To Study the Effect of Al/water Nanofluids on Thermo-hydraulic Performance of Single-pass Cross-flow Compact Heat Exchanger

A

Dissertation

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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JULY, 2015

CERTIFICATION

I, Suveg Singh, declare that this thesis report entitled **"To Study the Effect of Al/water Nanofluids on Thermo-hydraulic Performance of Single-pass Cross-flow Compact Heat Exchanger"** submitted towards fulfillment of the requirements for the award of Master's Degree in Thermal Engineering, in Mechanical Engineering Department of Thapar University, Patiala, India is entirely my own work. This document has not been submitted for any degree in any other institution.

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Dedicated to My Grand Parents

Late Mr. Bhajan Singh

Mrs. Gurdev Kaur

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Parents

Mr. Jagtar Singh

Mrs. Sarabjeet Kaur

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Mrs. Karamjeet Kaur

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ABSTRACT

Miniaturization of thermal systems along with ultra-high performance is one of the essential need for industrial, domestic and automobile cooling systems. Thermal conductivity of the heat transfer fluids have a vital role in the development of energy-efficient fluids. Poor thermal conductivity of conventional fluids such as water, ethylene glycol and oils put a constraint on the development of energy efficient thermal devices. Nanofluids, seem as panacea for thermal equipments, which are the suspension of nano sized particles (typically 1-100 nm) in base fluids provided by nanotechnology. In recent years, nanofluids gained a lot of attraction because of their superior thermal properties over base fluids. Nanofluids have unique features, which are significantly different form conventional heat transfer fluids prepared by millimeter or micrometer sized particles. A lot of work has been done on the metal oxide based nanofluids, however few researchers studied the effect of metal based nanofluids on the performance of thermal devices. In the present work, effect of Al/water nanofluids on the thermo-hydraulic performance of a single pass cross flow compact heat exchanger has been investigated. Nanofluids were prepared by dispersing metal basis aluminum nanoparticles of 100 nm size into double distilled water. Various thermo-physical properties such as density, viscosity and thermal conductivity of nanofluids and their variation with fluid temperature were measured experimentally. Experiments were performed on single pass cross flow compact heat exchanger by varying various parameters such as hot fluid flow rate, velocity of cold air, nanoparticle volume concentration and inlet temperature of hot fluid. Performance of heat exchanger was investigated by studying the effect of these parameters on hot and cold fluid side Nusselt number and friciton factor and Colburn factor was also studied for cold fluid side. It was observed that hot fluid side Nusselt number was improved by 3.23% and 4.65% for 0.1% and 0.2% concentration of nanofluid, respectively at 45°C inlet fluid temperature as compared to distilled water. Colburn factor was increased by 11.11% and 13.9% for 0.1% and 0.2% nanoparticle volume concentration of nanofluids, respectively at 45°C inlet fluid temperature with respect to base fluid. Hot fluid side friction factor was increased by 14.38% and 21.2748% for 0.1% and 0.2% nanoparticle volume concentrations of nanofluids, respectively with respect to distilled water but it was decreased by 2.63% and 9.50%, when temperature was increased from 45 to 50 and 55°C.

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NOMENCLATURE

A_t : total heat transfer surface area, m^2

A_o : free flow areas of the exchanger, or Cross section area of exchanger, m^2

A_f : surface area of fin exposed to heat transfer, m^2

A_{fr} : air side frontal area on one side of the exchanger, m^2

A_{nf} : non fin area, m^2

$A_{c,t}$: cross section area of tube, m^2

T_t : tube thickness, m

T_w : tube width, m

T_l : tube length for small cross section, m

$T_{t,1}$: total tube length in core dimension, m

T_s : tube sheet thickness, m

F_t : fin thickness, m

F_l : fin length, m

F_w : fin width, m

$F_{t,l}$: total fin length, m

N_t : number of tubes one side

N_f : number of fins in between two tube.

N_s : total number of fins one side

A : tube spacing, m

B : fin spacing, m

b : half of tube thickness for small cross section = (fin spacing)/2, m

W : fluid flow (air) length, m

Cf : circumference of fin exposed to heat transfer, m.

Cp : specific heat of fluid at constant pressure, J/kg°C.

D_h : hydraulic diameter of flow passage, m

f : friction factor, dimensionless

G : mass velocity, kg/m^2 s

C : heat capacity rate

H : total water flow length, m

h : heat transfer coefficient, W/m^2°C

J : Colburn factor, dimensionless

k : fluid thermal conductivity, W/m°C

kf : thermal conductivity of fin material, W/m°C

l : fin length for heat conduction from primary to the midpoint between plates, m

L : non-fluid flow length, m

LPH : Litres per hour.

fp : fin perimeter, m-1

P : pressure, Pa

Pr : Prandtl number, dimensionless

Nu : Nusselt number, dimensionless

Re : Reynolds number based on hydraulic diameter, dimensionless

r_h : flow passage hydraulic radius, m

T : fluid temperature, °C

U : overall heat transfer coefficient, W/m^2 °C.

V : volume, m^3

v : velocity, m/s

m : fluid mass flow rate, kg/s

R_c : fouling resistance, W/m^2°C

R_t : tube wall resistance, W/m^2 °C

R_h : fouling resistance, W/m^2 °C

α : ratio of total heat transfer area of one side to its volume, m^2/m^3

ρ : density, kg/m^3.

δ : thickness of fin, m.

μ : fluid dynamic viscosity, Pa.s.

σ : ratio of free flow area to frontal area, dimensionless.

ϕ **:** diameter, m.

φ **:** volume fraction.

ε **:** thermal conductivity enhancement.

η_f **:** fin efficiency

η_O **:** overall efficiency

Subscripts

a **:** air

b **:** bulk

c **:** cold fluid side

h **:** hot fluid side

w **:** water

1 **:** inlet condition

2 **:** outlet condition.

CHAPTER 1

INTRODUCTION & OBJECTIVES

Heat exchanger is a vital component of thermal systems. With ceaseless technological development of thermal systems, need arises to develop new strategies for improving the thermal performance of components along with compactness. Heat transfer performance of a heat exchanger highly depends on thermal conductivity of the coolant used. Low thermal conductivity of conventional fluids used in heat exchangers is a major limitation to achieve the aforementioned objective. Fluids having suspension of nano sized particles of various metals or their oxides into conventional fluids possess higher thermal conductivity as compared to base fluid, hence overall performance of heat exchanger can be greatly enhanced using nanofluids as coolant.

1.1 Heat exchangers

Heat exchanger is a basic component of thermal systems, used to transfer heat between two media. There may be solid separation between two media or they may be in direct contact, according to application. Heat exchanger have numerous applications including automobile cooling, space heating, power plants, petroleum refineries, air conditioning and refrigeration etc. An automobile cooling system is shown in Figure 1.1, where radiator is a cross flow compact type heat exchanger where coolant absorbs heat from engine walls and transfers it to the air circulating over the radiator.

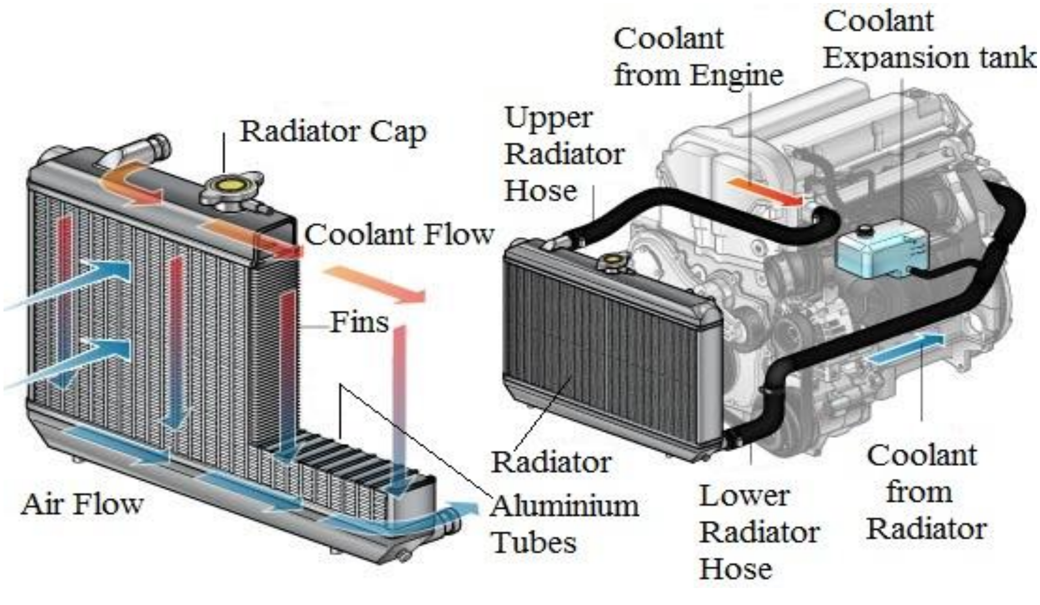

Figure 1.1: Automobile cooling system

In accordance to application area, heat exchangers are manufactured in a variety of configurations. Heat exchangers can be classified on the basis of following factors [1].

(i) Transfer processes (ii) Heat transfer mechanism (iii) Flow arrangement (iv) Surface Compactness (v) Construction (vi) Number of fluids etc.

This classification of heat exchangers is further elaborated in Figure 1.2.

CLASSIFICATION ACCORDING TO TRANSFER PROCESS

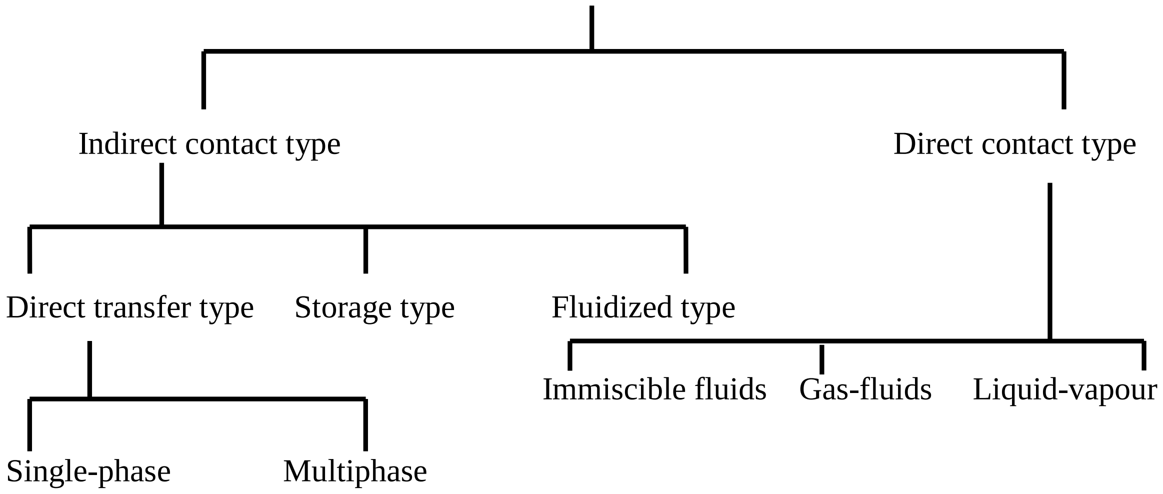

Figure 1.2 (a): Classification according to transfer processes

CLASSIFICATION ACCORDING TO HEAT TRANSFER MECHANISM

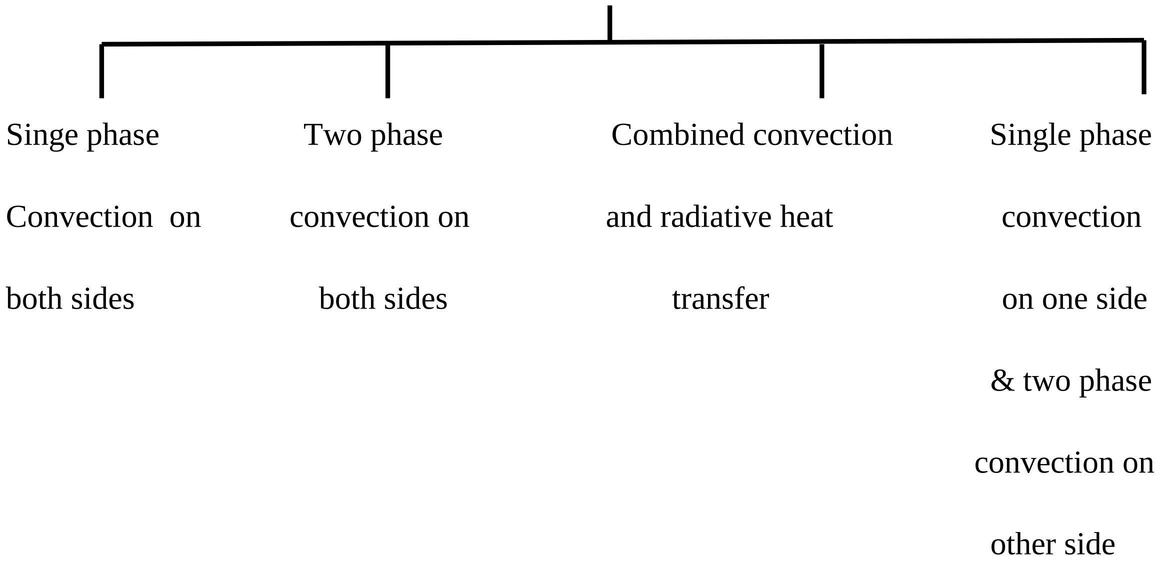

Figure 1.2 (b): Classification according to heat transfer mechanism

CLASSIFICATION ACCORDING TO FLOW ARRANGEMENT

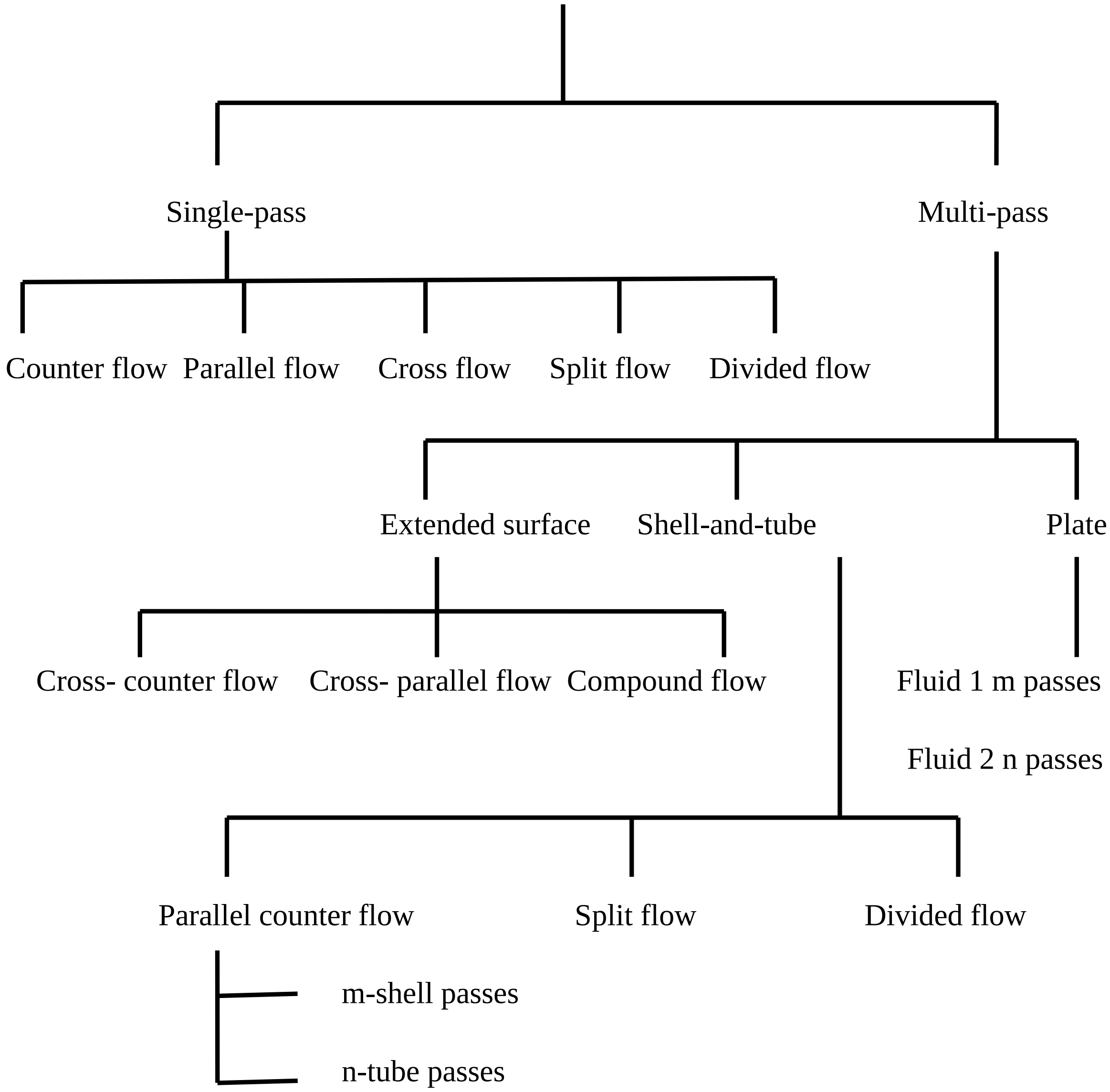

Figure 1.2 (c): Classification According to Flow Arrangement

CLASSIFICATION ACCORDING TO SURFACE COMPACTNESS

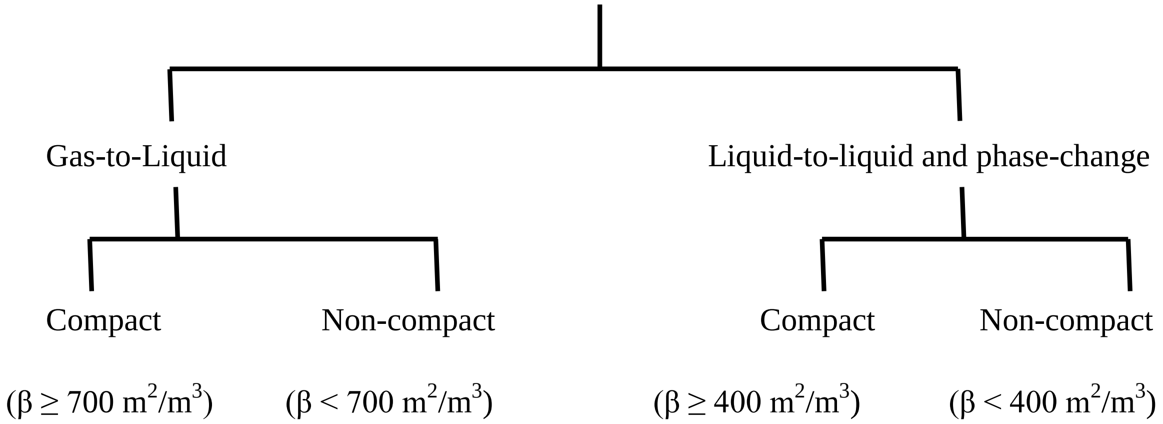

Figure 1.2 (d): Classification according to surface compactness

CLASSIFICATION ACCORDING TO CONSTRUCTION

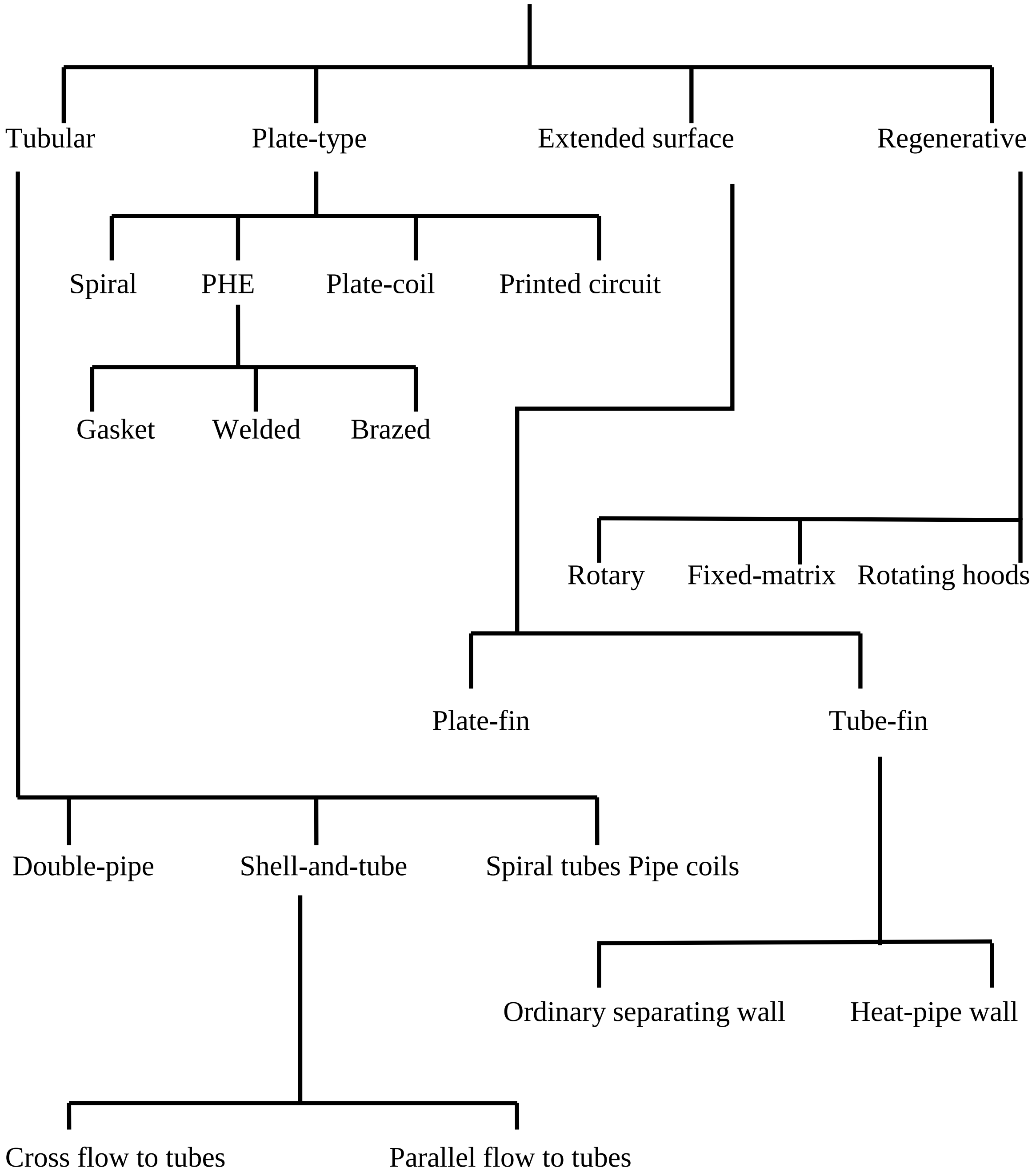

Figure 1.2 (e): Classification according to construction

CLASSIFICATION ACCORDING TO NUMBER OF FLUIDS

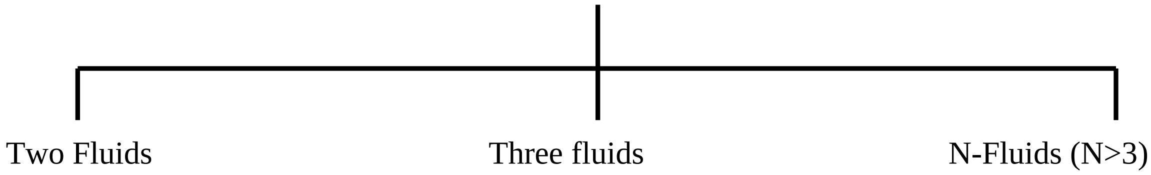

Figure 1.2 (f): Classification According to Number of Fluids

1.2 Finned tube compact heat exchangers

A very often used arrangement in automobile cooling system is cross-flow compact heat exchanger, where hot fluid from engine circulates in the tubes and air is made to pass over the tubes. In doing so, heat is transferred from hot fluid to cold fluid using heat exchanger. Since two fluids usually move at right angles to each other, such flow configuration is called cross-flow. Area density is defined as the ratio of heat exchanger surface area to its volume. Compact heat exchanger are characterized on the basis of area density, which is typically more than 700 m^2 per m^3 [2] and comparatively high heat transfer coefficient value. For liquid to gas heat exchangers, gas side heat transfer coefficient is comparatively very low, so secondary surfaces are attached to gas side to increase the heat transfer area on gas side so that its resistance to heat transfer can be reduced. Hence, compact heat exchangers are generally employed with fins. Finned tube compact heat exchanger configuration is shown in Figure 1.3.

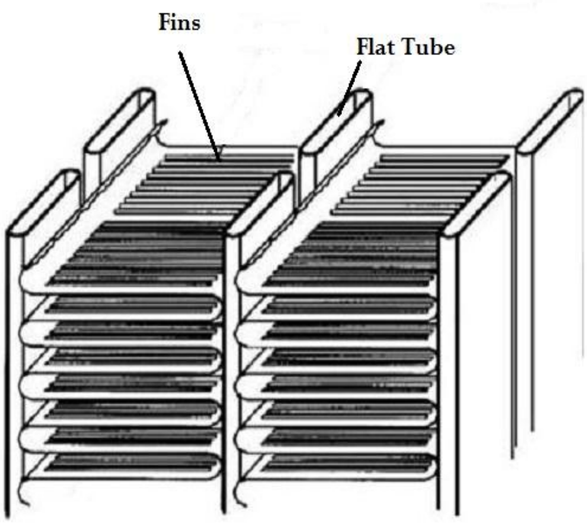

Figure 1.3: Finned tube Heat Exchanger

High area density caused the initial development and application of compact heat exchangers in automobile cooling systems, aerospace and marine engineering. Some common examples of compact heat exchangers are car radiators, gas turbine heat exchangers, the regenerator of a Stirling engine etc. Human lungs is also one of the examples, having area density of about 20000 m^2/m^3 [2]. Large area density of compact heat exchangers, indicating small hydraulic diameter for fluid flow, results in a higher efficiency than conventional heat exchanger in a significantly smaller volume. However, if the pressure drop and fouling phenomenon of such designs are taken into consideration, its versatility is limited. Compact heat exchangers are generally employed in gas-to-gas and liquid-to-gas heat exchangers to offset the low heat transfer coefficient related with gas flow with augmented surface area.

Finned tube heat exchangers have numerous configurations according to the application area. Finned tube heat exchanger is briefly classified as,

(a) Continuous fins on a array of tubes.

(b) Fins on inside of tube; internally attached fins, integral fins.

(c) Individual finned tubes; helical, annular fin geometries like spine, slotted, studded and wire loop fins, plain circular fins.

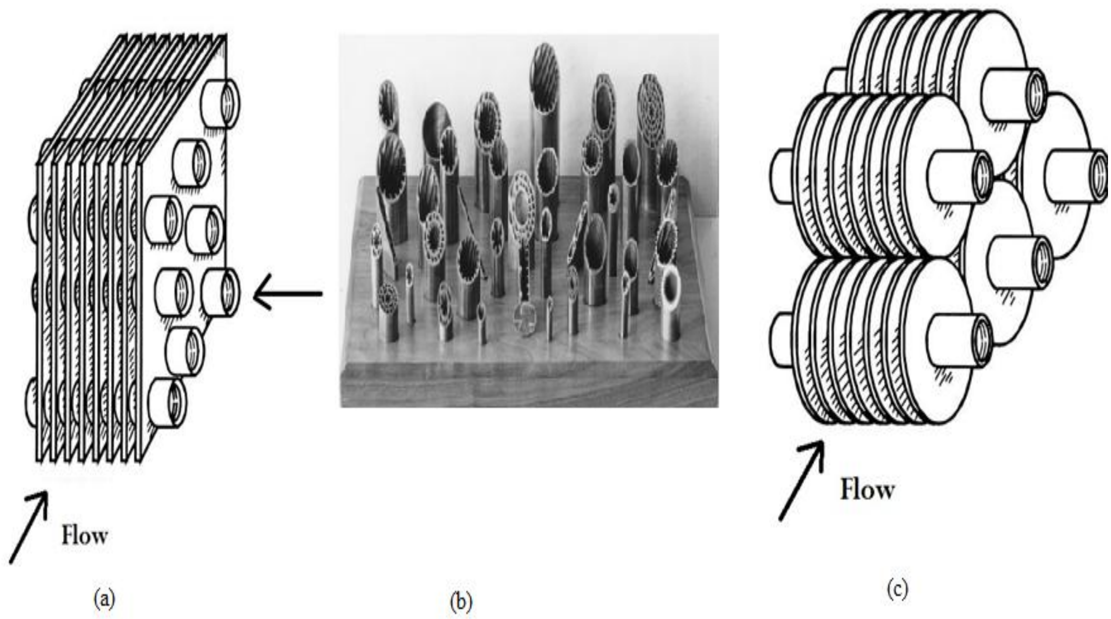

Figure 1.4: **(a)** Continuous fins on array of tubes **(b)** Internally attached fins (**c)** Individually finned tubes

1.3 Introduction to nanofluids

Ultrahigh-performance cooling is one of the most essential requirements of many engineering technologies. However, naturally low thermal conductivity is a primary constraint in developing energy-efficient heat transfer fluids that are required for ultrahigh-performance cooling. Metallic or non-metallic particles of nanometer dimensions can be produced by modern nanotechnology. Nanomaterials have unique mechanical, optical, electrical, magnetic, and thermal properties. Nanofluids are suspension of nanoparticles with average sizes below 100 nm in conventional heat transfer fluids such as water, oil, and ethylene glycol. A very small amount of nanoparticles, when dispersed uniformly and suspended stably in host fluids, can provide remarkable improvements in the thermal properties of base fluids.

Nanofluid is the term coined by Choi et. al. [3] to describe this new class of nanotechnology-based heat transfer fluids that exhibits thermal properties superior to those of their host fluids or conventional particle fluid suspensions. Nanofluid technology, a new interdisciplinary field of great importance where nanoscience, nanotechnology, and thermal engineering meet, has developed largely over the past decade. The goal of nanofluids is to achieve the highest possible thermal properties at the smallest possible concentrations by uniform dispersion and stable suspension of nanoparticles in base fluids.

1.4 Conventional methods of heat transfer enhancement & their limitations

Continuous development of technology requires the energy efficient thermal system along with their compactness. Overall performance of heat exchange device can be enhanced by increasing the heat transfer rate. Conventional methods used to increase the heat transfer rate with their limitations, are discussed below in brief.

1.4.1 Conventional solid-liquid suspensions

The conventional method used to increase cooling rates is to disperse millimetre or micrometer-sized particles into base fluids. The key problem with these suspensions is the fast settling of particles. Additionally, such particles are not appropriate to micro systems because they can cause clogging of micro channels. These conventional suspensions are not realistic because they require the addition of a large number of particles ensuing in drastic increase in pressure drop and pumping power.

1.4.2 Extended surfaces or Micro channel Cooling

A further approach to augment heat rejection rates is to employ extended surfaces, such as fins and micro channels, for air or liquid cooling. Micro scale heat exchangers have several attributes, including high thermal effectiveness, high heat transfer surface/volume ratio, small size, low weight, and design flexibility. Unfortunately, existing designs of thermal systems using this extended surface technology have reached its peak limits. Therefore, nanofluids seem to be a new alternative developed in order to meet these pressing needs for more efficient heat transfer fluids in several industries.

1.5 Concept of nanofluids

Thermal conductivity of heat transfer fluids plays imperative role in the preparation of energy-efficient heat transfer fluids. Inherent poor thermal conductivity of conventional fluids like water, oils, ethylene glycol etc. puts a constraint on the improvement of cooling performance of thermal systems. Thermal conductivity of metals in solid forms is significantly higher than that of fluids. Non-metallic liquids have thermal conductivity much lower than that of metallic liquids. Therefore, the thermal conductivity of metal based suspensions could likely be significantly higher than those of conventional heat transfer fluids. The thermal conductivity values for various metallic and non metallic liquids and solids at 300 K (otherwise mentioned) is tabulated by Das et al. [3] as shown in Table 1.1. Nanofluids are a new group of nanotechnology based heat transfer fluids engineered by dispersing nanometer-sized particles (generally less than 100 nm) into conventional heat transfer fluids. Heat transfer rate increases because nanoparticles have large surface area to volume ratio. Brownian motion between particles which depends on temperature and viscosity of base fluid, results in collision between particles which consequently results heat transfer to take place. Viscosity of the fluid decreases as temperature of the fluid increases which increase the Brownian motion and hence, conductivity increases this way. Particle size affects random motion of particles in the base fluid. Convection heat transfer phenomenon becomes dominant when particle size decreases [4].

Table 1.1: Thermal conductivity of different materials

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

1.6 Materials for nanoparticles and host fluids

Particles smaller than 100 nm size show evidence of properties diverse from those of traditional solids. Comparatively increased surface area/volume ratio results in the noble properties of nanophase materials. The thermal, optical, mechanical and electrical properties of nanophase materials are advanced to those of conventional materials with coarse grain structures. Base fluids and nano materials are available for the development of nanofluids are summarized as below.

I. **Host fluids:-** Many types of liquids, such as water, ethylene glycol, and oil, have been used as host liquids in nanofluids.

II. **Nanoparticle materials:-** Various materials used in the preparation of nanofluids are metal oxides like Al_2O_3, CuO, nitrides like AlN, SiN , carbides like SiC, TiC, metals like Cu, Ag, Au, Al, semiconductors like TiO2, SiC, and carbon nanotube.

1.7 Methods of nanofluid preparation

One-step and two-step production methods are used to prepare stable and highly conductive nanofluids. They are discussed below in brief.

1.7.1 The two-step method

This is the most common and extensively used method for preparation of nanofluid. In this method, firstly nanopowder is made by different methods viz. Inert-gas condensation (IGC), mechanical grinding, chemical vapour deposition (CVD), chemical emulsion; and then it is dispersed into base fluid with the aid of magnetic stirrer, ultra Sonication and intensive magnetic force agitation. Surfactants can be added to improve the dispersion.

1.7.2 The single-step method

In this process, nanoparticles are prepared and dissolve directly into host fluid simultaneously. Single step method is advanced from two step method. One-step chemical method is rapid and less costly than the one-step physical method. Process of drying, storage, transportation, and re-dispersion is eliminated in single step method. In this method, material vapours are condensed directly into nanoparticles with the help of low vapour pressure liquid flow.

1.8 Applications of nanofluids

Nanofluids are a promising new class of supplementary proficient heat transfer fluids that have the potential to significantly improve upon thermal systems in the numerous applications, due to their improved thermal conductivity and long period stability. Some common applications of nanofluids are discussed below in brief.

1.8.1 Transportation

Ordinarily used automotive coolant, is a relatively poor heat transfer fluid. Addition of nanoparticles to the conventional engine coolant highly improves the automobile engine cooling rates. Such enhancement can be used to eliminate engine heat with a compact cooling system.

1.8.2 Electronic applications

Higher density of electronic chips make design of electronic components more compact, which gives rise to heat dissipation problems. Heat removal can be improved either by improving the geometry or by using cooling fluids with higher thermal conductivity which, in turn, increases the convection heat transfer coefficient. Recent research has illustrated that nanofluids could increase the heat transfer coefficient by increasing the thermal conductivity of a coolant.

1.8.3 Industrial cooling systems

Great energy savings and emissions reductions can be achieved by replacing the conventional heat transfer fluids by nanofluids in various industrial cooling management systems. Addition of conductive particles greatly enhances the thermal conductivity of metal nanofluids.

1.8.4 Building heating systems

Nanofluids also can be used in the building heating systems. Larger heating system can be replaced by smaller with the help of nanofluids. Consequently, since smaller heating system requires less amount of power, environmental pollution can be reduced to a large extent.

1.8.5 Space and defence applications

Weight and space are always the limiting parameters in space and aircraft systems. Hence, highly efficient and compact cooling systems is a vital need of space cooling management. Nanofluids have the potential to provide required cooling in such applications as well as in other military systems, including military vehicles, submarines, and high-power laser diodes. Therefore, nanofluids have wide application in space and defence fields where power density is very high and the components should be smaller and weigh less.

1.8.6 Miscellaneous

Along with the major applications above discussed , nanofluids have energy applications viz. energy storage and solar absorption systems; mechanical applications viz. friction reduction and magnetic sealing; and numerous biomedical applications like antibacterial activity, nano drug delivery, cancer therapeutics. nano cryo-surgery. sensing and imaging, cryo-preservation nanofluid detergent, intensify micro reactors, nanofluids as vehicular brake fluids, nanofluids based microbial fuel cell, nanofluids as optical filters etc.

1.9 Challenges of nanofluids

Due to significantly improved thermal properties of nanofluid over conventional fluids, attention of researchers has been attracted to this field. Although, the thermal conductivity of base fluids enhances greatly with the addition of nanoparticles ,but the practical application of the field is hindered by:

i. Lack of agreement of results obtained by different researchers.
ii. Poor characterization of suspensions.
iii. Lack of theoretical understanding of the mechanisms responsible for changes in properties.
iv. Long term stability of nanofluid is a challenging task. Due to strong Van Der Waals forces between molecules, nanoparticles start to agglomerate when dispersed in the base fluid. Some physical or chemical treatments are used to get stable nanofluids, e.g., ultra sonication, addition of surfactants etc.
v. High cost of manufacture of nanoparticles.
vi. Increased pumping power requirements because of enhanced viscosity of nanofluids as compared to base fluids and with increasing particle concentration.

1.10 Nanofluids & thermal science

Energy efficient heat transfer fluids are one of the vital need of today's thermal devices to achieve high thermal efficiency along with their compactness. Conventional heat transfer fluids like water, ethylene glycol, oils, etc., inherently possess lower thermal conductivity, which greatly restricts the development of highly efficient and compact cooling systems. Particle size and volume concentration greatly affects the thermal properties of nanofluids. With small concentration of nanoparticles, heat transfer rates of thermal systems can be increased significantly, which allows to meet the desired cooling needs without increasing the size of heat transfer device. Also, recent research in the area of nanofluids has revealed that the thermal conductivity of nanofluids significantly increases with the temperature of the fluids, which makes the use of nanofluid in high energy density applications, feasible. For example, in automobile cooling system, the temperature of heat transfer fluids reaches a high value; therefore if conventional fluid is replaced with the nanofluid, its increased thermal conductivity at higher temperature results in greater heat transfer rates. Consequently for same heating load, the size and weight of the cooling system can be reduced significantly which is a major need in space and aircraft applications as well.

1.11 Objectives of present work

Miniaturization of thermal systems and Ultra-high performance is one of the crucial necessity for today's developing industrial, domestic and automobile cooling systems. Nanofluids appear as a panacea for thermal equipments, which are suspensions of nano sized particles (typically 1-100 nm) in host fluids provided by nanotechnology. Following are the objectives of the present study:

1. To study the effect of Aluminium/water Nanofluids on thermo-hydraulic performance of a single-pass cross-flow compact heat exchanger.
2. To study the effect of nanofluid concentration, hot and cold fluid Reynolds number, particle volume concentration and inlet temperature of the hot fluid on the performance parameters such as Nusselt number, Colburn factor and friction factor on both sides of tubes of the heat exchanger.

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CHAPTER 2

LITERATURE REVIEW

Numerous researchers have done a lot of research on nanofluid technology and its applications in the heat transfer devices. This chapter reviews the previously published literatures in this area, which lays foundation for further work and investigation. This helps to give a better understanding of the topic and also acts as a guideline for the present work. This section deals with the literature review on nanofluids, its various properties and the effect of various nanofluids on the thermo hydraulic performance of heat exchangers.

LotfizadehDehkordi et al. [1] carried out the experiments by taking 60:40 mass ratio of ethylene glycol and water. They used the surfactant SDBS and studied the effect of its concentration on viscosity and thermal conductivity of Al_2O_3 nanofluids. Lower concentrations provided better dispersion and thermal conductivity was enhanced, but higher concentrations of surfactant resulted in the drop in thermal conductivity of nanofluid and hasten the viscosity increase of nanofluid. Results revealed that the viscosity of nanofluid was strongly dependent on fluid temperature. Thermal conductivity was increased with nanoparticle volume concentration and temperature of the fluid.

Choi et al. [2] revealed that in the advancement of energy efficient fluids, thermal conductivity plays a vital role. They conducted experiments on Cu/water and Alumina/water nanofluids and showed that the thermal conductivity of nanofluids was significantly higher than that of base fluid. They also concluded that the rate of heat transfer could be increased by factor of 2 without significant increase in the pumping power required for the circulation of fluid through heat exchanger.

Xuan et al. [3] studied the effect of Cu/water nanofluids on heat transfer and fluid flow characteristics in a tube. They concluded that for the same Reynolds number, the rate of heat transfer was significantly enhanced with the suspension of nanoparticles into base fluid. Experiments were conducted in turbulent flow region. They proposed a convective heat transfer correlation for the flow of nanofluid in a tube, given as

$$Nu_{nf} = c_1(1.0 + c_2\emptyset^{m1}{Re_d}^{m2}){Re_{nf}}^{m3}{Pr_{nf}}^{0.4} \tag{2.1}$$

They also concluded that for low volume fraction of Copper nanoparticles suspended in the base fluid, friction factor increment was negligible.

Xuan et al. [4] measured the thermal conductivity of Cu/water nanofluids with hot wire method. They studied the effect of various parameters such as particle volume fraction, size and properties of nanoparticles on the thermal conductivity and revealed that thermal conductivity was highly dependent on these parameters. They concluded that for 2.5 % to 7.5 % nanoparticle volume fraction, the thermal conductivity was increased by factor of 1.24 to 1.78.

Kakaç et al. [5] reviewed that heat transfer capabilities of ordinary fluids such as water, oils and ethylene glycol can be increased significantly by addition of nanoparticles. They marked the importance of heat transfer fundamentals for a diverse advancement in the field of nanotechnology. Theoretical and experimental understanding of microscopic particle mechanism is vital.

Vajjha et al. [6] studied the effect of Al_2O_3 and CuO based nanofluids on the performance of an automobile radiator. Base fluid used was the mixture of water and ethylene glycol. Radiator under consideration was employed with flat tubes. Experiments were carried out in the laminar flow region. They concluded that the average heat transfer coefficient was increased considerably with particle volume concentrations. They showed that for 10 % Al_2O_3 nanofluid, the average heat transfer coefficient was improved by 94 %, while for 6 % CuO nanofluid, it was increased by 89 %. Also, for a fixed inlet velocity, average skin friction coefficient was increased by increasing the particle volume concentration. But, for the same amount of heat transfer, pumping power requirement with respect to base fluid was reduced by 82 % and 77 % for Al_2O_3 and CuO nanofluid, respectively.

Peyghambarzadeh et al. [7] studied the effect of Al_2O_3/water nanofluid on the cooling performance of an automobile radiator. Five different concentrations varying from 0.1 to 1 % (vol.) of Al_2O_3/water nanofluids were taken. Flow rate of fluid inside the tubes were changed from 2 to 5 litre per minute. Experiments were carried out in fully developed turbulent region. Inlet temperature of fluid through the tubes was varied from 37 °C to 49 °C. They concluded that the heat transfer performance of the heat exchanger was improved by increasing the flow rate of fluid flowing through the tubes. With respect to pure water, heat transfer was enhanced by 45 % by adding Al_2O_3 nanoparticles. By increasing the Reynolds number of working fluid, effective thermal conductivity was increased by 3%.

Naraki et al. [8] deliberated the effect of CuO/water nanofluids on the overall heat transfer coefficient of a car engine cooling system. Experiments were carried out in laminar flow regime. Two methods were employed to obtain the more stabilized nanofluids i.e. adjustment of pH value and addition of suitable surfactant. They concluded that highly stable and negligibly agglomerated nanofluids were prepared with pH value of 10.1 and with addition of SDS (Sodium Dodecyl sulfonate) as surfactant. Nanoparticle volume concentration was varied from 0.15 to 0.4 % (wt.). Nanofluid inlet temperature was taken 50 °C to 80 °C. They observed that, the overall heat transfer coefficient was improved by 6 % and 8% for 0.15 and 0.4 %(wt.) particle volume concentration, respectively. Reynolds number of air circulating over the tubes contributed to the overall heat transfer coefficient by 42 %. The overall heat transfer coefficient was also effect by nanofluid volumetric flow rate, inlet nanofluid temperature and particle volume concentration with contribution of 23 %, 22 % and 13 %, respectively.

Leong et al. [9] investigated the performance of an automotive car radiator by using Cu/EG nanofluids as coolant. Results were compared by taking Reynolds number of air and coolant as 6000 and 5000, respectively. They found that heat transfer rate was increased by 3.8 % by adding 2 % of Cu nanoparticles. Thermal performance of heat exchanger was found highly dependent on air and coolant Reynolds number. An increment of 42.7 % and 45.2 % was observed when air's Reynolds number was increased from 4000 to 6000 for ethylene glycol and Cu/EG nanofluid, respectively. While, thermal performance was increased by only 0.9 % and 0.4 % when coolant Reynolds number was increased from 5000 to 7000 for ethylene glycol and Cu/EG nanofluid, respectively. They observed that frontal area of heat exchanger was reduced by 18.7 % by adding 2% of Cu nanoparticles into the base fluid. Pumping power for nanofluid was found 12.13 % higher than that with pure ethylene glycol, while keeping volumetric flow rate of nanofluid constant to 0.2 m^3/s.

Ali et al. [10] studied the effect of particle volume concentration and particle material on the thermal conductivity and thermal diffusivity of nanofluids. Nanofluids were prepared using Al and Al_2O_3 nanoparticles, while three different base fluid namely, distilled water, ethylene glycol and ethanol were used to prepare nanofluids. Thermal conductivity and thermal diffusivity of various nanofluids were measured by using hot wire laser beam displacement method. Results showed that thermal properties vary linearly with particle volume concentration. They also revealed that metallic nanofluids always provide greater enhancement in thermal properties as compared to non-metallic nanofluids. They experimentally concluded that for a particle volume concentration of 0.42 % thermal conductivity of Al/water, Al/EG and Al/ethanol nanofluids was higher than that of base fluid by 18.63 %, 20.5 % and 24.27%, respectively. While at same concentration for Al_2O_3/water, Al_2O_3/EG and Al_2O_3/ethanol, it was increased by 9.56 %, 12.1 % and 15.1 %, respectively.

Huminic et al. [11] reviewed the various articles presenting the effect of nanofluids on the heat transfer characteristics of heat exchangers. They summarized the previous researches on nanofluids in two parts, in first part they focussed on the effect of nanoparticles on the thermo-physical properties of base fluids and heat transfer performance of thermal systems and in second part, they paid attention on the application of nanofluids in numerous heat exchanger configurations such as double pipe heat exchanger, cross flow compact heat exchanger, shell and tube heat exchanger , plate heat exchanger etc. They concluded that most of the experimental and numerical studies revealed that nanofluids display an improved heat transfer coefficient compared to its host fluid and it increases considerably with rising nanoparticle concentration as well as Reynolds number of hot and cold fluids. They observed that the augmentation of the heat transfer potential of nanofluids makes their utilization in heat exchangers an attractive alternative, leading to improved system performance.

Nieh et al. [12] employed Al_2O_3/water and TiO_2/water nanofluids in air cooled radiator to improve the performance. Thermo-physical properties of nanofluids were measured at different nanoparticle volume concentration and then pressure drop and heat dissipation rate were measured at different Reynolds number. Efficiency factor and heat dissipation rate was greater for nanofluids as compared to that with ethylene glycol/water solution. They concluded that the TiO_2/water nanofluids showed the greater enhancement than Al_2O_3/water nanofluids. Heat dissipation rate was enhanced by 25.6% , 6.1% improvement for pressure drop was seen , pumping power was increased by 2.5 % and efficiency factor has 27.2% enhancement as compared to ethylene glycol/water mixture.

Hong et al. [13] did experiments on Fe/ ethylene glycol and studied the effect of clustering of nanoparticles. They observed that as the time of sonication increases, thermal conductivity enhancement increases. Nanoparticles at high concentration easily agglomerate because distance is between the particles are reduced.

Murshed et al. [14] studied the effect of TiO_2 nanoparticles volume concentration and particle shape on the thermal conductivity of nanofluids. Two different shaped nanoparticles viz. cylindrical ($\emptyset\, 10\, nm \times 40\, nm$) and spherical ($\emptyset\, 15\, nm$) were used. De-ionised water was used as base fluid for nanofluid preparation. Thermal conductivity of different nanofluids was measured using transient hot wire approach. pH value and viscosity of nanofluid was also examined. They concluded that the particle shaped and size greatly effects the thermal conductivity enhancement. Nanofluid prepared with cylindrical shaped particles provided 33 % thermal conductivity enhancement while for spherical shaped it was neatly 30 % over base fluid [11]. Two different surfactants (CTAB and Oleic acid) were used to ensure better stability of nanofluids without affecting the thermo-physical properties of nanofluids. CTAB was found to be more effective. They concluded that adding a small amount of nanoparticles into the base fluid significantly increased its thermal conductivity. The thermal conductivity of nanofluids increased surprisingly with increasing volume concentrations of nanoparticles. Thermal conductivity augmentation of nanofluids was also influenced by particle size and shape of nanoparticles.

Das et al. [15] studied the effect of temperature on thermal conductivity enhancement of nanofluids. By using water as base fluid, two different nanofluid were prepared of Alumina and copper oxide nanoparticles. Thermal conductivity and thermal diffusivity of nanofluids were measured by using temperature oscillation method. They observed that thermal conductivity of nanofluids was enhanced by two to four times when its temperature increased from 21°C to 51°C. This observation makes the use of nanofluids feasible in the cooling application where heat transfer fluids reaches a very high temperature. It was observed that nanofluids containing smaller CuO particles demonstrate extra enhancement of conductivity with temperature.

Eastman et al. [16] revealed the effect of Cu/ethylene glycol nanofluids on the thermal conductivity. They concluded that Cu/ethylene glycol nanofluids provides higher thermal conductivity as compared to pure ethylene glycol or other oxides/ethylene glycol nanofluids. It was shown that thermal conductivity of nanofluids containing 3 % concentrations of Cu nanoparticles of particle size 10 nm was higher by 40 % as compared to base fluid. They concluded that metallic nanofluids exhibits higher thermal conductivity enhancement as compared to non-metallic nanofluids.

Nassan et al. [17] compared the effect of two different nanofluids such as Al_2O_3/water and CuO/water nanofluids on heat transfer characteristics of a square cross sectional duct of 100 cm length. Experiments were conducted in the laminar flow region and in uniform heat flux conditions. They concluded that a considerable increment in the heat transfer rate was found for both of the nanofluids. They also showed that CuO/water nanofluids provide greater enhancement as compared to Al_2O_3/water nanofluids.

Heris et al. [18] investigated the heat transfer characteristics of a circular tube while using Al_2O_3/water nanofluids as heat transfer fluid. Experiments were carried out at constant wall temperature boundary condition and in laminar flow region. Variation of Nusselt number with Reynolds number and Peclet number was investigated. Results showed that heat transfer coefficient increases with particle volume concentration and Peclet number. They concluded that along with increased thermal conductivity, Brownian motion of particles, chaotic movement and dispersion of particles plays a vital role in the heat transfer enhancement.

Wen et al. [19] studied the convective heat transfer of nanofluids in circular tube made of cooper material. They prepared nanofluids by using γ-Al_2O_3 nanoparticles and de-ionised water. Experiments were conducted in laminar flow region. They observed that the heat transfer enhancement was significant in entrance region of the tube which was mainly due to increased thermal conductivity of base fluid by adding nanoparticles. They revealed that the boundary layer growth in pipe flows influenced by presence of nanoparticles and heat transfer rate increases. They concluded that particle migration is also one of the factors responsible for heat transfer enhancement.

Huminic et al. [20] examined the cooling performance in the flat tubes of an automobile radiator. They used the Copper/ethylene glycol nanofluids as coolant. Experiments were conducted in laminar flow region with varying the Reynolds number, inlet temperature and particle volume concentration of nanofluids. Reynolds number varied from 10-125, inlet temperature from 348-368 K and particle volume concentration 1 % and 2%. They concluded that heat transfer coefficient of nanofluid is directly proportional to the Reynolds number and particle volume fraction. They showed that heat transfer coefficient was significantly higher than that of base fluid.

Chougule et al. [21] conducted the experiments on car radiator by using CNT/water and Al_2O_3/water nanofluids as coolant. Experiments were carried out with four different particle volume fractions varying from 0.15 to 1 %. Flow rate was regulated between 2 to 5 litre per minute. They studied the forced convective heat transfer performance and results revealed that with 1% particle volume concentration maximum heat transfer was enhanced by 90.76% and 52.03% for CNT/water and Al_2O_3/water nanofluids, respectively. As the nanofluids' Reynolds number increased, heat transfer performance was increased for both of nanofluids . CNT/water nanofluids provided huge enhancement as compared to Al_2O_3/waters nanofluids because of high aspect ratio, high thermal conductivity, and low thermal resistance of CNT. As the nanoparticle volume fraction was increased, thermal conductivity was also increased and consequently the cooling performance of radiator was also increased.

Chavan et al. [22] performed the experiments on automobile radiator to study the heat transfer characteristics by using Al_2O_3/water nanofluids as coolant. Pure water was used as base fluid to prepare nanofluids and the effect of nanofluids on the performance of heat exchanger was compared with the performance achieved with pure water. Flow rate of coolant was varied from 3 to 8 litres per minute and which provided the turbulent flow region for the experiments. Five different nanoparticles volume concentrations were used ranging from 0-1 %. They concluded that heat transfer performance was enhanced by increasing the Reynolds number of nanofluids. 40-45% improvement in heat transfer performance was observed with 1% nanoparticle volume concentration as compared to pure water. Brownian motion becomes more evident with decreasing particle size and consequently the heat transfer rate increased.

Hussein et al. [23] carried out the experiments to study the effect of tube's cross section on heat transfer performance of a car radiator to advance the radiator's heat transfer characteristics. Fluent software was used to developed CFD by using the finite volume technique. For experimentation TiO2/water nanofluids were prepared at 1 %, 1.5 %, 2 % and 2.5% nanoparticle volume concentration. Circular, elliptical and flat tube geometries of tube were considered having length 500 mm and hydraulic diameter of

3mm. Results revealed that for circular cross section given the higher values of friction factor as compared to elliptical and flat tube. With increasing value of Reynolds number, friction factor decreased. Heat transfer coefficient was found highest for flat tube cross configuration of tube because of more area of heat transfer.

Mohammed et al. [24] reviewed the effect of nanofluids on the heat transfer characteristics of micro channel heat exchanger. They reported that heat transfer rate can be increased significantly at the cost of increased friction factor. They recommended to understand the heat transfer phenomenon related to nanofluids more deliberately and proper study of nanofluid preparation techniques.

Garg et al. [25] investigated the thermal conductivity and viscosity of ethylene glycol based copper nanofluids. By using water as solvent, they prepared copper nanofluid with the help of chemical reaction method and then dispersed it into ethylene glycol by using sonication. They prepared nanofluid without adding any surfactant. Particle volume concentration was varied from 0.4 to 2%. Transient hot wire method was used to measure thermal conductivity of nanofluids of different concentrations. They concluded that because of higher increment in viscosity as compared to thermal conductivity, the nanofluids are not suitable in the existing thermal system. However, the advantages of increased thermal conductivity could be beneficial by increasing the tube diameter in the application where the size of thermal equipments is of lesser importance.

Amani et al. [26] studied the convective heat transfer and pressure drop characteristics of TiO_2/water nanofluids. Nanofluids were prepared using TiO_2 nanoparticle of 30 nm particle size by dispersing them in de-ionised water. The value of Reynolds number was varied from 8000 to 51000. Nusselt number was increased by increasing the Reynolds number. Unfortunately higher values of Reynolds number tends to increase the pumping power. Experimental results showed that for a given Reynolds number, the value of Nusselt number increased with particle volume concentration.

Bozorgan et al. [27] studied the effect of Al_2O_3/water nanofluids on the performance of an automotive diesel engine cooling system. Nanofluids were prepared by using γ-Al_2O_3 nanoparticles of 20 nm mean size and dispersing them in water. The nanofluids were used as coolant in automotive diesel engine radiator. Overall heat transfer enhancement was investigated at different particle volume concentrations in turbulent flow region. Results showed that while keeping the particle volume concentration as constant, the pumping power was decreased with the vehicle speed. Concentration of particles increased the viscosity and density of nanofluids which consequently increased the friction factor.

Davarnejad et al. [28] performed numerical and experimental study to investigate the heat transfer and fluid flow characteristics of horizontal tube. Working fluid used was the suspension of TiO_2 nanoparticles into distilled water. TiO_2/water nanofluids were prepared at different concentrations varying from 0.002 to 0.02 % by volume. Experiments were carried out in turbulent flow region with Reynolds number ranging from 8000 to 51000. Numerical results were validated by comparing with experimental results and found a good agreement in both results. They showed that Nusselt number, which is a dimensionless heat transfer coefficient, was increased with particle volume concentration and Reynolds number, but pressure drop also increased with increasing volume concentrations. Finally they concluded that heat transfer performance can be increased significantly at the cost of increased pumping power.

Vermahmoudi et al. [29] carried out the experiments under laminar flow region by using Fe_2O_3/water nanofluid with particle volume concentrations varying from 0.15 to 0.65% . Nanofluids flow rate was varied from 0.2 to 0.5 m3/ hr and inlet temperature of nanofluid was varied from 50^0C to 80^0C. Results showed that overall heat transfer coefficient increased with Reynolds number and nanoparticles volume concentrations. With increasing the air side's Reynolds number from 500 to 700, the overall heat transfer coefficient increased. As the inlet temperature of nanofluid increased, the overall heat transfer coefficient decreased because of increased log mean temperature difference. Enhancement of 13% was achieved with 0.65% particle volume concentration over base fluid.

Heris et al. [30] experimentally studied the effect of nanofluids on the heat transfer performance of a car radiator. Nanofluids were prepared by using CuO nanoparticles of APS 60 nm. Base fluid used was the mixture of ethylene glycol and distilled water. Particle volume concentration was varied from 0.05 to 0.8 %. Flow rate of nanofluid was regulated from 4 to 8 litres per minute. Three different nanofluid inlet temperatures were used i.e. 35, 44 and 54 °C. Results showed that heat transfer rate was increased considerably as compared to base fluid. It was observed that Nusselt number increases with flow rate of nanofluids and particle volume concentration. With a particle volume concentration of 0.8 %, heat transfer coefficient was improved by 55 % over base fluid.

Sheikhzadeh et al. [31] analyzed the thermal performance of a car radiator while using copper/ethylene glycol as coolant. It was found that the overall heat transfer coefficient of air side was increased considerably by increasing Reynolds number and particle volume fraction of nanofluids, consequently the heat transfer rate was increased. They observed that when particle volume concentration increased from 0 to 5 %, overall heat transfer coefficient and heat transfer rate were increased by 64.3 % and 26.9 %, respectively. They also found that when Reynolds number increased from 4000 to 6000, overall heat transfer coefficient of air and nanofluid were increased by 4.5 % and 12.4 %, respectively. They concluded that heat transfer performance of radiator was better in hot weather of 50°C as compared to weather of 20°C.

Bozorgan et al. [32] investigated the heat transfer and fluid flow characteristics of an automobile diesel engine radiator using CuO/water nanofluids. Nanofluids were prepared using CuO nanoparticles of particle size 20 nm. Experiments were carried out in turbulent flow region with two different particle volume concentrations viz. 0.1 % and 0.2 %. Effect of vehicle speed and nanofluid flow rate on thermal performance of radiator was studied. Results showed that at particle volume concentration of 2 %, Reynolds number of nanofluid equals to 6000 and automotive speed of 70 km per hour, the overall heat transfer coefficient and pumping power increased by 10 % and 23.8 %, respectively. It was observed that the overall heat transfer coefficient of nanofluid was higher than that of base fluid, hence the total heat transfer area of the radiator can be reduced. However, the significant increase in required pumping power may enforce some restrictions on the proficient utilization of CuO/water nanofluids in automotive diesel engine radiators.

Ali et al. [33] studied the effect of Alumina/water nanofluids on the thermal performance of cooling system of an automobile radiator. Al_2O_3/water nanofluids were prepared at five different concentrations viz. 0.1, 0.5, 1, 1.5 and 2 % by volume. Gradual enhancement in heat transfer was observed with particle volume concentration of 0.1, 0.5 and 1.0 % and was optimum at 1.0 % while it declined with further increment in particle volume concentration. The maximum percentage increase of the heat transfer rate, heat transfer coefficient, and Nusselt number of nanofluid was found 14.79, 14.72, and 9.51, respectively, which take place at maximum load of 1 KW and at particle volume concentration of 0.01.

Abdulhassan et al. [34] studied the heat transfer and fluid flow characteristics of a horizontal circular tube. Nanofluids were prepared by using three different nano materials viz. Al, Al_2O_3 and CuO with particle size of 25, 30 and 50 nm, respectively. Experiments were carried out with uniform heat flux boundary condition and in fully developed laminar flow conditions. Three nanofluids were prepared of different particle concentrations varying from 0.25 to 2.5 %. The rate of heat transfer was increased by 45%, 32 % and 25 % for Al, Al_2O_3 and CuO nanofluids, respectively with insulated circular tube, while without insulation, increment was 36 % , 23 % and 19 % , respectively. They concluded that the type of nanofluid plays vital role in heat transfer enhancement. The higher values were obtained when using Al, Al_2O_3 and CuO nanoparticles, respectively.

Jalal et al. [35] conducted the experiments to study the effect of CuO/water nanofluids on convective heat transfer performance of a heat sink. Four different nanoparticle volume concentrations i.e. 3.5, 4, 4.5 and 5 % were used. Experimental results verified that the overall heat transfer coefficient improved and thermal resistance of the heat sink declined. They concluded that increasing the particle volume concentration results in an increment in the heat transfer coefficient.

Nikkam et al. [36] investigated the thermal conductivity and rheological characteristics of CuO/diethylene glycol nanofluids. The physicochemical properties including thermal conductivity and viscosity of nanofluids were measured for the nanofluids with nanoparticle volume concentrations varying from 0.4 to 1.6 % (wt.) in the temperature range of 20 to 50ºC. Experimental results were compared with suitable theoretical models. Results showed that the increment in thermal conductivity was higher than that in viscosity, making them suitable heat transfer fluids for thermal systems.

Chougule et al. [37] studied the effect of CNT/water nanofluids on the heat transfer and hydraulic performance of automotive radiator. CNT/water nanofluid were prepared using different nanoparticle volume concentrations ranging from 0.15 to 1 %. Nanofluid were prepared using surface treatment (SCNT) and functionlization (FCNT) methods. Variation of Nusselt number with various parameters such as nanofluid preparation method, pH value and nanoparticle volume concentration was examined through experiments. It was observed that the thermal performance of automobile radiator was improved with the aid of nanofluids compared to water. It was found that for 1.0 % particle volume concentration and nanofluid flow rate of 5 litres per minute, the maximum improvement in heat transfer of FCNT/water nanofluid was found to be 90.76% higher as compared with water. The heat transfer performance SCNT/water and FCNT/water nanofluid increased with the increase in nanofluid flow rate.

Ebrahimi et al. [38] studied the forced convection heat transfer characteristics of car radiator using SiO_2/water nanofluids as coolant. The heat exchanger employed in the cooling system was compact heat exchanger. The effect of various parameters such as nanofluids flow rate, inlet temperature of nanofluids and particle volume concentration was studied by experiments. They found that Nusselt number increased with increasing inlet temperature, Reynolds number and particle volume concentration of nanofluids. Inlet temperature was varied by 43°C, 52°C and 60°C. They concluded that for a particle volume concentration of 0.04 %, heat transfer rate was increased by 3.8 % as compared to base fluid.

Elias et al. [39] experimentally investigated the thermo-hydraulic performance of car coolant system using nanofluids as coolant. Nanofluids were prepared using Al_2O_3 nanoparticles and base fluid as a mixture of water and ethylene glycol. Two step method was used to prepare nanofluids. Various thermo-physical properties of nanofluids such as density, viscosity, thermal conductivity and specific heat were measured at different temperatures ranging from 10°C to 50°C. Different volume concentrations of nanoparticles were used varying from 0 to 1 %. Results showed that density, viscosity and thermal conductivity were enhanced with particle volume concentration while specific heat of nanofluids was decreased. With increasing temperature, thermal conductivity and specific heat were increased while density and viscosity were decreased. Enhancement in average thermal conductivity was observed 3.26 % and 8.30 % with temperature and particle volume concentration, respectively.

Heyhat et al. [40] studied the effect of water based alumina nanofluids on heat transfer and pressure drop characteristics of laminar flow through horizontal tube. Experiments were carried out considering constant wall temperature boundary condition under fully developed region. Nanofluids were prepared by dispersing Al_2O_3 nanoparticles of 40 nm size into double distilled water. Experimentation was done by varying flow rate of nanofluids and particle volume concentration ranging from 0.1 to 2 %. Results showed that heat transfer rate was enhanced by nearly 32 % with 2 % concentration of nanoparticles. The maximum pressure drop was found about 5.7 times elevated than that of base fluid which was occurred in the highest volume fraction of nanofluid (2%) at Reynolds number of 360.

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CHAPTER 3

EXPERIMENTATION & CALCULATIONS

Literature reveals that the performance of heat exchangers can be improved by using the nanofluids as heat transfer fluids. The thermal conductivity of nanofluids is higher than that of usual heat transfer fluids like water, oil, ethylene glycol etc. Since the thermal conductivity of metals is much greater than that of non-metals and hence the thermal conductivity of fluids that contain suspended solid metallic particles is higher than those of conventional heat transfer fluids. So far, lot of research has been done where oxides of metals were used to prepare advanced heat transfer fluids in cooling applications. In presented work, metal basis aluminium nanoparticles were used to prepare the Al/water nanofluids and its effect on the thermo-hydraulic performance of a single-pass cross-flow compact heat Exchanger was investigated. Photographic view of experimental setup prepared for experimentation is shown below in Figure 3.1.

Figure 3.1: Photographic view of experimental setup

3.1 Layout of experimental setup

Experimental setup was fabricated to investigate the effect of Al/water nanofluids on the performance of a single-pass cross-flow compact heat exchanger. Experiments were carried out in the laminar flow region. The test setup was fabricated at Fluid Machinery Lab., Thapar University, Patiala, India. Experimental setup was consists of a cross flow compact heat exchanger with single pass of tubes, flow lines, forced draft fan, duct, fan speed regulator, temperature sensors, temperature display, rotameter, U-tube manometer, PID temperature controller, storage chamber, two centrifugal pumps, by-pass valve and heating element. Figure 3.2 shows the layout of fabricated experimental setup. Flow rate through the setup was regulated by gate valve provided at the bottom of rotameter. Inlet temperature of the hot fluid was regulated with the help of PID controller. Experiments were conducted at three different inlet temperatures of hot fluid i.e. 45°C, 50°C and 55°C. Five different flow rates of hot fluid were taken viz. 50, 60, 75,90 and 100LPH. Velocity of cold air circulating over heat exchanger was varied by five different values i.e. 3.38, 4.72, 5.88, 6.14 and 6.44m/s. To study the enhancement in the thermo-hydraulic performance of heat exchanger, experiments were first carried out by using distilled water as working medium and then Al/water nanofluids of 0.1% and 0.2% concentration by volume were used. Temperature sensors were located at different positions of the heat exchanger to measure the temperature of hot and cold fluid streams.

3.2 Requisites of the experimentation

Following were the major requirements for performing experiments on test rig:

1. 220V, 50 Hz, 3 Phase AC power supply with voltage regulation.
2. PID Controller to control the temperature of hot fluid.
3. Two centrifugal pumps to circulate the hot fluid through experimental setup.
4. Digital temperature display to show the temperature readings.

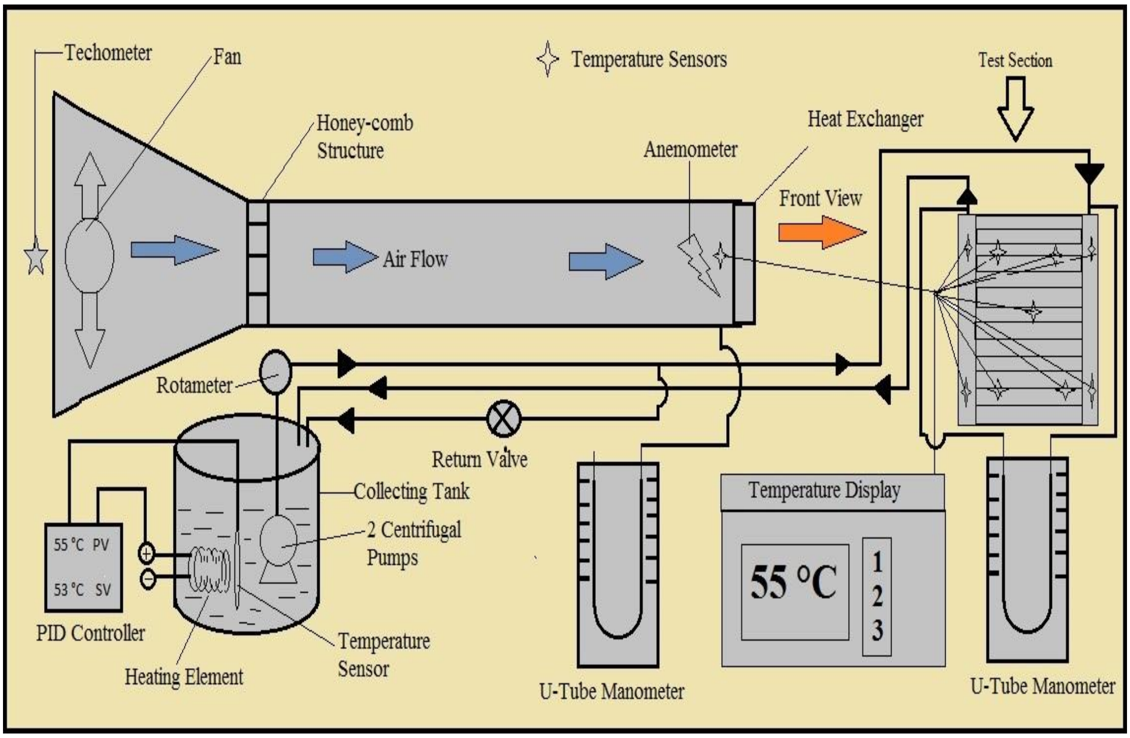

Figure 3.2: Layout of Experimental setup

3.3 Specification and function of various components of experimental setup

Various components associated with the experimental study along with their functions are described below.

3.3.1 Cross flow compact heat exchanger

To study the effect of Al/water nanofluid on the thermo-hydraulic performance of a cross flow heat exchanger, a model of single- pass cross- flow compact heat exchanger was purchased from B K Radiator Works, Patiala, India, which is generally employed as a heater core in TOYOTA HIACE, as shown in Figure 3.3. Two numbers of heat exchangers were purchased, one was used to measure all dimensions and other was used for experimentation. Geometrical specifications of the heat exchanger model used are tabulated below in Table 3.1.

Table 3.1: Geometrical specifications of heat exchanger

Configuration	Cross-flow Compact type
Number of passes	1
Number of tubes	38
Type of tubes	Flat tubes with semi circular ends
Fins	Multi-louvered fins
Core height	154 mm
Core length	193.5 mm
Core width	22 mm
Number of fins per tube	70
Area density	839 m^2/m^3

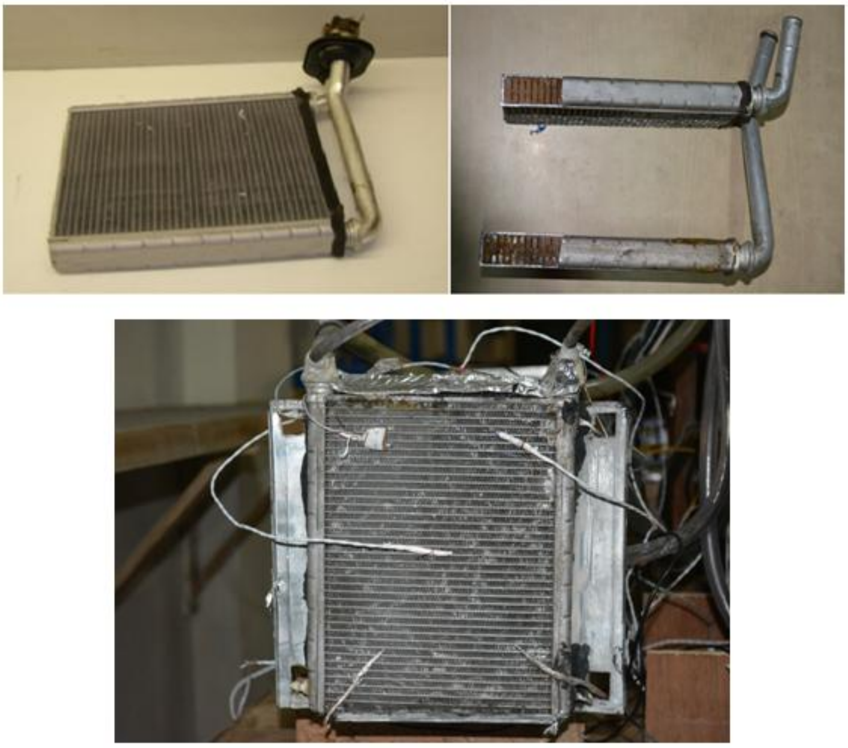

Figure 3.3: Cross-flow Compact Heat exchanger and it's cut section

3.3.2 Storage chamber fitted with heating element

Storage chamber of 5.5 litres capacity and made of stainless steel was used to store and circulate the hot fluid through the heat exchanger. A heating element of 3000 W power rating was fitted to the storage chamber to heat the fluid to a specified temperature. A temperature sensor was dipped into the chamber to measure the temperature of the liquid. Storage chamber and heating element is shown in Figure 3.4. and 3.5.

Figure 3.4: Storage chamber

Figure 3.5: Heating element

3.3.3 Centrifugal pump

Two numbers of centrifugal pumps were placed in the storage chamber, one was to make the circulation of hot fluid through the heat exchanger and another was to circulate the fluid within the storage tank, which helps to keep the temperature of the hot fluid uniform throughout the tank by diminishing the local heating effects of the fluid in the vicinity of heating element. Secondary pump was also beneficial because it helped to maintain the nanoparticles in suspension. Agglomeration of nanoparticles in the storage chamber is significantly eliminated with the aid of pump which acts as agitator in the fluid. Specifications and image of centrifugal pump used in the experiment is given in Table 3.2 and Figure 3.6 respectively.

Table 3.2: Specifications of the pump

Type	Centrifugal
Power rating	18W
Voltage	220V, AC, 50Hz
Size of outlet nozzle	0.5 inch
Maximum flow rate	750 LPH
Maximum head	1.55 m

Figure 3.6: Centrifugal pump

3.3.4 Duct fitted with forced draft fan

To make a uniform circulation of air over the heat exchanger, a duct of rectangular cross section was fabricated using galvanized iron sheets of 15 gauge thickness. A white stiff was filled in the joints to avoid the air leakage. A section of 0.25 m length of duct was convergent and then uniform rectangular cross section to 1.25 m length. A forced draft fan was fitted at the starting end of convergent section of the duct. Heat exchanger was fixed on the another side of the duct. A honey comb structure, visible in Figure 3.8, was placed in the duct at the place where converging section ends. It helped to obtain the uniform flow of air in the duct. Speed regulator was employed to the forced draft fan, which regulated the speed of the fan motor and consequently the flow rate of air passing over the heat exchanger was changed. Hence experiments were performed at different velocities of air over the heat exchanger.

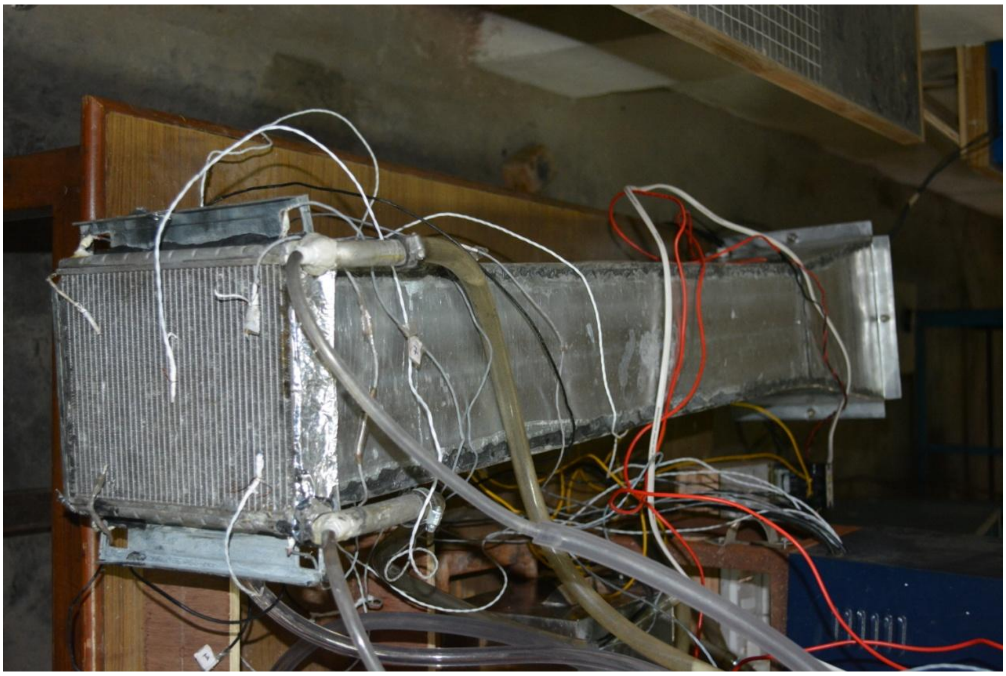

Figure 3.7: Duct made-up of GI Sheet

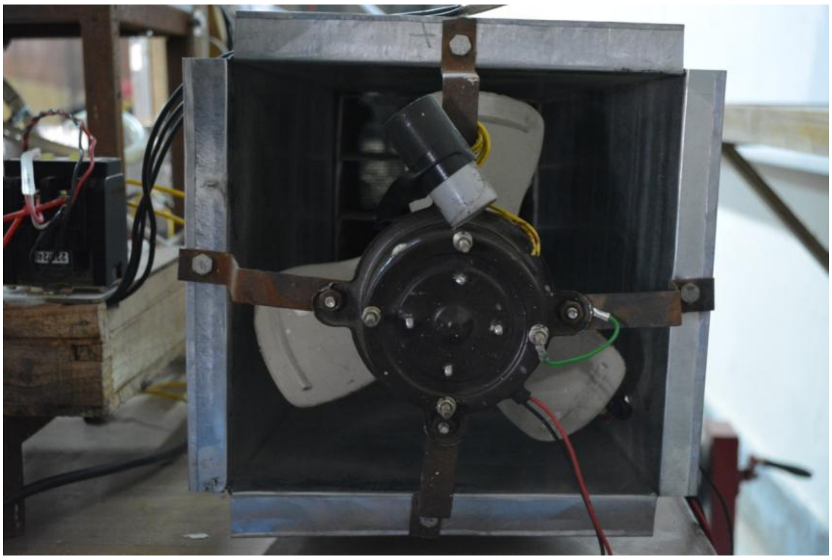

Figure 3.8: Forced draft fan with view of honey comb structure placed in the duct

3.3.5 Rotameter

Flow rate of hot fluid through the heat exchanger was measured with the help of rotameter. In this flow metering device, a float is guided in the conical shaped tube. A variable area flow meter has three basic elements namely, a float, a uniformly tapered tube and a measuring scale. Fluid enters from the bottom and float rises. The reading on the measuring scale corresponding to the float cap position gives the flow rate. Measuring scale gives the flow rate reading in generally in LPH. Rotameter used in this experimentation is shown in Figure 3.9.

Figure 3.9: Rotameter

3.3.5.1 Calibration of rotameter

Before performing the experiments, the rotameter was calibrated. A measuring beaker and a stop watch was used to calibrate the rotameter. Flow rate was fixed at different rates on the measuring scale and simultaneously the time was recorded with the help of stopwatch to fill the specified volume of the beaker. Reading were taken in time taken to fill 1000 ml fluid in the beaker and then converted into ml/min. flow rate was fixed at four different values on the measuring scale i.e. 50 LPH, 75 LPH, 100 LPH, 125 LPH, 150 LPH and corresponding flow rate with the help of stopwatch was recorded. Calibration data is given in annexure. Figure 3.10. shows the calibration graph.

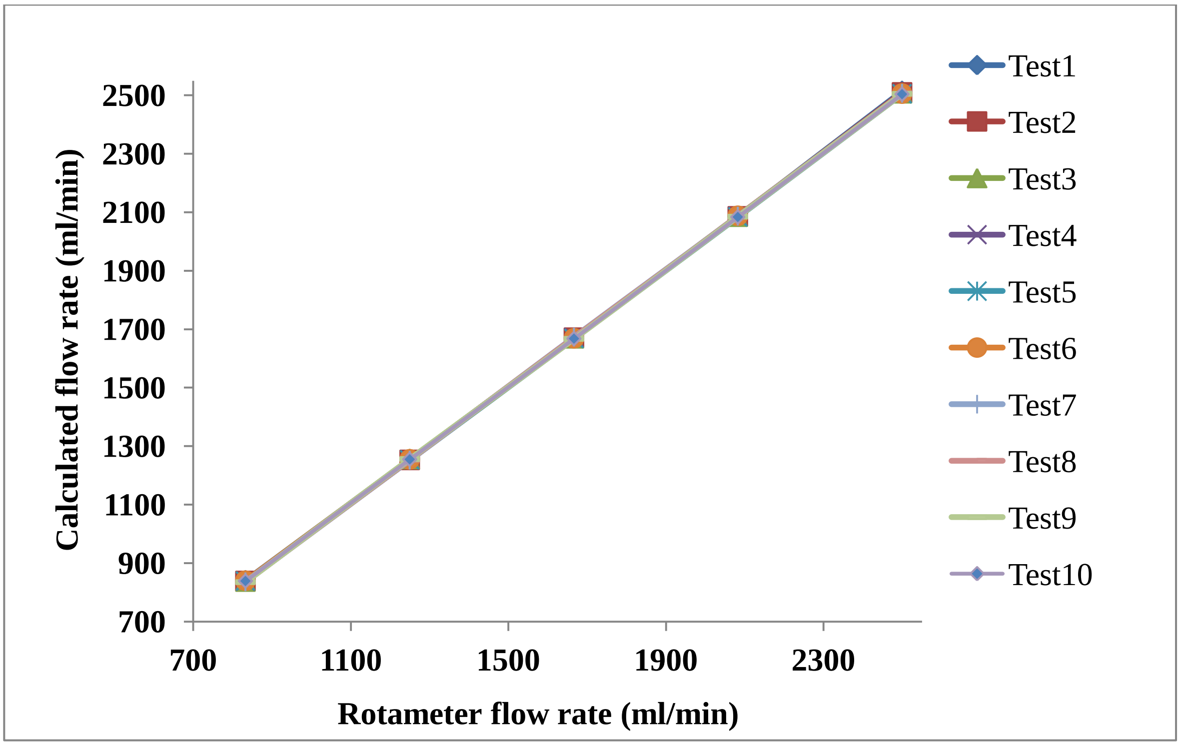

Figure 3.10: Rotameter calibration graph

3.3.6 U-tube manometer

Manometer of U-tube differential type (Figure 3.11) was used to measure the pressure drop of air when it flows across the heat exchanger core and of hot fluid when it passes through the tubes of the heat exchanger. For hot fluid side pressure drop measurements, two limbs were connected through tubes at inlet and exit of the hot fluids. The manometer liquid is decided on the basis of its density and immiscibility as compared to the density of the fluid whose pressure is to measure and by the pressure difference that is required to be measure. Small pressure differences are measured using water as the manometer liquid such that the height measured is satisfactorily large as to be approximated with adequate accuracy. Double distilled water was used in the manometer because it is highly sensitive for the small pressure drops. This manometer was used to measure pressure drop of air across the heat exchanger. The differential height of two water columns was recorded in mm of water. Similarly, another U-tube manometer with mercury as manometric fluid was used to measure the pressure drop of water through the heat exchanger.

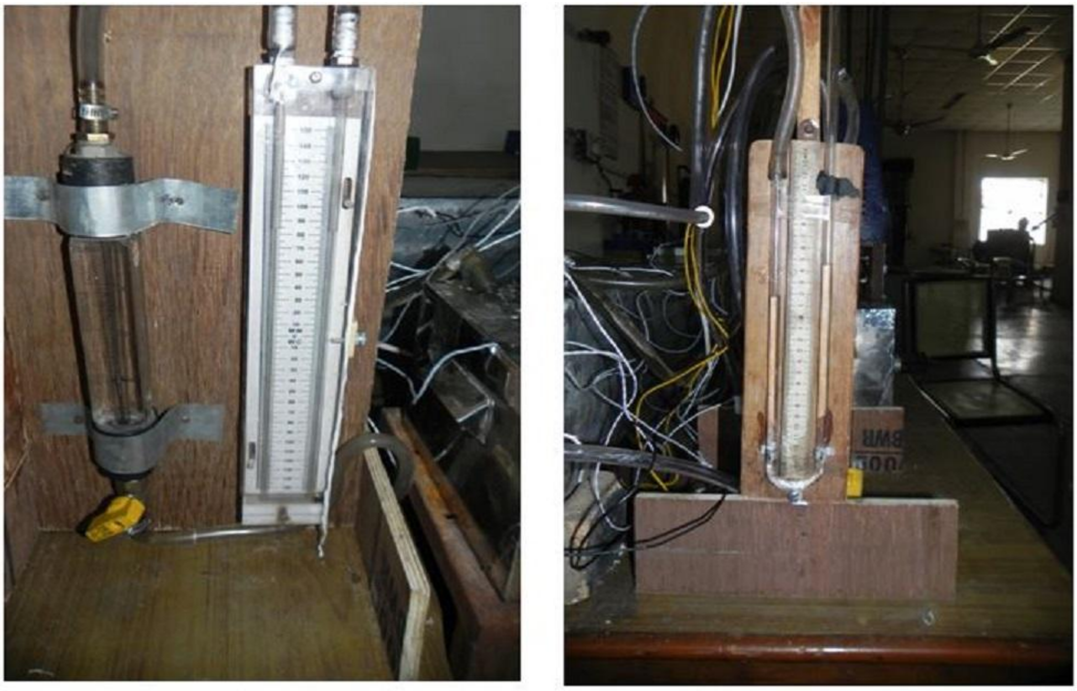

Figure 3.11: U-tube Manometer

3.3.7 PID temperature controller

To control the hot fluid inlet temperature, PID temperature controller Selec TC303 was used, as shown in Figure 3.13. PID controller works in a closed loop as shown by schematic diagram shown in Figure 3.12.

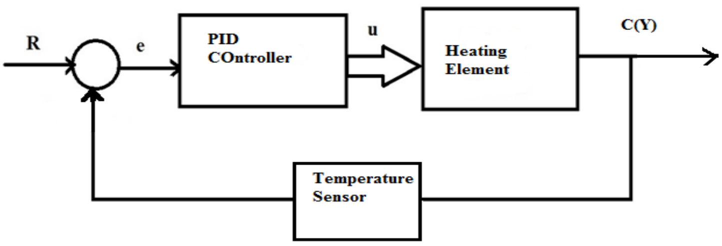

Figure 3.12: Schematic of PID controller working

The transfer function of the PID controller is written as,

$$Kp + \frac{K_I}{s} + K_D s$$

In above diagram, the variable (e) is the tracking error, which is the difference between the desired input value (R) and the actual output (Y). The error signal (e) is sent to the PID controller, and the controller computes both the derivative and the integral of this error signal. The signal (u) just past the controller is given by the following relation,

$$u = K_p e + K_I \int edt + K_D \frac{de}{dt}$$

This signal (u) is to the system, and the new output (Y) is obtained. This new output (Y) will be sent back to the sensor again to find the new error signal (e). The controller takes this new error signal and computes its derivative and its integral again. A temperature sensor and heating element was connected to the PID controller.

Figure 3.13: PID temperature controller device

3.3.8 Temperature sensors

RTD Pt-100 type of temperature sensors were used to measure the temperature of hot and cold fluids at various locations of the heat exchanger. Here RTD is resistance temperature detector and Pt holds for platinum. Significance of value 100 is that the resistance of platinum at 0°C is 100Ω. Ten temperature sensors were used in this experimentation. Two sensors were inserted in the inlet header of the heat exchanger to measure inlet temperature of the hot fluid. Another two sensor were inserted in the exit header to measure the exit temperature of the hot fluid. One sensor was placed near the duct side surface of the heat exchanger to measure inlet temperature of the cold air. Five sensor was placed at different locations near the exit surface of the heat exchanger to measure the exit temperature of hot air.

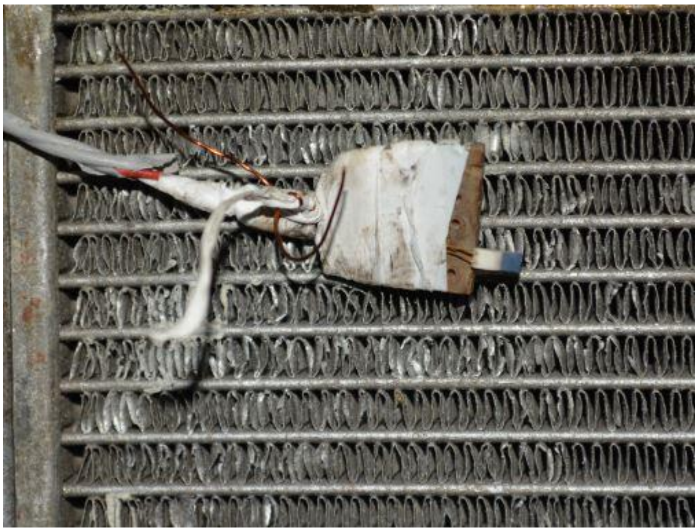

Figure 3.14: RTD Pt-100 temperature sensor

A temperature sensor of RTD Pt-100 type is shown in Figure 3.14. Figure 3.15 shows the location of various temperature sensors fixed on the heat exchanger.

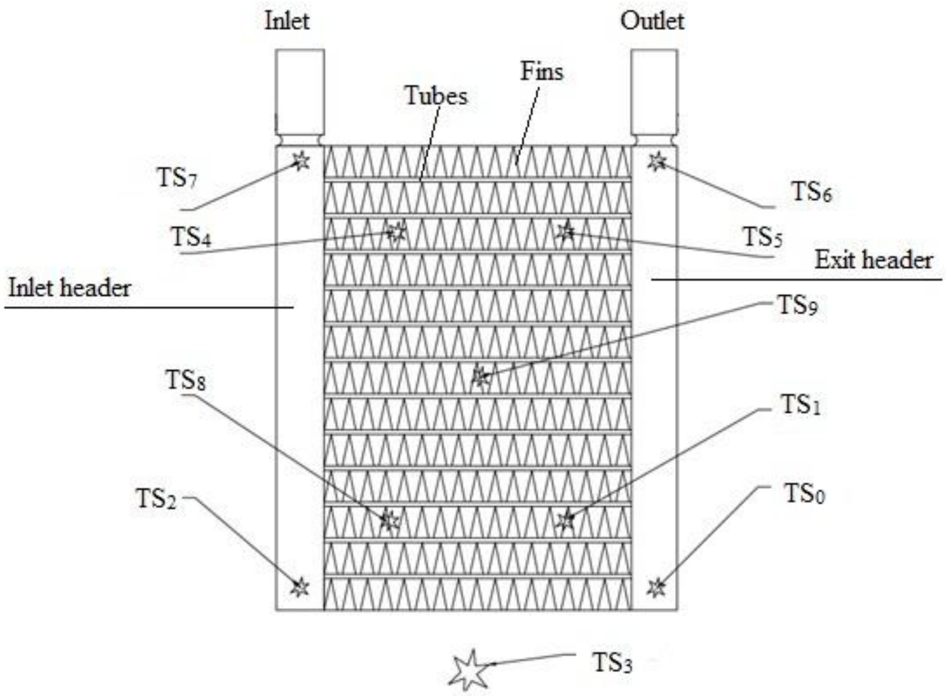

Figure 3.15: Location of various temperature sensors on the heat exchanger

3.3.8.1 Calibration of temperature sensors

Before employing these temperature sensors in the experimental setup, these were calibrated with the help of a water bath and thermometer. All the temperature sensors were connected to temperature display and were inserted in the water bath. Temperature of the bath was varied, and the readings of various temperature sensors were compared with the reading of thermometer. Deviation found ±1°C to 1.5°C . From the data obtained, calibration graph is obtained is shown below in Figure 3.16. Calibration data is given in annexure.

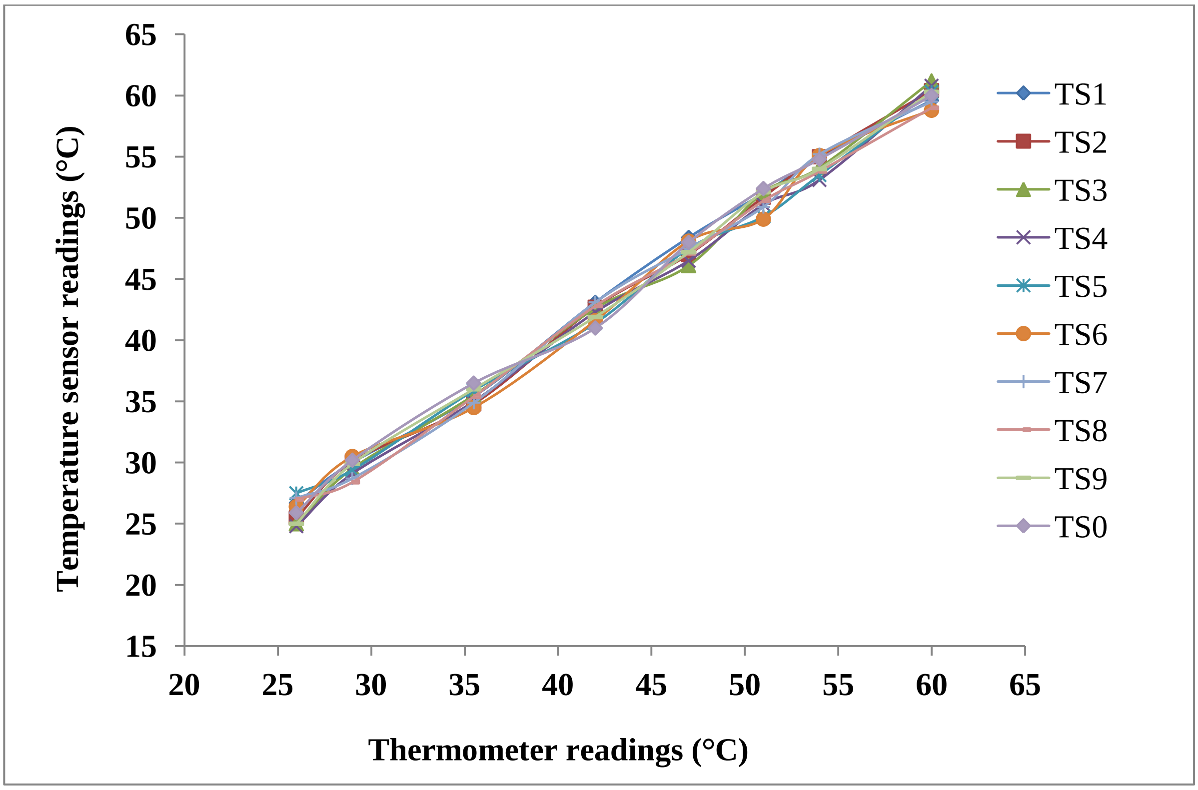

Figure 3.16: Calibration graph of temperature sensors

3.4 Nanofluid preparation & measurement of its thermo-physical properties

Preparation of nanofluids endure agglomeration of nanoparticles, which is a major problem in all technologies concerning nanopowders. Therefore proper synthesis and suspension of nanoparticles in liquids is the key to considerable improvement in the thermal properties of nanofluids. Stability of nanofluids is the major concern of nanofluid preparation. Nanofluids are thermodynamically unstable because of high surface energies associated with multi-phase dispersion system. Strong Brownian motion is one of the aspect of nanofluid instability. Aluminium/water nanofluid was prepared at Mass transfer Lab., Chemical Engineering Department, Thapar University, Patiala, Punjab, India. and thereafter thermo-physical properties i.e. thermal conductivity, density and viscosity were measured experimentally. Experimental values were compared with that obtained by mathematical models available in literature.

3.4.1 Preparation of nanofluids

The Al (metal basis) nanopowder of average particle size 100 nm was purchased from Intelligent Materials Pvt. Ltd, Panchkula, Haryana, India. The properties of Aluminium nanoparticles are tabulated in Table 3.3 and TEM and XRD images are shown in Figure 3.17 and 3.18, respectively.

Table 3.3 Properties of the aluminium nanoparticles

Chemical Name	Al
Appearance	Black gray Powder
Morphology	Spherical
Purity	99.9+% metal basis
Average Particle Size	100 nm (80-125nm)
Specific Surface Area	10.84 m^2/g
Bulk Density	0.08-0.20 g/m^3
True Density	2700 kg/m^3
Thermal Conductivity	71.1 W/mK

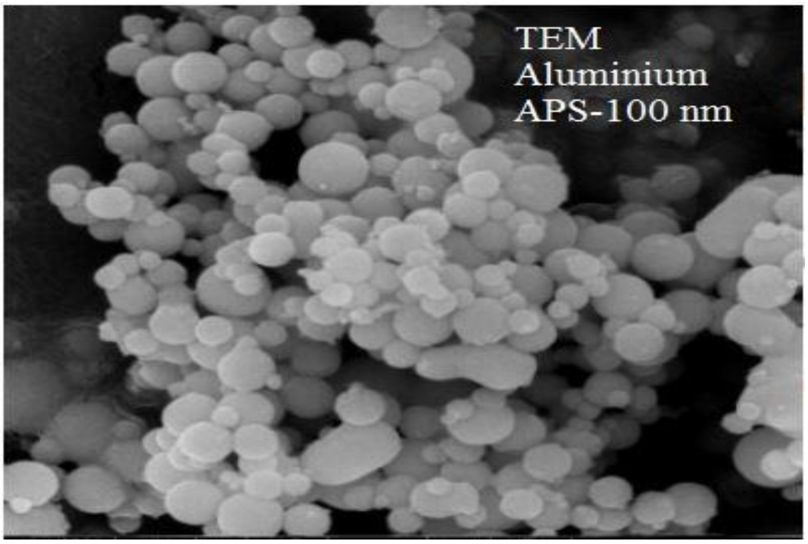

Figure 3.17: TEM Image of Al Nanoparticles

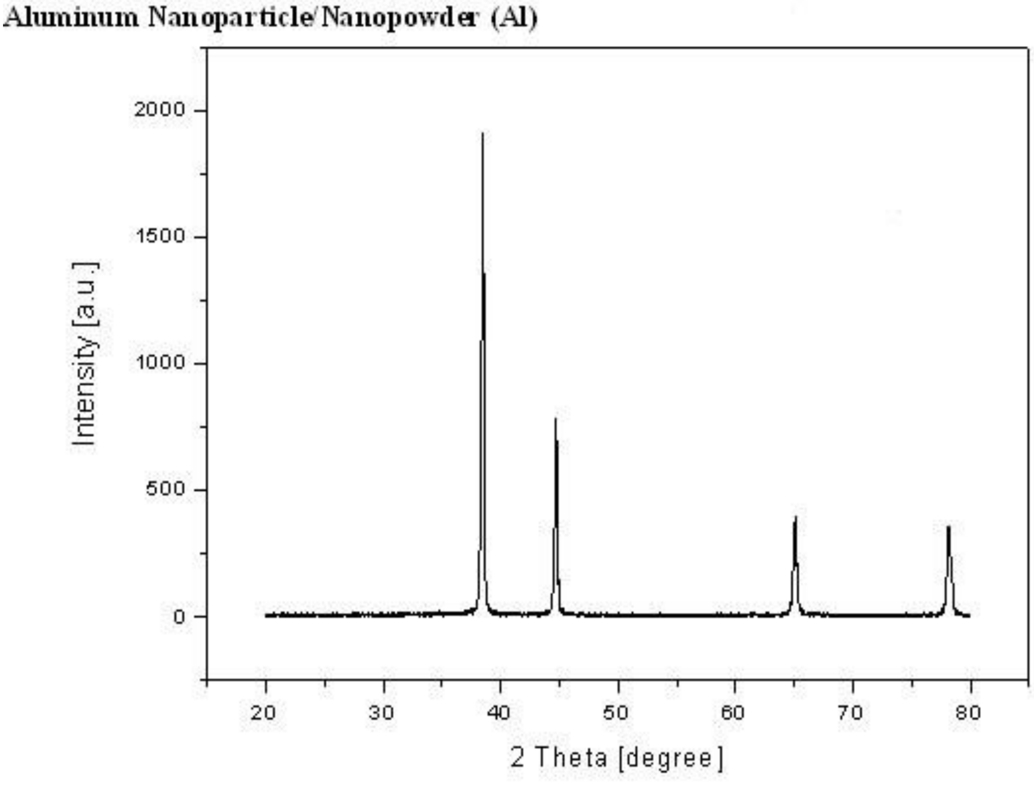

Figure 3.18: XRD image of Aluminium nanoparticles

In the presented work, nanofluids were prepared by two step method. Double distilled water was used to prepare nanofluid. Nanofluids were prepared at two different concentrations by volume 0.1% and 0.2 %. Since the Aluminium nanoparticles are not stable for longer time in water hence to increase the stability and to make them in suspension for more time the surfactant CTAB (Cetyl Trimethyl ammonium bromide) was added in the same amount as that of nanoparticles, which make them stable and in suspension for longer time, otherwise they settled down in few minutes. Two samples, one with surfactant and other without surfactant are shown in Figure 3.19, which clearly showed the effect of surfactant.

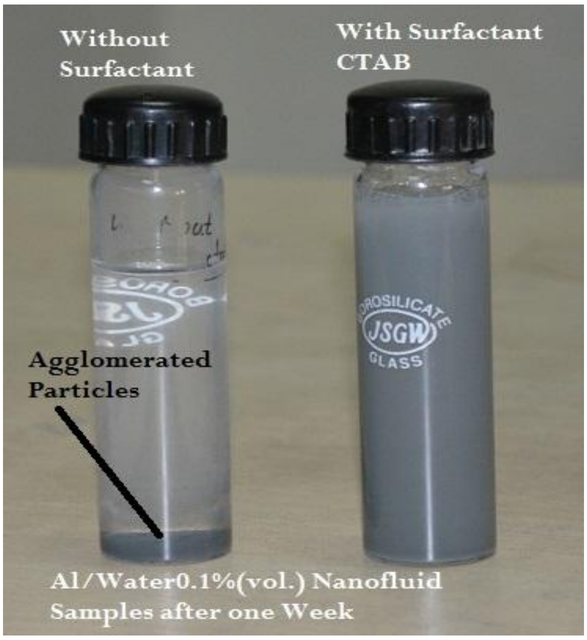

Figure 3.19: Effect of Surfactant on Nanofluid stability

Also to make the nanoparticles more stable and remain more dispersed in water, sonicated with the help of ultra sonicator. Sonication was done for 3 hours before testing thermo physical properties. These properties were measured at various temperatures. Nanofluid was heated by hot plate magnetic stirrer till the steady state achieved at a particular temperature. Following steps were followed for nanofluid preparation. A beaker of double distilled water was placed on the hot plate magnetic stirrer (Figure 3.20) and then Al nanopowder in required amount was slowly poured into the beaker. True density of 2.7g/cm^3 (Table.3.3) was used to calculate the amount of nanopowder to be added. Nanoparticles were mixed into double distilled water by magnetic stirrer for 30 minutes. Further, to reduce the particle agglomeration and to prepare more stable nanofluids, the solution was placed in Ultrasonic water bath (Figure.3.21) for 3 hours. Surfactant CTAB was added in same amount that of nanopowder, which provided more dispersed and stable solution.

Figure 3.20: Hot plate magnetic stirrer

Figure 3.21: Ultrasonicator Water Bath

3.4.2 Measurement of thermal conductivity of nanofluid

Thermal conductivity of Al/water nanofluid of 0.1% (Vol.) concentration was measured at different temperatures. Various methods are available for thermal conductivity measurement, like; transient hot wire method, temperature oscillation method etc.

Thermal conductivity of nanofluid was measured using KD2 Pro thermal property analyzer (Decagon Devices, Inc., USA) shown in Figure 3.22.

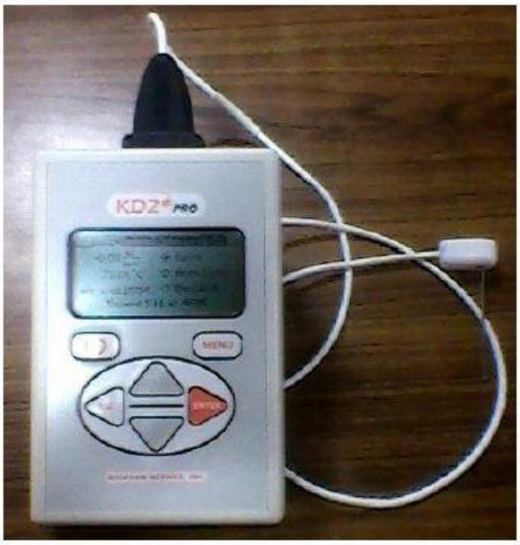

Figure 3.22: KD2 Pro with KS-1 needle

KD2 Pro comprises of sensor needles and hand-operated micro-controller. Sensor consists of thermostat and a heating element. To measure the thermal conductivity of fluids of range 0.2-2.0 W/mK, KS-1 needle can be used, having accuracy of ±0.5%. Thermal conductivity of nanofluid was measured at different temperatures, which was controlled by PID controller. One experimental cycle was accomplished in 90 sec. For first 60 sec. needle equilibrate to the surrounding fluid temperature. Then, for 30 sec. heating and cooling of sensor needle takes place. KD2 Pro gives the thermal conductivity values using the following formula,

$$k = \frac{q(lnt_2 - lnt_2)}{4\pi(\Delta T_2 - \Delta T_1)}$$

Where, k is the thermal conductivity, q is the constant heat rate supplied to an infinitely long and small line source. ΔT_1 & ΔT_2 are the temperature changes at time t_1 and t_2 respectively.

Experimental data obtained for thermal conductivity was compared with Hamilton and Crosser model illustrated by Mujumdar et al. [1] given by the following expression.

$$\frac{K_{nf}}{K_{bf}} = \frac{\alpha + (n-1) - (n-1)(1-\alpha)\emptyset}{\alpha + (n-1) + (1-\alpha)\emptyset}$$

Where $\alpha = \frac{K_p}{K_{bf}}$ and $n = \frac{3}{\varphi}$ is shows the sphericity of the particles, since Aluminium nanoparticles are spherical in shape hence φ=1.

3.4.3 Measurement of density of nanofluid

Density of prepared nanofluid was measured using Pycnometer (Figure 3.23) or specific gravity bottle. Ratio of density of any fluid to density of distilled water at 4^0C temperature is termed as specific gravity. Pycnometer holds a specific volume at a particular temperature. There is a capillary hole in the closely fixed ground glass cork of Pycnometer. Extra liquid is released from the capillary for top filled bottle and remaining liquid's volume and weight was measured. Now weight of empty bottle is measured, the difference in weight is divided by the weight of distilled water of equal volume, which will give specific gravity of given liquid. Density of double distilled water and Al/water nanofluid of 0.1% (Vol.) concentration was measured using Pycnometer for a temperature range of 25°C to 70°C.

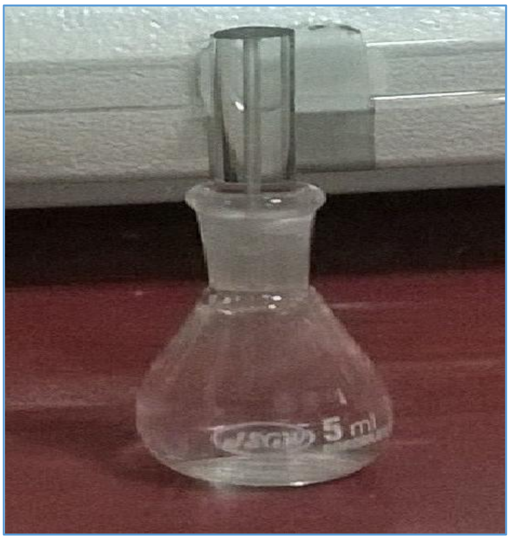

Figure 3.23: Pycnometer

The mathematical model given by Pak et al. [2] was used to compare the experimental values of the density. Model is expressed by the formula,

$$\rho_{nf} = \rho_p \emptyset + \rho_{bf}(1 - \emptyset)$$

Where ρ_{nf} is the density of nanofluid, ρ_p is the density of nanoparticles, ρ_{bf} is the density of base fluid and $\emptyset$ is the nanofluid concentration by volume.

3.4.4 Measurement of viscosity of nanofluid

A measure of resistance occurring during flow is termed as Viscosity. Brookfield DV-III Rheometer (Figure 3.24) was used to measure the viscosity of prepared nanofluid sample. It provides flow factors like shear stress and viscosity at given shear rate. The device operates on the principle that the spindle is driven through a calibrated spring, which measures the viscous drag of the fluid against the spindle by the spring deflection. Spring deflection is measured by a rotary transducer. Rotational speed of the spindle measures the range of a DV-III in centipoises.

Figure 3.24: Brookfield DV-III Rheometer

Measurements was done for various temperature values ranging from 30°C to 80°C. Experimental results were compared with the Einstein model available in the literature [3]. Model is given by,

$$\mu_{nf} = \mu_{bf}(1 + 2.5\emptyset + 6.5\emptyset^2)$$

Where μ_{nf} is the viscosity of nanofluid in cP, μ_{bf} is the viscosity of base fluid and $\emptyset$ is the nanofluid concentration by volume.

3.5 Methodology

As described earlier, experimental setup was consists of a cross flow compact heat exchanger with single pass of 38 tubes with multi-louvered fins. Attention was focussed on the heat exchanger which was the major component of the experiment. To study the effect of replacing the conventional heat transfer fluid by Al/water nanofluids, firstly, the experiments were performed using distilled water as heat transfer fluid and then it was replaced by aluminium/water nanofluid. Experiments were performed by varying inlet temperature of the hot fluid, flow rate of hot fluid, velocity of air passing over the heat exchanger and the concentration of Al/water nanofluids. Different values of these parameters are tabulated below in Table 3.4.

Table 3.4: Parameters to be varied

Flow rate of hot fluid (LPH)	50	60	75	90	100
Velocity of air (m/s)	3.38	4.72	5.88	6.14	6.44
Hot fluid inlet temperature (°C)	45	50		55	
Nanofluid concentration (φ)	Φ=0	Φ=0.1		Φ=0.2	

Velocity of air at the exit of duct was measured with the help of anemometer at different speeds of the fan motor. Speed of fan motor was regulated with the help of speed regulator. Temperature of fluid inlet was controlled with the help of PID temperature controller. Flow rate of hot fluid was fixed at different values with the help of valve provided at the bottom of rotameter. Nanofluids were prepared at different concentrations by volume. Temperature sensors were placed at different locations of the heat exchanger to measure the temperature of hot and cold fluids. U-tube manometers were attached to measure the pressure drop of cold and hot fluid across and through the heat exchanger, respectively. Experiments were performed in the following manner.

1. Power supply to the experimental setup was turned on.
2. Distilled water is taken as hot fluid in the storage chamber.
3. Inlet temperature of the hot fluid was fixed at 45°C with the help of PID temperature controller.

4. Flow rate of the hot fluid was fixed at 50 LPH with the help of rotameter and valve provided at its bottom.
5. While keeping inlet temperature and flow rate of hot fluid constant, readings for temperature at various locations of the heat exchanger, as shown in figure 3.15, with a digital temperature display and for pressure drop provided by two U-tube manometers were recorded at 3.38, 4.72, 5.88, 6.14 and 6.44 m/s velocity of air. Air velocity was varied with the help of speed regulator provided with fan motor.
6. Now the flow rate was changed to 60, 75, 90 and 100 LPH and inlet temperature was kept 45°C. Again the reading were recorded at different velocities of air.
7. Step 4 & 5 was repeated for hot fluid temperature 50°C and 55°C.
8. Step 3 to 6 were repeated by replacing the distilled water with 0.1% (vol.) and 0.2% (vol.) concentrations of Al/water nanofluids.
9. To achieve the steady state operation, all reading were recorded at 15 minutes time interval for a fixed set of parameters. One of the last three similar readings was taken as temperature and pressure drop value at a particular set of parameters.

3.6 Experimental calculations

Experimental calculations were performed for both tube side and fin side. Working media for tube side were distilled water and Al/water nanofluid at 0.1% and 0.2% concentration by volume, respectively. Air was working fluid on fin side. Thermo-physical properties such as thermal conductivity, density, dynamic viscosity, specific heat, Prandtl number of working fluids were taken at bulk mean temperature. Thermo physical properties of nanofluid were measured experimentally.

3.6.1 Tube side calculations

Heat exchanger employed in this experimental work has 38 flat tubes with semi-circular ends. Tubes were separated into two parts by a simple dividing wall. For the analysis of tube side, one part of the tube was considered. Flow of hot fluid was assumed to be equally divided in all the tubes. Figure 3.25 shows the cross sectional view of one part of the tube. Sample calculations for water at 45°C inlet temperatures are given below. Averaged Nusselt number and friction factor was calculated for water side with varying parameters.

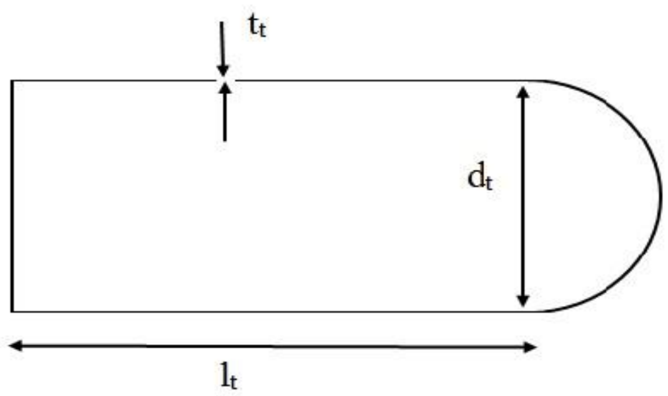

Figure 3.25: Cross section of a heat exchanger tube

$T_{bm} = 39.625$°C

Water flow rate $\dot{Q} = 50\ LPH$

Flow rate per tube, $\dot{q} = \frac{\dot{Q}}{76}$

$\dot{q} = 1.82749\text{E-}07$ m^3/s

Mean velocity of water $V_w = \frac{\dot{q}}{A_c}$

$V_w = 0.014462$ m/s

Thermo-physical properties of water at 39.625°C are as given below [4].

$\rho_w = 992.015259$ kg/m^3

$\mu_w = 0.0006614$ Ns/m^2

$C_{pw} = 4178.525$ kJ/kgK

$Pr_w = 4.3785$

$k_w = 0.6316$ W/mK

Hydraulic diameter of tube, $D_{hw} = \frac{4 \times A_c}{P}$.

$D_{hw} = 0.002149$ m^2

Reynolds number, $R_{ew} = \frac{\rho_w \times V_w \times D_{hw}}{\mu_w}$

$R_{ew} = 46.46$

Nusselt number given by Shah-London equation as follows[5],

$Nu_w = 4.364 + 0.0722(Re_{D_{hw}} \Pr \frac{D_{hw}}{L})$ for $(Re_{D_{hw}} \Pr \frac{D_{hw}}{L}) < 33.33$

$Nu_w = 4.569565648$

Heat transfer coefficient, $h_w = \frac{Nu_w \times K_w}{D_{hw}}$

$h_w = 1343.419101$ W/mk

Friction factor $f_w = \frac{82}{Re_w}$

$f_w = 1.75939046$

3.6.2 Fin side calculations

Multi-louvered fins were employed on the air side of the heat exchanger. To calculate the various performance parameters on air side, a control volume was chosen as shown in Figure 3.26. Also the various dimensions of a louvered fin are illustrated in Figure 3.27. Sample calculations for air side performace are given as follows. Properties of air were also taken at bulk mean temperature of the air.

Stream heat capacity rate, $C_a = \dot{m} \times C_{pa}$

$\dot{m}$ = Air mass flow rate in kg/s

Air side reynolds number, $R_{ea} = \frac{\rho_a \times V_a \times D_{ha}}{\mu_a}$

$R_{ea} = 1074.444$

Heat transfer coefficient [6], $h_a = j_a \times G_a \times C_{pa}/(Pr_a)^{2/3}$

Where,

$j_a = 0.249 \times Re_{lp}{}^{-0.42} \times l_h{}^{0.33} \times H_f{}^{0.25} \times (\frac{l_l}{H_f})^{1.1}$

lp = louver pitch $\quad l_h$ = louver height

l_l = louver length $\quad H_f$ = fin height

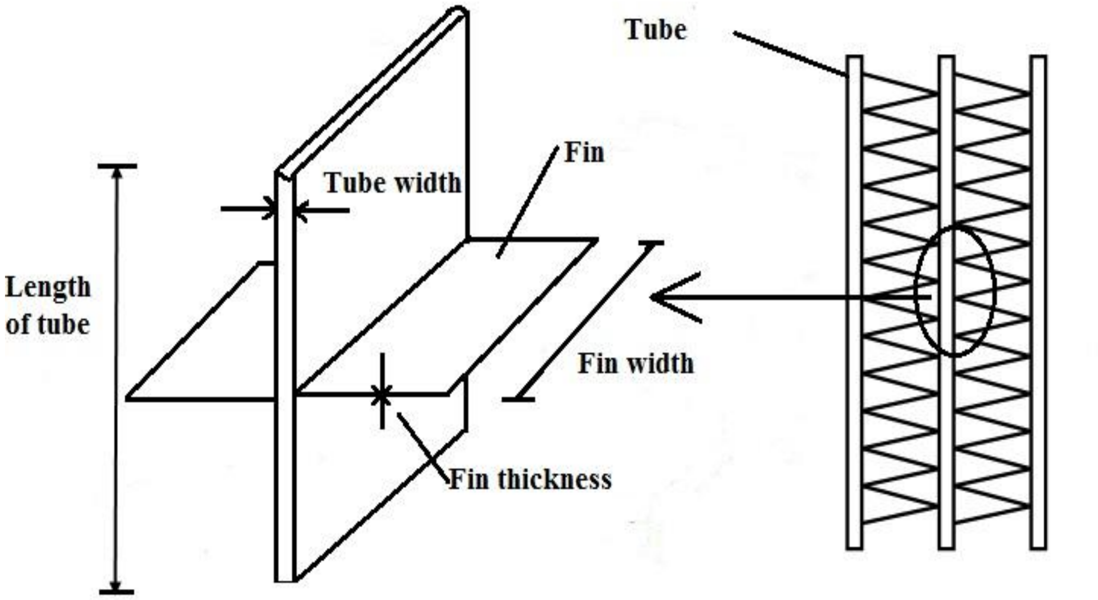

Figure 3.26: Tube fin control volume

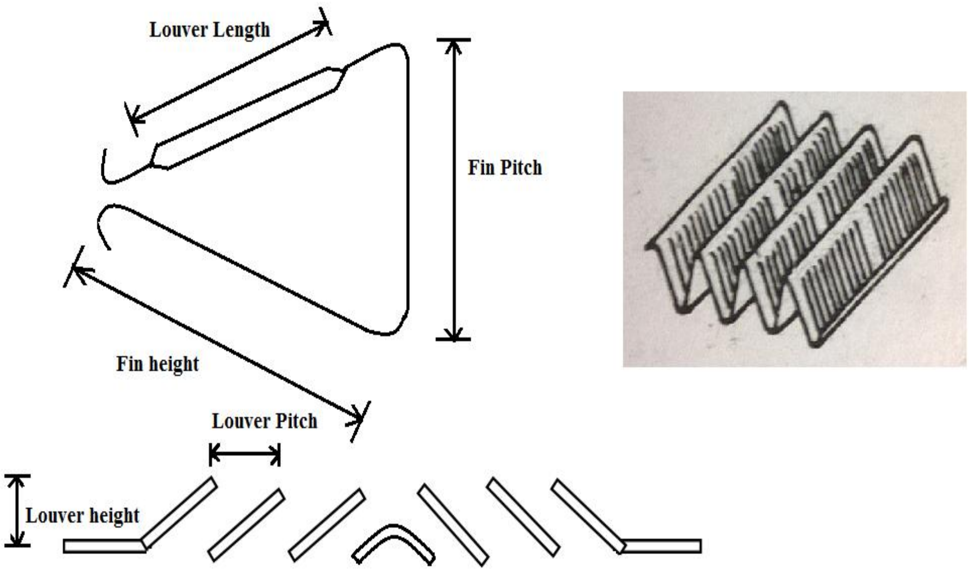

Figure 3.27: Louvered fin geometry

Re_{lp} = Reynolds number calculated on the base of louver pitch

G = core mass velocity = $\frac{\dot{m}}{A_c}$

$j_a = 0.006777$

$h_a = 33.17263$ W/m^{2}K

Fin efficiency, $\eta_f = \tanh(ml)\,/(ml)$

where $m = [2 \times \frac{h_a}{K_{fin} \times \delta}]^{1/2}$

$\eta_f = 0.971447$

Total surface temperature effectiveness of the fin, $\eta_o = 1.0 - (1.0 - \eta_f) \times \frac{A_f}{A}$

$\eta_o = 0.978183$

Friction factor is given by the following correlation [6],

$f_a = 0.494 \times {Re_{lp}}^{-0.39} \times (\frac{l_h}{H_f})^{0.33} \times (\frac{l_l}{H_f})^{1.1} \times {H_f}^{0.46}$; for 1000$< Re_a < 4000$

$f_a = 0.048465$

3.6.3 Heat Exchanger effectiveness

Heat Exchanger effectiveness was calculated with the help of following formula.[6]

Heat exchanger effectiveness, $\epsilon = \; C_h \times (T_{h,in} - T_{h,out})/[C_{min} \times (T_{h,in} - T_{c,in})$

$\epsilon = 0.672164948$

3.6.4 Overall heat transfer coefficient

Neglecting very small wall resistance, overall heat transfer coefficient based on fin side heat transfer area (U_a) was calculated using the following relation.

$$\left(\frac{1}{U_a}\right) = \left(\frac{1}{\eta_o h_a}\right) + 1/\left(\frac{\alpha_w}{\alpha_a}\right) h_w$$

U_a = 32.25614 W/m^{2}K

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CHAPTER 4

RESULTS & DISCUSSIONS

To study the effect of Al/water nanofluid on the thermo-hydraulic performance of a single-pass cross-flow compact heat exchanger, initially nanofluid was prepared at 0.1% and 0.2% volume concentrations by adding aluminium nanopowder into distilled water. Experiments were performed to study the effect of Al/water nanofluid concentration on the performance of a cross flow compact heat exchanger. Effect of Reynolds number of hot and cold fluids on the performance parameters such as Nusselt number and friction factor on the both hot and cold fluids was studied. Results obtained from experimental study are illustrated with the aid of graphs as follows.

4.1 Temperature dependence of thermo-physical properties of nanofluid

Various thermo physical properties of nanofluid such as thermal conductivity, density and viscosity was measured experimentally with the help of KD2 Pro thermal property analyzer, gravity bottle and Brookfield DV-III Rheometer respectively. Temperature dependence of various properties was also studied experimentally. Experimental values for thermo physical properties were compared with the mathematical models available in the literature.

4.1.1 Temperature dependence of thermal conductivity of nanofluid

From experimental data and theoretical model, it was found that the thermal conductivity of Al/water nanofluid was significantly higher than that of water and strongly dependent on temperature of the fluid. Figure 4.1 exhibit that the experimental values of thermal conductivity of nanofluid increased significantly with the fluid temperature. The reason is that, fluid temperature strengthens the Brownian motion of nanoparticles and also drops the viscosity of the base fluid. With a strengthened Brownian motion, the influence of micro convection in heat transport rises and in consequence increased enhancement of the thermal conductivity of nanofluids. Results obtained were compared with the Hamilton and crosser model of thermal conductivity available in the literature [1].

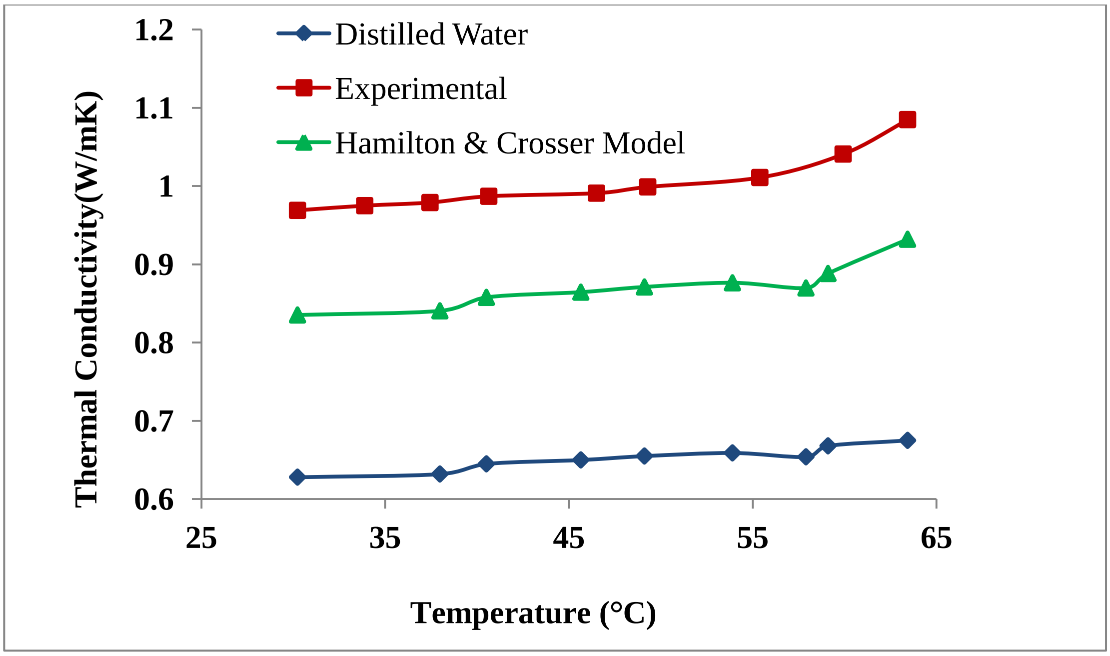

Figure 4.1: Variation of Thermal Conductivity of Al/water nanofluids with temperature

4.1.2 Influence of temperature on the density of nanofluid

From Figure 4.2, it was concluded that, the density of nanofluid was significantly higher than that of water but it was declined slightly as temperature of the fluid was increased.

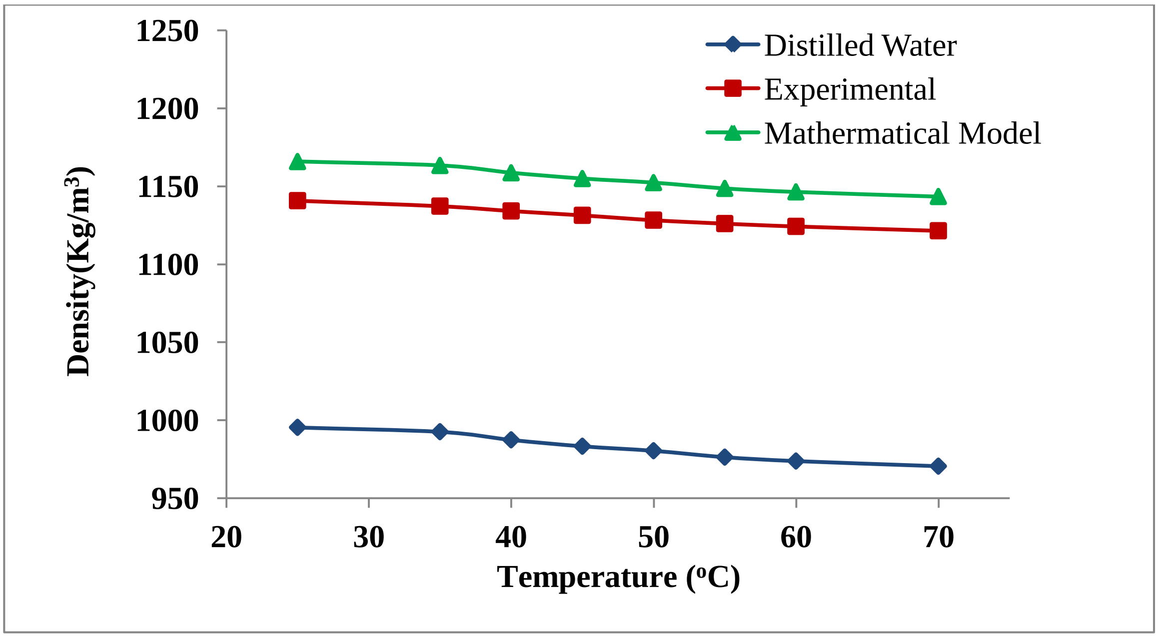

Figure 4.2: Effect of temperature changes on density of Al/water nanofluids

There was a variation of only 1.92% when temperature increased from 25℃ to 70℃. There was similar trend of variation in the density form theoretical model [2].

4.1.3 Temperature dependent viscosity data for Al/water nanofluid

From above data obtained by experiment and theoretical model following graph had been obtained. From Figure 4.3, it was concluded that viscosity of Al/water nanofluid at 0.1% (vol.) concentration was slightly higher than that of water, simply because when solid particles are added to the liquid it increases the density of the mixture and consequently more force will be required to overcome the inertial forces, as a result viscosity increases but there was significant decrement of viscosity with temperature. Similar trends were obtained from experimental data when compared with mathematical model available in literature [3].

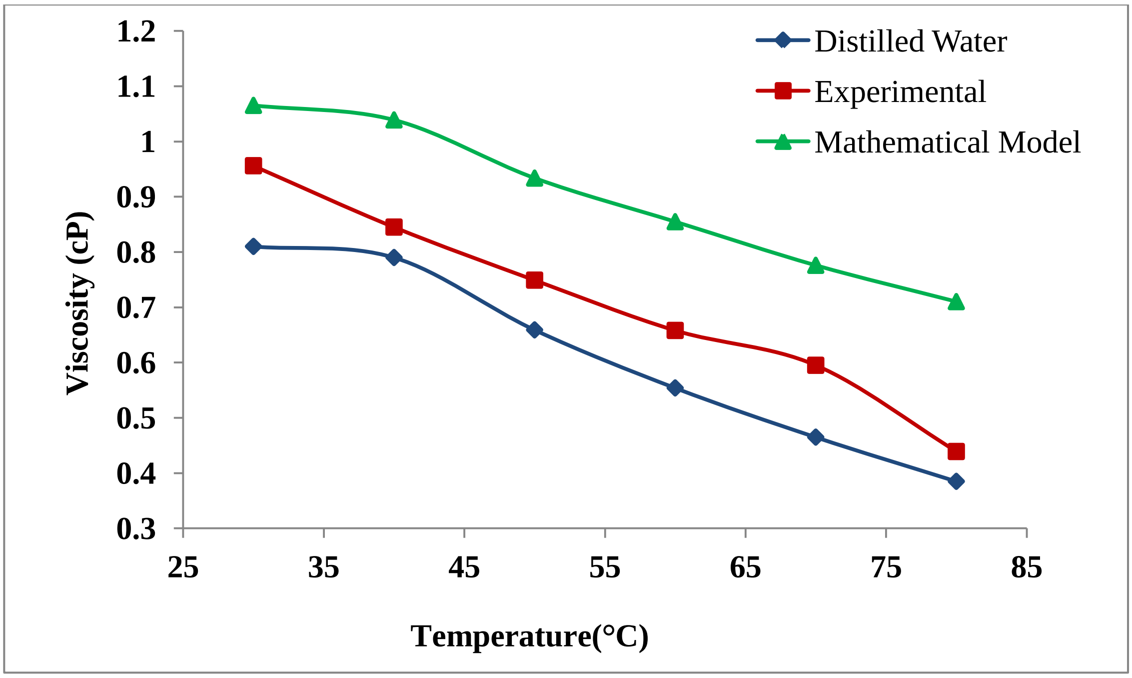

Figure 4.3: Influence of temperature on viscosity of Al/water nanofluid

4.2 Hot fluid side analysis

Experiments were performed at different temperatures and flow rates of hot fluid, using distilled water and two different concentrations of Al/water nanofluid. Heat transfer rate was increased with addition of nanoparticles into base fluid due to increased thermal conductivity of hot fluid. Nusselt number is a dimensionless heat transfer co-efficient. Friction factor, a measure of pressure drop is also another important parameter to be studied. Thermal performance of the heat exchanger increased significantly while a slight increase in the pumping power required with addition of nanoparticles. Tube side performance was studied by analyzing these two parameters. Effect of Reynolds number of hot fluid, inlet temperature of hot fluid and particles volume concentration on tube side Nusselt number and friction factor is illustrated as follows.

4.2.1 Influence of Reynolds number and nanofluid concentration on the tube side Nusselt number

Nusselt number, a dimensionless heat transfer coefficient and is a function of Reynolds number and Prandtl number. Reynolds number is a measure of flow pattern and Prandtl number represents the fluid properties. Hence, Nusselt number directly dependent on flow and fluid properties. Figure 4.4 shows the effect of hot fluid Reynolds number and particle volume concentration on hot fluid side Nusselt number at 45°C inlet fluid temperature, similar trends were found for higher values of temperature. Nusselt numbers significantly increased with increasing Reynolds number. The enhancement of heat transfer with rising Reynolds number was observed by reduction of the thermal boundary thickness caused by increased turbulent intensity.

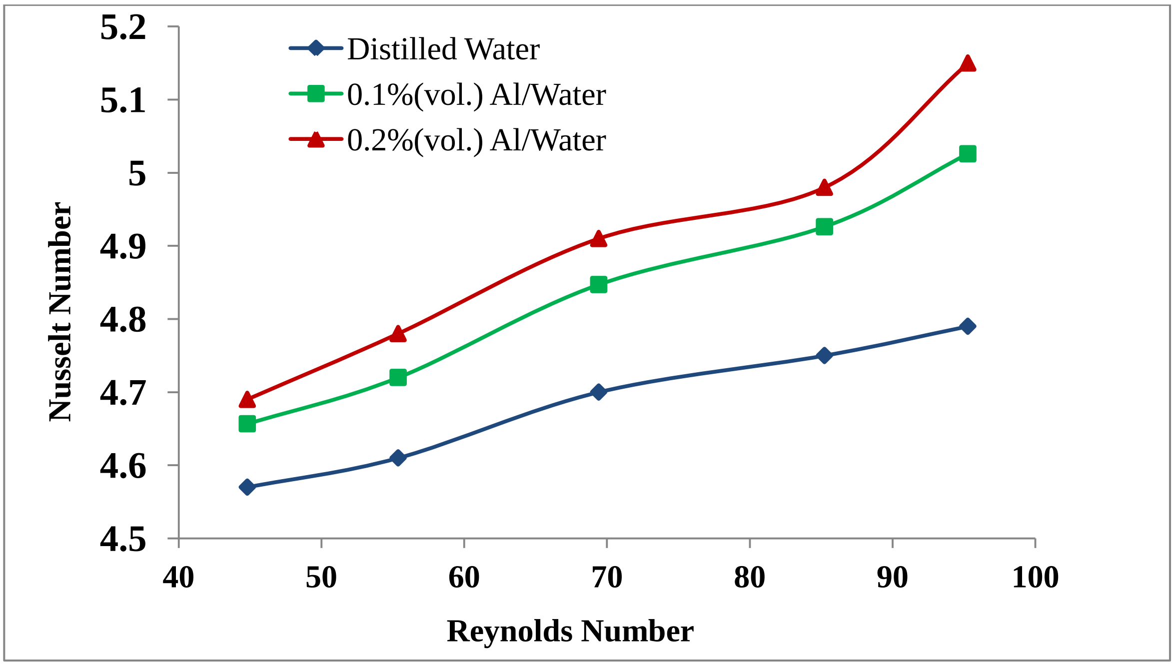

Figure 4.4: Influence of particle volume concentration and Reynolds number on tube side Nusselt number at 45°C inlet fluid temperature

Also, Nusselt number of Al/water nanofluid was higher than that of the distilled water and increased with nanoparticle concentration. The addition of nanoparticles into base fluid increased the thermal conductivity of fluid and also increased the nanoparticles collisions which is responsible for heat transfer enhancement. At same moment, addition of nanoparticles into base fluid increased the viscosity of fluid which declined the heat transfer rate. But the effect of increased thermal conductivity was dominant, hence Nusselt number increased with particle volume concentration.

4.2.2 Influence of Reynolds number and nanofluid concentration on tube side friction factor

Friction factor is a measure of pressure drop and consequently the pumping power required to circulate the hot fluid through heat exchanger. Friction factor decreased considerably with increasing Reynolds number, simply because at higher Reynolds number the inertial forces becomes dominant as compared to viscous forces. Addition of nanoparticles slightly increased the friction factor, which revealed that utilization of nanofluids for heat transfer enhancement have penalty of slightly increased pumping power. Figure 4.5 shows the effect of Reynolds number and particle volume concentrations on hot fluid side friction factor at 45°C inlet fluid temperature, similar trends were found for higher temperature values.

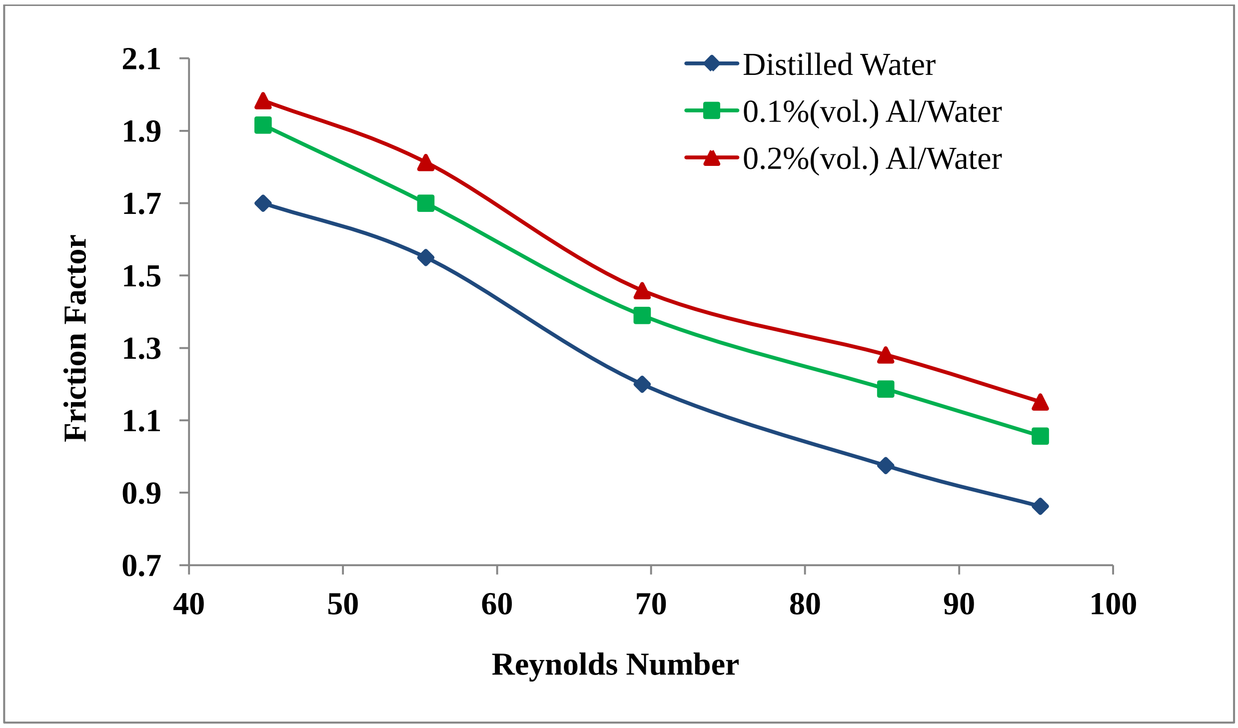

Figure 4.5: Influence of particle volume concentration and Reynolds number on tube side friction factor at 45°C inlet fluid temperature

4.2.3 Effect of inlet fluid temperature on tube side Nusselt number

Nusselt number is a function of Reynolds number and Prandtl number. When inlet temperature of hot fluid was increased, Reynolds number increased because of significant effect of temperature on viscosity of fluid and at the same time Prandtl number decreased because of dominantly increased thermal conductivity of hot fluid and decrease of viscosity.

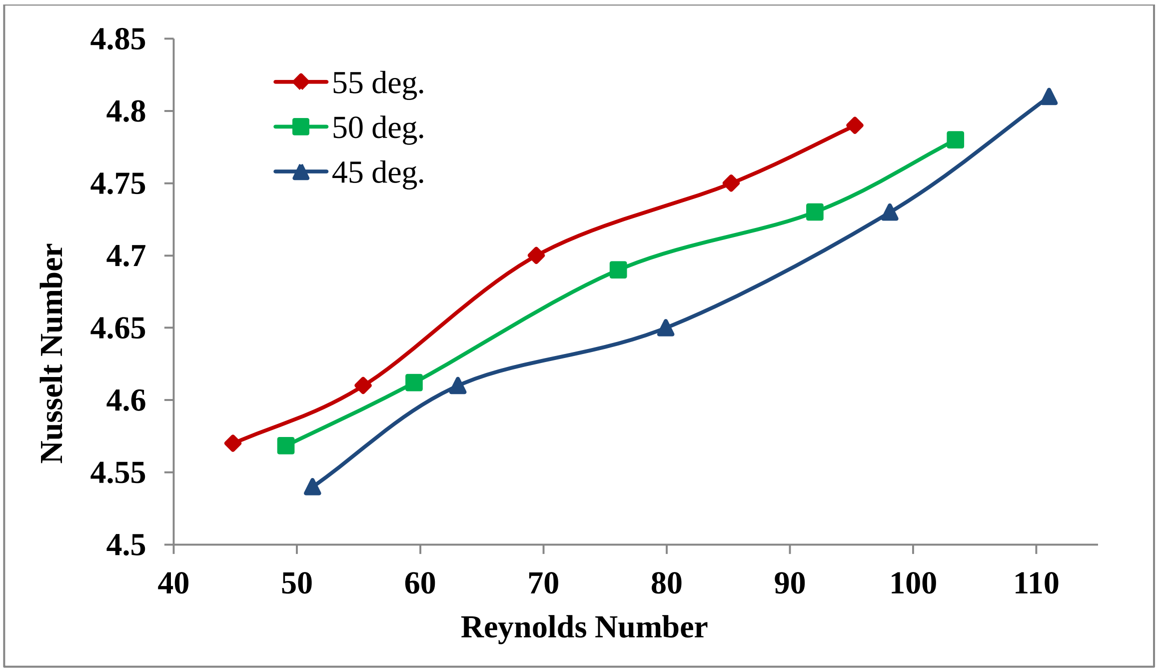

Figure 4.6: Effect of inlet temperature of water on tube side Nusselt number

Nusselt number was increased due to dominant effect of increased Reynolds number. Figure 4.6 shows the effect of inlet temperature of water on Nusselt number, similar trends were found for 0.1% and 0.2% concentrations of Al/water nanofluid.

4.2.4 Effect of inlet fluid temperature on tube side friction factor

As inlet temperature of hot fluid increased, viscosity decreased slightly, consequently the friction factor decreased with increasing temperature of hot fluid. Viscosity of nanofluid was slightly higher than that of water but similar variation of friction factor with temperature were found for 0.1% and 0.2% particle volume concentrations. Figure 4.7 shows the variation of friction factor with inlet temperature of distilled water.

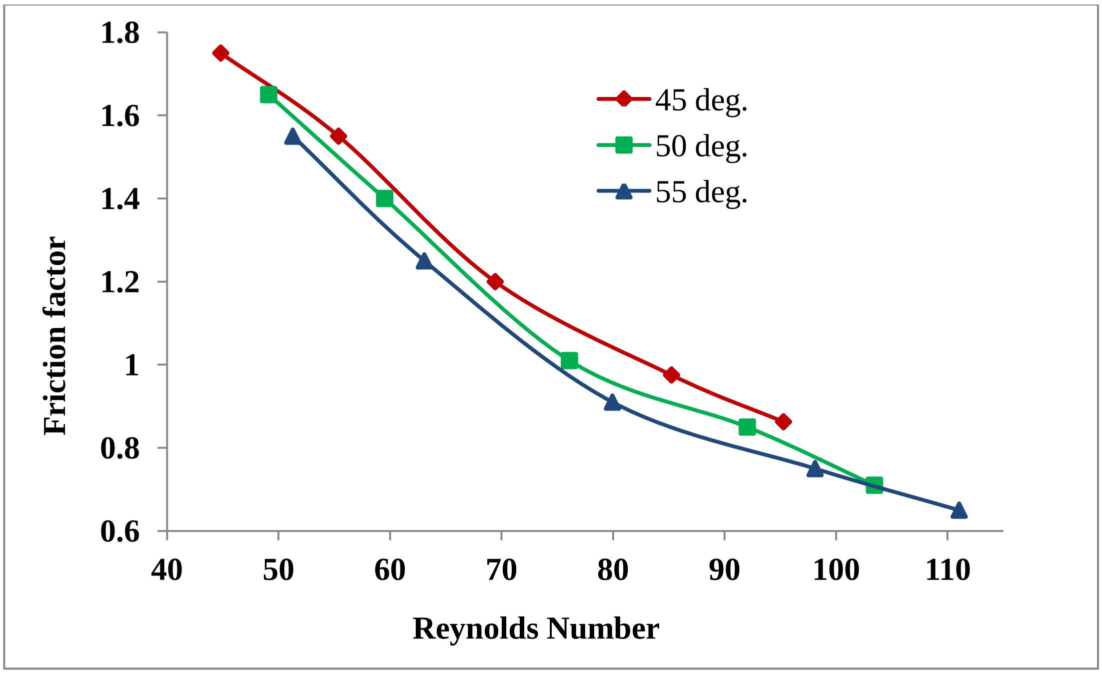

Figure 4.7: Effect of inlet temperature of water on tube side friction factor

4.3 Cold fluid side analysis

Fins of heat exchanger under consideration are multi-louvered type and made of aluminium metal. Heat of hot fluid flowing inside the tubes carried away by air (cold fluid) passing over the fins. Friction between fins and air passing over the fins is also important parameters to be studied. Cold fluid side performance of heat exchanger was studied by Nusselt number, Colburn factor and friction factor. Effect of Reynolds number of cold fluid, nanoparticle volume concentration and inlet temperature of hot fluid on cold fluid side performance is illustrated as follows.

4.3.1 Influence of air Reynolds number and nanofluid concentration on cold fluid side Nusselt Number

Heat transfer rate is highly depends upon the thickness of thermal boundary layer. Increasing the air velocity makes the boundary layer thick and thinner boundary layer leads to the increased heat transfer rate. Multi-louvered fins breaks the thermal boundary layer and hence increased rate of heat transfer. Nusselt number increased with increasing particle volume concentration because it increased thermal conductivity of hot fluid and consequently higher rates of heat loss to the air.

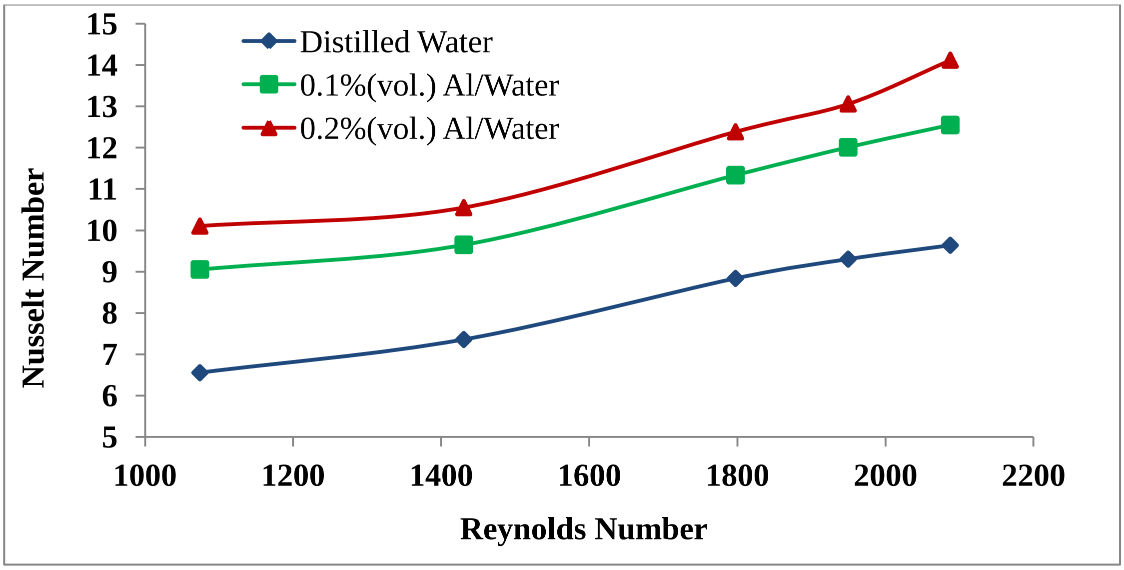

Figure 4.8: Influence of particle volume concentration and Reynolds number on cold fluid side Nusselt number at 45°C hot fluid inlet temperature

4.3.2 Influence of air Reynolds number and nanofluid concentration on cold fluid side Colburn factor

Colburn factor is also a dimensionless heat transfer coefficient and a function of Stanton number and Prandtl number [4]. Stanton number is a modified Nusselt number. Cold fluid side performance of the heat exchanger is usually studied by Colburn factor.

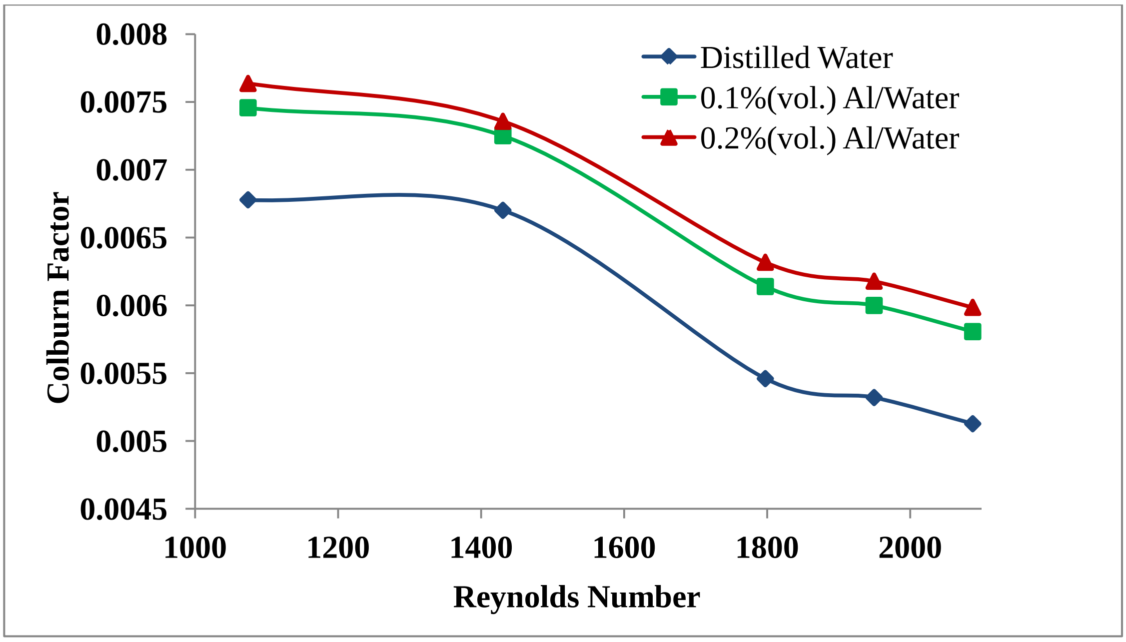

Figure 4.9: Influence of particle volume concentration and Reynolds number on cold fluid side Colburn factor at 45°C hot fluid inlet temperature

As Reynolds number increased Colburn factor decreased due to decreased Stanton number. Colburn factor was increased slightly with increasing particle volume concentrations. Figure 4.9 shows the effect of Reynolds number of air and nanoparticles volume concentration on Colburn factor for 45°C hot fluid inlet temperature, analogues trends were found for higher temperatures.

4.3.3 Influence of Reynolds number of air and nanofluid concentration on cold fluid side friction factor

Friction between air and fin is quite significant because of increased surface area in the case of multi-louvered fins. Multi-louvered configuration increases the heat transfer rate at the cost of slightly increased pressure drop.

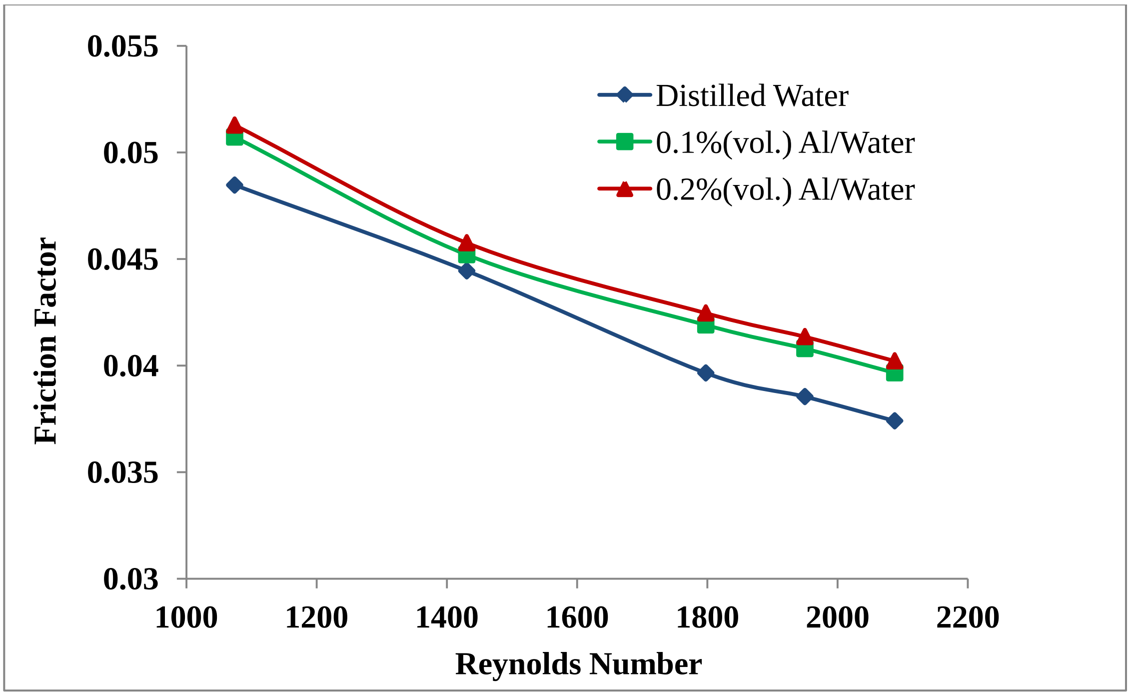

Figure 4.10: Influence of particle volume concentration and Reynolds number on cold fluid side Friction factor at 45°C inlet fluid temperature

Friction factor decreased with increasing Reynolds number of air because at higher Reynolds number, inertial effects were dominant as compared to viscous effects. With increasing particles volume concentration heat was lost to the higher rates consequently air temperature was increased. Viscosity of gases increases with temperature, hence friction between air and fin increased because of increased viscosity of air with increasing particle volume concentrations. Figure 4.10 shows the influence of particle volume concentration and Reynolds number on air side Friction factor at 45°C inlet fluid temperature, similar trends were observed for 50°C and 55°C temperature.

4.3.4 Effect of inlet hot fluid temperature on cold fluid side Nusselt Number

Nusselt number can be viewed as dimensionless temperature gradient at heat transfer surface. As inlet temperature of hot fluid inside the tube increased, consequently the surface temperature was increased. Due to increased surface temperature, the density of air was decreased and viscosity was increased [5]. Nusselt number is a function of both Reynolds number and Prandtl number. Prandtl number also decreased with increased inlet temperature of hot fluid. As a result, Nusselt number decreased with increasing hot fluid inlet temperature. Figure 4.11 shows the effect of inlet temperature of water on fin side Nusselt number, similar trends were obtained for 0.1% and 0.2% concentrations of nanofluid.

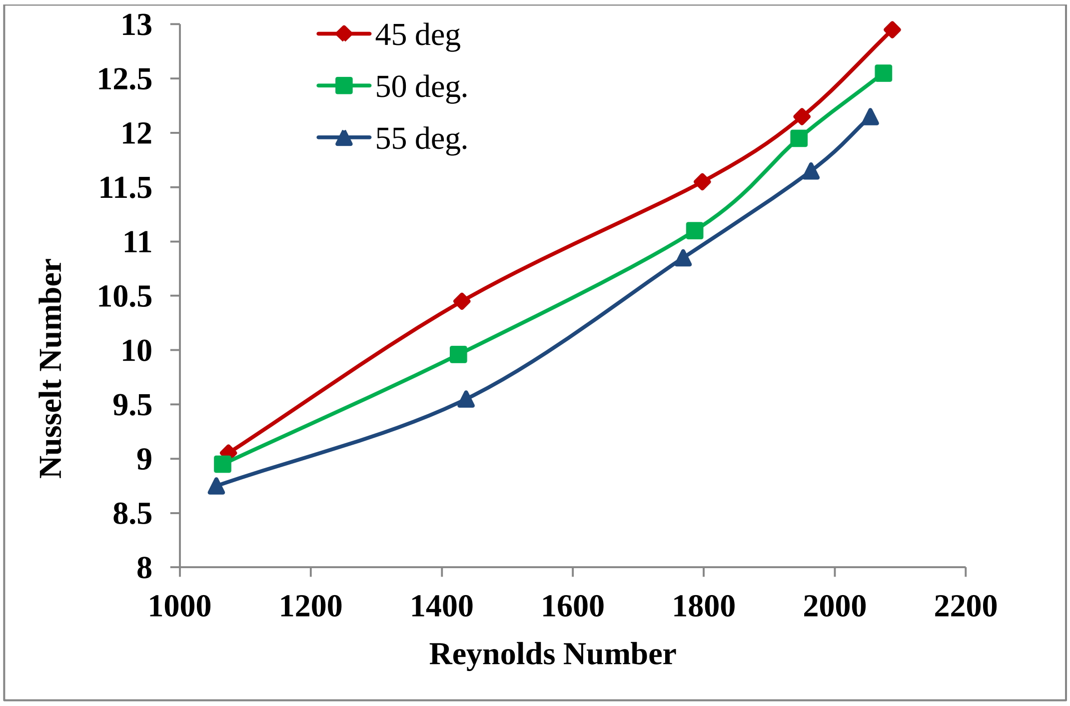

Figure 4.11: Effect of inlet temperature of water on cold fluid side Nusselt number

4.3.5 Inlet fluid temperature dependence of cold fluid side friction factor

As temperature of hot fluid was increased, surface temperature of the heat exchanger was also increased. Viscosity of air increased due to increased temperature of air. Consequently the friction between fins and air passing over the fins was increased with increasing temperature of hot fluid.

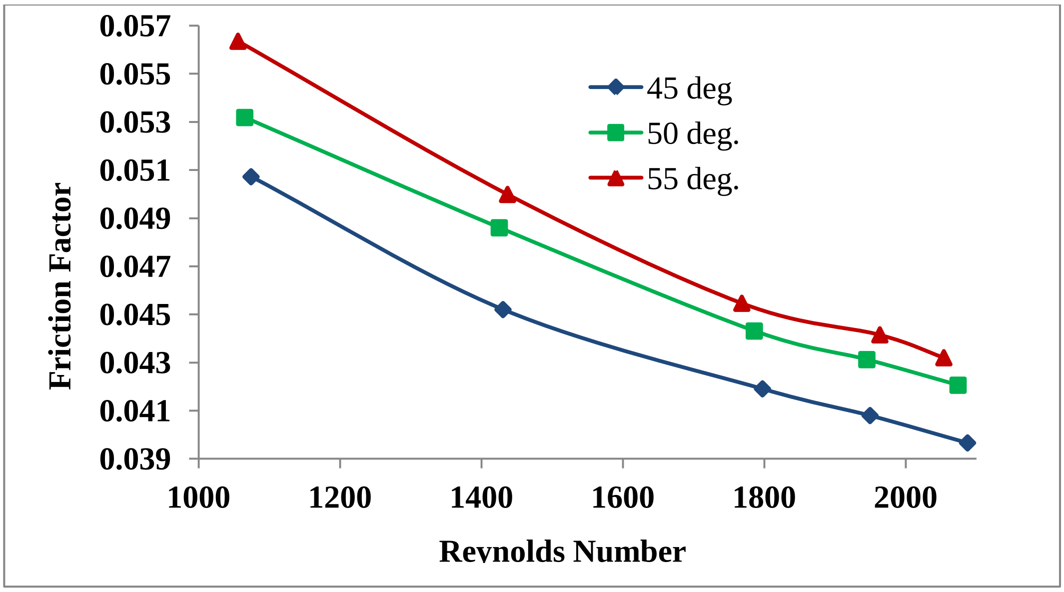

Figure 4.12: Effect of inlet temperature of water on cold fluid side friction factor

Figure 4.12 shows the effect of inlet temperature of water on air side friction factor, similar trends were observed for 0.1% and 0.2% particle volume concentrations of nanofluid.

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CHAPTER 5

CONCLUSIONS

Experimental work was performed on single-pass cross-flow compact heat exchanger to study the effect of Al/water nanofluid on the thermo hydraulic performance of the heat exchanger. Experiments work performed using distilled water, 0.1% (vol.) and 0.2% (vol.) concentration Al/water nanofluid as hot fluid flowing inside the tubes. Experiments were carried out at 45°C, 50°C and 55°C inlet fluid temperature. Reynolds number of hot fluid and air flowing over the heat exchanger was varied by five different values. Experiments were carried out in laminar flow regime. Following conclusions were made from the experimental work performed.

5.1 Thermo physical Properties of nanofluid

- Thermal conductivity of base fluid was enhanced significantly with addition of nanoparticles. Also thermal conductivity observed to be a strong function of temperature. Enhancement of 24.299 % in thermal conductivity was seen at 30.24°C while it was 29.079 % at 61.33°C.
- Density of nanofluid was slightly higher than the base fluid. With increasing temperature density was decreased. Density was decreased by 1.92% as temperature increased from 25°C to 70°C.
- Viscosity of nanofluid was slightly higher than that of base fluid, but it was decreased significantly with temperature. Viscosity of nanofluid was decreased by 54.079% as temperature rises from 30°C to 80°C.

5.2 Tube side performance of the heat exchanger

- Nusselt number of hot fluid was increased with Reynolds number of hot fluid and nanoparticles volume concentration. For 45°C inlet fluid temperature, Nusselt number was higher than that of distilled water by 3.23% and 4.65% for 0.1% and 0.2% concentration of nanofluid, respectively.
- Nusselt number was also increased with increasing inlet temperature of hot fluid. For 0.1% particle volume concentration, Average value of Nusselt number was

increased by 0.898% and 1.75% when temperature increased to 50°C and 55°C respectively with respect to 45°C.

- Tube side heat transfer coefficient was increased by 13.72% and 18.97% for 0.1% and 0.2% particle volume concentrations respectively with respect to distilled water.
- Friction factor of hot fluid increased with addition of nanoparticles into base fluid. At 45°C inlet temperature of hot fluid, friction factor was increased by 14.38% and 21.2748% for 0.1% and 0.2% nanoparticle volume concentrations respectively with respect to distilled water.
- Average friction factor of hot fluid was decreased by increasing inlet temperature of hot fluid. For 0.1% nanoparticle volume concentration, friction factor was decreased by 2.63% and 9.50%, when temperature was increased to 50°C and 55°C respectively with respect to 45°C.

5.3 Air side performance of the heat exchanger

- Nusselt number of air was increased with increasing Reynolds number and particle volume concentration in hot fluid in tube. Average value of Nusselt number was increased by 30.97% and 44.46% with 0.1% and 0.2% concentration of nanofluid respectively with respect to distilled water at 45°C inlet temperature of hot fluid.
- Nusselt number of air was decreased by increasing the inlet hot fluid temperature. Nusselt number was decreased by 2.92% and 4.94% for 50°C and 55°C inlet fluid temperature respectively with respect to 45°C.
- Colburn factor of air decreased with Reynolds number but it was increased with particle volume concentration. Colburn factor was increased by 11.11% and 13.9% for 0.1% and 0.2% nanoparticle volume concentration respectively at 45°C inlet fluid temperature with respect to distilled water.
- Air side friction factor was increased with nanoparticles volume concentration while it was decreased with Reynolds number. Friction factor was increased by 4.69% and 9.5% with 0.1% and 0.2% particle volume concentration respectively with respect to distilled water at 45°C inlet fluid temperature.

- Friction factor was increased with increasing temperature of hot fluid. When temperature of hot fluid was increased to 50°C and 55°C, friction factor was increased by 5.95% and 10.52% respectively with respect to 45°C.

5.4 Overall performance of the heat exchanger

- Effectiveness of the heat exchanger was also increased significantly with the aid of Al/Water nanofluids. It was 67.22% for distilled water and increased to 74.56% and 79.65% for 0.1% and 0.2% particle volume concentration respectively.
- Overall heat transfer co-efficient based on fin side heat transfer area was increased by 14.25% and 16.67% for 0.1% and 0.2% nanoparticle volume concentration respectively with respect to base fluid.

FUTURE SCOPES

In the presented work, Al/water nanofluid was prepared by adding Al (metal basis) nanoparticles of average particle size 100 nm into distilled water at two different particle volume concentrations i.e. 0.1% and 0.2%. Experimental work was performed to study the effect of nanofluid on thermo-hydraulic performance of a single-pass cross-flow compact heat exchanger in laminar flow regime. Future scopes of the presented work are listed as follows.

- Effect of average particle size on the performance of heat exchanger can be studied using nanoparticles of different particle size.
- Experiments can be performed for wide range of particle volume concentrations.
- Numerical analysis can be performed to obtain more precised results and to validate the experimental results.
- Experimental work can be performed in turbulent flow regime by using high flow rate capacity pump.
- Different base fluids can be used to prepared nanofluid and the effect of various base fluid-nanoparticle combination on the heat exchanger performance can be studied.

ANNEXURE

Table A1: Calibration data of rotameter

Flow rate (LPH)	Flow rate (ml/min)	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7	Test 8	Test 9	Test 10
50	833	841	839	837	838	839	840	836	837	835	839
75	1250	1255	1253	1256	1254	1253	1255	1252	1254	1256	1255
100	1667	1670	1671	1669	1672	1668	1669	1673	1672	1667	1668
125	2083	2087	2086	2085	2086	2084	2088	2087	2086	2085	2084
150	2500	2515	2510	2507	2506	2505	2506	2507	2507	2505	2504

Table A2: Calibration data for temperature sensors (all values are in °C)

Thermometer readings	TS_1	TS_2	TS_3	TS_4	TS_5	TS_6	TS_7	TS_8	TS_9	TS_0
21	19.6	20.4	21.3	21.5	22.4	23.1	21.4	22.2	21.5	22.4
26	26.7	25.5	25	24.8	27.5	26.4	27.1	27	25	25.9
29	30	30.1	29.5	29.1	29.4	30.5	28.7	28.4	29.9	30.2
35.5	35.4	34.8	35.5	35	35.9	34.5	34.9	35.4	36	36.5
42	43.1	42.7	42.6	42.3	41.4	41.6	43.1	42.8	41.9	41
47	48.4	47	46.1	46.5	47.5	48.1	47.5	47.1	47.2	48
51	52	51.7	52.1	51.1	50.1	49.9	50.9	51.4	52.1	52.4
54	54.9	55	54.1	53.1	53.5	55.1	55.2	53.8	54	54.8
60	59.6	60.4	61.2	60.8	60.3	58.8	59.5	59	60.3	60

Table A3: Experimental data for thermal conductivity measurements (W/mK)

Sr. No.	Temperature(°C)	Thermal conductivity Of water	Temperature (°C)	Thermal conductivity of Al/water nanofluid ($\emptyset = 0.1$)
1	30.24	0.628	28.51	0.969
2	37.99	0.632	33.90	0.975
3	40.52	0.645	37.45	0.979
4	45.66	0.65	40.65	0.987
5	49.12	0.655	46.51	0.991
6	53.91	0.659	49.30	0.999
7	57.91	0.654	55.40	1.011
8	59.11	0.668	59.93	1.041
9	61.33	0.675	63.44	1.085

Table A4: Experimental data for density (kg/m^3)

Sr. No.	Temperature (^{o}C)	Density of distilled water	Density of nanofluid ($\emptyset = 0.1$)
1	25	995.4	1140.8
2	35	992.6	1137.3
3	40	987.4	1134.2
4	45	983.3	1131.4
5	50	980.4	1128.3
6	55	976.3	1126.1
7	60	973.8	1124.3
8	70	970.5	1121.5

Table A5: Experimental data for Viscosity (cP)

Sr. No.	Temperature (°C)	Distilled water	Nanofluid ($\emptyset = 0.1$)
1	30	0.810	0.956
2	40	0.790	0.845
3	50	0.659	0.749
4	60	0.554	0.658
5	70	0.465	0.595
6	80	0.385	0.439

Table A6: Properties of air at 1 atm pressure

Temperature (°C)	Density (kg/m^3)	Specific heat (c_p) (J/kg.K)	Thermal conductivity (k) (W/m.K)	Dynamic Viscosity (μ) (kg/m.s) $\times 10^6$	Prandtl number (Pr)
0	1.292	1006	0.02364	1.179	0.7362
5	1.269	1006	0.02401	1.754	0.7350
10	1.246	1006	0.02439	1.778	0.7336
15	1.225	1007	0.02476	1.802	0.7323
20	1.204	1007	0.02514	1.825	0.7309
25	1.184	1007	0.02551	1.849	0.7296
30	1.164	1007	0.02588	1.872	0.7282
35	1.145	1007	0.02625	1.895	0.7268
40	1.127	1007	0.02662	1.918	0.7255
45	1.109	1007	0.02699	1.941	0.7241
50	1.092	1007	0.02735	1.963	0.7228
60	1.059	1007	0.02808	2.008	0.7202
70	1.028	1007	0.02881	2.052	0.7177
80	0.9994	1008	0.02953	2.096	0.7154

Table A7: Properties of saturated water

Temperature (°C)	Density (kg/m^3)	Specific heat (c_p) (J/kg.K)	Thermal conductivity (k) (W/m.K)	Dynamic Viscosity (μ) (kg/m.s) $\times 10^3$	Prandtl number (Pr)
0.01	999.8	4217	0.561	1.792	13.5
5	999.9	4205	0.571	1.519	11.2
10	999.7	4194	0.580	1.307	9.45
15	999.1	4185	0.589	1.138	8.09
20	998.0	4182	0.598	1.002	7.01
25	997.0	4180	0607	0.891	6.14
30	996.0	4178	0.615	0.798	5.42
35	994.0	4178	0.623	0.720	4.83
40	992.1	4179	0.631	0.653	4.32
45	990.1	4180	0.644	0.596	3.91
50	988.1	4181	0.649	0.547	3.55
55	985.2	4183	0.654	0.504	3.25
60	983.3	4185	0.659	0.467	2.99
65	980.4	4187	0.663	0.433	2.75
70	977.5	4190	0.667	0.404	2.55
75	974.5	4193	0.670	0.378	2.38
80	971.8	4197	0.673	0.355	2.22
85	968.1	4201	0.675	0.333	2.08
90	965.3	4206	0.677	0.315	1.96
95	961.5	4212	0.679	0.297	1.85
100	957.9	4217	0.682	0.282	1.75

Table A8: Temperature readings at Inlet fluid temperature of 45°C, Water flow rate of 50 LPH and velocity of air as 3.38 m/s

Sr. No.	Time(PM)	T0	T1	T2	T3	T4	T5	T6	T7	T8	T9
1	5:55	31.7	32.3	45.1	22.4	44.3	40.3	38.8	46	40.8	41.6
2	6:10	30.7	31.5	43.9	21.9	43	39.2	40.5	45.1	40.5	40.3
3	6:25	30.2	31.3	44.2	21.5	43	39.4	39.8	44.7	40.3	39.8
4	6:40	30.4	31.3	43.8	21.1	42.6	38.9	39.8	45	39.5	39.8
5	6:55	30.4	31.3	44.2	21.1	42.7	38.9	39.6	44.5	39.2	39.6
6	7:10	29.7	30.8	44.3	21.1	42.5	38.3	39.3	44.7	39.2	39.3
7	7:25	30.4	31	43.6	20.8	42.5	38.7	39.7	44.8	39.2	39.6

Table A9: Temperature readings at Inlet fluid temperature of 50°C, Water flow rate of 50 LPH and velocity of air as 3.38 m/s

No.	Time(PM)	T0	T1	T2	T3	T4	T5	T6	T7	T8	T9
1	7:45	25.4	26	49.1	20.2	45.3	43	42.6	50.5	37.7	38.4
2	8:00	25.2	25.8	49.2	20.1	44.9	42.8	42.4	50.7	37.6	38.2
3	8:15	25.3	25.7	48.9	20.1	44.9	42.7	42.5	50.7	37.5	38
4	8:30	25	25.4	48.8	20	44.9	42.6	42.6	50.8	37.2	37.8
5	8:45	25	25.4	49.3	20	44.9	42.7	42.6	50.7	37.3	37.9
6	9:00	25	25.4	48.8	20	44.9	42.7	42.6	50.7	37.3	37.7
7	9.15	25	25.4	48.8	20	44.9	42.7	42.6	50.7	37.3	37.7

Table A10: Temperature readings at Inlet fluid temperature of 55°C, Water flow rate of 50 LPH and velocity of air as 3.38 m/s

No.	Time(PM)	T0	T1	T2	T3	T4	T5	T6	T7	T8	T9
1	2:40	32.2	32.9	53.0	21.3	52.4	49.4	50.7	56.0	45.4	46.4
2	2:55	32.3	33.1	53.2	21.4	52.3	48.6	49.2	55.9	45.5	46.5
3	3:10	32.1	32.7	53.0	21.2	52.2	48.4	47.4	55.8	45.2	46.2
4	3:25	32.1	32.7	53.2	21.3	52.2	48.4	47.5	55.8	45.2	46.2
5	3:40	32.1	32.4	52.9	21.2	52.1	48.0	47.5	55.8	45.2	46.1
6	3:55	32.1	32.7	53.0	21.2	52.2	48.4	47.4	55.8	45.2	46.2
7	4:10	32.1	32.7	53.0	21.2	52.2	48.0	47.4	55.8	45.2	46.2