.34	–
0.302	–	29.7	–	22698.84	–	6.12	–
0.3319	–	29.7	–	24946.18	–	6.07	–
0.3431	–	29.7	–	25787.97	–	6.03	–

A.2.9: Friction Factor vs. Reynolds number for four wire insert with baffle spacing $\boldsymbol{\beta = 6cm}$

m (kg/sec)	Δh (m)	$T(°C)$	ΔP (N/m^2)	Re	$f_a \times 1000$	$f_0 \times 1000$	f_a/f_0
0.0927	0.139	32.0	822.24	7059.17	54.5	7.82	6.9683
0.1081	0.357	32.0	2111.81	8231.89	102.75	7.58	13.5548
0.1357	0.868	32.1	5134.59	10333.95	158.37	7.24	21.8744
0.1622	–	32.1	–	12351.64	–	6.99	–
0.1901	–	32.1	–	14669.26	–	6.73	–
0.2205	–	32.1	–	17015.11	–	6.56	–
0.2471	–	32.4	–	19067.73	–	6.41	–
0.2752	–	32.4	–	21236.09	–	6.27	–
0.302	–	32.4	–	23304.14	–	6.16	–
0.3319	–	32.4	–	26313.09	–	6.00	–
0.3431	–	32.4	–	27201.03	–	5.97	–