

**Performance Evaluation of a Nanofluid (CuO-H<sub>2</sub>O) Based  
Heat Exchanger with Turbulators**

**A**

**THESIS**

*submitted in partial fulfillment of the requirements for the award of degree of*

**Master of Engineering**

**In**

**Thermal Engineering**

Submitted by:

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Under the guidance of

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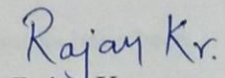
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July 2015**

## CERTIFICATION

I, Rajan Kumar, declare that this thesis report entitled "*Performance Evaluation of a Nanofluid (CuO-H<sub>2</sub>O) Based Heat Exchanger with Turbulators*", submitted towards fulfillment of the requirements for the award of Master's Degree in Thermal Engineering, in Mechanical Engineering Department of Thapar University, Patiala, is entirely my own work. This document has not been submitted for any degree in any other institution.

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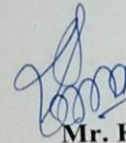
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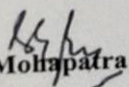
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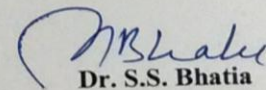
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## ABSTRACT

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An accurate estimation of heat exchanger performance parameters for different types of working fluids is an important aspect of heat exchanger design. Systems can be made to perform effectively with conventional fluids like water or with special fluids with improved performance called nanofluids. This report presents the results of an investigation carried out to provide these important parameters (heat transfer coefficient, pressure drop) for counter flow double tube heat exchangers using CuO-H<sub>2</sub>O (DI) nanofluid. Different volumetric concentrations (0.1%, 0.2%, 0.3%) of CuO (25-55nm) have been used in the preparation of nanofluid as the working fluid for the heat exchanger. System performance has been evaluated using two different turbulators (spring and twister tape). The twisted tape and springs inserts of twist ratio of 5 and 10 and pitch of 5mm and 10 mm respectively are used. The experiments are carried out for the wide range of Reynolds Number ranging from 5000 to 25000. Water at volumetric flow rate of 5 lpm as a cooling medium is made to flow through the annular section of the concentric tube heat exchanger whereas, hot fluids are made to flow through the inner tube section. It has been observed that when water is replaced with nanofluid, an improvement in heat transfer coefficients is observed, which found to be increasing with increase in the volumetric concentrations of nanoparticles in the nanofluid. Moreover, with the application of turbulators, Nusselt number also increases. An improvement around 41% in the heat transfer coefficient is observed when CuO-H<sub>2</sub>O (0.3% vol. conc.) nanofluid is used with twisted tape of twist ratio 5 compared to nanofluid of same concentration without any inserts at Reynolds number of 25000. An enhancement in Nusselt number of about 59% is reported when water is replaced by 0.3% vol. conc. CuO-H<sub>2</sub>O nanofluid with twisted tape of twist ratio 5. Overall, the performance of the heat exchanger is found to be improved with the use of nanofluids and also with different types of inserts.

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## NOMENCLATURE

### List of Symbols

$A_c$	Cross sectional area, $m^2$
$d_i$	Internal diameter of pipe, m
$d_o$	Outer diameter of pipe, $\mu m$
$Re$	Reynolds number
$Pr$	Prantl number
$Nu$	Nusselt number
$F$	Friction factor
$\mu_w$	Dynamic viscosity of water, $kg s^{-1} m^{-1}$
$\mu_{nf}$	Dynamic viscosity of nanofluid, $kg s^{-1} m^{-1}$
$\phi$	Percentage concentration of nanofluid by volume
$L$	Total test section length, m
$k_w$	Thermal conductivity of water, $W m^{-1} K^{-1}$
$K_{nf}$	Thermal conductivity of nanofluid, $W m^{-1} K^{-1}$
$M$	Mass flow rate of water, $kg s^{-1}$
$m_{nf}$	Mass flow rate of nanofluid, $kg s^{-1}$
$P$	Density of water, $kg m^{-3}$
$\rho_{nf}$	Density of nanofluid, $kg m^{-3}$
$P$	Pressure, Pa
$V$	Velocity of the fluid, $ms^{-1}$
$G$	Acceleration due to gravity, $ms^{-2}$

$Q$	Volume flow rate, $m^3$
$U$	Overall heat transfer coefficient, $Wm^{-2}K^{-1}$
$H$	Convective heat transfer coefficient, $Wm^{-2}K^{-1}$
$t_h$	Temperature of hot fluid, K
$t_c$	Temperature of cold fluid, K
$h_f$	Pressure drop head, m
$R_{ov}$	Overall thermal resistance
$R_i$	Thermal resistance corresponding to the internal convection
$R_w$	Thermal resistance corresponding to the tube wall
$R_o$	Thermal resistance corresponding to the external convection
$c_{pw}$	Specific heat capacity of water, $KJkg^{-1}K^{-1}$
$c_{p\ nf}$	Specific heat capacity of nanofluid, $KJkg^{-1}K^{-1}$
$V_{np}$	Quantity of nanoparticle
$V_{bf}$	Quantity of base fluid
$V_{nf}$	Quantity of nanofluid
$W_{np}$	Weight of the nanoparticle
$W_{bf}$	Weight of nanofluid

### **Subscripts**

I	Inlet
O	Outlet
C	Cold
H	Hot

### **Abbreviations**

Tape 1	Twisted tape of twist ratio 5,
Tape 2	Twisted tape of twist ratio 10,
Spring 1	Spring pitch of 5 mm pitch
Spring 2	Spring pitch of 10 mm pitch
nf	Nanofluid
LMTD	Log mean temperature difference
bf	Base fluid
avg	Average
ov	Over all
lpm	Litre per minute

## INTRODUCTION

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### 1.1 Introduction

#### 1.1.1 Heat exchanger

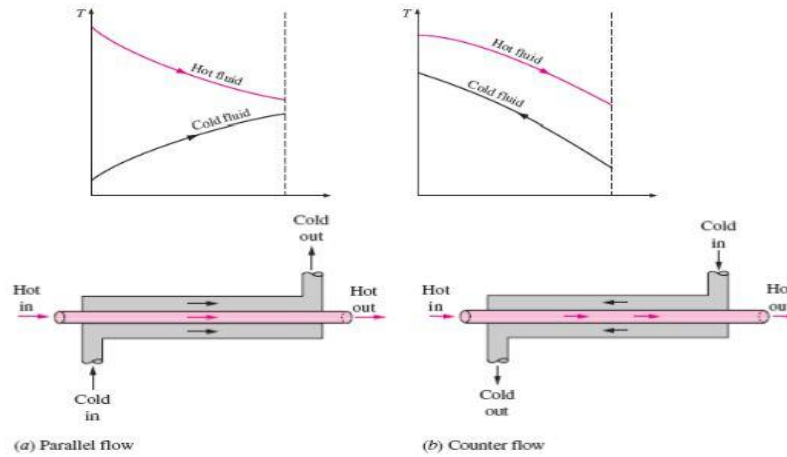
A heat exchanger is equipment that is used for efficient heat transfer between two media that are at different temperatures. The media may be separated by a solid wall to prevent mixing or they may be in direct contact with each other.

Flow arrangement in heat exchangers – Flow arrangement refers to the relative movement of two fluids taking part in heat transfer process. There are three primary classifications of heat exchangers according to their flow arrangement

1. Parallel flow heat exchangers in which the two fluids enter at the same end and travel in parallel to one another to the other end.
2. Counter flow heat exchangers in which the fluids enter in the heat exchanger from opposite end.
3. Cross flow heat exchangers in which fluids travel roughly perpendicular to one another.

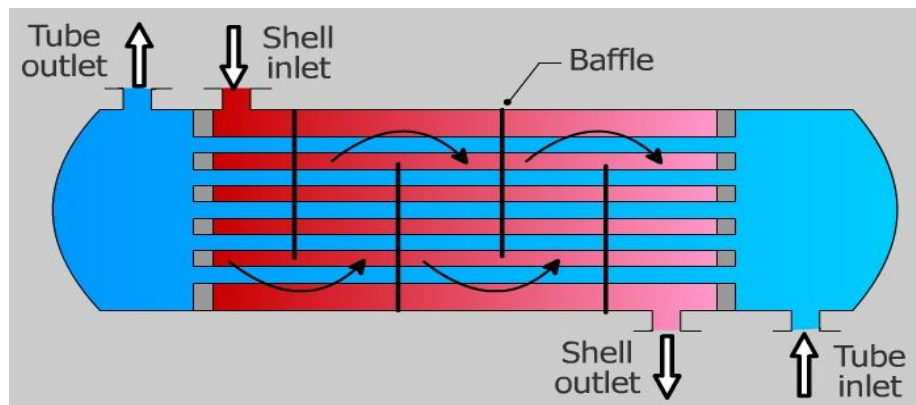
#### 1.1.2 Types of heat exchangers

1. Double pipe heat exchangers - A double pipe heat exchanger consists of one pipe inside another pipe. One fluid flows through the inside pipe and other flows through the annulus between two pipes. The wall of inner pipe is the heat transfer surface. These heat exchangers are cheap for both design and maintenance and are widely used in small scale industries. Owing to their low efficiency, these are rarely used in large scale applications.



**Figure 1.1 Double pipe heat exchanger**

2. Shell and tube heat exchanger – Shell and tube heat exchangers consists of a series of tubes. One set of these tubes contains the fluid which needs to be heated or cooled and the other fluid runs over the tubes. These heat exchangers are normally used for high pressure (> 30 bar) and high temperature(>260 c) applications.



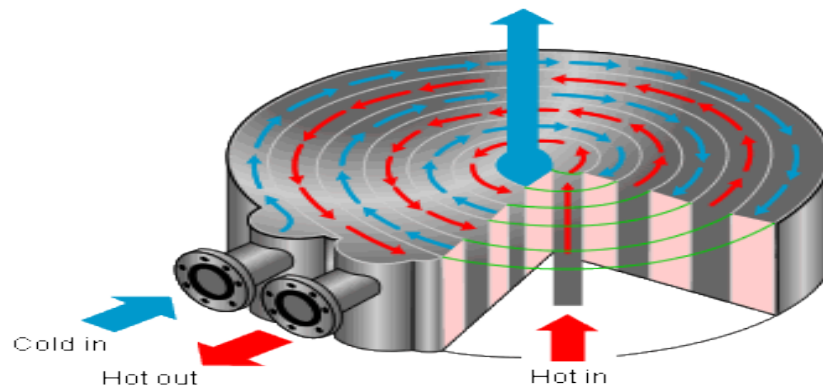
**Figure 2.2 Shell and tube heat exchanger**

3. Plate heat exchanger – It is composed of multiple, thin, slightly separated plates that have very large surface areas and fluid flow passages for heat transfer. This stacked-plate arrangement can be more effective, in a given space.



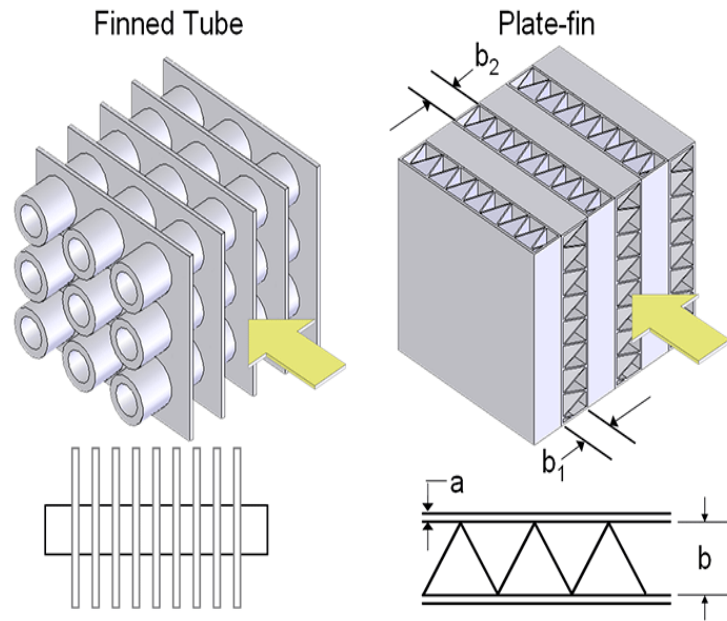
**Figure 3.3 Plate heat exchanger**

4. Spiral heat exchanger - They have helical tube configuration and are manufactured by coiling the flat plates. They provide a very compact design and are very useful in those situations where there are space limitations. These heat exchangers are usually used for viscous liquids and slurries and their classical example is the use in pasteurization of milk.



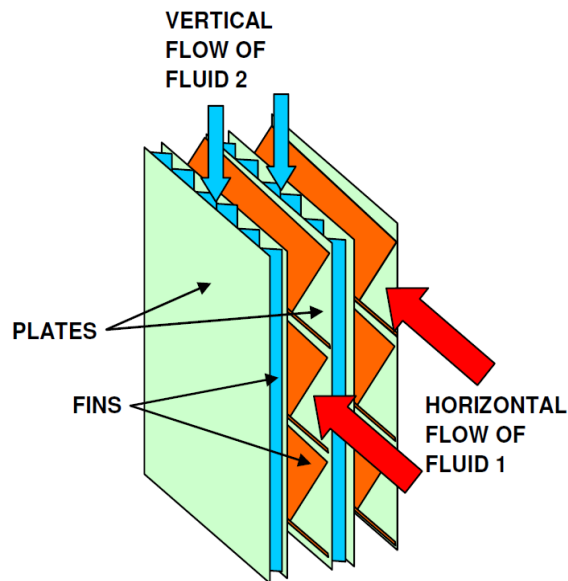
**Figure 4.4 Spiral heat exchanger**

5. Plate Fin heat exchanger- These heat exchangers are widely used for gas to gas applications . In their design, corrugated fins are sandwiched between flat plates resulting in a very compact design.They are usually made from aluminium and provide very high heat transfer rate per unit volume. They are normally used in gas turbine plants and nuclear power plants.



**Figure 5.5 Plate fin heat exchanger**

6. Tubular fin heat exchanger- They are used for gas to liquid applications. Regarding their design, the fins are attached to outside of tubes. They can have helical, longitudinal or transverse arrangements depending upon the orientation of fins. Their applications includes waste heat boilers and air cooled condensers.



**Figure 6.6 Tubular fin heat exchanger**

### Enhancement methods need

1. To reduce capital cost of new heat exchangers
2. To reduce size of heat exchanger
3. To meet limitations of space availability
4. To increase efficiency of existing systems
5. To mitigate fouling of heat exchangers
6. To eliminate cleaning and fouling removal costs
7. To minimise maintenance costs
8. To fight ill effects such as scaling and corrosion.

### 1.1.3 Classification of enhancement techniques

Improvement of thermo-hydraulic performance of heat exchangers is the primary aim of heat transfer enhancement or augmentation techniques. These augmentation techniques are currently classified in two broad categories.

1. Passive techniques
2. Active techniques

#### 1. Passive Techniques

These techniques alter geometry of the heat exchanger by making modification such as incorporation of inserts or additional devices. These devices increase the heat transfer coefficients by changing the existing flow behavior. But such alteration leads to increase of pressure drops in the heat exchangers. In passive techniques, for extended surfaces, heat transfer area is increased on the side of the extended surface to enhance the heat transfer coefficients.

Heat transfer enhancement by passive techniques can be achieved in following ways

- Treatment of surfaces-

This technique alters the surfaces by invoking pits, cavities and scratches to change the heat transfer area. They find their primary application in boiling and condensation.

- Treated surfaces-

Treated surfaces consist of various surfaces such as continuous or discontinuous surfaces, roughness alterations, variety of coatings etc. The roughness invoked in the surface due to the treatment does not affect the heat transfer rate in case of single phase. These treated surfaces are mostly use to enhance the heat transfer coefficient of two phase heat transfer.

- Surface treatment for boiling-

For boiling applications, the passive techniques involve the following treatments on the surfaces-

- a. Machined or grooved surfaces
- b. Formed or modified low-fin surfaces
- c. Multilayered surfaces
- d. Coated surfaces

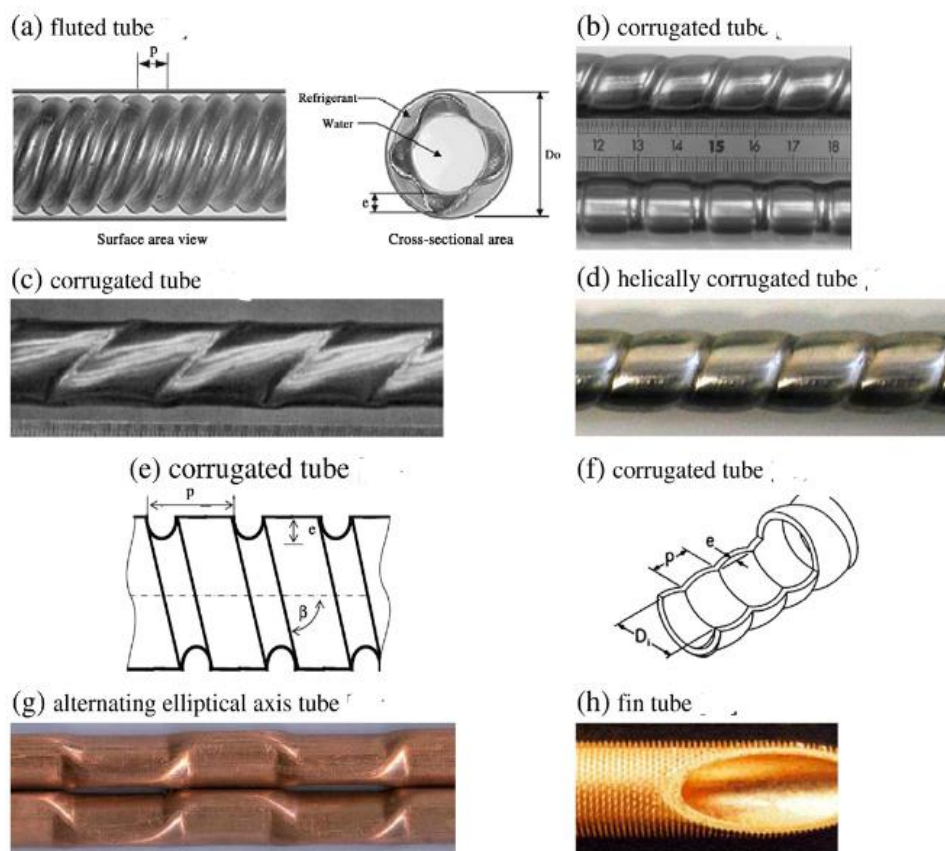
Treated surfaces for boiling enhance the heat transfer rate by providing large number of vapor traps called as nucleation sites on the surfaces for bubble formation. For fluids with high wetting tendency such as refrigerants, cryogenes, alkali liquid metals etc, the cavities present on the heated surfaces experience sub-cooled liquid accumulation. For fluids having high surface tension, non-wetting material coatings are found to be effective in case of nucleate boiling. This coating is either done on the heated surface or in the pits and cavities. Teflon coated stainless steel surfaces are used as heated surfaces which results in increase of heat transfer coefficients by three to four times.

- Surface treatment for Condensing-

Treated surface enhance drop wise condensation, an ideal way which prevents surface wetting and breaking up the condensation film into droplets. This process enhances drainage and vapor removal at the cold interface. Heat transfer is increased by 10 to 100 times in case of drop wise condensation when compared with that of film wise condensation. Non-wetting inorganic compounds, noble metals, organic polymers are effectively used for coating the heat transfer surfaces. Organic coatings are widely used in steam systems.

- Rough surfaces-

Rough surfaces modifications create disturbances in viscous sub-layer region of the flow. Hence these techniques are applicable in single phase turbulent flows. Surface roughness promotes turbulence near the wall region where viscous laminar sub layer is present. This turbulence causes high momentum and heat transfer. These rough surfaces have high effectiveness in case of turbulent single phase flows compared to laminar flows.



**Figure 7.7 Different types of corrugated tubes**

External surface roughness is created by creating grooves in the heat transfer surfaces and are used in double pipe and shell and tube type heat exchangers. Bergles and Champagne proposed the concept of variable roughness which is obtained by using wire coiled insertions made of an alloy called as a shape memory alloy (SMA). SMA changes its geometry in accordance with change in temperature.

- Extended surfaces-

Plain fins are one of the oldest types of extended surfaces tracing its use in many heat exchangers. Finned surfaces have tendency to disturb the flow field as well as increase the surface area. The disturbance in flow field increases the heat and momentum exchange.

The increase in heat transfer area using finned surfaces is very effective with fluids having low heat transfer coefficients. The examples of finned type heat exchangers are shell and tube type exchangers, plate fins for compact heat exchangers, finned heat sinks for electronic cooling. In electrical and electronic devices, finned surfaces enhance heat transfer in natural and forced convection.

Boundary layer separation of the fluids enhances heat transfer; this is done by use of segmented or interrupted longitudinal fins. This fin promotes boundary layer separations of the fluid and disturbs the bulk flow field inside circular tubes. Plate fin or tube and plate fin type of compact heat exchangers, find their increasing use in automobiles, waste heat recovery systems, refrigeration and air conditioning, cryogenic, propulsion system etc. An offset strip fins, louvered fins, perforated fins, wavy fins are the typical shapes of fin used in extended type heat exchangers.



**Figure 8.8 Tubes with different twist ratio and rib ratio**

- Swirl flow devices-

Swirl flow type insertion devices produce swirls or secondary circulations on the axial flow in the channel. The common examples of swirl flow devices are helical twisted tape, twisted ducts etc. These swirl flow devices can be used for both single phase and two-phase flows.

Swirl flow inducing devices includes tube inserts, altered or differently arranged tube flow arrangements and duct geometry modifications. Dimples, ribs, helically twisted tubes are different types of duct geometry modifications. Tube inserts include twisted-tape inserts, helical, strip, wire coils etc. Twisted type inserts are amongst the popular swirl flow devices. This is because of their performance in thermal-hydraulic systems in single phase, boiling and condensation, forced convection.

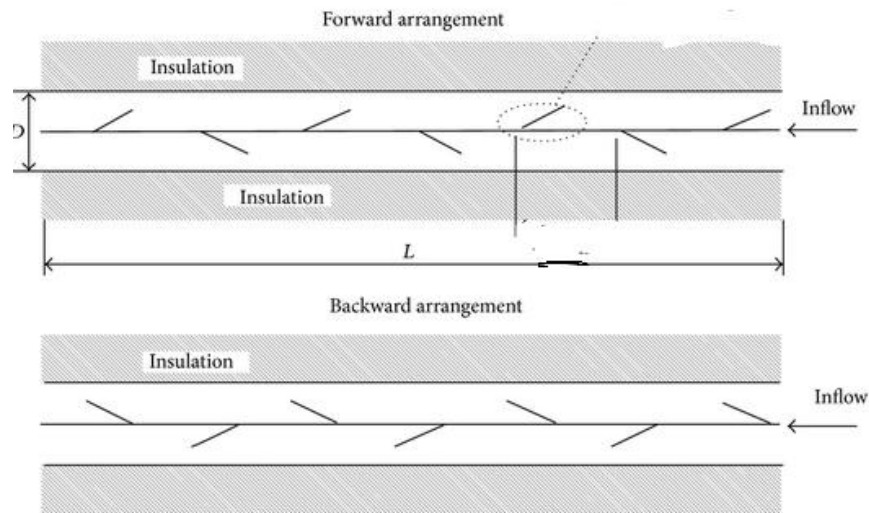
Twisted tape inserts are known to increase heat transfer coefficients with small increase in pressure drop. This type of swirl flow devices are used in single phase processes. Twisted tape type inserts in heat exchangers reduces the size significantly for same heat load. They are widely used in industries to generate swirl flow in the fluid.

Twisted tapes increase the heat duties when used in the existing heat exchangers. They come with multiple-tube bundles which are easy to fit and remove, helping cleaning the tubes during fouling conditions.

Twisted tape, wire coils, ribs, dimples separate the primary flow from the secondary flows causing increase of heat transfer. The effective reduction in flow area causes increase of velocity of flow thus increasing the pressure drop as well. This pressure drop leads to increased secondary flow. The secondary flow creates swirl and mixing of fluid, enhancing the temperature gradients leading high heat transfer.



**Figure 9.9 Twisted tape**



**Figure 10.10 Forward and backward arrangements of louvered strips**

- Coiled tubes inserts

Curvature of coils promotes high heat transfer coefficients in single phase flows by creating secondary flow and vortices. This type of arrangement of coils leads to a highly compact type heat exchangers.

At the curved surface of the duct, continuous change in the bulk velocity occurs. This causes secondary flow to occur in the curved tube. These tubes are used in domestic water heaters, chemical process reactors, solar heating system, industrial and marine boilers, kidney dialysis devices etc. Centrifugal force is induced on the fluid in motion due to the curvature of the coil, this generates secondary flow. This flow depends on various geometric characteristics of the duct such as its radius of curvature, helical number etc.

## 2. Active Techniques

Active techniques when compared to passive ones are more complex from its use and design point of view. This method requires external power source to modify the flow s as to improve the rate of heat transfer. Its application is limited because of the need of external power and hence in comparison to passive techniques, not much of its potential has been explored.

#### 1.1.4 Introduction to nanofluids

For various industrial applications, one of the most challenging part is designing of energy efficient and enhancement methods in heat transfer rate. Cooling of heat transfer equipment's such as higher power engines, fuel cell, microelectronic devices operating at high speeds, super conducting magnets, ultrahigh-heat flux optical devices, x-ray machines, and high-powered lasers etc. for maintaining the desired performance as well as safe operations of these equipment. Due to compact size and high power to weight ratio, the heat load exceeds  $2000 \text{ W/cm}^2$ .

Cooling agents such as water, ethylene glycol, and oil etc. are characterized with having low heat transfer properties. Hence there is ever increasing demand to develop fluids having high heat transfer properties. Thermo-fluid properties play an important role in development of such high heat transfer efficient devices. Dispersion of small size particles of the order of millimeter or micrometer size was first done by Maxwell in the year 1873. The major problem of this experimentation was settling down of large particles in suspension. The particles used in dispersion had high thermal conductivity, even of the order of hundreds and thousands time greater than the conventional heat transfer fluids. A table of comparison is shown below [table no-]

Nanofluids consist of solid nanoparticles of the size ranging from 1-100 nm, suspended in liquid. Nanofluid hence forms a solid liquid composite material. They enhance the thermal and heat transfer properties of the fluid greatly and hence this is one of the most researched fields of technology. Modern nanotechnology has ability to produce nanofluids having metallic as well as non-metallic particles and carbon nanotube having dimensions of the range of nanometers having extensively high thermal conductivity. Nano particles are characterized by having unique mechanical, optical electrical as well as thermal properties and are engineered by forming suspension of nanoparticles having average size below 100 nm in conventional heat transfer such as water, ethylene glycol, oil etc. metallic liquids have greater thermal conductivity as compared to non-metallic liquids as shown in (table 1.2). Hence suspended metallic solid particles in a fluid increases the thermal conductivity of the nanofluid compared to that of the traditional heat transfer fluids.

### 1.1.5 Nano size significance

The major problem that arose with the Maxwell's experimentation of adding micro-sized particles was that these particles settled rapidly in the base fluid which formed sediment layer when the low velocity was low and thus decreasing the thermal conductivity. They also cause clogging problems by forming agglomeration in the fluid during its circulation, especially in the micro channels. These particles coagulate to form high massed particles which at high kinetic energy cause erosions. Nanoparticle concentration plays an important role in the enhancement of thermal conductivity. As the concentration of the particles increases, the pressure drop also increases. Nanoparticles have less momentum transfer to the device wall and hence reduce the common problems of erosion and clogging. Nanosize particles have very good thermal properties with high surface to volume ratio.

**Table 1.1 Thermal conductivity of various materials**

Materials		Thermal conductivity (W/m.K)
Metallic solids	Silver	429
	Copper	401
	Aluminium	237
Non-metallic solids	Diamond	3300
	Carbon nanotube	3000
	Silicon	148
	Alumina( $Al_2O_3$ )	40
Metallic liquids	Sodium at 644 K	72.3
Non-metallic liquids	Water	0.613
	Ethylene glycol	0.253
	Engine	0.145

**Table 1.2 Comparison of micro size and nanosize particles at the same operating conditions**

Properties	Micro particles	Nanoparticles
Stability	Settle	Kinetically stable (long lived in suspension )
Surface/volume ratio	1	1000 times larger than micro particles.
Conductivity	Low	High
Erosion & clogging	Yes	No
Pump power	High	Low

### **1.1.6 Type of host Fluids**

Metallic fluids

- Sodium

Non-metallic fluids

- Water
- Ethylene glycol
- Oil

**Table 1.3 Different material for nanoparticle**

Material	Form	Thermal conductivity
Carbon	Nanotube	1800-6600
	Diamond	2300
	Graphite	110-190
Metallic solids	Silver	429
	Copper	401
	Nickel	237
Non metallic	Silicone	148
Metallic liquids	Aluminium	40
	Sodium at 644K	72.3
Others	Water	0.613
	Ethylene	0.253
	Engine oil	0.145

### **1.1.7 Preparation of nanofluid**

A two-step process is followed to synthesize Nanofluids. In the first step, nanoparticles are created by mechanical comminuting followed by chemical reaction, vapor condensation or decomposition of decomposition of organic compound. The second step then involves dispersion of the produced nanoparticles into the heat transfer fluids. This is done by mechanical agitation or ultra-sonication. Depending on the type of interfaces between nanoparticles and the heat transfer fluid if required, nanofluids are homogenized using surfactants. This two-step synthesis method is

desirable as it produces nanoparticles such that no undesirable coatings or other contaminants are produced. But agglomeration of nanoparticles may occur. The particles formed by this method clump to each other, which can be easily dispersed again in liquids by mechanical dispersion. But though they re-disperse, they again clump very soon to form large particles and these particles settle quickly in the suspension.

Uniform dispersions are stabilized by steric barrier which surrounds the nanoparticles and forms a coating layer. When this layer is absorbed by the solid particle, the surfactant molecules produce a barrier to prevent agglomeration of nanoparticles.

#### **1.1.8 Advantages of Nanofluid**

1. They have higher stability when compared to micro size to mili-sized particles.
2. They have high heat conducting capacity
3. Due to nanosize of the particles, they have high surface to volume ratio and hence provides large area for heat interaction between nanoparticles and base fluid.
4. Negligible or small amount of pressure drop, hence it requires less pump work because of the small particle size and low volume fraction.
5. Miniaturization is possible as no erosion or clogging takes place in device channels.
6. By changing the volume fraction of nanoparticles in base fluid thermal conductivity can be adjusted.

#### **1.1.9 Application of nanofluid**

##### **1. Transformer cooling**

Transformer cooling application of nanofluids helps in reduction of transformer size and weight. The increased thermal transport of transformer oils increasing into either reduction in the size of new transformers at the same level of power transmitted or an increase in the performance of existing transformers.

##### **2. Vehicles cooling**

Nanofluids are used in engine oil and coolants to assist cooling. They are

also used in lubricant, oil and transmission fluid

### 3. Defense applications

Nanofluids provides advanced cooling or military vehicles, submarines, and high-power laser diodes. High powered military electronics, military vehicle components, radars, and lasers use nanofluids for high-heat-flux cooling of the level of thousands of  $W/cm^2$ .

### 4. Other applications

Other possible areas for the application of nanofluids technology include cooling a new class of super powerful and small computers and other electronic devices for use in military systems, airplanes, or spacecraft as well as for large-scale cooling.

#### **1.1.10 Challenges of nanofluids**

Long term stability of nanofluid is a challenging task. They make agglomerate after sometime due to strong Van Der Waals forces between molecules when dispersed. To get metastability in nanofluids some physical or chemical treatment is given. For example ultra-sonication, surfactants. High cost of production of nanoparticles .High nanoparticle concentration and nanoparticle size pump power is more because viscosity is increased. As Reynolds number of nanofluid increases, the pumping power is increases.

### LITERATURE REVIEW

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**Akpinar et al. (2004)** investigated the effect on heat transfer rates, friction factor and exergy loss of swirl generators with holes for the entrance of fluid by placing them at the entrance section of inner pipe of heat exchanger. Various swirl generators having circular holes of different numbers and diameters were used. Cold water and hot air were passed through the annulus and inner pipe respectively. Experiments were performed for both parallel and counter flow type heat exchangers by varying the Reynolds numbers from 8500 to 17500. Analysis was made for heat transfer, friction factor and exergy with and without swirl generators as condition and compared with each other. It was seen that Nusselt number was increased up to 130% compared with the empty tube heat exchanger. While the Nusselt number was increased up to 113% in swirl generator having 20 circular holes with 6 mm diameter, the increase was 109% with 9 mm diameter holes. The dimensionless NTU and exergy loss were increased by increasing the number of holes and decreased with increasing the holes diameter. The increase was about 1.25 times higher than that of the empty tube for swirl element having 20 holes with 3 mm diameter at the highest Reynolds number. The average increase friction factors for swirl element having 20 holes with 3 mm diameter was about 2.9 times in comparison with that for the inner pipe without swirl generators at the highest Reynolds number.

**Viedma et al. (2005)** studied the thermo hydraulic characteristics in laminar, transition and turbulent flow by using water and water–propylene glycol mixtures in circular tube with helical wire coils as inserts. Six wire coils were tested within a geometrical range of helical pitch  $1.17 < p/d < 2.68$  and wire diameter  $0.07 < e/d < 0.10$ . During experiment, he varied Reynolds numbers from 80 to 90,000 and Prandtl numbers from 2.8 to 150. In laminar region, the performance improved with Reynolds number. When wire coils are inserted in the tube, at  $Re = 500$ , heat transfer increased from 30% to 40% and when  $Re = 1500$ , heat transfer increased from 140% to 160%. Wire coils increased pressure drop up to nine times and heat transfer up to four times

compared to the smooth tube independently in turbulent flow. Wire coil inserts offer their best performance within the transition region.

**Solano et al. (2007)** studied enhancement of heat transfer rate by inserting wire coils of different pitch in a smooth tube in laminar and transition regimes. Experiment was performed by varying Reynolds numbers from 10 to 2500, Prandtl number from 200 to 700 and Rayleigh number from  $3 \cdot 10^6$  to  $3 \cdot 10^8$ . Heat transfer experiments were carried out under uniform heat flux (UHF) conditions. Three wire coils were used having same diameter and different pitch. All the three wire coils were given their names according to their pitch length as short pitch, intermediate pitch and long pitch. The friction factor increases between 5% and 40% in the fully laminar region. The transition from laminar flow to turbulent flow is smooth and without any drop in pressure that a smooth tube presents. The fully laminar region covers up to  $Re = 200$ . Wire coils did not enhance the heat transfer with respect to the smooth tube in this region. In this low Reynolds number region, the heat transfer increase is quite remarkable. For short pitched wire coil at  $Re = 1000$  and  $Pr = 360$ , the Nusselt number is eight times as compared to the one of a smooth tube. In the range of Reynolds numbers between 700 and 2500, the increment in heat transfer is very higher with the wire coils than the one which we get with the twisted tapes. At Reynolds number ranging between 200 and 1000, there is a remarkable increase in heat transfer with wire coils.

**Promvongse et al. (2008)** studied the effect of insertion of wire coils in conjunction with twisted tapes on heat transfer and friction factor in a uniform heat-flux. Flow regime was taken as turbulent flow; there was a circular pipe within which experiment was conducted using air as the test fluid. The wire coil used as a turbulator is placed inside the test tube while the twisted tape is inserted into the wire coil to create a continuous swirl flow along the tube wall. The effects of insertion of the two turbulators with different coil pitch and twist ratios on heat transfer and friction loss in the tube was experimented. The flow rate was varied with Reynolds number which was in the range from 3000 to 18,000. The experimental results are compared with those obtained from using wire coil/twisted tape alone, apart from the smooth tube. The heat transfer was found to be 130–250% and 180–400% more from both the twisted tape and wire coil inserts over those from the twisted tape and wire coil alone respectively which depended on the Reynolds number, coil spring pitch ratio, and

twist ratio of twisted tape values. The increase in Nusselt number from using the twisted tape alone is found to be about 20–50% above the smooth tube. Also by using the wire coil alone is around 100–110% increment in Nusselt number. The use of combined twisted tape and the wire coil inserts have higher heat transfer rate than that of the wire coil alone around 150–300% which was dependent on Reynolds number values. The friction factor value decreased with the increase of Reynolds number. The friction factor value increase is much higher than in the smooth tube. The average increase in friction losses for using the twisted tape alone is around five times of the smooth tube. The friction factor value for both the wire coil and the twisted tape is around 2–4.5 times above that for the wire coil alone about 6–12.5 times over that for the twisted tape. This shows that the insertion of the twisted tape and the wire coil leads to a substantial increase in friction losses in the tube. The best operating regime for combined both the turbulators was found at lower Reynolds number values for the lowest values of the twist ratio coil spring pitch.

**Eiamsa-ard et al. (2008)** investigated heat transfer and friction characteristics by using louvered strips as turbulator which was inserted in a concentric tube heat exchanger. The louvered strip helped to create turbulent flow which increase the heat transfer rate of the heat exchanger. The flow rate of the tube was adjusted by varying the Reynolds number from 6000 to 42000. There were two ways to employ the experiment to create turbulent flow either the louvered strips with forward or backward arrangements or the louvered strip with various inclined angles ( $\theta=15^\circ$ ,  $25^\circ$  and  $30^\circ$ ) inserted in the inner tube of the heat exchanger. In their experiment, cold water was flowed in the annulus and hot water was flowed through the inner tube. Experimental results showed that the use of louvered strips leads to a higher heat transfer rate over the plain tube. The increases in Nusselt number and friction loss for the Louvered strip insertions can be used efficiently to increase heat transfer rate because the turbulence intensity induced could enhance the heat transfer. The highest heat transfer rate was reached for the backward inclined angle of  $30^\circ$  due to the increase of strong turbulence intensity. All inclined louvered strip arrangements would significantly enhance the heat transfer in comparison with the plain tube. The forward louvered strip arrangements could increase the heat transfer rate by approximately 150% to 284%, while the backward arrangements could increase the heat transfer rate by approximately 133% to 264%. Due to blockage effect of fluid

flow, friction factors were approximately 280% to 413% over the plain tube for the forward arrangement and about 155% to 233%, for the backward arrangements. Moreover, the use of the louvered strip with backward arrangement led to better overall enhancement ratio than that of forward arrangement around 9% to 24% for all inclined angles. The mean overall enhancement ratio was around 1.17 to 1.98 for backward arrangement and 1.11 to 1.8 for forward arrangement. In addition, the use of the louvered strip with backward arrangement leads to better overall enhancement ratio than that with forward arrangement around 9% to 24%.

**Sharma et al. (2009)** evaluated the heat transfer coefficient and friction factor for flow in a tube and with twisted tape as insert with  $\text{Al}_2\text{O}_3$  nanofluid in the transition flow regime. The flow rate is varied between Reynolds number 3000 to 9000. The experimental results showed increment of convective heat transfer with  $\text{Al}_2\text{O}_3$  nanofluids compared to flow with water. It is observed that the equation of Gleninski applicable in transitional flow range for single-phase fluids showed considerable deviation when compared with values obtained with nanofluid. The heat transfer coefficient of nanofluid with 0.1% volume concentration is 23.7% higher when compared with water at Reynolds number of 9000. Heat transfer coefficient and pressure drop with nanofluid has been experimentally determined with tapes of different twist ratios and found to deviate with values obtained from equations developed for single-phase flow. The maximum friction factor with twisted tape at 0.1% nanofluid volume concentration is 1.21 times that of water flowing in a plain tube. The heat transfer enhancement of  $\text{Al}_2\text{O}_3$  nanofluid in circular tube with 0.1% volume concentration is 13.77% at 3000 Reynolds number and 23.69% at 9000 Reynolds number when compared to water. Heat transfer rates with twisted tape insert are obtained with nanofluid of 0.1% volume concentration 36.96% at Reynolds number 3000 and 44.71% at Reynolds number 9000 compared to flow of nanofluid in a plain tube.

**Sundar et al. (2009)** studied thermo physical properties like viscosity and thermal conductivity of  $\text{Al}_2\text{O}_3$  nanofluid through experiments at different volume concentrations and temperatures. Experiments were conducted in the Reynolds number range of 10000 to 22000 with tapes of different twist ratios in the range of  $0 < H/D < 83$ . Convective heat transfer coefficient and friction factor for flow in a plain tube and with twisted tape insert was found experimentally for  $\text{Al}_2\text{O}_3$  nanofluid at

various volume concentrations. At Reynolds number 10000, the heat transfer coefficient and friction factor of 0.5% volume concentration of Al<sub>2</sub>O<sub>3</sub> nanofluid with twist ratio of 5 is 33.51% and 1.096 times respectively higher compared to flow of water alone in a tube and at Reynolds number 22000, friction factor with nanofluid of same volume concentration for the same twist ratio is higher when compared to water in a plain tube 1.2657 times.

**Zhou et al. (2010)** reviewed the definition of heat capacity and clarifies the defined specific heat capacity and volumetric heat capacity. The specific heat capacity and volumetric heat capacity, with our measured experimental data for CuO nanofluids were discussed. For illustration, the specific heat capacity of nanofluid made by ethylene glycol and copper dioxide nanoparticle inclusions measured at room temperature were compared with two kinds of models for determination of the specific heat capacity of nanofluid. The result indicates that the specific heat capacity of CuO nanofluid decreases gradually with increasing volume concentration of nanoparticles. The other simple mixing model fails to predict the specific heat capacity of CuO nanofluid. The nanoparticle size effect and solid-liquid interface effect on the specific heat capacity of nanofluid were discussed. The effect of liquid adsorption on suspended nanoparticles surface will also increase the specific heat capacity of nanofluid to some extent with increasing nanoparticles' volume concentration.

**Eiamsa-ard et al. (2010)** studied the heat transfer, friction factor and thermal performance behaviors in a tube equipped with the combined devices between the twisted tape (TT) and constant/periodically varying wire coil pitch ratio. The periodically varying three coil pitch ratios were arranged into two different forms: (1) D-coil (decreasing coil pitch ratio arrangement) and (2) DI-coil (decreasing/increasing coil pitch ratio arrangement) while the twisted tapes were prepared with two different twist ratios. Each device alone is also tested and the results are subjected for comparison with those from the combined devices. The experiments were conducted in a turbulent flow regime with Reynolds numbers ranging from 4600 to 20000 using air as the test fluid. At low Reynolds number, the experimental results reveal that the compound devices of the TT with Y=3 and the DI-coil, provide the highest thermal performance which is around 6.3%, 13.7%, 2.4% and 3.7% higher than the wire coil alone, the TT alone, the TT with uniform wire coil, and the TT with D-coil

respectively. Over the range investigated, the highest thermal performance factor of around 1.25 is found by using DI-coil in common with the TT at lower Reynolds number.

**Kongkaitpaiboon et al. (2010)** studied the effect of the circular-ring turbulator (CRT) on the heat transfer, fluid friction characteristics and thermal performance factor behaviors in a heat exchanger tube. The experiments were conducted by insertion of CRTs with various geometries, including three different diameter ratios ( $DR = d/D = 0.5, 0.6$  and  $0.7$ ) and three different pitch ratios ( $PR = p/D = 6, 8$  and  $12$ ). During the test air at  $27\text{ }^{\circ}\text{C}$  was passed through the test tube which was controlled under uniform wall heat flux condition. The Reynolds number was varied from 4000 to 20000. According to the experimental results, heat transfer rates in the tube fitted with CRTs are augmented around 57% to 195% compared to that in the plain tube. The maximum thermal performance factor of 1.07 is found by the use of the CRT with  $DR=0.7$  and  $PR=6$ . In addition, the results also reveal the CRT with the smallest pitch and diameter ratios offers the highest heat transfer rate in accompany with the largest pressure loss.

**Wongcharee et al. (2011)** investigated heat transfer, friction and thermal performance characteristics of CuO/water nanofluid. The nanofluid was employed in a circular tube equipped with modified twisted tape with alternate axis (TA). The three concentration variation of nanofluid was of 0.3%, 0.5% and 0.7% by volume while the twisted ratio ( $y/W$ ) of TA was kept constant at 3. The experiments were performed in laminar regime in the range of Reynolds number between 830 to 1990. The uses of nanofluid together with typical twisted tape (TT), TA alone and TT alone were also examined. The obtained results reveal that Nusselt number increases with increasing Reynolds number and nanofluid concentration. By the individual uses of TA and TT, Nusselt numbers increase up to 12.8 and 7.2 times of the plain tube respectively. The simultaneous use of nanofluid and TA improves Nusselt number up to 13.8 times of the plain tube. Over the range investigated, the maximum thermal performance factor of 5.53 is found with the simultaneous employment of the CuO/water nanofluid at 0.7% volume and the TA at Reynolds number of 1990.

**Suresh et al. (2012)** compared the thermal characteristics of  $\text{Al}_2\text{O}_3/\text{water}$  and CuO/water nanofluids in transition flow through a circular pipe fitted with helical

screw tape inserts. The helical screw tape inserts with twist ratios  $Y = 1.78, 2.44$  and  $3$  were used in the experimental study using  $0.1\%$  volume concentration  $\text{Al}_2\text{O}_3/\text{water}$  and  $\text{CuO}/\text{water}$  nanofluid. The average enhancements in Nusselt number for water with twist ratios of  $1.78, 2.44$  and  $3$  were  $156.24\%, 122.16\%$  and  $89.22\%$  respectively when compared to plain tube. The average increase in Nusselt number corresponding to the twist ratios of  $1.78, 2.44$  and  $3$  were  $166.84\%, 128.67\%$  and  $89.22\%$  respectively for  $\text{Al}_2\text{O}_3/\text{water}$  nanofluid. In the case of  $\text{CuO}/\text{water}$  nanofluid, the enhancements in Nusselt number were  $179.82\%, 144.29\%$  and  $105.63\%$  for twist ratios  $1.78, 2.44$  and  $3$  respectively. The experimental results show that  $\text{CuO}/\text{water}$  nanofluid gives better enhancement in heat transfer than  $\text{Al}_2\text{O}_3/\text{water}$  nanofluid. It can be concluded that helical screw tape inserts give better thermal performance when used with  $\text{CuO}/\text{water}$  nanofluid than with  $\text{Al}_2\text{O}_3/\text{water}$  nanofluid. Thermal performance based on the constant pumping power criteria showed that the thermal performance factor varies between  $1.22$  and  $1.14$  for twist ratio  $Y = 1.78$ , between  $1.15$  and  $1.02$  for twist ratio  $Y = 2.44$ , and between  $1.07$  and  $0.94$  for twist ratio  $Y = 3$  when used with  $0.1\%$   $\text{CuO}/\text{water}$  nanofluid. The thermal performance factors are  $1.2$  to  $1.02$  for  $Y = 1.78$ ,  $1.13$  to  $0.95$  for  $Y = 2.44$ ,  $1.06$  to  $0.87$  for  $Y = 3$  when used with  $\text{Al}_2\text{O}_3/\text{water}$  nanofluid of  $0.1\%$  volume concentration.

**Bhuiya et al. (2012)** investigated the heat transfer and friction factor characteristics in a circular tube fitted with twisted wire brush inserts for the turbulent flow region. The twisted wire brush inserts were made with four different twisted wire densities of  $100, 150, 200,$  and  $250$  wires per centimeter by winding a  $1$  mm diameter of the copper wire over a  $5$  mm diameter of two twisted iron core-rods. Flow rates are varied from Reynolds number  $7200$  to  $50200$ . The results indicated that the presence of twisted wire brush inserts led to a large effect on the enhancement of heat transfer with corresponding increase in friction factor over the plain tube. It was found that the Nusselt number, friction factor, and thermal performance factor increased with the increase of twisted wire densities. The Nusselt number and friction factor of using the twisted wire brush inserts were found to be increased up to  $2.15$  and  $2.0$  times respectively than those over the plain tube values. Furthermore, the heat transfer performance was calculated to assess the real benefits of using those type of inserts and the performance was achieved  $1.85$  times higher compared to the plain tube based on the constant blower power.

**Bhuiya et al. (2012)** studied the influence of triple helical tapes inserted through a tube on heat transfer enhancement for turbulent flow. The triple helical tapes made of mild steel with different helix angles  $9^\circ$ ,  $13^\circ$ ,  $17^\circ$ , and  $21^\circ$  were used for experimentation. Flow rate is varied between the Reynolds number ranging from 22,000 to 51,000. The experiment revealed that the triple helical tape inserts caused an increase of heat transfer at the cost of increased blower power. The experiment showed that the Nusselt number, effectiveness and friction factor for the inserts were found to be up to 4.5, 3.45 and 3.0 times respectively over the plain tube. The highest enhancement efficiency achieved was 3.7 for the inserts based on constant blower power.

**Bhuiya et al. (2012)** investigated the heat transfer enhancement, thermal performance and friction factor characteristics of double helical tape inserted tube for turbulent fluid flow experimentally. The effects of insertion of the helical tape turbulators with different helix angles ( $9^\circ$ ,  $15^\circ$ ,  $21^\circ$  and  $28^\circ$ ) on heat transfer and pressure drop in the tube for Reynolds number ranging from 22000 to 51000 were examined. The Nusselt number, thermal enhancement efficiency and friction factor increased with decreasing helix angles. The Nusselt number obtained for the tube with double helical tape inserts was 305% higher in comparison to those of the plain tube values. Helical tape insert of helix angle  $9^\circ$  showed the highest heat flux among the inserts tested and was around 60% higher the value of the plain tube. The friction factor for the tube equipped with double helical tape inserts was 170% higher, whereas the blower power was 160% higher than those of the plain tube. The maximum thermal enhancement efficiency ( $\eta$ ) of 215% was obtained for the double helical tape inserted tube with helix angle  $9^\circ$ .

**Thianpong et al. (2012)** investigated heat transfer rate in heat exchanger tubes by using perforated twisted tapes (PTT) as inserts experimentally. This article reports an experimental investigation on heat transfer and pressure drop characteristics of turbulent flow in a heating tube equipped with perforated twisted tapes with parallel wings for Reynolds number between 5500 to 20500. There were two concepts behind the design of PTT wings, i.e., induce an extra turbulence near tube wall and thus efficiently disrupt a thermal boundary layer and holes existing along a core tube, diminish pressure loss within the tube. The parameters investigated were the hole diameter ratio ( $d/W = 0.11, 0.33$  and  $0.55$ ) and wing depth ratio ( $w/W = 0.11, 0.22$  and

0.33). A typical twisted tape was also tested for assessment. Compared to the plain tube, the tubes with PTT and TT yielded heat transfer enhancement up to 208% and 190%, respectively. The evaluation of overall performance under the same pumping power revealed that the PTT with  $d/W = 0.11$  and  $w/W = 0.33$ , gave the maximum thermal performance factor of 1.32, at Reynolds number of 5500.

**Sundar et al. (2012)** estimated the convective heat transfer and friction factor characteristics of magnetic  $\text{Fe}_3\text{O}_4$  nanofluid flowing through a uniformly heated horizontal circular tube with and without twisted tape inserts. Experiment was conducted in turbulent flow regime. Three twist ratios of  $H/D = 5, 10$  and  $15$  were taken. Flow ranged from Reynolds number 3000 to 22000. Heat transfer coefficients and friction factors at 0.02%, 0.1%, 0.3% and 0.6% volume concentration were calculated. Heat transfer and friction factor enhancement of 0.6% volume concentration of  $\text{Fe}_3\text{O}_4$  nanofluid in a plain tube with twisted tape insert of twist ratio  $H/D = 5$  is 51.88% and 1.231 times compared to water flowing in a plain tube under same Reynolds number.

**Thianpong et al. (2012)** investigated the heat transfer, friction factor and thermal performance characteristics in a tube equipped with twisted-rings are experimentally investigated. The experiments were conducted using twisted-rings with three different width ratios ( $W/D = 0.05, 0.1$  and  $0.15$ ) and three pitch ratios of ( $p/D = 1, 1.5$  and  $2$ ) for Reynolds numbers ranging from 6000 to 20,000 using air as a test fluid. The typical circular rings were also tested for an assessment. The experimental results reveal that most twisted-rings yield lower Nusselt numbers and friction factor than circular rings, except at the largest width ratio ( $W/D = 0.15$ ) and the smallest pitch ratio ( $p/D = 1.0$ ). Nusselt number and friction factor increase as width ratio increases and pitch ratio decreases. Although the twisted-rings at small width ratios ( $W/D = 0.05$  and  $0.1$ ) yield lower Nusselt numbers, they give higher thermal performance factors as the results of their dramatically lower friction factors. twisted-rings are feasible in terms of energy saving at the lowest Reynolds number and the smallest width ratio. A maximum thermal performance factor is associated by twisted-rings with the smallest width ratio ( $W/D = 0.05$ ) and pitch ratio ( $p/D = 1.0$ ) at a Reynolds number of 6000.

**Bas et al. (2012)** studied heat transfer and flow friction in a twisted tape swirl generator inserted tube. The twisted tapes are inserted separately from the tube wall. The effects of twist ratios ( $y/D = 2, 2.5, 3, 3.5$  and  $4$ ) and clearance ratios ( $c/D = 0.0178$  and  $0.0357$ ) are experimented in the range of Reynolds number from 5132 to 24989 and the typical one ( $c/D = 0$ ) is also tested for comparison. Uniform heat flux is applied to the external surface of the tube wall. The air is selected as a working fluid. The using of twisted tapes supplies considerable increase on heat transfer and pressure drop when compared with those from the plain tube. The Nusselt number increases with the decrease of clearance ratio ( $c/D$ ) and twist ratio ( $y/D$ ), also increase of Reynolds number. For all investigated cases, heat transfer enhancement ( $f$ ) tends to decrease with the increase of Reynolds number and to be nearly uniform for Reynolds number over 15000 and  $y/D$  lower than 3.0. The highest heat transfer enhancement is achieved as 1.756 for  $c/D = 0.0178$  and  $y/D = 2$  at Reynolds number of 5183.

**Wongcharee et al. (2012)** investigated the heat transfer enhancement by using CuO/water nanofluid in corrugated tube equipped with twisted tape. The investigated ranges are (1) three different CuO concentrations: 0.3, 0.5 and 0.7% by volume (2) three different twist ratios of twisted tape:  $y/w=2.7, 3.6$  and  $5.3$  (3) two different arrangements of twisted direction of twisted tape relative to spiral direction of corrugated tube: parallel and counter arrangements, and (4) Reynolds number from 6200 to 24000. The results achieved from the use of the nanofluid and twisted tape, are compared with those obtained from the uses of nanofluid alone and twisted tape alone. The experimental results reveal that at similar operating conditions, heat transfer rate, friction factor as well as thermal performance factor associated with the simultaneous application of CuO/water nanofluid and twisted tape are higher than those associated with the individual techniques. Evidently, heat transfer rate increases with increasing CuO/water nanofluid concentration and decreasing twist ratio. In addition, the twisted tape coupled with corrugated tube in counter pattern offer higher heat transfer performances than the ones in parallel pattern. Over the range studied, the maximum thermal performance factor 1.57 is found with the use of CuO/water nanofluid at concentration of 0.7% by volume in corrugated tube together with twisted tape at twist ratio ( $y/w$ ) of 2.7 (in counter arrangement), for Reynolds number of 6200 where heat transfer rate and friction factor increase to 2.67 times and 5.76 times of those in the plain corrugated tube.

**Darzi et al. (2013)** investigated the turbulent heat transfer in heated helically corrugated tube for pure water and water–alumina nanofluid using two phase approach. The study was carried out for different corrugating pitch and height ratios at various Reynolds numbers ranging from 10000 to 40000. The effect of nano-particles in heat transfer augmentation for smooth tube and helically corrugation tubes (HCT) was discussed and their relative Nusselt number was compared. . The overall enhancement of heat transfer using nano-particles and helical corrugation roughness can promote this factor up to 3.31. Increasing the Reynolds number enhances the heat transfer but it causes the reduction of the relative Nusselt number. Results show that the heat transfer enhancement is promoted extremely by increasing the volume fraction of nano-particles. Adding 2% and 4% nano-particles by volume to water enhances the heat transfer by 21% and 58% respectively. Also, the overall enhancement in heat transfer using two mechanisms simultaneously compared to using pure fluid within smooth tube exceeds over 330%. Results indicate that using nano-particles yields different enhancement in heat transfer of tube for different corrugation height and pitch.

**Naik et al. (2013)** investigated Convective heat transfer and friction factor characteristics of water/propylene glycol (70:30% by volume) based CuO nanofluids flowing in a plain tube under constant heat flux boundary condition. Glycols are normally used as an anti-freezing heat transfer fluids in cold climatic regions. Nanofluids are prepared by dispersing 50 nm diameter of CuO nanoparticles in the base fluid. Experiments are conducted using CuO nanofluids with 0.025%, 0.1% and 0.5% volume concentration in the Reynolds numbers ranging from 1000 to 10000 and considerable heat transfer enhancement in CuO nanofluids is observed. The effect of twisted tape inserts with twist ratios (H/D) in the range of 0 to 15 on nanofluids is studied. Considerable enhancement in the Nusselt number is observed with CuO nanofluids over the base fluids and heat transfer enhancement is linearly proportional to the nanoparticle volume concentration in the base fluid. The increment in the pressure drop in the CuO nanofluids over the base fluid is negligible but the experimental results have shown a significant increment in the convective heat transfer coefficient of CuO nanofluids. The use of twisted tape inserts in CuO nanofluids, enhances the heat transfer coefficient with little increment of friction factor and transfer enhancement is proportional to the number of twists on inserts. The

convective heat transfer coefficient increased up to 27.95% in the 0.5% CuO nanofluid in plain tube and with a twisted tape insert of  $H/D = 5$  it is further increased to 76.06% over the base fluid at a particular Reynolds number. The friction factor enhancement of 10.08% is noticed and increased to 26.57% with the same twisted tape, when compared with the base fluid friction factor at the same Reynolds number.

**Eiamsa-ard et al. (2013)** investigated the combined effects of nanofluids, dual twisted-tapes (DTs) and a micro-fin tube (MF) on the heat transfer rate, friction factor and thermal performance factor characteristics. Nanofluids consisting of CuO and water at CuO concentrations between 0.3% and 1.0% by volume, were utilized as working fluids in the MF equipped with DTs, for Reynolds number between 5650 and 17000. The experiments using the MF alone as well as the MF equipped with a single twisted tape (ST), were also conducted for comparison. The experimental results revealed that the heat transfer rate increased with increasing nanofluid concentration. At similar operating conditions, the micro-fin tube equipped with dual twisted-tapes (MF-DTs) consistently gave superior thermal performance factor to the one equipped with a single twisted-tape (MF-ST) as well as the micro-fin tube alone (MF). For all cases, thermal performance factors were apparently above unity. This indicates the beneficial effect for the energy saving by the uses of the combined techniques, especially at low Reynolds number.

**Mohammed et al. (2013)** studied the effect of using louvered strip inserts placed in a circular double pipe heat exchanger on the thermal and flow fields utilizing various types of nanofluids. The continuity, momentum and energy equations were solved by means of a finite volume method (FVM). The top and the bottom walls of the pipe were heated with a uniform heat flux boundary condition. Two different louvered strip insert arrangements (forward and backward) were used in this study with a Reynolds number range of 10,000 to 50,000. The effects of various louvered strip slant angles and pitches were also investigated. Four different types of nanoparticles  $Al_2O_3$ , CuO,  $SiO_2$ , and ZnO with different volume fractions in the range of 1% to 4% and different nanoparticle diameters in the range of 20 nm to 50 nm, dispersed in a base fluid (water) are used. The numerical results indicate that the forward louvered strip arrangement can promote the heat transfer by approximately 367% to 411% at the highest slant angle of  $\alpha=30^\circ$  and lowest pitch of  $S=30$  mm. The maximal skin friction coefficient of the enhanced tube is around 10 times than that of the smooth

tube and the value of performance evaluation criterion (PEC) lies in the range of 1.28–1.56. It is found that SiO<sub>2</sub> nanofluid has the highest Nusselt number value, followed by Al<sub>2</sub>O<sub>3</sub>, ZnO, and CuO while pure water has the lowest Nusselt number. The results show that the Nusselt number increases with decreasing the nanoparticle diameter and it increases slightly with increasing the volume fraction of nanoparticles.

**Nanan et al. (2014)** studied the Influence of perforated helical twisted-tapes (P-HTTs) on the heat transfer, friction loss and thermal performance characteristics under a uniform heat flux condition. The P-HTTs were obtained by perforating typical helical twisted-tapes (HTTs) with a prospect to reduce the friction loss of fluid flow. The experiments were conducted using P-HTTs' three different diameter ratios ( $d/w$ ) of 0.2, 0.4 and 0.6 and three different perforation pitch ratios ( $s/w$ ) of 1, 1.5 and 2. The helical pitch ratio and twist ratio were fixed at  $P/D = 2$  and  $y/w = 3$ . Tests were performed for Reynolds number between 6000 and 20000. The experiments using the plain tube and the tubes with HTTs were also carried out for assessment. The experimental results reveal that the use of P-HTTs leads to the reduction of friction loss as compare to that of HTT due to a diminishing fluid flow blockage and turbulence intensity. Heat transfer, friction loss and thermal performance factor increase as  $d/w$  decreases and  $s/w$  increases. For the present range, the maximum thermal performance factor of 1.28 is obtained by using the P-HTT with  $d/w = 0.2$  and  $s/w = 2.0$  at the Reynolds number of 6000.

**Azmi et al. (2014)** studied the enhancement in heat transfer coefficients in combination with the addition of tape inserts. Experiments are undertaken to determine heat transfer coefficients and friction factor of TiO<sub>2</sub>/water nanofluid up to 3.0% volume concentration at an average temperature of 30 °C. The investigations are undertaken in the Reynolds number range of 8000 to 30000 for flow in tubes and with tapes of different twist ratios. A significant enhancement of 23.2% in the heat transfer coefficients is observed at 1.0% concentration for flow in a tube. With the use of twisted tapes, the heat transfer coefficient increased with decrease in twist ratio for water and nanofluid. The heat transfer coefficient and friction factor are respectively 81.1% and 1.5 times greater at  $Re = 23558$  with 1.0% concentration and twist ratio of 5, compared to values with flow of water in a tube. An increase in the nanofluid concentration to 3.0% decreased heat transfer coefficients to values lower than water for flow in a tube and with tape inserts. A thermal system with tape insert of twist

ratio 15 and 1.0% TiO<sub>2</sub> concentration gives maximum advantage ratio, if pressure drop is considered along with enhancement in heat transfer coefficient.

**Blanco et al. (2014)** studied the thermal properties of CuO dispersed in water and ethylene glycol as a function of the particle volume fraction. Experiments were performed between temperatures ranging between 298 to 338 K. A micro calorimetric technique has been used. It is particularly suitable for studying nanofluids because the measurements are made with very small temperature gradients and with practical absence of natural convection. Thermal conductivities have been studied by the steady-state coaxial cylinders method, using a C80D micro calorimeter equipped with special calorimetric vessels. Heat capacities have been measured with a Micro DSC II micro calorimeter with batch cells designed in their laboratory and the “scanning or continuous method.” Results for thermal conductivities can be well justified using a classical model (Hamilton–Crosser), and experimental measurements of heat capacities can be justified with a model of particles in thermal equilibrium with the base fluid. They concluded that the observed growth of the thermal conductivity of their nanofluids with increasing temperature was mainly due to the base fluids water and EG rather than to the nanoparticles. this thermal study had been completed with very precise specific heat capacity measurements of the same nanofluids.

**Naik et al. (2014)** studied heat transfer and friction factor analysis of CuO/water nanofluid flowing through a tube under turbulent flow conditions and with twisted tape (TT) and wire coil (WC) inserts. The experimental investigations were performed in the Reynolds number range from 4000 to 20000, volume concentrations of 0.1% and 0.3%, twisted tape inserts of  $h/d = 5$  and 10 and wire coil inserts of  $p/d = 1.97$  and 2.95. The experimental results indicated that under same operating conditions and flow rates, heat transfer coefficient, friction factor and thermal performance factor associated with nanofluid in a tube with wire coil inserts are higher than those with the twisted tape inserts. The Nusselt number enhancement for 0.3% nanofluid in a tube without inserts is 17.62%, 0.3% nanofluid in a tube with TT-2 is 31.88% and 0.3% nanofluid in a tube with WC-2 is 44.45% at a Reynolds number of 20000 compared to water. Whereas, the friction factor enhancement for 0.3% nanofluid in a tube without inserts is 1.149-times 0.3% nanofluid in a tube with TT-2 is 1.179-times and 0.3% nanofluid in a tube with WC-2 is 1.198-times at a Reynolds number of 20000 compared to water. The thermal performance factor of 0.3% nanofluid in tube

with twisted tape and wire coil inserts are 1.24 and 1.36 compared against water data respectively.

**Azmi et al. (2014)** investigated the heat transfer coefficient and friction factor of TiO<sub>2</sub> and SiO<sub>2</sub> water based nanofluids flowing in a circular tube under turbulent flow under constant heat flux boundary condition. TiO<sub>2</sub> and SiO<sub>2</sub> nanofluids with an average particle size of 50 nm and 22 nm respectively are used in the working fluid for volume concentrations up to 3.0%. Experiments are conducted at a bulk temperature of 30 °C in the turbulent Reynolds number range of 5000 to 25000. The enhancements in viscosity and thermal conductivity of TiO<sub>2</sub> are greater than SiO<sub>2</sub> nanofluid. However, a maximum enhancement of 26% in heat transfer coefficients is obtained with TiO<sub>2</sub> nanofluid at 1.0% concentration, while SiO<sub>2</sub> nanofluid gave 33% enhancement at 3.0% concentration. The heat transfer coefficients are lower at all other concentrations. It is observed that the pressure drop is directly proportional to the concentration and density of the nanoparticle.

**Bhuiya et al. (2014)** studied the effects of the double counter twisted tapes on heat transfer and fluid friction characteristics in a heat exchanger tube. The double counter twisted tapes were used as counter-swirl flow generators in the test section. The experiments were performed with double counter twisted tapes of four different twist ratios ( $y = 1.95, 3.85, 5.92$  and  $7.75$ ) using air as the testing fluid in a circular tube turbulent flow regime where the Reynolds number was varied from 6950 to 50050. The experimental results demonstrated that the Nusselt number, friction factor and thermal enhancement efficiency were increased with decreasing twist ratio. The results also revealed that the heat transfer rate in the tube fitted with double counter twisted tape was significantly increased with corresponding increase in pressure drop. The heat transfer rate and friction factor were obtained to be around 60 to 240% and 91 to 286% higher than those of the plain tube values respectively. The thermal enhancement efficiency for all the cases was more than one, which indicated that the effect of heat transfer enhancement due to the enhancing tool was more dominant than the effect of the rising friction factor and vice versa. The maximum thermal enhancement efficiency of 1.34 was achieved by the use of double counter twisted tapes at constant blower power.

**Azmi et al. (2014)** developed numerical model for turbulent flow of SiO<sub>2</sub>/water nanofluids in a tube with twisted tape inserts. The model is based on the assumption that van Driest eddy diffusivity equation can be applied by considering the coefficient and the Prandtl index in momentum and heat respectively as a variable. The results from the numerical analysis are compared with experiments undertaken with SiO<sub>2</sub>/water nanofluid for a wide range of Reynolds number. Reynolds number varied from 6000 to 10000. Generalized equation for the estimation of nanofluid friction factor and Nusselt number is proposed with the experimental data for twisted tapes. The coefficient and the Prandtl index in the eddy diffusivity equation of momentum and heat is obtained from the numerical values as a function of Reynolds number, concentration and twist ratio. The experimental results indicate 29.6% enhancement in heat transfer coefficient and 23.7% greater friction factor at 3.0% concentration when compared to water at a twist ratio of 5.0. An enhancement of 94.1% in heat transfer coefficient and 160% higher friction factor at  $Re = 19046$  is observed at a twist ratio of five with 3.0% volumetric concentration when compared to flow of water in a tube. The heat transfer coefficients decrease when undertaken for concentrations greater than 3.0% at the nanofluid bulk temperature of 30<sup>0</sup>C.

**Esmailzadeh et al. (2014)** studied heat transfer and friction factor characteristics of g- Al<sub>2</sub>O<sub>3</sub>/water nanofluid through circular tube with twisted tape inserts with various thicknesses at constant heat flux. In this work, g- Al<sub>2</sub>O<sub>3</sub>/water nanofluids with two volume concentrations of 0.5% and 1% were used as the working fluid. The twist ratio of twisted tape remained constant at 3.21, while the thicknesses were changed through three values of 0.5 mm, 1 mm and 2 mm. The experiments were performed in laminar flow regime from 150 to 1600 Reynolds numbers. Results indicated that twisted tape inserts enhanced the average convective heat transfer coefficient and also more the thickness of twisted tape is more the enhancement of convective heat transfer coefficient was. Also, the highest enhancement was achieved at maximum volume concentration. Results showed that nanofluids have better heat transfer performance when utilized with thicker twisted tapes. At the same time, the increase in twisted tape thickness leads to an increase in friction factor. The use of twisted tapes increase friction factor due to larger contact surface and reduction of fluid free flow area which causes high speed swirl flow. More the thickness of twisted tape is more the increase of friction factor is. Ultimately, the convective heat transfer

enhancement outweighs the effect of friction factor increase leading enhanced thermal performance.

**Azmi et al. (2014)** determines the heat transfer coefficients of SiO<sub>2</sub>/water and TiO<sub>2</sub>/water nanofluid up to 3% volume concentration flowing in a circular tube. The experiments were conducted in the Reynolds number ranging from 5000 to 25000 at a bulk temperature of 30°C. In this experiment a circular tube is taken with twisted tapes of different twist ratios which was varied between 5 to 93. It was seen that the heat transfer enhancement of nanofluids with twisted tape is higher than the water flow for the same twist ratio. The experimental result showed that with twisted tape inserted, SiO<sub>2</sub> nanofluid had more heat transfer coefficient values compared to the values of TiO<sub>2</sub> nanofluid at 3% concentration. The heat transfer coefficients of SiO<sub>2</sub>/water nanofluid is 27.9% higher than water at 3% concentration. The heat transfer coefficient values of TiO<sub>2</sub>/water nanofluid calculated at the same concentration is 11.4% greater than water for twist ratio as 5. The heat transfer coefficient inversely increased with twist ratio. The maximum heat transfer enhancement with twisted tape for TiO<sub>2</sub>/water and SiO<sub>2</sub>/water nanofluids was found at 1.0% and 3.0% volume concentration respectively.

**Maddah et al. (2014)** studied fluid flow of the Al<sub>2</sub>O<sub>3</sub> nanofluid in a horizontal double pipe heat exchanger fitted with modified twisted tapes under turbulent flow conditions. Range of Reynolds number was taken between 5000 to 21000 with a wide range of solid volume fraction ranging from 0.2% to 0.9%. The experiments with different geometrical progression ratio (GPR) of twists as the new modified twisted tapes and different nanofluid concentration were performed under similar operation condition. Pitch length of the proposed twisted tapes and consequently the twist ratios changed along the twists with respect to the geometrical progression ratio (GPR) whether reducer (RGPR < 1) or increaser (IGPR > 1). Regarding the experimental data, utilization of RGPR twists together with nanofluids tends to increase heat transfer and friction factor by 12% to 52% and 5% to 28% as compared with the tube with typical twisted tapes (GPR = 1) and nanofluid. Contrarily, performances were weakened by using for IGPR twists 0.6 to 0.92 and 0.75 to 0.95.

**Darzi et al. (2014)** studied the effect of nanoparticles and helical corrugation on turbulent heat transfer and dimensionless friction factor. Thermal conductivity and

viscosity of nanofluids with various volume fractions were experimentally measured at different temperatures. Reynolds number and nanoparticle concentration varied from 5000 to 20000 and 0% to 1% by volume, respectively. Nusselt number and friction factor of nanofluids were obtained for different concentrations as well as various Reynolds and Prandtl numbers. Results show that the heat transfer and friction factor increase by increasing nanofluid concentrations in plain and helical corrugated tube while its effect are more significant in helical corrugated tubes. The effect of nanoparticles on heat transfer is intensified at higher corrugation height and smaller corrugation pitch. Heat transfer enhancement at same pumping power, heat transfer performance, have been improved by increasing the nanoparticles concentration not as well as by increasing the corrugation height or reducing the corrugation pitch. By using nano-particles and helical corrugation simultaneously enhanced the heat transfers by factor of 3.2 for higher concentration of nanoparticles, higher height and smaller pitch of corrugation.

**Eiamsa-ard et al. (2015)** Titanium dioxide ( $\text{TiO}_2$ ) in water as nanofluid was employed for heat transfer enhancement together with overlapped dual twisted tapes (O-DTs). The study encompassed Reynolds numbers from 5400 to 15200, O-DTs with overlapped twist ratios ( $y_0/y$ ) of 1.5, 2.0 and 2.5 and nanofluids with  $\text{TiO}_2$  volume concentrations ( $\phi$ ) of 0.07%, 0.14% and 0.21%. The experimental and numerical results indicated that ODTs with smaller overlapped twisted ratio delivered a stronger swirl intensity and higher turbulent kinetic energy (TKE). The use of O-DTs at the smallest overlapped twist ratio of 1.5 enhanced heat transfer rates up to 89%, friction factor by 5.43 times and thermal performance up to 1.13 times as compared to those of plain tube. In addition, heat transfer increased as  $\text{TiO}_2$  volume concentration of nanofluid increased, owing to the increases of contact surface and thermal conductivity. Nusselt number, friction factor and thermal performance increased with decreasing overlapped twist ratio and increasing  $\text{TiO}_2$  volume concentration. The simultaneous use of the O-DTs having twist ratios 1.5 with the nanofluid with  $\text{TiO}_2$  volume concentration of 0.21% resulted in heat transfer enhancement around 9.9 to 11.2% and thermal performance improvement up to 4.5% as compared to the use of O-DTs alone.

### GAP STUDY AND OBJECTIVES

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#### 3.1 Gap study

After going through the literature review and previous theoretical and experimental work related to the field of using nanofluids and heat transfer enhancement devices in heat exchangers, it was found that the no solid conclusion has been made using experimental analysis and the is lot of variability in the results proposed by various researchers. Basically, following conclusions could be drawn from the literature review

- The use of nanofluids in heat exchangers has been proposed by many researchers but most of the work is based on theoretical observations or numerical simulation and very less literature regarding experimental calculations is available.
- Since, nanofluids is a newer field and the validation of results at this stage is mainly through experimental procedure and different researchers have proposed different effects caused by using nanofluids which indicates that further research is required to reach out on results which are within permissible tolerances and accepted by all researchers in the field.
- The results obtained from experimental procedures using nanofluids depend highly on the state of nanofluid i.e. whether the nano fluid concentration is uniform or not, nanofluid has undergone proper sonification or not and many other factors. Better defined are the properties of nanofluid, more accurate and reliable will be the results obtained. Hence this factor also demands for further research in this field.
- Coming towards heat transfer enhancement devices, here also the case is more or less same. Although research work is available but there is wide variation between research work of different researchers. The validation is only possible

once huge numbers of sample results are available to decide the actual performance enhancement and possible errors.

- Different types of heat enhancement devices have been proposed and utilized by various researchers but the comparative performance has not been studied. There are two sides of coin in using heat transfer enhancement devices. These devices increase the rate of heat transfer but at the same time, the pressure drop increases which adds to the cost of pumping. The best device will be the one where the heat transfer enhancement obtained is maximum and the pressure drop is minimum. Thus, this further enhances the need of more research in field of heat transfer enhancement devices.
- In case of systems, where temperatures encountered are low or where space availability is a problem, we want to utilize the same heat exchanger with maximum efficiency. For this purpose, combination of two methods i.e. nanofluid and heat transfer enhancement devices can be utilized. This is also a field where research needs to be carried out to understand the relative merits and demerits of such a hybrid system.

Mentioned above were the gaps from which the need for further research in this field has to be identified. Hence, this study has to be carried out to study the effects of nanofluids and various heat transfer enhancement devices and comparative analysis of various heat transfer devices has to be done.

### **3.2 Objectives**

- To evaluate the performance parameters of double tube counter flow heat exchanger using CuO-water based nanofluids at different nanoparticle concentrations (0.1%, 0.2% and 0.3% vol. conc.)
- To compare the performance parameters of double tube counter flow heat exchanger using two different turbulence promoters (twisted tape of twist ratio 5 and 10 and spring having pitch 5 and 10 mm inserts) at different concentrations and flow rates of nanofluids used as hot fluid.

**EXPERIMENTAL SET UP AND METHODOLOGY**

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Experimental studies are conducted to determine the effectiveness of various heat transfer enhancement devices and methods used in double pipe heat exchanger. Comparisons are made between different methods on the basis of increase in heat transfer and friction factor or pressure drop across the hot fluid side. To be specific, following heat transfer enhancement devices or methods are used for the experimental investigation:

1. CuO nanofluid
2. Twisted tape inserts with different pitch
3. Spring insert with different pitch
4. Combination of CuO nanofluid and twisted tape inserts
5. Combination of CuO nanofluid and spring insert

The experimental studies are carried out on a setup which was fabricated in the heat transfer laboratory and the details of the setup are given below.

**4.1 Experimental setup**

The details of various components and the heat transfer enhancement devices used for the experimental work in this study are as following:

1. Support frame: It carries all the components and the double pipe heat exchanger. Iron channels and sheets were used to fabricate the support frame. All the measuring instruments such as rotameters, manometer, electrical components and pumps are mounted on the support frame.
2. Double pipe heat exchanger: A double pipe heat exchanger has been used in this study to investigate the effect of heat transfer methods experimentally. The material of inner pipe of the heat exchanger has been chosen as copper due to very high thermal conductivity, resistance towards fouling and also because it is the most

widely used material in the heat exchanger industry. The dimensions of inner copper pipe are

Inner Diameter of copper pipe: 12 mm

Thickness of copper pipe: 2 mm



**Figure 4.1 Inner copper pipe used in exchanger**

The material of the outer pipe of the double pipe heat exchanger has been chosen to be galvanized iron due to its corrosion resistance. The dimensions of the galvanized iron outer pipe are:

Inner diameter of outer pipe: 25.4 mm

Thickness of outer pipe: 3 mm



**Figure 4.2 Galvanized iron outer pipe of heat exchanger**

The length of the double pipe heat exchanger is 1500 mm. Connectors are provided at both ends of inner and outer pipes for easy dismantling so that inserts can be easily introduced into the exchanger. The heat exchanger is insulated from surroundings using an asbestos rope so that there are no heat losses which can cause an error in the experimental readings. Also, the radiation heat losses are minimised by providing the outer surface of galvanized iron pipe with aluminium tape. The fabricated heat exchanger is shown as under:



**Figure 4.3 Fabricated heat exchanger**

3. Hot fluid supply: The hot fluid is supplied from the hot water tank where a 3 KW electric heater is installed to heat up the fluid. A PID controller is used to maintain the constant temperature of the hot fluid to be supplied at the inner pipe of the double pipe heat exchanger. Hot fluid tank is also insulated from the surroundings by using the asbestos rope and aluminium tape to minimise convections and radiation losses respectively. Hot fluid is circulated into the tank after passing through the heat exchanger and is re circulated again and again. A centrifugal pump is used to circulate the hot fluid through the exchanger.

4. Heating element: A coil type electric heater is used as the heating element in this experiment for heating up the hot fluid that is to be passed through the annulus section of the double pipe heat exchanger. The capacity of the heater is calculated as:

Amount of Fluid (assuming water, density=  $100 \text{ kg/m}^3$ ) to be heated = 4Kg

Specific heat = 4.18 KJ/Kg K

Temperature difference =  $80 - 25 = 55$

Amount of heat required = 920Kj

Assuming 5 minutes of heating time,

Capacity of heater required =  $920/5 \times 60 = 3.06 \text{ KW}$

Hence a standard coil type heater of capacity 3 KW readily available in the market is used for heating the fluid. The heater is installed in the hot water tank by drilling and is properly sealed to avoid any sort of leaking. Proper earthing is provided to eliminate any chances of a shock. Power supply at 220 V is provided and heater is also connected to PID controller so as to maintain the constant temperature of the hot fluid.



**Figure 4.4 Heating element**

5. Cold fluid supply: The cold fluid i.e. water is supplied at the ambient temperature from the supply line. Centrifugal pump is used to circulate the cold fluid. The cold fluid is not circulated as there is no provision of a chiller and once the cold fluid passes through the exchanger, it is diverted towards the drain side.

6. PID temperature controller: In order to maintain the constant temperature of the hot fluid or the nanofluid, PID temperature controller has been installed along with the equipment. Since, the hot fluid to the heat exchanger is to be supplied at the constant temperature of 80 degree Celsius; the controller is connected to the heating unit. PID controller combines proportional control with two additional adjustment sand that helps the unit automatically compensate for changes in the system. These adjustments, integral and derivative, are expressed in time-based units. The heater gets cut off once the temperature of the hot fluid reaches the value of 80 degree celcius. Once the temperature of hot fluid falls down due to heat transfer in the double pipe heat exchanger, once again the PID controller sends the signal to switch onn the heater. Thus this process is continued and the hot fluid is maintained at constant temperature to maintain the uniformity in test results.



**Figure 4.5 PID controller**

7. Pumps: two numbers of centrifugal pumps have been used in the study to pump the hot and cold fluids through the inner pipe and annulus section of the heat exchanger respectively.

The specifications of the centrifugal pumps are as under:

Maximum Discharge: 12 m

Maximum head: 1100 lph

Both the pumps are providing with by pass valves to control the mass flow rate of the fluid passing through the exchanger. Screens were used to avoid the entrance of fouling particles into the pump and heat exchanger.



**Figure 4.6 Centrifugal pumps used for circulating fluids**

8. Rotameters: 2 No's rotameters are used to measure the mass flow rate of hot and cold fluid passing through the heat exchanger. The specifications of rotameters are as under which were decided on the basis of the desired mass flow rates:

Flow range of hot fluid rotameter: 1-5 lpm

Flow range of cold fluid rotameter: 5 lpm

Control valves are also available on both the rotameters so that mass flow rate of the fluid flowing through the exchanger can be varied thereby varying the Reynolds number.



**Figure 4.7 Rotameters used in study**

9. Resistance temperature detectors: 4 No's PT100 resistance temperature detectors have been used to measure the temperatures. One RTD each is installed on the inlet and outlet of both hot and cold fluid carrying pipes. RTD have been chosen to measure the temperature as they are highly accurate in the temperature range of -50 to 300 degree celcius. RTD have been preferred over thermocouples as thermocouples give poor repeatability that can cause the occurrence of substantial error in the experimental calculations. RTD were fixed using the thermal paste in the connectors fabricated for installing the temperature detectors.

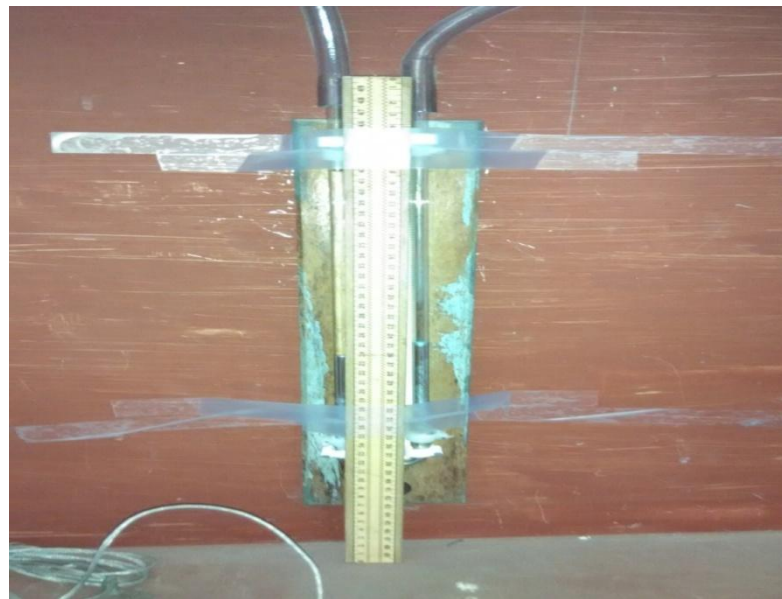
10. Temperature display/ Selector switch: A display is used to indicate the temperature of any of the RTD. The temperature which is to be displayed on the display is controlled by a 4 way selector switch. RTD were numbered and the connections were accordingly made with the selector switch so that there is no confusion regarding any particular temperature reading.

RTD were calibrated before installing so that necessary temperature corrections can be applied to obtain the accurate results.



**Figure 4.8 Temperature display and selector switch**

11. Pressure drop measurement: To measure the pressure drop across the hot fluid side i.e. copper inner pipe of the heat exchanger, a U tube manometer with mercury as the manometric fluid has been provided in the experimental setup. A scale is installed to indicate the pressure drop across the hot fluid side of the heat exchanger.



**Figure 4.9 U-tube manometer with mercury as manometric fluid for pressure drop measurement.**

12. Twisted tape inserts: Twisted tape inserts that are to be introduced in the exchanger were manufactured at the machine shop. A primary condition for any form of insert is that it should not touch the heat exchanger body. Since the inner diameter of the copper pipe is 12 mm, hence to provide adequate spacing between the twisted

tape insert and pipe, the width of twisted tape has been taken as 10 mm. The thickness of the tape is 1mm. The material of tape is steel as it can be easily twisted and also is corrosion resistance as a corrosion catching component in longer runs for a heat exchanger can cause substantial amount of fouling by providing the active sites for accumulation of fouling components. The plain strip was taken and one of its ends was fixed in the tool post of the lathe machine. The other end was given a slow rotation in order to obtain the required pitch of the twisted tape. The strip was kept under constant tension so that there is not any distortion of the metal strip. Following pitches of twisted tape were fabricated:

Pitch 1: 5 cm

Pitch 2: 10 cm



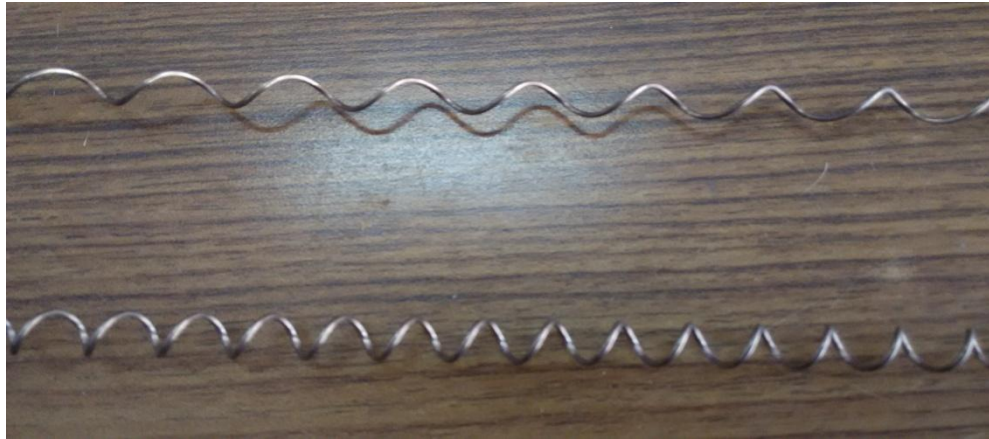
**Figure 4.10 Twisted tape inserts**

13. Spring inserts: Spring inserts were readily available in the market. The specifications of spring type inserts used in this experiment are as under:

Diameter of spring insert: 10 cm

Pitch 1: 5 mm

Pitch 2: 10 mm



**Figure 4.11 spring inserts**

14. Nanofluid:

#### Preparation of Nanofluids

For the preparation of nanofluid, two step method is adopted. Following methodology is adopted for the nanofluid preparation:

- First of all, required amount of copper oxide nanoparticle is calculated using following depicted formula, for the preparation of nanofluid of desired volumetric concentration. Table shows the calculated amount of nanoparticles for the preparation of nanofluid of desired volumetric concentration.

$$f_v = \frac{V_{np}}{V_{np} + V_{bf}} = \frac{W_{np} / \rho_{np}}{(W_{np} / \rho_{np}) + V_{nf}}$$

**Table 4.1 calculated value of required weight of nanoparticle for desired volumetric calculation**

Volumetric concentration	Amount of nanoparticles per litre (gm)
$\phi = 0.1$	6.4
$\phi = 0.2$	12.8
$\phi = 0.3$	19.2

- Then weight of copper oxide nanoparticles is made to measure in measuring device, which conforms to calculated weight of the nanoparticles
- Then measured amount of copper oxide nanoparticles is then made to disperse in the base fluid
- In order to have a proper dispersing of the nanoparticles, sample prepared in above step is stirred in magnetic stirrer, so that nanoparticles is properly mixed with the base fluid
- Then prepared sample is then placed in the ultrasonicator, where ultrasonic rays are made to traverse through the sample, which in turns breaks the nanoparticles. Due to which nanoparticles is made to suspend properly in the base fluid.



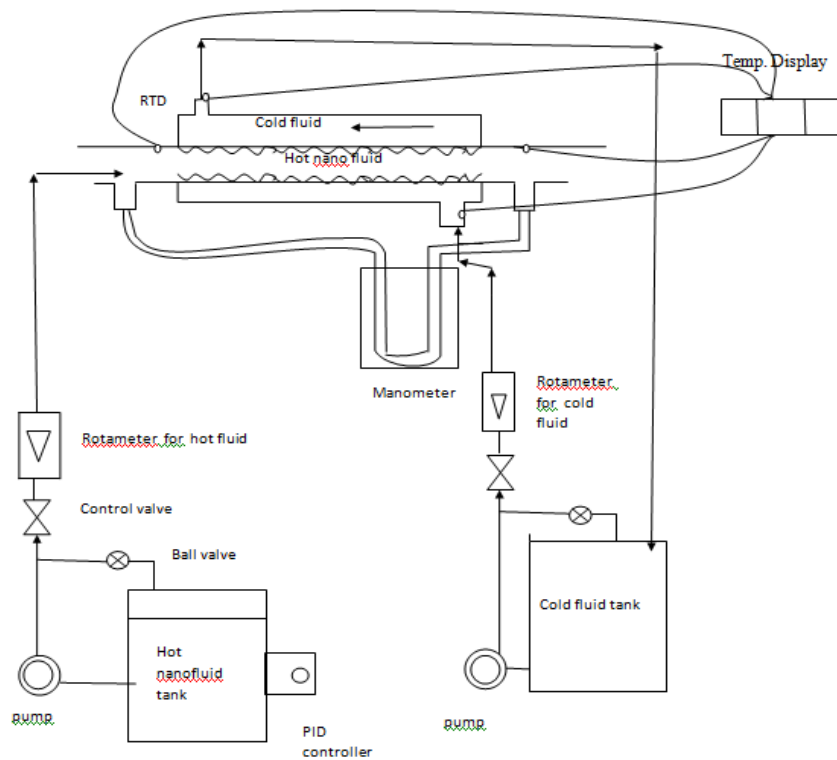
**Figure 4.12 Pump used in hot tank and weighing machine for measuring the weight of nanoparticles.**



**Figure 4.13 Ultrasonicator and magnetic stirrer**

Surfactants has been added to stabilise the nano fluid so that the particles of the fluid does not settle down that can cause a non homogeneous mixture which will lead to wrong results. A pump has also been used in the hot fluid tank to keep the nano fluid in circulation which further helps in maintaining its stability and homogeneity.

The layout and the actual experimental setup are shown as under



**Figure 4.14 Experimental setup layouts**



**Figure 4.14 Experimental setup**

#### **4.2 Parameters varied in experimental investigation**

Following parameters were varied to perform the experimental investigation so that relative effectiveness of various enhancement methods can be compared:

1. Reynolds number of hot fluid: the Reynolds number of hot fluid was varied between 5000 and 25000 by varying the mass flow rate of the fluid passing through the heat exchanger controlled by control valves provided on the centrifugal pump and the rotameter itself.
2. The pitch of twisted tape was varied. Tape1 has twist ratio 5 and Tape 2 has twist ratio 10.
3. Pitch of spring insert was varied. Spring1 has 5 mm and spring2 has 10 mm.
4. The concentration of CuO nanofluid was varied. Three compositions i.e. 0.1 %, 0.2 % and 0.3 % vol. Conc. were chosen for the study. Empirical relations were used to find out the properties such as density and viscosity of nanofluid at different concentrations.

### **4.3 Calibration**

Before installing the measuring instruments and starting the actual experiment, the instruments were calibrated to phase out the possibility of any unknown error in the observations as under:

1. Rotameter calibration: Rotameter was calibrated by using the traditional technique of noting the time taken to fill a certain amount of container. A 1 litre beaker was used for calibration. Three readings were taken for each mass flow rate and the average mass flow rate was used for the purpose of calibration.
2. RTD calibration: RTD were calibrated by using a constant water bath. After dipping all the four RTD in the water bath, readings were taken from the display using the selector switch. One of the RTD was taken as reference and the corrections were applied for other three RTD.

### **4.4 Experimental procedure**

1. First of all the rotameter and RTD were calibrated
2. Standardization of the setup: The setup was first standardized using the smooth tube heat exchanger without introducing any insert and using plain hot water as the hot fluid. The values of the heat transfer coefficient and friction factor for such arrangement were compared with the results available from the standard equations in order to ensure that all the components of the setup are fine tuned and the corresponding results are in match with the literature findings.
3. Once the standardization was complete, the experiments were conducted for modified methods by following the steps mentioned ahead
4. Hot fluid was pumped by the centrifugal pump from the hot fluid tank to the heat exchanger through rotameter. The desired mass flow rate was obtained using the control valves available on centrifugal pump and the rotameter. After passing through the heat exchanger, hot fluid comes back into the tank where it is again heated to the desired temperature and re circulated.
5. Cold fluid was pumped by centrifugal pump through rotameter into the annular section of the double pipe heat exchanger. The mass flow rate of the cold fluids is

kept constant at 5 lpm throughout the whole experiment. The outlet of the cold fluid is connected to the drain.

6. The system is made to run as long as the steady state conditions are not reached. Usually, it took 45-50 minutes for steady state conditions to prevail.

7. Once, the steady state condition is achieved; all the four temperatures are recorded by using the selector switch.

8. The pressure drop across the hot fluid side is recorded from the reading of the u tube manometer. It is always ensured that there are no air bubbles in the manometric line

9. After completion of one cycle, Parameters such as mass flow rate are varied and whole procedure is repeated until steady state conditions are again reached.

10. A similar approach is adapted for each of the enhancements techniques i.e. nanofluid at varied concentration, twisted tape at varied pitch, spring insert at varied pitch and their combination

#### **4.5 Precautions**

1. During fabrication of twisted tape, the strip should be kept under constant tension and it should be rotated slowly so that there is no distortion.

2. Rotameter should be carefully calibrated as the Reynolds number depends upon mass flow rate and a small error in mass flow rate can upset the whole calculation

3. Necessary correction should be applied to RTD readings.

4. There should not be any air bubble in manometric line during pressure measurement

5. Nano fluid should be periodically observed to check whether there is any sort of particle settlement

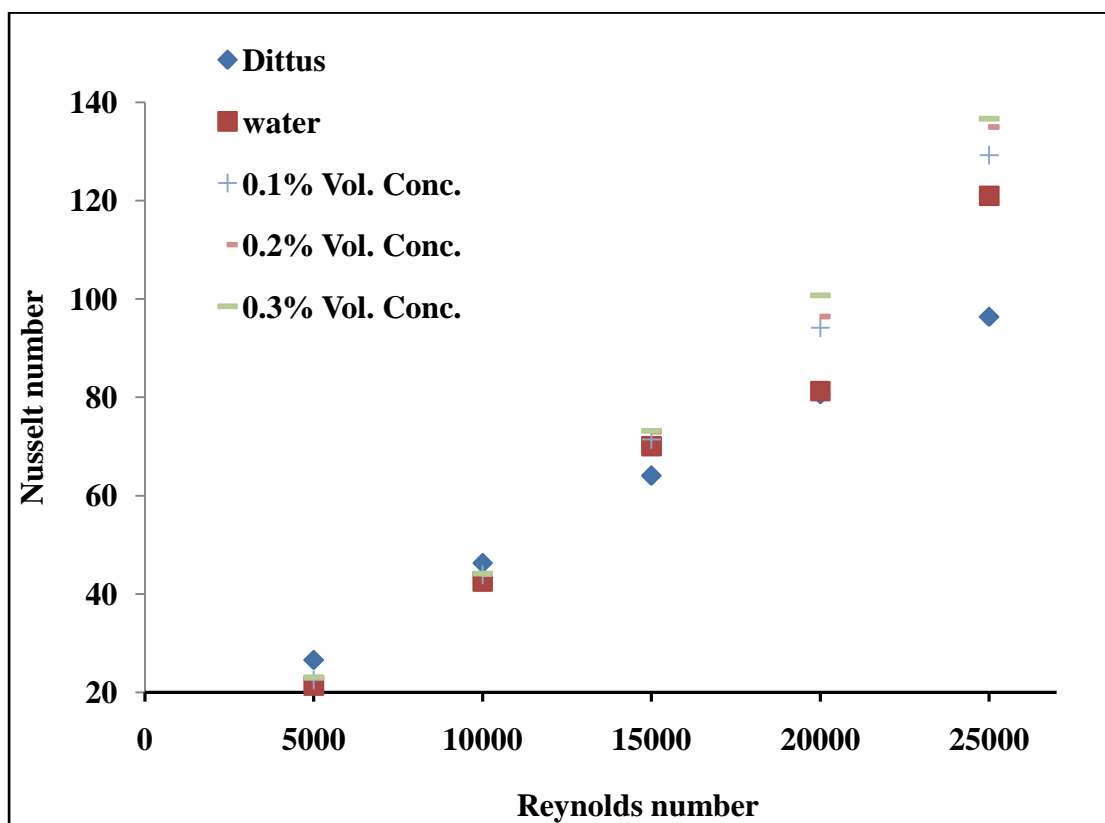
6. Readings for temperature should only be taken after the steady state conditions are achieved.

7. Inserts should not at all touch the heat exchanger body.

## RESULTS AND DISCUSSIONS

### 5.1 Effect of nanofluids of different concentration on Nusselt number and friction factor

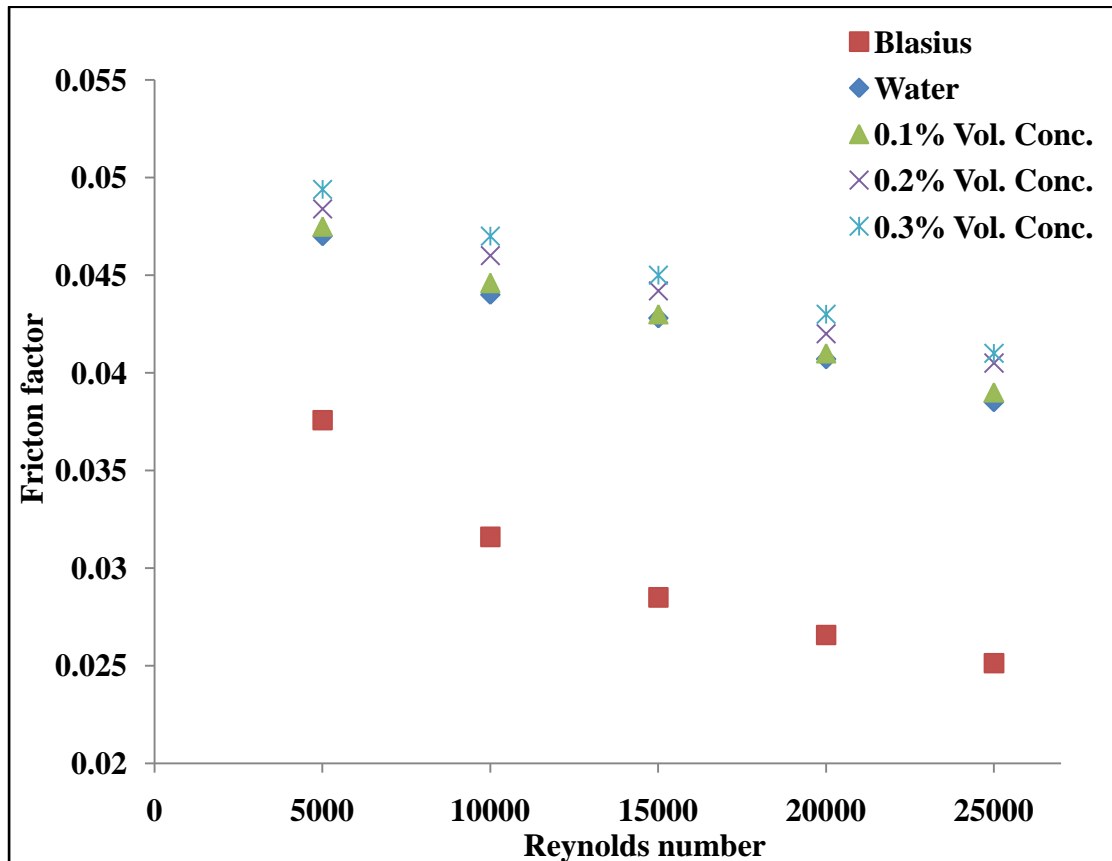
Heat transfer characteristics of double tube heat exchanger, employing plain tube with various working fluids (water, nanofluids of 0.1%, 0.2%, 0.3% vol. conc.), are depicted through variation of Nusselt number and friction factor as shown in figure 4.1 and figure 4.2 respectively.



**Figure 5.1 Nusselt number Vs Reynolds number for water and CuO-H<sub>2</sub>O nanofluid with 0.1%, 0.2% and 0.3% vol. conc.**

Figure 5.1 shows the variation of Nusselt number with Reynolds number for various working fluids (water, nanofluids of 0.1%, 0.2%, 0.3% vol. conc.). It is seen that as the Reynolds number is made to increase then, the corresponding enhancement in the Nusselt number with various working fluid (water, nanofluids of 0.1%, 0.2%, 0.3% vol. conc.) takes place. Also, it is observed that, as the volume concentration of the nanofluid increases then, the Nusselt number also increases. For the same Reynolds

number value, maximum Nussult number value is seen for the nanofluid of 0.3% vol. conc. At a Reynlods No. of 2500, an improvement of 13% is seen when water is replaced by naonofluid of 0.3% vol. conc.

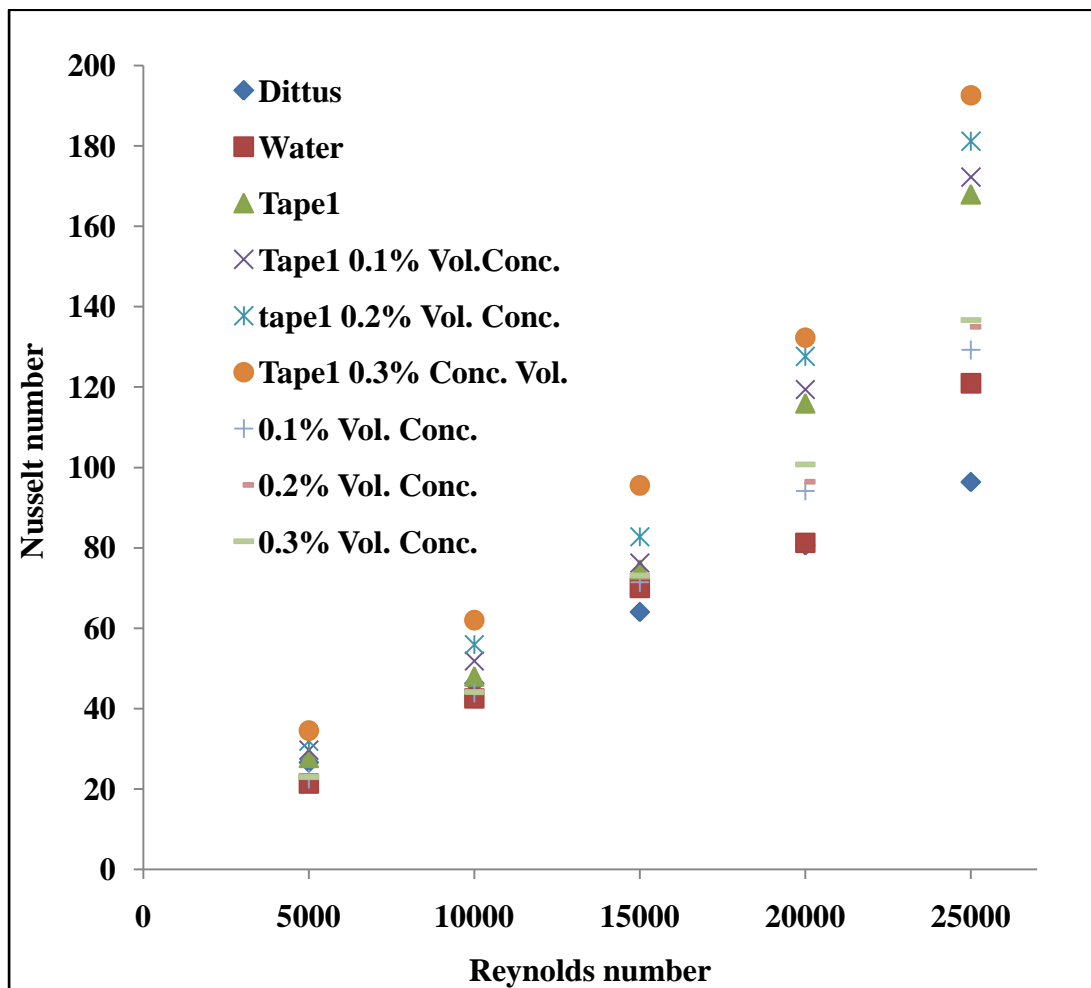


**Figure 5.2 Friction factor Vs Reynolds number for water and CuO-H<sub>2</sub>O nanofluid with 0.1%, 0.2% and 0.3% concentration by volume**

Figure 5.2 shows the variation of Friction factor with Reynolds number for various working fluids (water, naonofluids of 0.1%, 0.2%, 0.3% vol. conc.). It is observed that friction factor decreases with increase in Reynolds number, and highest value of friction factor come from lowest value of Reynolds number. From the graph, it is clear that as the volumetric concentration of CuO nanofluid is increased, and then increase in the value of pressure drop takes place. Pressure drop is maximum for the CuO nanofluid of 0.3% vol. conc. at the lowest Reynolds number of 5000. Percentage increase in friction factor for the CuO nanofluid of 0.3% vol. conc. is noted around 5% as compare to water at same Reynolds number of 5000.

## 5.2 Effect of twisted tape inserts on Nusselt number and friction factor

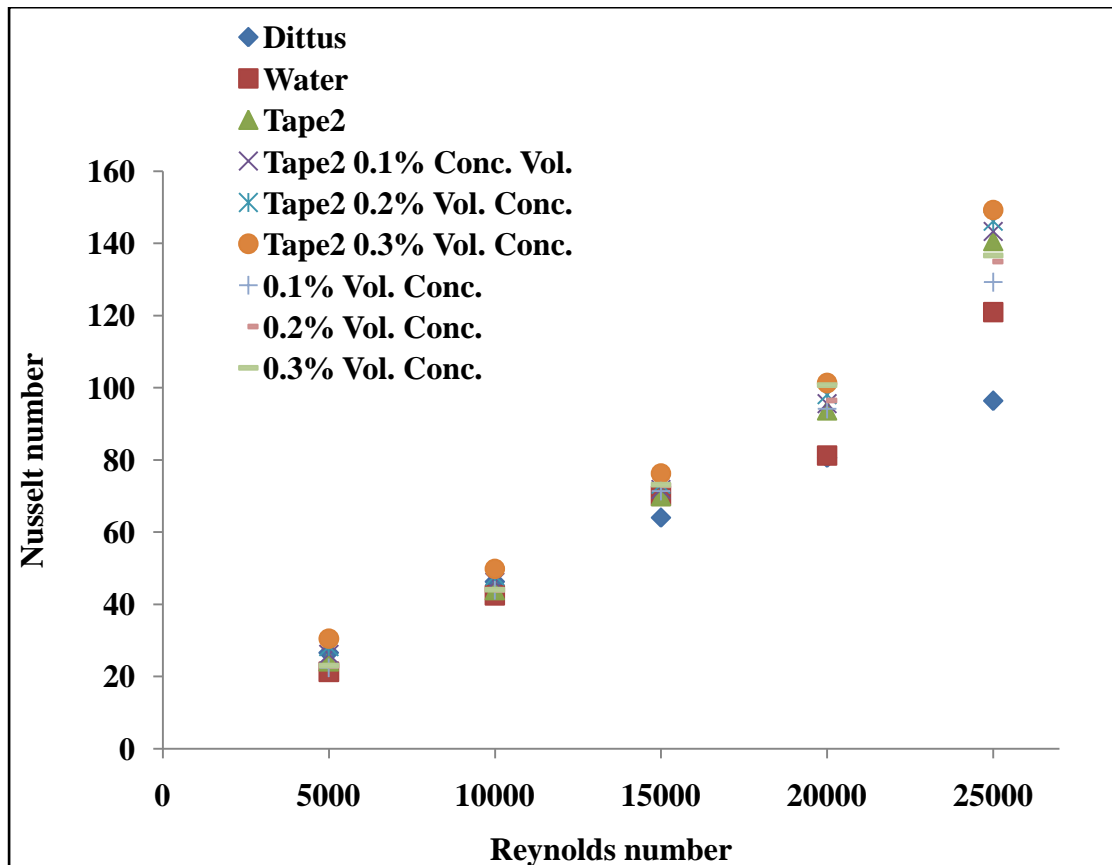
Variation of Nusselt number and friction factor with Reynolds number for a different working fluids like water, water with twisted tape (twist ratio 5 and 10), various nanofluid of different vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with twisted tape (twist ratio of 5 and 10) is shown in figure 5.3, 5.4 and 5.5.



**Figure 5.3 Reynolds number Vs Nusselt number for water, water with twisted tape of twist ratio 5, CuO-H<sub>2</sub>O nanofluid of 0.1%, 0.2% and 0.3% vol. conc., CuO-H<sub>2</sub>O based nanofluids combined with twisted tape**

Plot of variation of Nusselt number with Reynolds number is depicted in figure 5.3. Maximum value of Nusselt number is observed with nanofluid of 0.3% vol. Conc. having inserts of twisted tape (twist ratio 5). From graph, it is inferred that heat transfer rate is increased with corresponding increase in the vol. conc. of nanofluid, but significant improvement is witnessed when twisted tape (twist ratio 5) is inserted

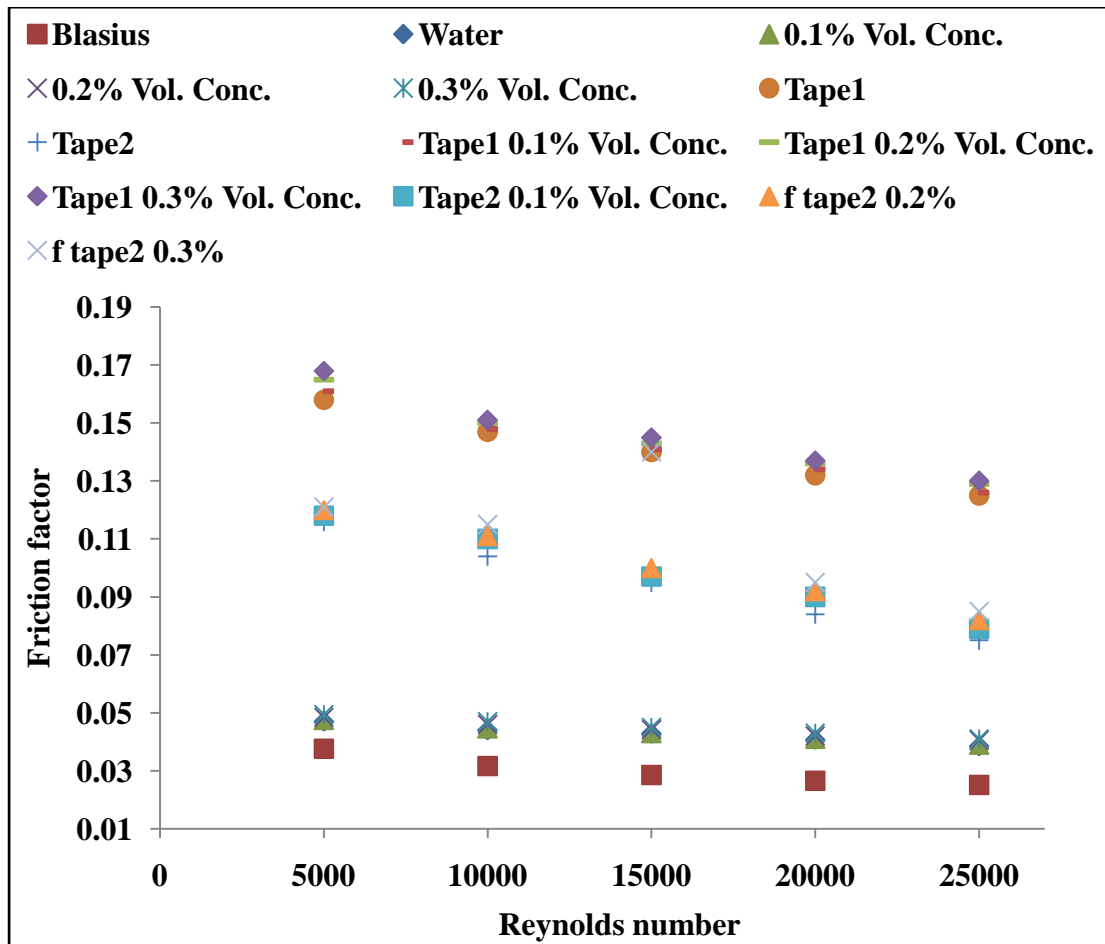
within the fluid region. Improvement of about 41% is observed in Nusselt number when twisted tape of twist ratio 5 is used with 0.3% vol. conc. nanofluid at a Reynolds number of 25000. Also, when water is replaced with 0.3% vol. conc. nanofluid having inserts of twisted tape (twist ratio 5), an improvement of 59% in the value of Nusselt number is observed at the same Reynolds number.



**Figure 5.4 Reynolds number Vs Nusselt number for plain water, water with twisted tape of twist ratio 10, nanofluids of 0.1%, 0.2% and 0.3% by volume, CuO-H<sub>2</sub>O nanofluids combined with twisted tape**

Variation of Nusselt number with Reynolds number for a different working fluids like water, water with twisted tape (twist ratio 10), various nanofluid of different vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with twisted tape is shown in figure 5.4. It is reported that Nusselt number increases with increase in Reynolds number for all used working fluids. Maximum value of Nusselt number is seen with nanofluid of 0.3% vol. conc. With the application of twisted tape having a twist ratio of 10, more improvement in the heat transfer is observed as compared with tube without having twisted tape. Around 9% improvement in the

Nusselt number is observed when twisted tape (twist ratio 10) is used with nanofluid of 0.3% vol. concentration as compared to nanofluid of same volumetric conc. without twisted tape at a Reynolds number of 25000. When water is replaced by nanofluid of 0.3% vol. conc. with twisted tape, an enhancement of 23% in Nusselt number is observed.



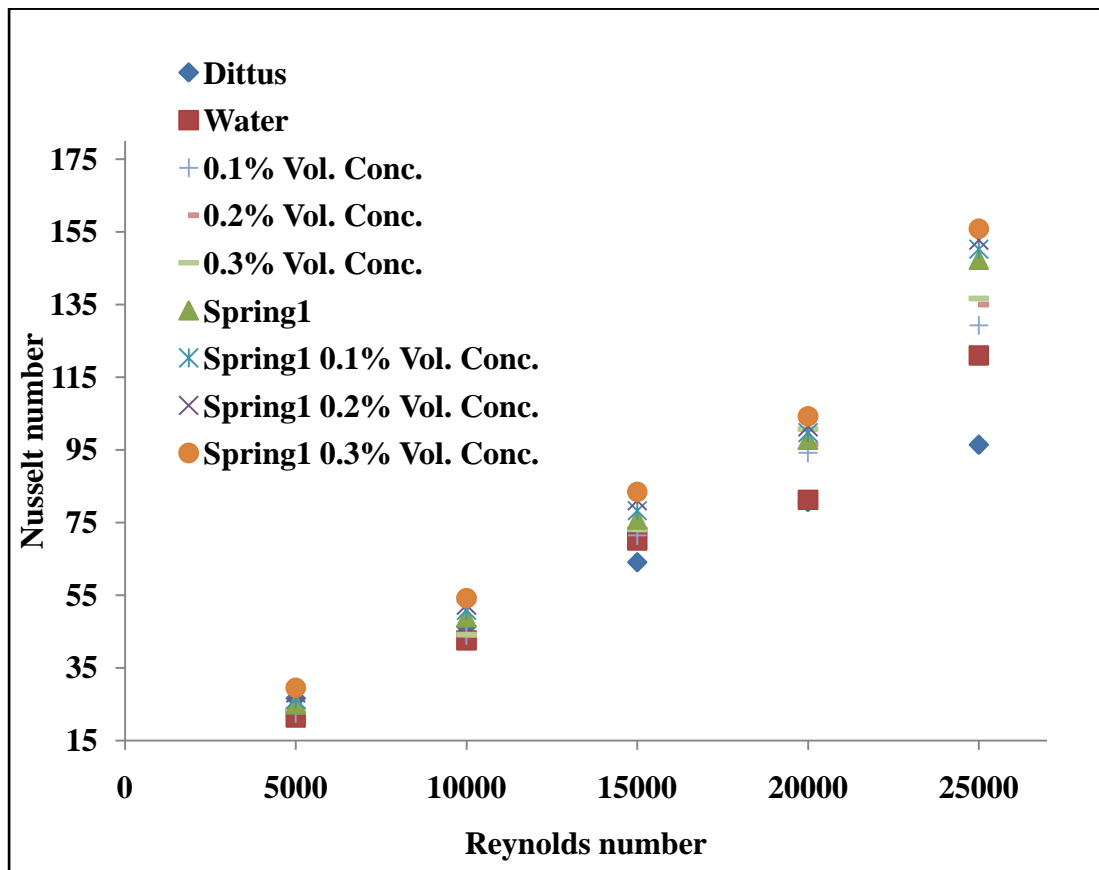
**Figure 5.5 Reynolds number Vs friction factor for plain water, water with twisted tape of twist ratio 5 and 10, nanofluids of 0.1%, 0.2% and 0.3% by volume, CuO-H<sub>2</sub>O based nanofluids combined with twisted tape**

Variation of friction factor with Reynolds number for a different working fluids like water, water with twisted tape (twist ratio 5 and 10), various nanofluid of different vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with twisted tape (twist ratio of 5 and 10) is shown in figure 5.5. It is reported that, there is drop in the value of pressure with corresponding increment in the Reynolds number. For a particular flow rate, value of friction factor increases, as the volume

conc. of nanofluid increases. Maximum value of friction factor is observed when 0.3% vol. conc. nanofluid with twisted tape is used. Also, as the pitch value of the twisted tape increases, drop in the value of pressure drop is seen with nanofluid of different volumetric concentration at a same Reynolds number.

#### . 4.3 Effect of spring inserts on Nusselt number and friction factor

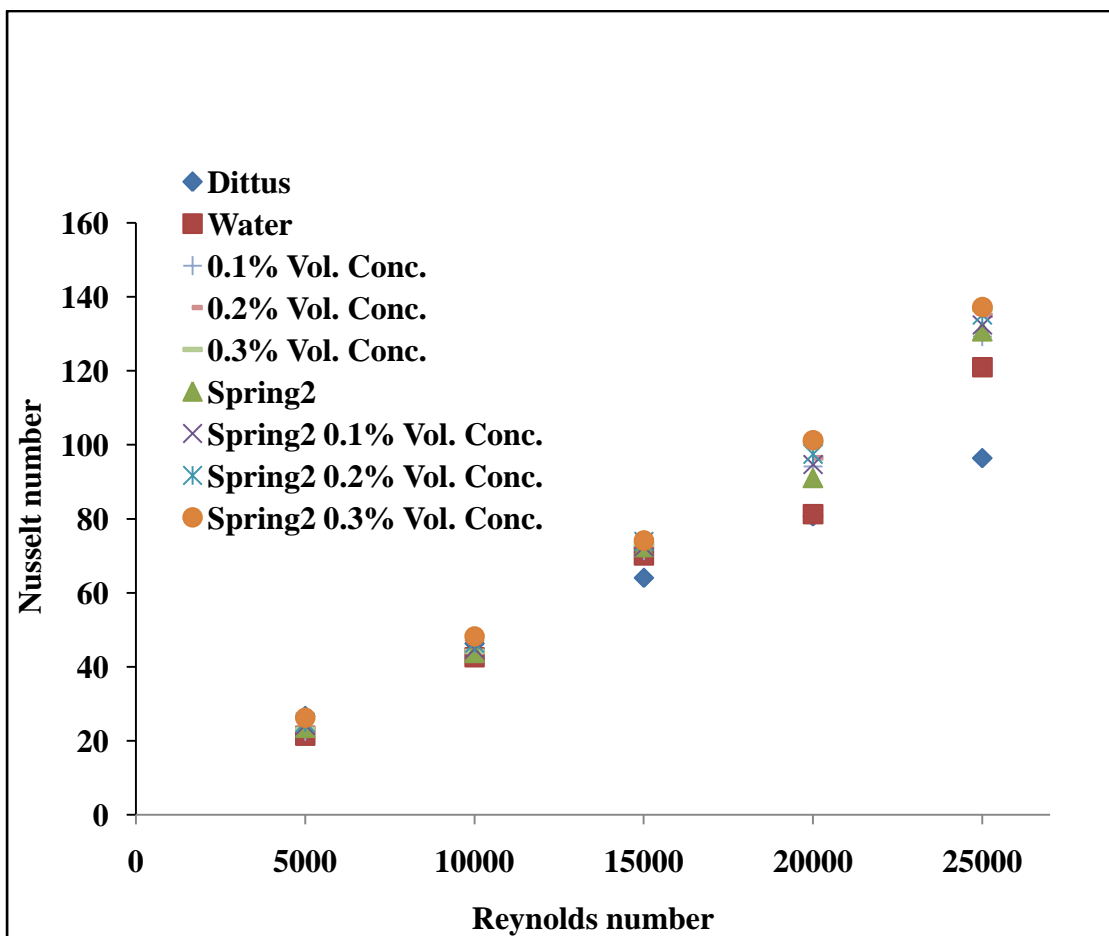
Variation of Nusselt number and friction factor with Reynolds number for a different working fluids like water, water with spring inserts with pitch of 5mm and 10mm, various nanofluid of different vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with spring insert (pitch 5 mm and 10 mm) is shown in figure 5.6, 5.7 and figure 5.8.



**Figure 5.6 Reynolds number Vs Nusselt number for plain water, water with spring insert of pitch 5 mm, nanofluids of 0.1%, 0.2% and 0.3% by volume, CuO-H<sub>2</sub>O based nanofluids combined with spring insert**

Variation of friction factor with Reynolds number for a different working fluids like water, water with spring inserts with pitch of 5mm, various nanofluid of different

vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with spring insert (pitch 5 mm) is shown in figure 5.6. It is observed that there is an improvement in the heat transfer rate when spring inserts is used, which is due to better mixing of fluid, increased turbulence and generation of swirl flow. With the usage of nanofluid with spring insert, an improvement in the heat transfer rate takes place, which increases with increasing vol. conc. of nanofluid. An improvement of about 28% in the heat transfer rate is observed when 0.3% vol. conc. nanofluid is replaced with water. Also, with the application of spring insert improvement in the heat transfer rate is observed.



**Figure 5.7 Reynolds number Vs Nusselt number plot for plain water, water with spring insert of pitch 10 mm, nanofluids of 0.1%, 0.2% and 0.3% by volume, CuO-H<sub>2</sub>O based nanofluids combined with spring insert**

Variation of Nusselt number with Reynolds number for a different working fluids like water, water with spring inserts with pitch of 10 mm, various nanofluid of different

vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with spring insert (pitch 10 mm) is shown in figure 5.7. It is seen that, value of Nusselt number increases with increasing Reynolds no. for all working fluids. Maximum value of Nusselt number is observed with 0.3% vol. conc. nanofluid employing spring insert. Improvement of about 13% in the value of Nusselt number is observed, when 0.3% vol. conc. nanofluid with spring insert is replaced with water. Also, the value of Nusselt number is enhanced with the application of spring insert.

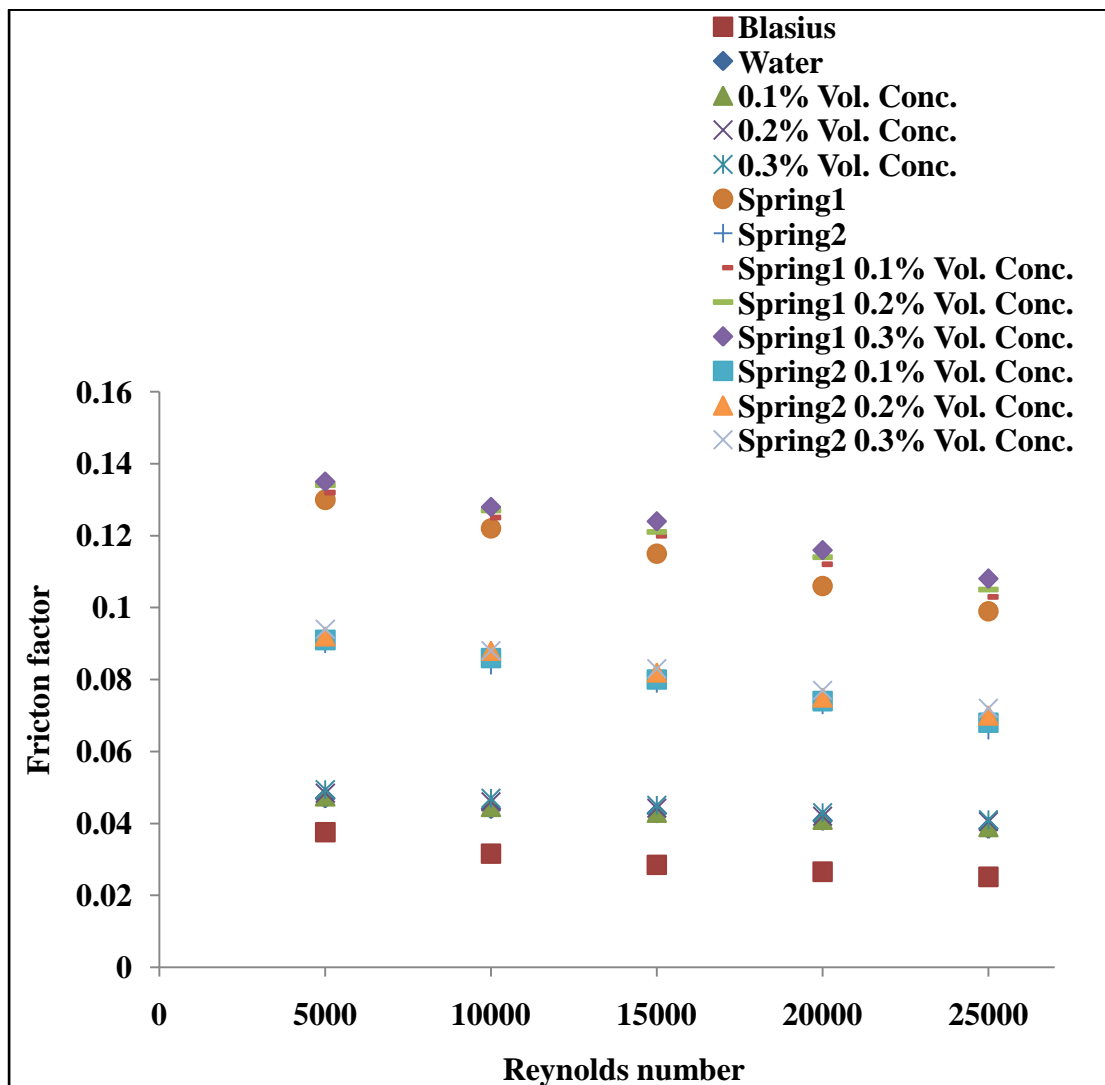


Figure 5.8 Reynolds number Vs Friction factor plot for plain water, water with spring inserts having pitch 5 mm and 10 mm, CuO-H<sub>2</sub>O based nanofluids with concentrations 0.1%, 0.2% and 0.3% by volume and combination of CuO-H<sub>2</sub>O based nanofluids and spring inserts.

Variation of friction factor with Reynolds number for a different working fluids like water, water with spring inserts (pitch 5 and 10), various nanofluid of different vol. concentrations (0.1%, 0.2% and 0.3%) and nanofluid of same vol. concentrations with spring inserts (pitch 5 and 10) is shown in figure 5.8. Higher value of friction factor is observed when spring of 5 mm pitch is used as compared to spring of 10 mm pitch. Also, as the volume concentration of nanofluid is increased, then value of friction factor shows increasing trend. The highest value of friction factor is obtained for the configuration where the combination of spring insert with pitch 5 mm is used with nanofluid of 0.3 % vol. conc. when the usage of spring insert having pitch 5 mm, an improvement of about 176% is seen in friction factor, but when both spring insert (5 mm pitch) and 0.3% vol. conc. nanofluid are used altogether, improvement of 187% is seen in the friction factor at the Reynolds number of 5000.

### CONCLUSIONS AND FUTURE SCOPES

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#### 6.1 Conclusions

The experimental results of the heat transfer enhancement by using CuO-water nanofluid in double tube counter flow heat exchanger which was equipped with twisted tape and spring of different twist ratio and spring pitch, respectively as turbulators lead to the following conclusions:

- Nusselt number, convective heat transfer coefficient and friction factor associated with application of CuO-water nanofluid, twisted tape and spring are higher than those associated with the individual techniques.
- Convective heat transfer coefficient and friction factor increases with increasing concentration of CuO-water nanofluid.
- Convective heat transfer coefficient and friction factor are higher for twisted tape of low twist ratio and for spring of low spring pitch.
- Heat transfer rate is higher for twisted tape than spring within given flow regime.
- Within the selected range of flow (5000 to 25000), maximum value of Nusselt number was found with the use of 0.3% by volume concentration, equipped with twisted tape of twist ratio 5 at Reynolds number 25000.
- Improvement of about 41% is observed in Nusselt number when twisted tape of twist ratio 5 is used with 0.3% vol. conc. nanofluid at a Reynolds number of 25000.

#### 6.2 Future scopes

- Different types of inserts such as louvered strip inserts and wire mesh inserts can be used to carry out experimental investigation.
- Experiments can be conducted to compare the relative effectiveness of two or more different nanofluids such as CuO, Al<sub>2</sub>O<sub>3</sub> and Aluminium in metal form.
- The experiments can be conducted for longer periods to know the effect of these heat transfer enhancement methods on the fouling characteristics of the

heat exchanger as increased turbulence and swirl will result in lower depositions and that aspect can be highly useful in reducing the maintenance costs of heat exchangers that the users have to bear in case of conventional systems.

- Empirical correlations can be derived for each of the enhancement techniques.
- Application of enhancement techniques for other exchangers such as shell and tube or plate heat exchangers can be considered.
- Experimental results can be verified by using CFD simulation.
- The better method can be applied at a practical situation to verify the experimental observations.

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## Appendix A1

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**Table A1.1 Temperature at inlet and outlet of both hot and cold fluid for simple double heat exchanger where is used as working fluid**

Reynolds number	M	Thi	tho	tci	tc0
5000	0.016588	80	58.1	27.4	31.8
10000	0.033175	80	63.6	27.4	33.9
15000	0.049763	80	66.6	27.4	35.3
20000	0.06635	80	69.3	27.4	35.9
25000	0.082938	80	70.1	27.4	36.3

**Table A1.2 Temperature at inlet and outlet of both hot and cold fluid for simple double heat exchanger with twisted tape of twist ratio 5 and 10 where water is used as working fluid**

Thi	Tho	tci	tc0
80	54.1	27.4	32.9
80	61	27.4	35.2
80	64.4	27.4	36.9
80	66.1	27.4	38
80	67.2	27.4	38.4

thi	tho	tci	tc0
80	56.2	27.4	32.4
80	62.1	27.4	34.4
80	65.5	27.4	36.1
80	67.8	27.4	37
80	68.7	27.4	37.5

**Table A1.3 Temperature at inlet and outlet of both hot and cold fluid for simple double heat exchanger with spring of pitch 5 and 10 mm where water is used as working fluid**

thi	Tho	tci	tc0
80	55.4	27.4	32.6
80	61.8	27.4	34.9
80	65.2	27.4	36.6
80	67.4	27.4	37.8
80	68.2	27.4	38.3

thi	tho	tci	tc0
80	56.6	27.4	32.3
80	62.4	27.4	34.4
80	65.9	27.4	35.9
80	68.8	27.4	36.9
80	69.5	27.4	37.7

## Appendix A2

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### A2.1 Sample calculations

Inner diameter of the test section,  $d_i = 12$  mm

outer diameter of the test section,  $d_o = 25.4$  mm

length of test section = 1.5m

$\mu_{water}$  at  $80^\circ\text{C} = 3.52 \times 10^{-4}$  pas

$\rho_{water}$  at  $80^\circ\text{C} = 972$  kg/m<sup>3</sup>

mass flow rate,  $m = \rho A v$

$$= \rho A \times \frac{R_e \mu}{D \rho}$$

$$= \mu \times \frac{\pi}{4} \times d \times R_e$$

$$m = 3.3175 \times 10^{-6} R_e$$

$$Q = \frac{m}{\rho} = 3.413 \times 10^{-9} R_e \text{ Type equation here.}$$

**Table A2.1 Calculated value of flow rate**

$R_e$	m(kg/s)	Q(lpm)
5000	0.0165875	1.0239
10000	0.033175	2.0478
15000	0.0497625	3.0717
20000	0.06635	4.03567
25000	0.0829375	5.1195

Sample calculation for  $Re = 5000$

for inner pipe (hot),

$$Pr = \frac{\mu C_p}{k} = 2.22$$

$$Nu = 0.023 Re^{0.8} Pr^{0.3} = 26.59$$

$$Nu = \frac{h_i D_i}{k}$$

$$h_i = 1484.608 \text{ w/m}^2 \text{ k}$$

For Annular section,

Q=5 lpm (constant)

$$m = \rho Q = 0.0830833 \text{ kg/s}$$

$$m = e \times \frac{\pi}{4} (d_0^2 - d_i^2) \times \frac{Re \times \mu}{e \times (d_0 - d_i)}$$
$$= 0.083083 \text{ kg/s}$$

Therefore,

$$Re = 2529.892$$

Assuming isothermal inside and adiabatic outside

By interpolation,

$$\text{for } \frac{d_i}{d_0} = 0.063, \quad Nu = 5.5112$$

$$h_0 = \frac{Nu \cdot k}{D_h} = 357.6417 \text{ W/m}^2\text{K}$$

$$\text{Thermal Resistance, } R_{th} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{D_0}{D_i}\right)}{2\pi k_{cu}} + \frac{1}{h_0 A_0}$$

$$R_{th} = 0.056607842$$

$$R_{th} = \frac{1}{U_i A_i} = U_i = \frac{1}{R_{th} A_i} = 702.8838 \text{ W/m}^2\text{K}$$

For the Reynolds Number,  $Re = 5000$

mass flow rate of hot fluid,  $m_h = 0.0165875 \text{ kg/s}$ ,

mass flow rate of cold fluid,  $m_c = 0.083083 \text{ kg/s}$

temperature of hot fluid at inlet,  $t_{h_i} = 80^\circ\text{C}$

temperature of hot fluid at outlet,  $t_{h_0} = 58.1^\circ\text{C}$

temperature of cold fluid at inlet,  $t_{c_i} = 27.4^\circ\text{C}$

temperature of cold fluid at outlet,  $t_{c_0} = 32^\circ\text{C}$

$$Q_h = m c_p (\Delta T_h) = 1.522 \text{ kW}$$

$$Q_c = m c_p (\Delta T_c) = 1.666977 \text{ kW}$$

$$Q_{Avg} = \frac{Q_h + Q_c}{2} = 1.5944886kw$$

$$Q_{Avg} = U_i A_i \Delta T_{lmtd}$$

$$\Delta T_{lmtd} = 38.62^\circ C$$

$$U_i = \frac{Q_{Avg}}{A_i \Delta T_{lmtd}} = 730.08853w/m^2k$$

Blasius correlation for friction factor

$$f = \frac{0.316}{Re^{0.25}}, Re \leq 10^5$$

for  $Re = 5000$ ,  $f=0.03757$

$$f = \frac{12.1 \times D^5 \times h_{f1}}{LQ^2} = 0.041356$$

## Calculation for thermo physical properties of nanofluids

### Thermal conductivity

$$k_{nf} = \left[ \frac{k_{np} + 2k_{bf} + 2\theta(k_{np} - k_{bf})}{k_{np} + 2k_{bf} - \theta(k_{np} - k_{bf})} \right] k_{bf}$$

For CuO nanofluids with 0.1% concentration by volume

$k_{nf}=78w/mk$ ,  $k_{bf}=0.61w/mk$

$$k_{nf} = 0.881037$$

### Dynamic viscosity

$$\eta = 0.355 \times 10^{-3} kg/m^{-s}$$

$$\mu_{nf} = \frac{1}{[1 - \phi]^{2.5}} = 4.6197 \times 10^{-4}$$

### Density

true density= $6.4g/cm^3$

$$\rho_{nf} = \phi \cdot \rho_{np} + (1 - \phi)\rho_{bf}$$

for 0.1%,  $\rho_{nf} = 1540\text{kg/m}^3$

**specific heat capacity**

$$Cp_{np} = 531.8 \text{ J/kgk}$$

$$Cp_{nf} = \frac{(1-\phi)\rho_{bf} Cp_{bf} + \phi \cdot \rho_{nf} \cdot Cp_{np}}{\rho_{nf}} = 2500.128 \text{ J/kgk}$$

For heat transfer coefficient calculation, the Wilson chart is Prepared.

$$R_{ov} = R_i + R_w + R_e$$

$$R_{ov} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi K_w L_w} + \frac{1}{h_o A_o}$$

Where,

$h_i$  = internal convection coefficient

$h_o$  = external convection coefficient

$A_o$  = outer tube surface area

$A_i$  = inner tube surface area

the overall thermal resistance can also be repressed as a function of the overall heat transfer coefficient

$$R_{ov} = \frac{1}{U_{i/o} A_{i/o}}$$

William theorized that if the mass flow of the cooling liquid was modified, then the change in the overall thermal resistance would be mainly due to the variation of the in-tube convection coefficient, while the remaining thermal resistance remained nearly constant. Therefore, the thermal resistances outside of the tubes and the tube wall could be considered constant.

$$R_w + R_o = c_1$$

Therefore,

$$\frac{1}{U} = \frac{1}{h_i} + B$$

And also, the heat transfer coefficient is related to Reynolds number as

$$h_i = CRe^m$$

Now,

$$\frac{1}{U} = \frac{1}{CRe^m} + B = A Re^{-m} + B$$

From above equation, it implies that the plot between

$\frac{1}{U}$  and  $Re^{-m}$  is a straight line with its slope of A and intercept of B in yaxis

Therefore, after arranging the above equation,

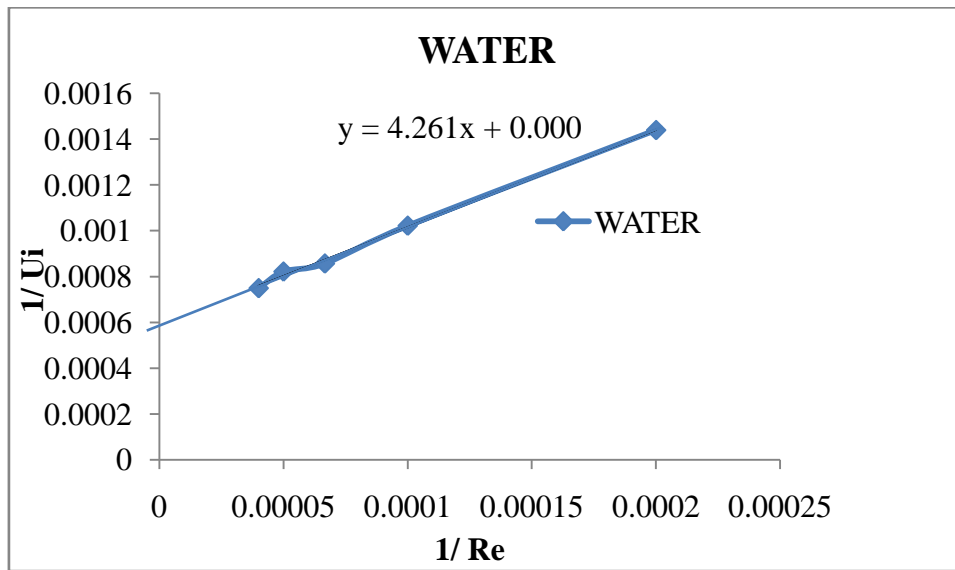
we can write it as

$$\frac{1}{h_i} = \frac{1}{\left[\frac{1}{U} - B\right]}$$

The average Nusselt number based on the inner diameter of the tube will be calculated as

$$Nu = \frac{h_i d_i}{k}$$

One of the sample graphs from the Wilson chart is shown below.



**Figure A2.1 Reynolds number Vs Overall heat transfer coefficient**