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Experimental And Theoretical Studies of Nanofluid Al_2O_3/SiC In Shell And Tube Heat Exchanger

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PROJECT REPORT

On

EXPERIMENTAL AND THEORETICAL STUDIES OF NANOFLUID Al_2O_3/SiC IN SHELL AND TUBE HEAT EXCHANGER

Submitted in partial fulfilment of the requirements

for the award of degree of

Bachelor of Technology

In

Chemical Engineering

Submitted By

PALAK K. LAKHANI (AUR1358008)

DEPARTMENT OF CHEMICAL ENGINEERING

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DECLARATION

I do hereby declare that the training report entitled **“Experimental and theoretical studies of nanofluid Al_2O_3/SiC in shell and tube heat exchanger”** is an authentic work developed by me under the guidance of Dr. Pankaj Kumar Pandey, Department of Chemical Engineering and submitted for evaluation during Final Year Project at Amity University, Rajasthan.

I also declare that any or all contents incorporated in this report have not been submitted in any form for the award of degree or diploma of any other institution or university.

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Signature and Name of the Supervisor:

Dr. Pankaj Kumar Pandey,

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CERTIFICATE

Certified that this project report "**Experimental and theoretical studies of nanofluid Al_2O_3/SiC in shell and tube heat exchanger"** is the bonafide work of **Palak K. Lakhani** who carried out the project work under my supervision.

Signature and Name of the Supervisor:

Dr. Pankaj Kumar Pandey,

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Wisdom knows what to do next…

Skill knows how to do it…

And Virtue is doing it…!

First and foremost, I most gratefully thank my research examiner and supervisor, Dr. Pankaj Kumar Pandey, Coordinator (Chemical Engineering Department). Without his assistance and dedicated involvement in every step throughout the process, this dissertation would have never been accomplished.

A number of gems at the Department of Chemical Engineering in Amity University deserve special mention for all contributions on experimental and laboratory work they made to my Bachelor's thesis project, Mr. Rohit Singh Rajput who guided me with the experiments on Heat Exchanger. Mr. Mahesh Gurjar and Mr. Sukhbir who helped me by providing necessary chemicals for the project.

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Palak K. Lakhani (AUR1358008)

PREFACE

Technical studies cannot be completed without practical knowledge. Hence, final year project is of great importance for an engineering student. Teaching gives theoretical aspects of technology but practical knowledge is important for better understanding of the theory and giving exposure to the student to face actual situations. Our aim during project was to study in detail about the heat transfer concepts, shell and tube heat exchanger, nanofluids and to prepare a project that uses this knowledge.

This report comprises the summary of work we achieved during our final year project (FYP). The task we chose to carry out during the year is the experimental study of nanofluid in shell and tube heat exchanger and validating those results by using various theoretical models. The effort is conducted under the supervision of Dr. Pankaj Kumar Pandey. This report provides a detailed exposition of the tasks we completed towards accomplishing our project goals. We begin by introducing the topic we selected and explaining the primary reasons that encouraged us to select this particular subject as our FYP. We provide all the necessary background on the topic through an exhaustive literature survey. We then go over the primary goals we achieved and provide a brief explanation of the path that was followed in order to achieve these goals. We then present the phases for implementing the various tasks of the project, and provide test results reflecting the improvement in performance of the project. Afterwards we provide a description of the way we divided the various tasks amongst ourselves, and provide tables portraying the timeline of our project as well as all the important milestones. We conclude our report by mentioning the results we obtained throughout the life of the FYP and how these results were addressed. We have prepared this report upon all the knowledge that we acquired during the project. At the end, we would like to say that we have tried our best to fill the gap between theoretical and practical knowledge.

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INTRODUCTION

1.1. HEAT EXCHANGERS

Heat exchangers are devices that facilitate the exchange of heat between two fluids that are at different temperatures without direct contact of fluids. They are widely used in space heating, refrigeration, air conditioning, power stations, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. Another example is the heat sink, which is a passive heat exchanger that transfers the heat generated by an electronic or a mechanical device to a fluid medium, often air or a liquid coolant. Heat exchangers differ from mixing chambers in the way that they do not allow the two fluids involved to mix with each other. For example, in a car radiator, heat is transferred from the hot water flowing through the radiator tubes to the air flowing through the closely spaced thin plates attached outside of the tubes.

Heat transfer in a heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids. In the analysis of heat exchangers, it is convenient to work with an overall heat transfer coefficient U that accounts for the contribution of all these effects on heat transfer. The rate of heat transfer between the two fluids at a location in a heat exchanger depends on the magnitude of the temperature difference at that location, which varies along the heat exchanger. In the analysis of heat exchangers, it is usually convenient to work with the logarithmic mean temperature difference LMTD, which is an equivalent mean temperature difference between the two fluids for the entire heat exchanger.

Heat exchangers are used for:

- Heat exchangers are practical devices used to transfer energy from one fluid to another
- To get fluid streams to the right temperature for the next process
    - reactions often require feeds at high temp
- To condense vapours
- To evaporate liquids
- To recover heat to use elsewhere
- To reject low-grade heat
- To drive a power cycle

1.1.1. Classification

The heat exchangers are classified into four types, according to the following:

- Nature of heat exchanger process
- Relative direction of motion of fluids
- Mechanical design of heat exchanger surface
- Physical state of heat exchanging

1.2. THE OVERALL HEAT TRANSFER COEFFICIENT

The overall heat transfer coefficient, or U-value, refers to how well heat is conducted over a series of mediums. Its units are the W/(m^2 °C). The overall heat transfer coefficient is influenced by the thickness and thermal conductivity of the mediums through which heat is transferred. The larger the coefficient, the easier heat is transferred from its source to the product being heated. In a heat exchanger, the relationship between the overall heat transfer coefficient (U) and the heat transfer rate (Q) can be demonstrated by the following equation:

$$Q = U \times A \times \Delta T_{LMTD}$$

where,

Q = Heat transfer rate, W=J/s

A = Heat transfer surface area, m^2

U = Overall heat transfer coefficient, W/(m^2·°C)

ΔT_{LMTD} = Logarithmic mean temperature difference, °C

From this equation we can see that the U value is directly proportional to Q, the heat transfer rate. Assuming the heat transfer surface and temperature difference remain unchanged, the greater the U value, the greater the heat transfer rate. In other words, this means that for a same kettle and product, a higher U value could lead to shorter batch times.

1.3. FOULING FACTOR OR DIRT FACTOR

Over a time period of heat exchanger operation the surface of the heat exchanger may be coated by the various deposits present in the flow system. Moreover, the surfaces may become corroded or eroded over the time. Therefore, the thickness of the surface may get changed due to these deposits. These deposits are known as scale. These scales provide another resistance and usually decrease the performance of the heat exchangers. The overall effect is usually represented by dirt factor or fouling factor, or fouling resistance, R_f (Table 8.1) which must have included all the resistances along with the resistances due to scales for the calculation of overall heat transfer coefficient. The fouling factor must be determined experimentally using the following equation:

$$R_f = \frac{1}{U_{dirty}} - \frac{1}{U_{clean}}$$

Thus to determine the R_f, it is very important to know U_{clean} for the new heat exchanger. The U_{clean} must be kept securely to obtain the R_f, at any time of the exchanger’s life.

Fouling can be caused by the following sources:

1. Scaling is the most common form of fouling and is associated with inverse solubility salts. Examples of such salts are $CaCO_3$, $CaSO_4$, $Ca_3(PO_4)_2$, $CaSiO_3$, $Ca(OH)_2$, $Mg(OH)_2$, $MgSiO_3$, Na_2SO_4, $LiSO_4$, and Li_2CO_3.
2. Corrosion fouling is caused by chemical reaction of some fluid constituents with the heat exchanger tube material.
3. Chemical reaction fouling involves chemical reactions in the process stream which results in deposition of material on the heat exchanger tubes. This commonly occurs in food processing industries.
4. Freezing fouling is occurs when a portion of the hot stream is cooled to near the freezing point for one of its components. This commonly occurs in refineries where

paraffin frequently solidifies from petroleum products at various stages in the refining process, obstructing both flow and heat transfer.

5. Biological fouling is common where untreated water from natural resources such as rivers and lakes is used as a coolant. Biological microorganisms such as algae or other microbes can grow inside the heat exchanger and hinder heat transfer.
6. Particulate fouling results from the presence of micro-scale sized particles in solution. When such particles accumulate on a heat exchanger surface they sometimes fuse and harden. Like scale these deposits are difficult to remove.

1.4. TEMPERATURE PROFILES IN HEAT EXCHANGERS

Figure 1.1 shows the temperature profile along the length of a 1-2 exchangers and 2-4 exchangers.

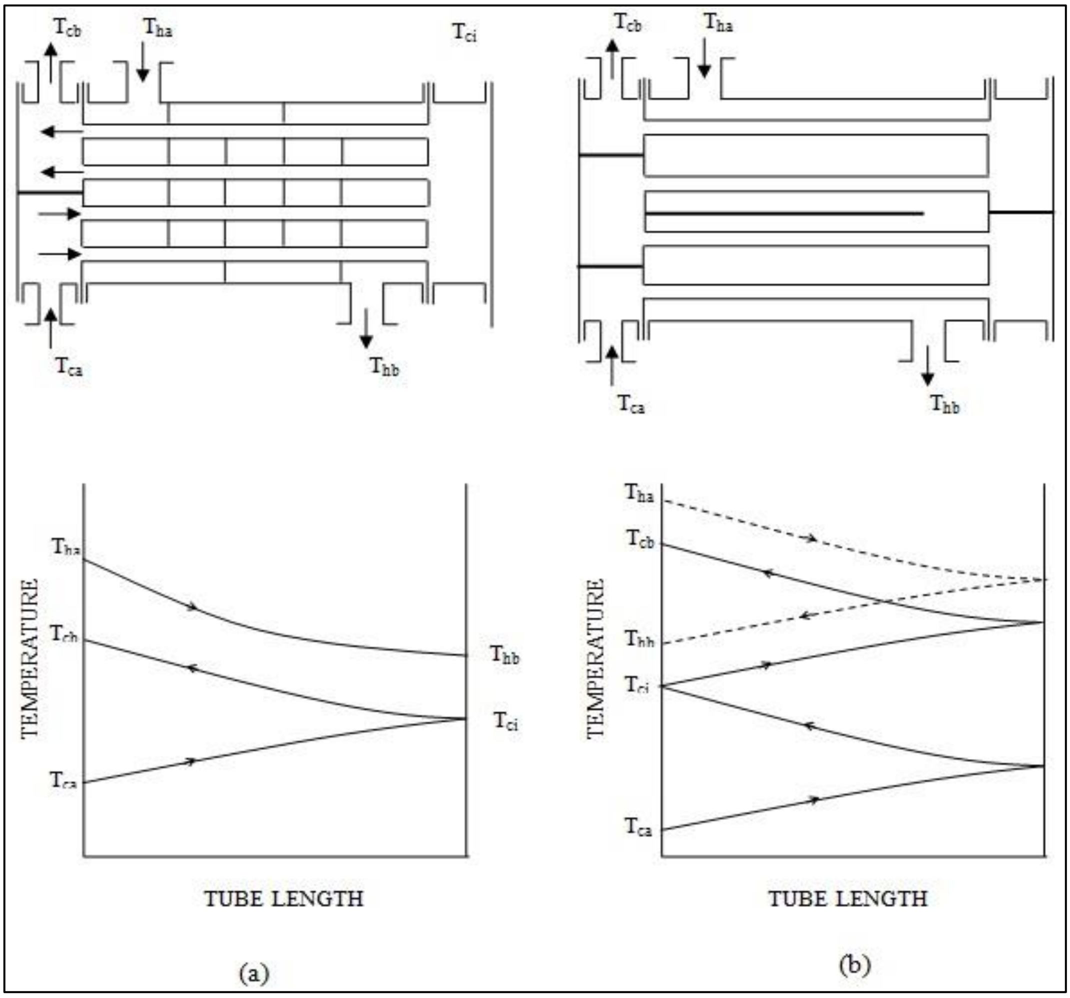


Figure 1.1. Temperature Profiles (a) 1-2 exchanger (b) 2-4 exchanger

The nomenclature used in the figure 1.1 is described below:

T_{ha} = Inlet temperature of hot fluid, °C

T_{hb} = Outlet temperature of hot fluid, °C

T_{ca} = Inlet temperature of cold fluid, °C

T_{cb} = Outlet temperature of cold fluid, °C

T_{ci} = Intermediate temperature of cold fluid, °C

In the above arrangement it is assumed that the hot fluid is flowing in the shell side and cold fluid is flowing in the tube side of the exchangers. The figure 1.1 (a) shows the 1-2 exchangers in which the hot fluid enter into the exchanger from the left side and exits from the right side. The cold fluid enters concurrently that is from the left side to the tube of the exchangers and goes up to right end of the exchangers and returns back to make two tube pass, and exits from the left end of the exchangers. The temperature profile all along the length of the exchanger is shown in the corresponding temperature length profile. Figure 1.1 (b) shows the flow direction and corresponding temperature length profile for 2-4 exchangers. The shell side fluid has 2 passes and the tube side fluid has 4 passes in the exchangers.

It can be easily understood that whenever the number of passes is more than one, the flow cannot be truly co-current or counter current. Thus it will be a mix of co-current and counter current flows in any multi-pass heat exchangers.

1.4.1. Log Mean Temperature Difference

The logarithmic mean temperature difference (also known as log mean temperature difference or simply by LMTD) is used to determine the temperature driving force for heat transfer in flow systems, most notably in heat exchangers. The LMTD is a logarithmic average of the temperature difference between the hot and cold feeds at each end of the double pipe or shell and tube heat exchanger. The larger the LMTD, the more heat is transferred. The use of the LMTD arises straightforwardly from the analysis of a heat exchanger with constant flow rate and fluid thermal properties.

$$LMTD = \frac{(\Delta T1 - \Delta T2)}{\ln(\frac{\Delta T1}{\Delta T2})}$$

Counter-current flow	Co-current flow
$\Delta T1 = T_{ha}$-T_{cb}	$\Delta T1 = T_{ha}$-T_{ca}
$\Delta T2 = T_{hb}$-T_{ca}	$\Delta T2 = T_{hb}$-T_{cb}

1.5. TYPES OF HEAT EXCHANGERS

We come across heat management and controlling during a design of any mechanical or chemical unit equipment. A physical and chemical change involves somehow the application of heat exchanger in order to maintain the temperature as per the condition to run the system. Exchange of heat of the fluid system from a point to other point of interest is done using a piece of equipment called heat exchanger and of course it may differ from the application but ultimate principle behind it working is same.

Theory behind heat exchanger is simple, just the allowing heat energy to flow in perfectly designed system by means of conduction, convection in case of fluids and when dealt with direct firing radiation will be involved. We discuss about four main types of heat exchangers by comparing their application, capacity and range of duty which are classified based on the flow pattern of fluid in them:

1. Plate heat exchanger
2. Shell and tube heat exchanger
3. Fin type heat exchanger
4. Tubular heat exchanger

1.5.1. Plate type heat exchanger

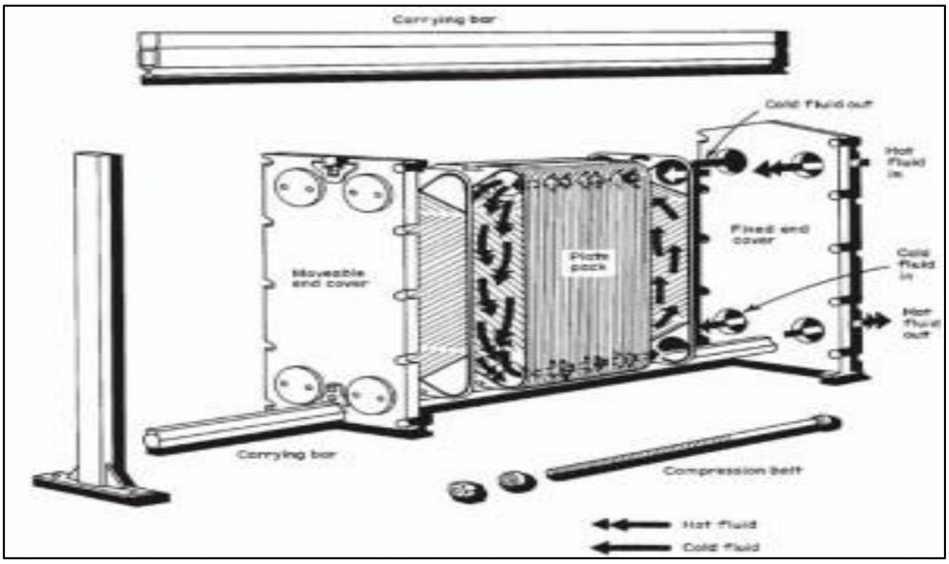


Figure 1.2. Plate type heat exchanger

It is an arrangement of plates side by side fixed along with gaskets. Each plate will be having four opening ports, a gasket which is fitted between two adjacent plate will make the fluid to flow between two ports by block remaining ports, to imagine a rectangle plate having four holes near at its edges and these are fixed in such a way that gasket between plate make a enough room for fluid to flow and change the direction of as required. This operates fluids in counter current wise for attaining maximum heat exchange.

It has advantages which overcome other heat exchanger are:

- Negligible heat loss
- Overall weight of set is less
- Fits in less space
- Less maintenance cost and
- Overall heat transfer coefficient is more
- Easy installation

It has following disadvantage:

- Not compatible for higher temperature and pressure say above 200°C and 20 bars

Applications:

- Used in oil cooling system in automobiles, steam condensations, swimming water cooling systems, refrigeration systems

1.5.2. Shell and tube heat exchanger

Figure 1.3. Shell and tube heat exchanger

It contains tubes inside a shell which is basic and basic design of heat exchangers, it is heavy duty equipment which is widely used in process industries it can handle higher temperature up to 900˚C and more. Temperature of fluids are maximum at wall than the flowing stream and the heat has to redistribute evenly, baffle arrangement make the flow somewhat turbulent for better heat transfer disturbing the boundary layer and supporting the tube also. In most cases hot fluid like steam if fed towards tube side and cold fluid is sent into shell side of shell and tube heat exchanger.

Based on the design and pattern of flow of fluid internally it can be classified as single pass, 1-2 pass, 2-4 pass, floating head heat exchangers(internal floating head and outside packed floating head), fixed tube sheet, U-tube types . It accomplish the duty of condenser when it function is to cool a stream and similarly it operates as re-boiler when used for changing the phase of the stream. It operates in counter-current and co-current flow pattern and their approach will differ for same equipment.

It has following advantages:

- Can handle heavy duty and high temperature and pressures
- Easy control and operate-able

It has following disadvantages:

- Requires large space
- Maintenance cost is more

Applications:

- Used as regular heat exchanger for distillation columns, chemical equipment's stream cooling and heating purpose
- Used internally in re-boilers and evaporators

1.5.3. Fin type heat exchanger

Figure 1.4. Fin type heat exchanger

These are called as extended surface exchangers it is used in the conditions where when a process fluid is having very low heat transfer coefficient when compared to other fluids because the overall heat transfer coefficient decreases which affect the capacity of heat transfer by the available surface area so the area of contact is to be increase. We come across this type of problem when heat transfer is dealt with heating of air or gas streams and for cooling viscous liquids like engine oil. A metal piece is welded to the surface of the tube to increase the surface are which are called as fin and mostly found on outside of the tube. Fin type heat exchanger is classified into longitudinal fins type and transverse fins type.

It has following advantages:

- Used to handle low heat transfer coefficient fluids
- Used for cooling and heating of vast quantity of gases

It has following disadvantages:

- Slurry fluids cannot be handled
- Deposition of particle at fin corner
- Cleaning difficulty
- High pressure drop

Applications:

- Used as economizers, automobile radiators, air cooled condensers

1.5.4. Tubular heat exchanger

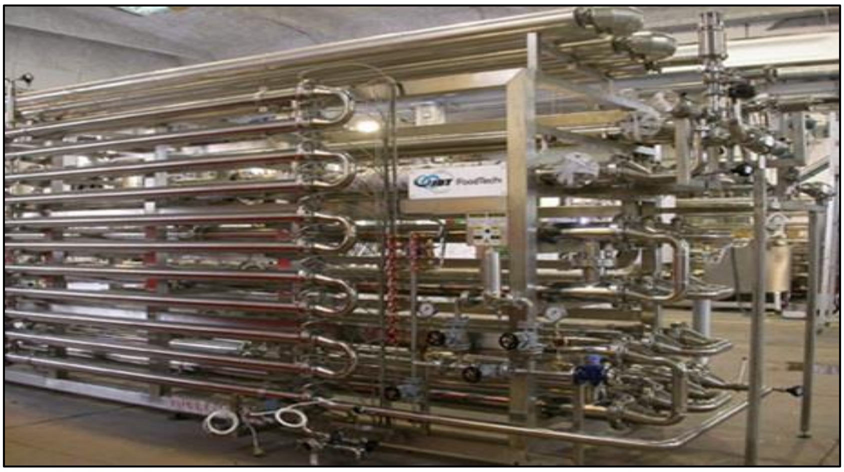


Figure 1.5. Tubular heat exchanger

It is a pipe in pipe heat exchanger commonly known as double pipe heat transfer equipment which is named by its construction because two pipes are pipes are fitted in such way that one pipe is fitted into other inside space. By cross section they look concentric by coaxial view. It is extended to the required length and bent like hair pin shape at the edges to make it fit in particular area. Hot fluid is fed into the inner tube and cold fluid is fed into the space between inner and outer pipes.

The tubular condenser consists of:

1. A bundle of parallel tubes whose ends are expanded into tube sheets.
2. The tube bundle is inside a cylindrical shell and is provided with two channels, one at each end and two channel covers.
3. Steam or other pure vapour is introduced into the shell-side surrounding the tubes, condensate is withdrawn from the shell.
4. Any non-condensable gas is removed through vent.
5. The fluid to be heated is pumped through the tubes.
6. Two fluids (fluid, steam) are physically separated but are in thermal contact with the thin metal tube separating them.
7. Heat flows through the tube walls from the condensing vapour to the cooler fluid in the tubes.
8. Fluid flowing in the tube side gain heat and increase it temperature from inlet to outlet of the tubes.
9. The temperature of the shell side fluid is constant for the pressure maintained on the shell side.
10. Types of flows in heat exchanger.

It has following advantages:

- Easy in construction
- Can handle slurries
- Can be used at low heat transfer areas
- Cheap
- Easy cleaning
- Concurrent and counter current flow can be adjusted easily

It has following disadvantages:

- Leakage corners
- Maintenance is time consuming
- Occupy more floor space compared to others

Applications:

- Used for refrigeration process in appliance like refrigerators, domestic heating systems and car radiators etc.

1.6. SHELL AND TUBE HEAT EXCHANGER

Shell and tube heat exchanger is a type of heat exchanger and it comes to its one of its designs. This shell and tube heat exchanger is most commonly used heat exchanger used in many oil refineries and thermal processes. This shell and tube heat exchanger is commonly employed for high pressure applications. The heat exchanger consists of two names in the main name, namely shell and tube. So we can easily conclude that the shell and tube heat exchanger is composed on tubes and shells in it for the transfer of the heat energy.

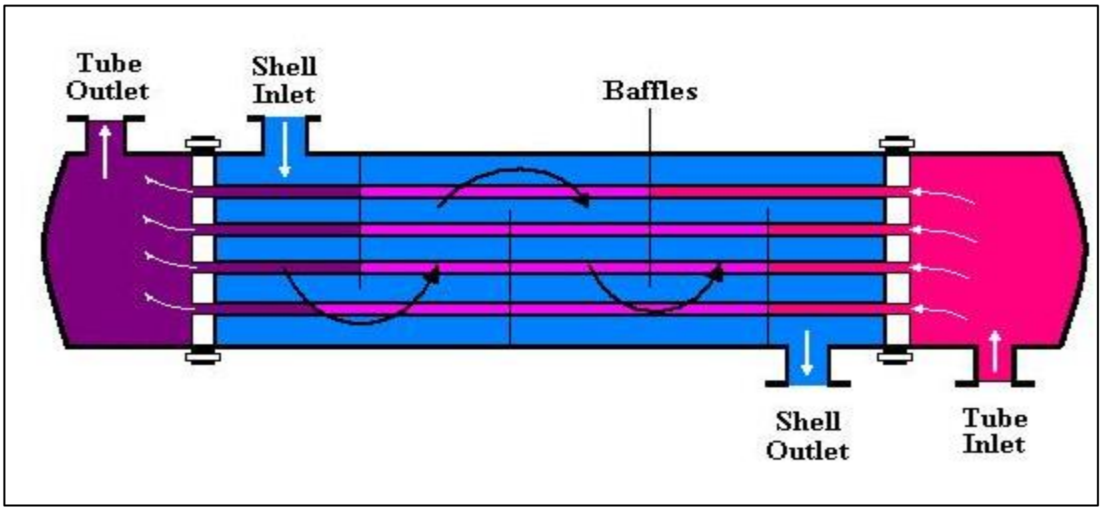


Figure 1.6. Major components of shell and tube heat exchanger

A shell and tube heat exchanger consists of large number of tubes in a large pressure vessel. Figure 1.6 illustrates a typical unit that may be found in a petrochemical plant.

One type of fluid runs in the tubes and the other type of fluid runs in the shell. The tube may be of any type and are likely to be plain and longitudinal but less finned. The Shell and Tube heat exchanger consists of two fluids. The fluid may be liquid or gas depending on the temperature and the type of liquid. The liquids may enter from the same direction or different direction into the shell and tube heat exchanger.

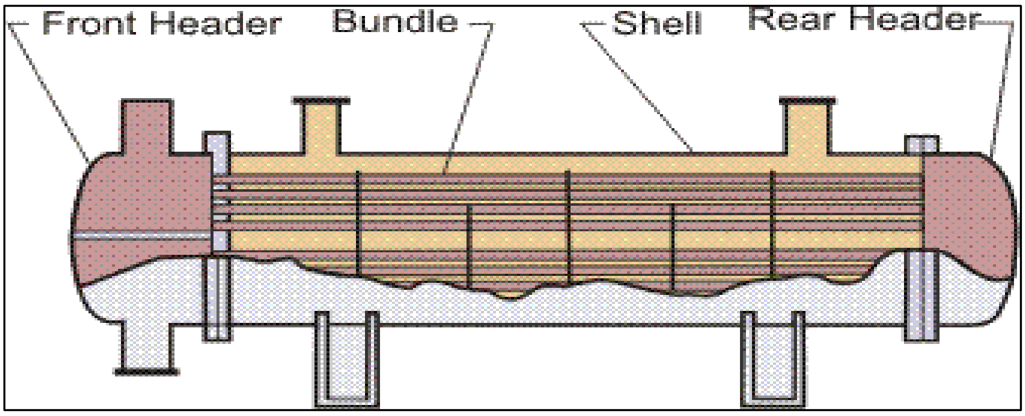


Figure 1.7. Major parts of shell and tube heat exchanger

The shell and tube exchanger consists of four major parts:

- Front Header—this is where the fluid enters the tube side of the exchanger. It is sometimes referred to as the Stationary Header.
- Rear Header—this is where the tube side fluid leaves the exchanger or where it is returned to the front header in exchangers with multiple tube side passes.
- Tube bundle—this comprises of the tubes, tube sheets, baffles and tie rods etc. to hold the bundle together.
- Shell—this contains the tube bundle.

The heat is transferred from the tubes inside the shell and the fluid in the shell. The more will be the efficiency of the shell and tube heat exchanger with more number of tubes. The other reason to use more number of tubes in that it reduces the heat transfer waste. This is very efficient and can be used in many situations. When the flow of fluids is from the same direction, more heat transfer takes place but when the flow of fluids is from opposite directions, less heat transfer takes place.

There are two types of heat exchangers:

1. Heat Exchangers with one phase
2. Heat Exchangers with two phases

The heat exchangers with one phase can consist of any fluid that is liquid or gas. The heat transfer cannot take place when the conversion of the phase takes place, i.e., from liquid to gas or gas to liquid. But in the heat exchangers with two phases, the heat transfer will be more efficient than the first one and heat transfer can also take place by converting the water to gas by boiling or converting the liquid to gas by heating.

The shell and tube heat exchanger will also be U shape and consist of inlet and outlet at the same point but two points. The shell consists of the inlet at the opposite direction as the tubes. When the fluid moves in the tubes bundle, the other fluid moves from the shell side. The heat exchange occurs between them efficiently if it is a heat exchanger with two phase and shell and tube heat exchanger.

The heat exchanger may also be the straight tube heat exchanger with one pass tube side and the straight tube heat exchanger with two pass tube side. The selection of material must be in such a way that more heat transfer takes place, i.e., the material should have more thermal conductivity.

Some common heat exchanger terms are:

1. Tube side: Inside the tubes
2. Shell side: Outside the tubes, between the tubes and the shell
3. Tube sheet: A thick plate provided with holes (one per tube) in which the tubes are fixed
4. Tube bundle: Consists of tubes, tube sheet and baffle plates
5. Shell: A cylinder of plate in which the tube bundle is placed

1.6.1. Flow Arrangement

1.6.1.1. Cross Flow

The cold and the hot fluid flow axis is at an angle to each other and hence, the fluids cross each other in this arrangement. The most common type of crossed flow exchanges has the angle between axes as 90 degrees.

1.6.1.2. Counter-current Flow

In this setup, the hot fluid enters from one end of the exchanger and the cold from the opposite end. This results in nearly constant temperature difference between the hot and the cold fluid. This is a significant aspect and makes counter current exchangers preferable over co-current exchangers.

1.6.1.3. Co-current (Parallel) Flow

As the name suggests, the flow of the hot and the cold fluid is taking place in the same direction in this case.

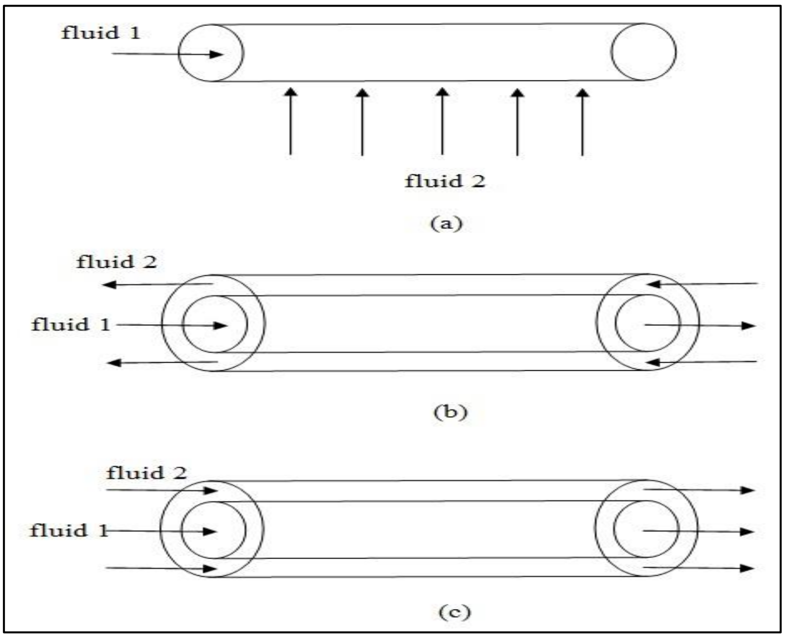


Figure 1.8. Orientation of fluid stream in heat exchanger (a) cross flow (b) counter-current flow (c) parallel flow

The heat exchanger as shown in figure 1.9 consists of a bundle of tubes properly secured at either ends in tube sheets. The tube sheets are drilled plates into which the tubes are fixed up using different technique to have leak proof joints. The entire tube bundles shown in the figure 1.10 (a) is placed inside a closed shell, which seals around the tube sheet periphery to form the two immiscible zones for hot and cold fluids are shown in figure 1.10 (b).

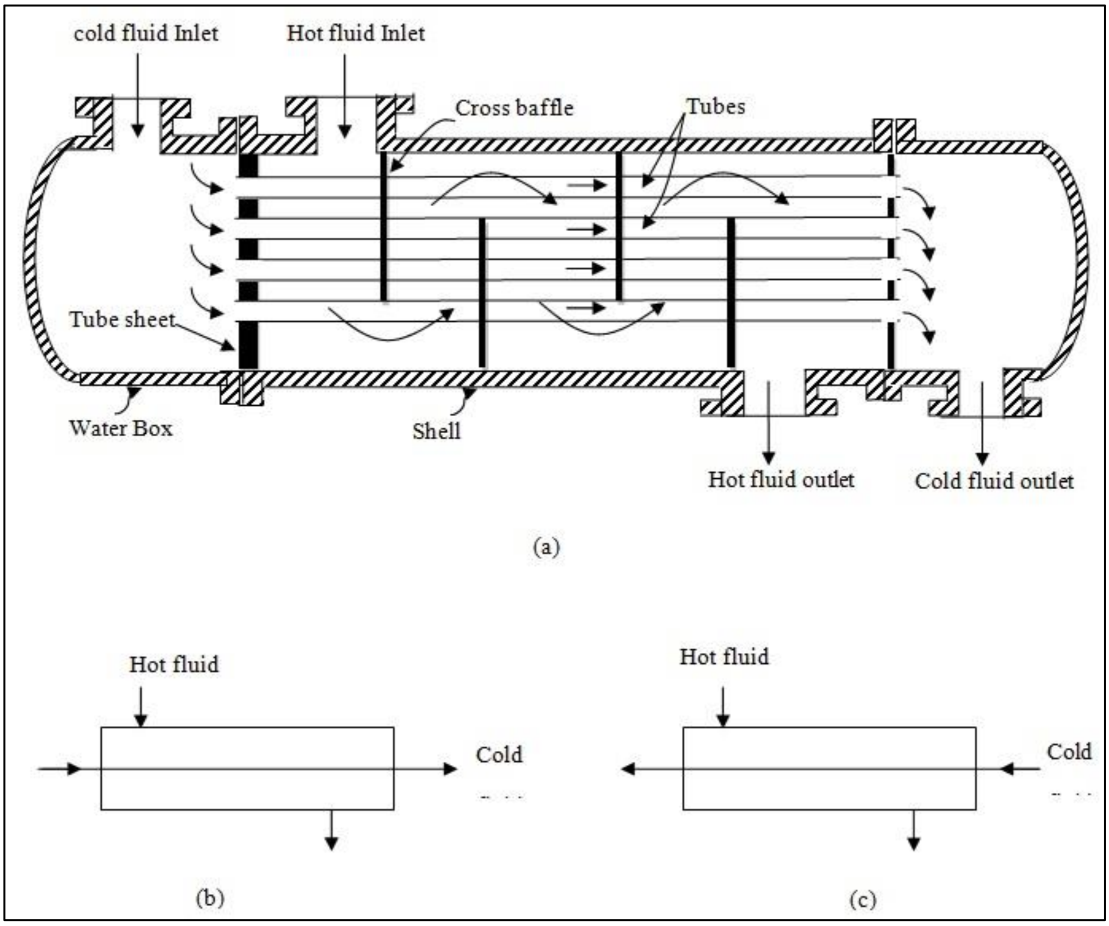


Figure 1.9. A schematic of (a) one-shell pass, one tube pass heat exchanger (b) parallel flow (c) counter flow

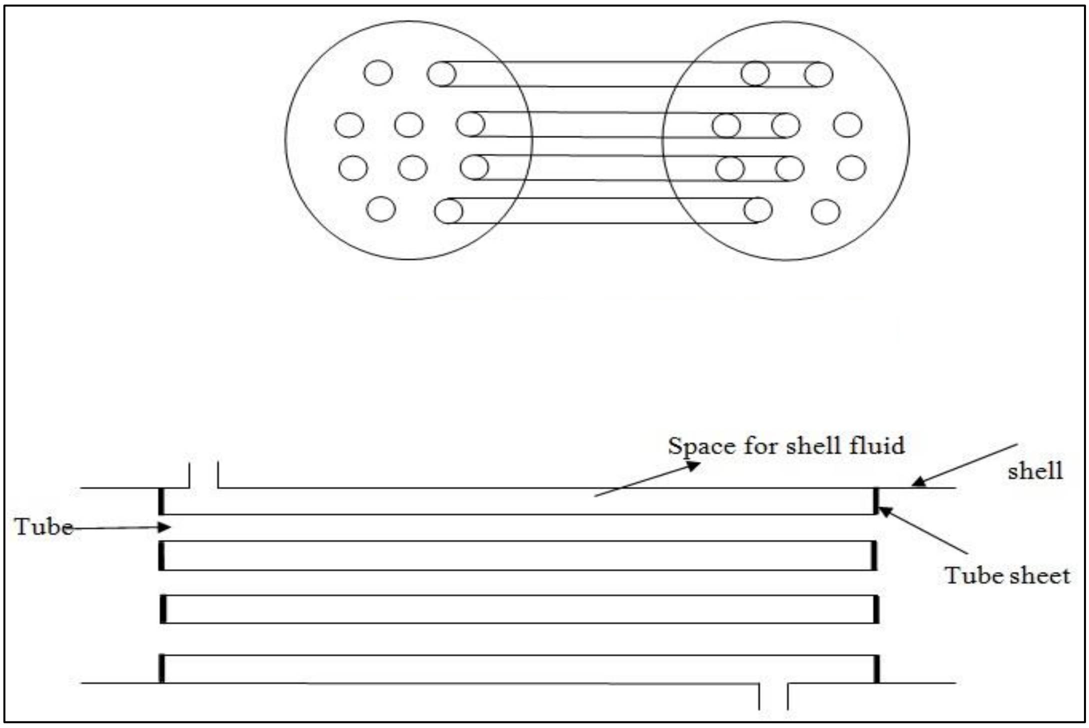


Figure 1.10. (a) Tube bundle fitted in two sheets (b) Tube bundle inside a shell

One fluid flows through the tubes while the other fluid flows around the outside of the tubes, it is the space between the tube sheets and enclosed by the outer shell.

For a thorough distribution of the shell side fluid, baffles are placed normal to the tube bundle. This baffle creates turbulence in the shell side fluid and enhances the transfer coefficients for the shell side flow.

Figure 1.9 shows the simplified diagram of a shell and tube heat exchanger, showing a few of the important components. In fact, the present heat exchanger used in the process industry are quite complex and have improved design such as factors for thermal expansion stresses, tube fouling due to contaminated fluids, ease of assembly and disassembly, size, weight, etc.

The heat exchanger shown in figure 1.9 is having one shell and one tube pass since both the shell and tube side fluid make a single traverse through the heat exchanger. Thus, this type of shell-and-tube heat exchangers is designated as 1-1 exchanger. If we desire to pass the tube fluid twice, then it is designated as 1-2 exchangers. Similarly, if there are 2 shell pass and 4 tube pass, the designation will be 2-4 exchanger. The number of pass in tube side is done by the pass partition plate. A pass particular plate or pass divider is shown in figure 1.11 (a). The shell side pass can be created by a flat plate as shown in figure 1.11 (b).

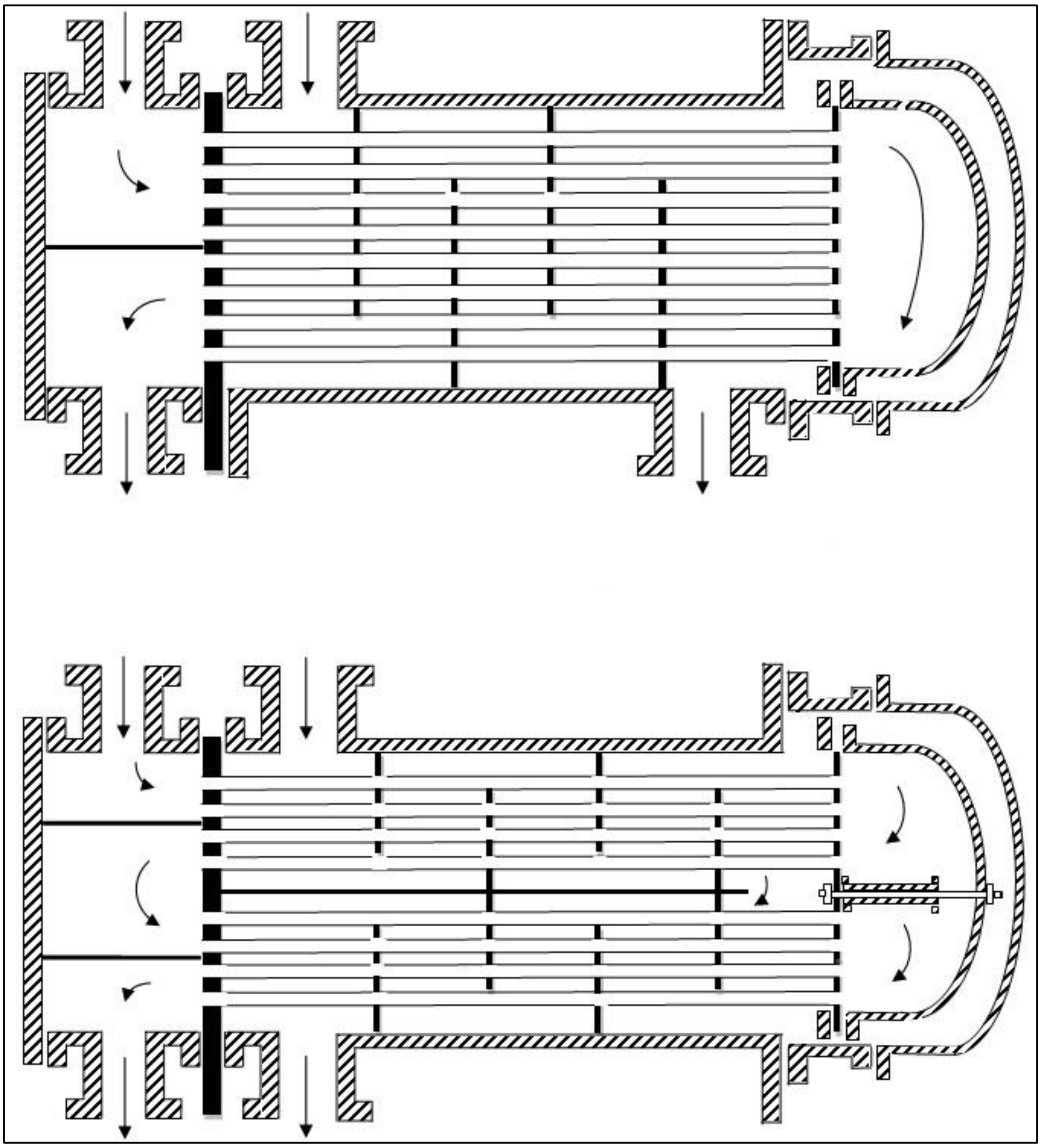


Figure 1.11. (a) 1-2 exchanger showing pass partition plate (b) 2-4 exchanger showing shell and tube passes

It can be understood that for a given number of tubes; the area available for flow of the tube side fluid is inversely proportional to the number of passes. Thus, on increasing the pass the area reduces and as a result the velocity of fluid in the tube increases and henceforth the Reynolds number increases. It would result in increased heat transfer coefficient but at the expense of high pressure drop. Generally, even numbers of tube passes are used for the multi-pass heat exchangers.

Figure 1.9 shows some baffles. These baffles (or shell-side baffles) are a metal plate usually in the form of the segment of a circular disc having holes to accommodate tubes. Shell-side baffles have two functions. The first is to create turbulence in the shell side fluid by changing the flow pattern parallel or cross flow to the tube bundles and thus increases the shell side heat transfer coefficient. The second major function of these baffles is to support the tube all along its length otherwise the tube may bend. Moreover, these baffles may have horizontal or vertical cuts (segmental baffle) as shown in figure 1.12.

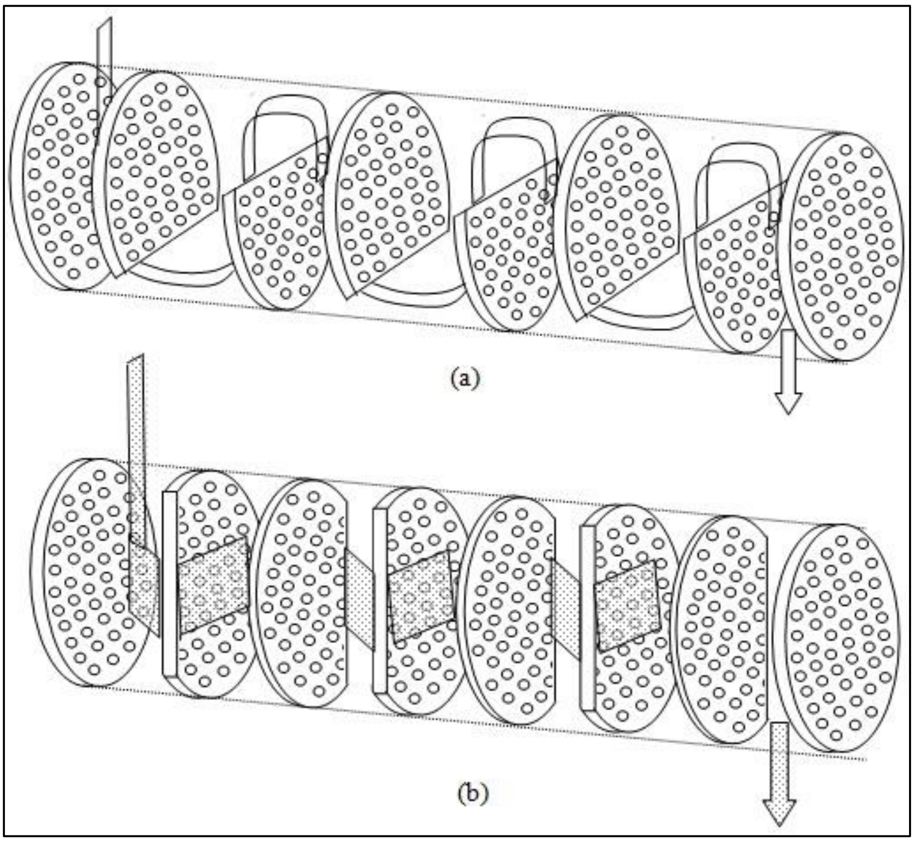


Figure 1.12. Baffles (a) horizontal cut baffles (b) vertical cut baffles

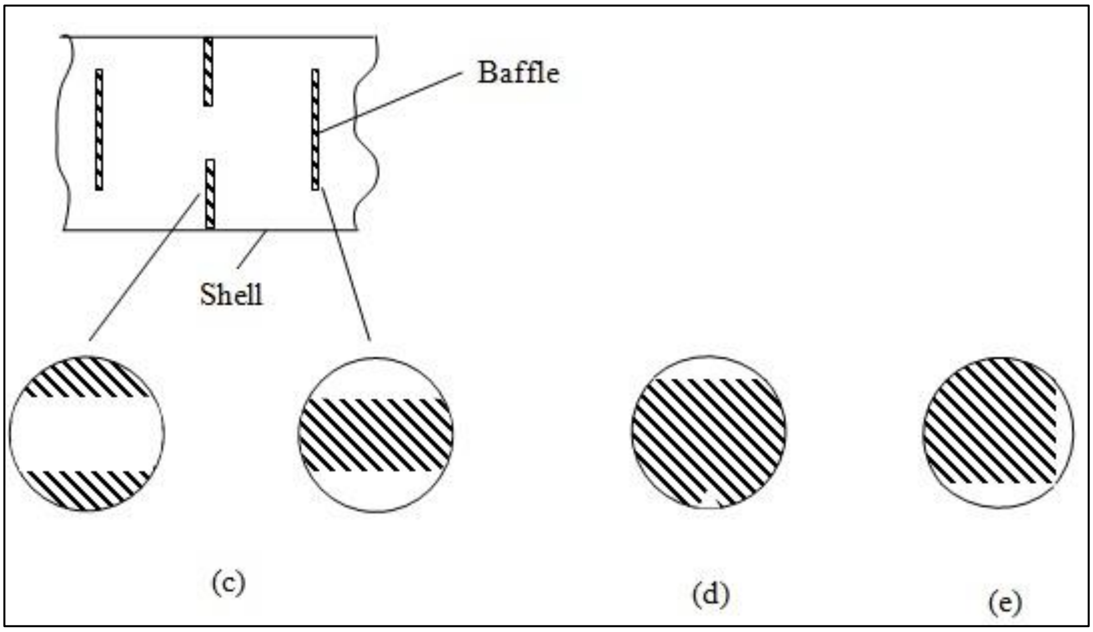


Figure 1.13. (c, d, e) The shaded region show the baffle area

The cut portion of the baffle, which is called baffle window, provides the area for flow of the shell fluid. The baffle window area ranges from 15% to 50%. At 25% cut segmental baffle means that the area of the cut-out portion is 25% of the area of the baffle. The spacing between the baffles is an important aspect. A larger baffles spacing reduces the shell side pressure drop but at the same time decreases the turbulence and heat transfer coefficient. Smaller baffle spacing increases the turbulence and heat transfer coefficient. However, the pressure drop may increase significantly, thus the advantage attained due to the higher heat transfer coefficient may be nullified. Therefore baffle spacing is selected considering the allowable shell side pressure drop and the heat transfer coefficient desired. A rule of thumb is that the minimum spacing of segmental baffles is one by fifth of the shell diameter or 5 cm, whichever is larger.

1.6.2. Geometric Terminology

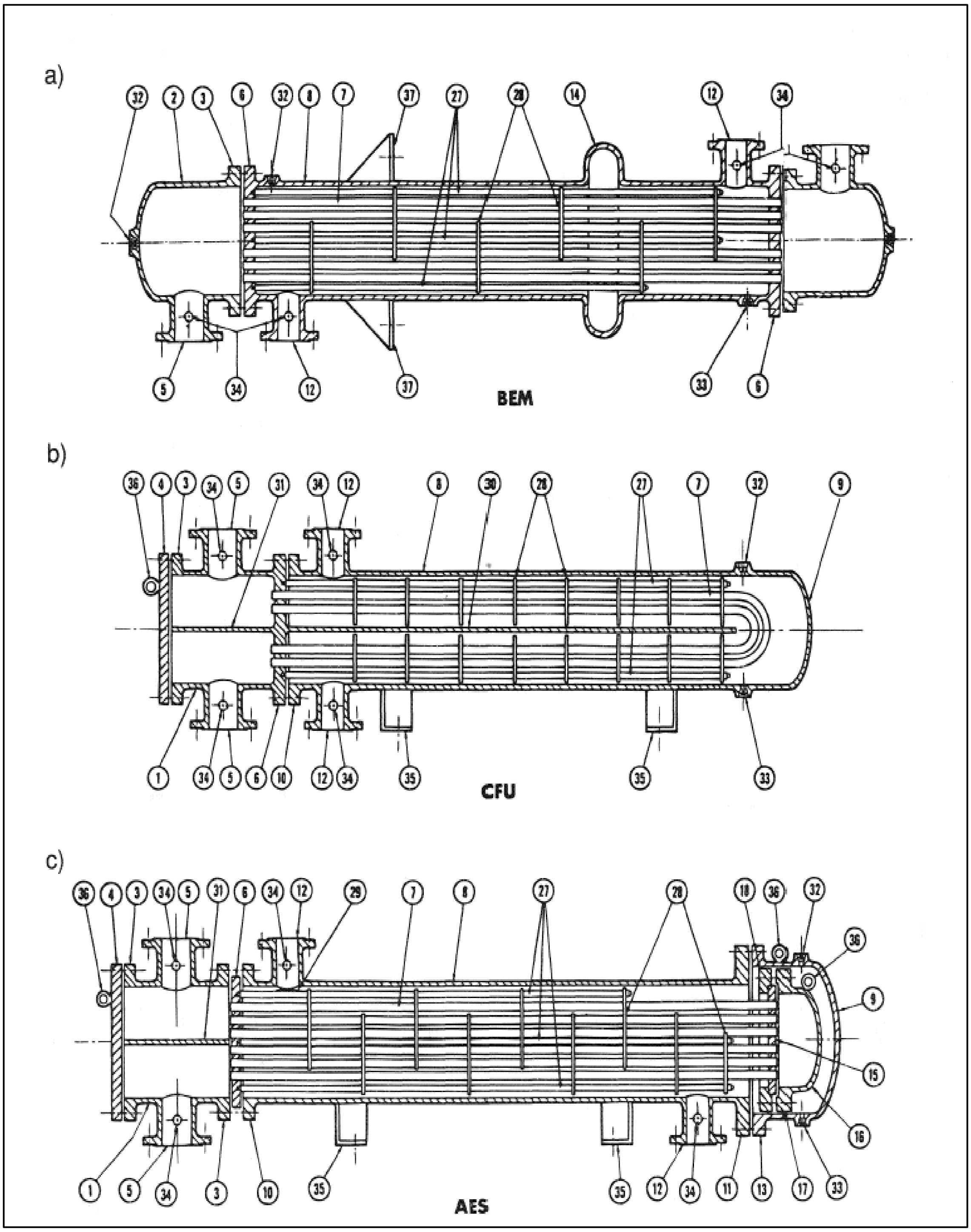


Figure 1.14. Type BEM, CFU and AES exchangers (©1988 by TEMA)

Table 1.1. Shell and tube geometric terminology

1	Stationary (Front) Head-Channel	20	Slip-on Backing Flange
2	Stationary (Front) Head-Bonnet	21	Floating Tube sheet Skirt
3	Stationary (Front) Head Flange	22	Floating Tube sheet Skirt
4	Channel Cover	23	Packing Box Flange
5	Stationary Head Nozzle	24	Packing
6	Stationary Tube sheet	25	Packing Follower Ring
7	Tubes	26	Lantern Ring
8	Shell	27	Tie Rods and Spacers
9	Shell Cover	28	Transverse Baffles or Support Plates
10	Shell Flange-Stationary Head End	29	Impingement Baffle or Plate
11	Shell Flange-Rear Head End	30	Longitudinal Baffle
12	Shell Nozzle	31	Pass Partition
13	Shell Cover Flange	32	Vent Connection
14	Expansion Joint	33	Drain Connection
15	Floating Tube sheet	34	Instrument Connection
16	Floating Head Cover	35	Support Saddle
17	Floating Head Flange	36	Lifting Lug
18	Floating Head Backing Device	37	Support Bracket
19	Split Shear Ring		

1.6.2.1. TEMA Designations

The popularity of shell and tube exchangers has resulted in a standard nomenclature being developed for their designation and use by the Tubular Exchanger Manufactures Association (TEMA). This nomenclature is defined in terms letters and diagrams. The first letter describes the front header type, the second letter the shell type and the third letter the rear header type. Figure 1.14 shows examples of a BEM, CFU, and AES exchangers while figure 1.15 illustrates the full TEMA nomenclature.

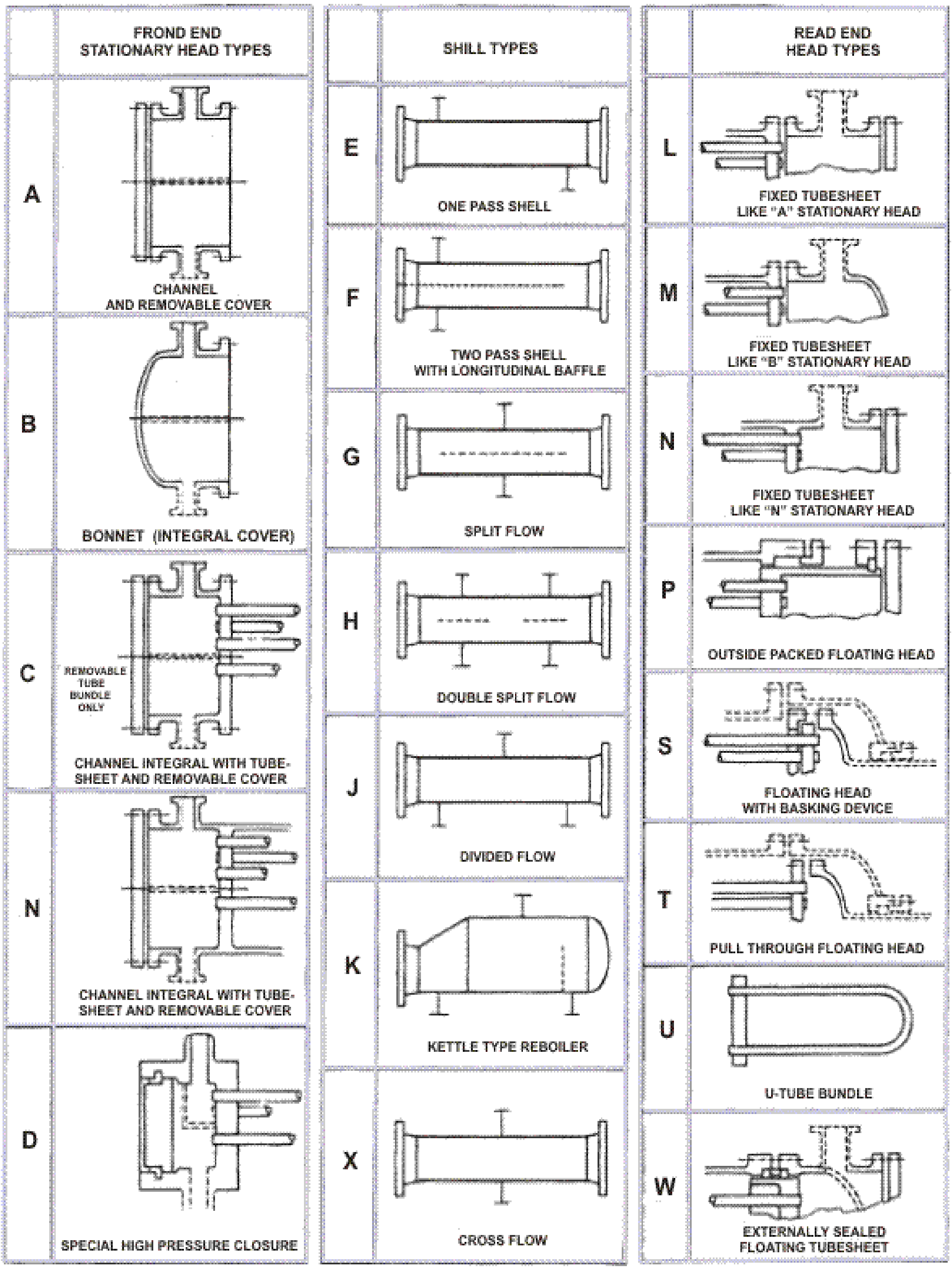


Figure 1.15. TEMA nomenclature (©1988 by TEMA)

Many combinations of front header, shell and rear header can be made. The most common combinations for an E-Type Shell are given in table 1.2 but other combinations are also used.

Table 1.2. Common combinations for E-type shell

Fixed tube sheet exchangers	U-tube exchangers	Floating head exchangers
AEL	AEU	AES
AEM	CEU	BES
AEN	DEU	
BEL		
BEM		
BEN		

Essentially there are three main combinations:

1. Fixed tube sheet exchangers
2. U-tube exchangers
3. Floating header exchangers

1. Fixed Tube sheet Exchanger (L, M, and N Type Rear Headers)

In a fixed tube sheet exchanger, the tube sheet is welded to the shell. This results in a simple and economical construction and the tube bores can be cleaned mechanically or chemically. However, the outside surfaces of the tubes are inaccessible except to chemical cleaning.

If large temperature differences exist between the shell and tube materials, it may be necessary to incorporate an expansion bellows in the shell, to eliminate excessive stresses caused by expansion. Such bellows are often a source of weakness and failure in operation. In circumstances where the consequences of failure are particularly grave U-Tube or Floating Header units are normally used.

This is the cheapest of all removable bundle designs, but is generally slightly more expensive than a fixed tube sheet design at low pressures.

2. U-Tube Exchangers

In a U-Tube exchanger any of the front header types may be used and the rear header is normally M-Type. The U-tubes permit unlimited thermal expansion, the tube bundle can be removed for cleaning and small bundle to shell clearances can be achieved. However, since internal cleaning of the tubes by mechanical means is difficult, it is normal only to use this type where the tube side fluids are clean.

3. Floating Head Exchanger (P, S, T and W Type Rear Headers)

In this type of exchanger the tube sheet at the Rear Header end is not welded to the shell but allowed to move or float. The tube sheet at the Front Header (tube side fluid inlet end) is of a larger diameter than the shell and is sealed in a similar manner to that used in the fixed tube sheet design. The tube sheet at the rear header end of the shell is of slightly smaller diameter than the shell, allowing the bundle to be pulled through the shell. The use of a floating head means that thermal expansion can be allowed for and the tube bundle can be removed for cleaning. There are several rear header types that can be used but the S-Type Rear Head is the most popular. A floating head exchanger is suitable for the rigorous duties associated with high temperatures and pressures but is more expensive (typically of order of 25% for carbon steel construction) than the equivalent fixed tube sheet exchanger.

Considering each header and shell type in turn:

1. A-Type front header

This type of header is easy to repair and replace. It also gives access to the tubes for cleaning or repair without having to disturb the pipe work. It does however have two seals (one between the tube sheet and header and the other between the header and the end plate). This increases the risk of leakage and the cost of the header over a B-Type Front Header.

2. B-Type front header

This is the cheapest type of front header. It also is more suitable than the A-Type Front Header for high pressure duties because the header has only one seal. A disadvantage is that to gain access to the tubes requires disturbance to the pipe work in order to remove the header.

3. C-Type front header

This type of header is for high pressure applications (>100 bar). It does allow access to the tube without disturbing the pipe work but is difficult to repair and replace because the tube bundle is an integral part of the header.

4. D-Type front header

This is the most expensive type of front header. It is for very high pressures (> 150 bar). It does allow access to the tubes without disturbing the pipe work but is difficult to repair and replace because the tube bundle is an integral part of the header.

5. N-Type front header

The advantage of this type of header is that the tubes can be accessed without disturbing the pipe work and it is cheaper than an A-Type Front Header. However, they are difficult to maintain and replace as the header and tube sheet are an integral part of the shell.

6. Y-Type front header

Strictly speaking this is not a TEMA designated type but is generally recognized. It can be used as a front or rear header and is used when the exchanger is to be used in a pipe line. It is cheaper than other types of headers as it reduces piping costs. It is mainly used with single tube pass units although with suitable partitioning any odd number of passes can be allowed.

7. E-Type shell

This is most commonly used shell type, suitable for most duties and applications. Other shell types only tend to be used for special duties or applications.

8. F-Type shell

This is generally used when pure countercurrent flow is required in a two tube side pass unit. This is achieved by having two shells side passes—the two passes being separated by a longitudinal baffle. The main problem with this type of unit is thermal and hydraulic leakage across this longitudinal baffle unless special precautions are taken.

9. G-Type shell

This is used for horizontal thermosiphon reboilers and applications where the shell side pressure drop needs to be kept small. This is achieved by splitting the shell side flow.

10. H-Type shell

This is used for similar applications to G-Type Shell but tends to be used when larger units are required.

11. J-Type shell

This tends to be used when the maximum allowable pressure drop is exceeded in an E-Type Shell even when double segmental baffles are used. It is also used when tube vibration is a problem. The divided flow on the shell side reduces the flow velocities over the tubes and hence reduces the pressure drop and the likelihood of tube vibration. When there are two inlet nozzles and one outlet nozzle this is sometimes referred to as an I-Type Shell.

12. K-Type shell

This is used only for reboilers to provide a large disengagement space in order to minimize shell side liquid carry over. Alternatively a K-Type Shell may be used as a chiller. In this case the main process is to cool the tube side fluid by boiling a fluid on the shell side.

13. X-Type shell

This is used if the maximum shell side pressure drop is exceeded by all other shell and baffle type combinations. The main applications are shell side condensers and gas coolers.

14. L-Type rear header

This type of header is for use with fixed tube sheets only, since the tube sheet is welded to the shell and access to the outside of the tubes is not possible. The main advantages of this type of header are that access can be gained to the inside of the tubes without having to remove any pipework and the bundle to shell clearances are small. The main disadvantage is that a bellows or an expansion roll are required to allow for large thermal expansions and this limits the permitted operating temperature and pressure.

15. M-Type rear header

This type of header is similar to the L-Type Rear Header but it is slightly cheaper. However, the header has to be removed to gain access to the inside of the tubes. Again, special measures have to be taken to cope with large thermal expansions and this limits the permitted operating temperature and pressure.

16. N-Type rear header

The advantage of this type of header is that the tubes can be accessed without disturbing the pipe work. However, they are difficult to maintain and replace since the header and tube sheet are an integral part of the shell.

17. P-Type rear header

This is an outside packed floating rear header. It is, in theory, a low cost floating head design which allows access to the inside of the tubes for cleaning and also allows the bundle to be removed for cleaning. The main problems with this type of header are:

- large bundle to shell clearances required in order to pull the bundle
- it is limited to low pressure nonhazardous fluids, because it is possible for the shell side fluid to leak via the packing rings
- only small thermal expansions are permitted

In practice it is not a low cost design, because the shell has to be rolled to small tolerances for the packing to be effective.

18. S-Type rear header

This is a floating rear header with backing device. It is the most expensive of the floating head types but does allow the bundle to be removed and unlimited thermal expansion is possible. It also has smaller shell to bundle clearances than the other floating head types. However, it is difficult to dismantle for bundle pulling and the shell diameter and bundle to shell clearances are larger than for fixed head type exchangers.

19. T-Type rear header

This is a pull through floating head. It is cheaper and easier to remove the bundle than with the S-Type Rear Header, but still allows for unlimited thermal expansion. It does, however, have the largest bundle to shell clearance of all the floating head types and is more expensive than fixed header and U-tube types.

20. U-tube

This is the cheapest of all removable bundle designs, but is generally slightly more expensive than a fixed tube sheet design at low pressures. However, it permits unlimited thermal expansion, allows the bundle to be removed to clean the outside of the tubes, has the tightest bundle to shell clearances and is the simplest design. A disadvantage of the U-tube design is that it cannot normally have pure counter-flow unless an F-Type Shell is used. Also, U-tube designs are limited to even numbers of tube passes.

21. W-Type rear header

This is a packed floating tube sheet with lantern ring. It is the cheapest of the floating head designs, allows for unlimited thermal expansion and allows the tube bundle to be removed for cleaning. The main problems with this type of head are:

- the large bundle to shell clearances required to pull the bundle and
- the limitation to low pressure nonhazardous fluids (because it is possible for both the fluids to leak via the packing rings)

It is also possible for the shell and tube side fluids to become mixed if leakage occurs.

1.6.3. Geometric Options

1. Tube diameter layout and pitch

Tubes may range in diameter from 12.7 mm (0.5 in) to 50.8 mm (2 in), but 19.05 mm (0.75 in) and 25.4 mm (1 in) are the most common sizes. The tubes are laid out in triangular or square patterns in the tube sheets. See figure 1.16.

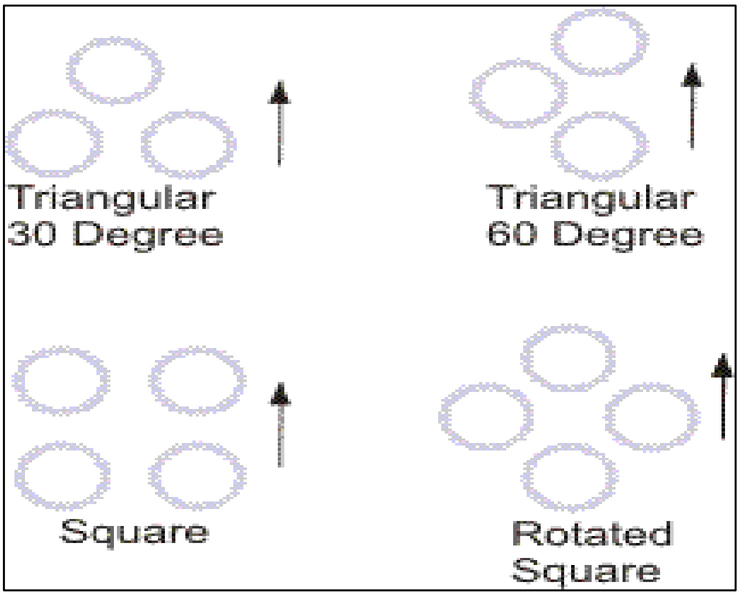


Figure 1.16. Tube layouts

The square layouts are required where it is necessary to get at the tube surface for mechanical cleaning. The triangular arrangement allows more tubes in a given space. The tube pitch is the shortest center-to-center distance between tubes. The tube spacing is given by the tube pitch/tube diameter ratio, which is normally 1.25 or 1.33. Since a square layout is used for cleaning purposes, a minimum gap of 6.35 mm (0.25 in) is allowed between tubes.

2. Baffle Types

Baffles are installed on the shell side to give a higher heat-transfer rate due to increased turbulence and to support the tubes thus reducing the chance of damage due to vibration. There are a number of different baffle types, which support the tubes and promote flow across the tubes. Figure 1.17 shows the following baffle arrangements:

- Single Segmental (this is the most common)
- Double Segmental (this is used to obtain a lower shell side velocity and pressure drop)
- Disc and Doughnut

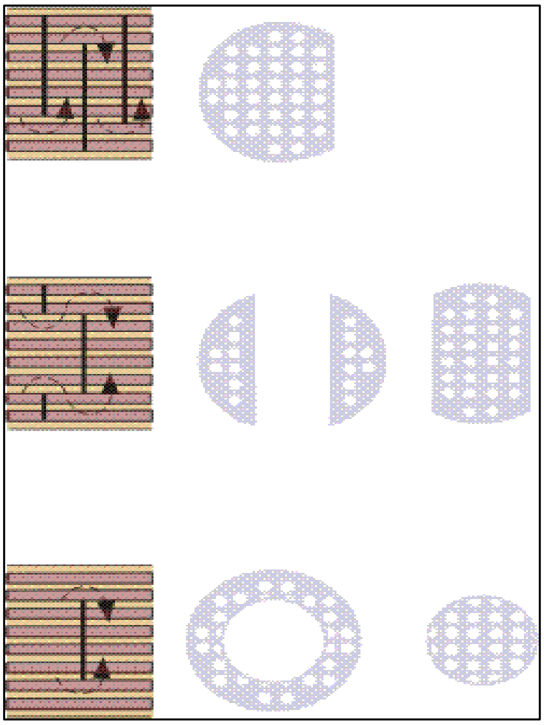


Figure 1.17. Baffle arrangements

The center-to-center distance between baffles is called the baffle-pitch and this can be adjusted to vary the cross flow velocity. In practice the baffle pitch is not normally greater than a distance equal to the inside diameter of the shell or closer than a distance equal to one-fifth the diameter or 50.8 mm (2 in) whichever is greater. In order to allow the fluid to flow backwards and forwards across the tubes part of the baffle is cut away. The height of this part is referred to as the baffle-cut and is measured as a percentage of the shell diameter, e.g., 25 per cent baffle-cut. The size of the baffle-cut (or baffle window) needs to be considered along with the baffle pitch. It is normal to size the baffle-cut and baffle pitch to approximately equalize the velocities through the window and in cross flow, respectively.

There are two main types of baffle which give longitudinal flow:

- Orifice Baffle
- Rod Baffle

In these types of baffle the turbulence is generated as the flow crosses the baffle.

1.6.4. Heat Transfer Enhancement Devices

There are three main types:

1. Special surfaces

These tend to be used to promote nucleate boiling when the temperature driving force is small.

2. Tube inserts

These are normally wire wound inserts or twisted tapes. They are normally used with medium to high viscosity fluids to improve heat transfer by increasing turbulence. There is also some evidence that they reduce fouling. In order to use these most effectively the exchanger should be designed for their use. This usually entails increasing the shell diameter, reducing the tube length and the number of tube side passes in order to allow for the increased pressure loss characteristics of the devices.

3. Extended surfaces

These are used to increase the heat transfer area when a stream has a low heat transfer coefficient. The most common type is “low fin tubing” where typically the fins are 1.5 mm high at 19 fins per inch.

1.6.5. Fluid Allocation

When deciding which side to allocate the hot and cold fluids the following need to be taken into account, in order of priority:

1. Consider any and every safety and reliability aspect and allocate fluids accordingly. Never allocate hazardous fluids such they are contained by anything other than conventional bolted and gasketted—or welded—joints.
2. Ensure that the allocation of fluids complies with established engineering practices, particularly those laid down in customer specifications.
3. Having complied with the above, allocate the fluid likely to cause the most severe mechanical cleaning problems (if any) to the tube side.
4. If neither of the above are applicable, the allocation of the fluids should be decided only after running two alternative designs and selecting the cheapest (this is time

consuming if hand calculations are used but programs such as TASC from the Heat Transfer and Fluid Flow Service (HTFS) make this a trivial task).

1.6.6. Selection of Exchanger Geometry

1. Tube outside diameter

For the process industry, 19.05 mm (3/4") tends to be the most common.

2. Tube wall thickness

Reference must be made to a recognized pressure vessel code to decide this.

3. Tube length

For a given surface area, the longer the tube length the cheaper the exchanger, although a long thin exchanger may not be feasible.

4. Tube layout

45 or 90 degree layouts are chosen if mechanical cleaning is required, otherwise a 30 degree layout is often selected, because it provides a higher heat transfer and hence smaller exchanger.

5. Tube pitch

The smallest allowable pitch of 1.25 times the tube outside diameter is normally used unless there is a requirement to use a larger pitch due to mechanical cleaning or tube end welding.

6. Number of tube passes

This is usually one or an even number (not normally greater than 16). Increasing the number of passes increases the heat transfer coefficient but care must be taken to ensure that the tube side ρv^2 is not greater than about 10,000 $kg/m \cdot s^2$.

7. Shell diameter

Standard pipe is normally used for shell diameters up to 610 mm (24"). Above this the shell is made from rolled plate. Typically shell diameters range from 152 mm to 3000 mm (6" to 120").

8. Baffle type

Single segmental baffles are used by default but other types are considered if pressure drop constraints or vibration is a problem.

9. Baffle spacing

This is decided after trying to balance the desire for increased cross flow velocity and tube support (smaller baffle pitch) and pressure drop constraints (larger baffle pitch). TEMA provides guidance on the maximum and minimum baffle pitch.

10. Baffle cut

This depends on the baffle type but is typically 45% for single segmental baffles and 25% for double segmental baffles.

11. Nozzles and impingement

For shell side nozzles the ρv^2 should not be greater than about 9000 $kg/m \cdot s^2$. For tube side nozzles the maximum ρv^2 should not exceed 2230 $kg/m \cdot s^2$ for noncorrosive, nonabrasive single phase fluids and 740 $kg/m \cdot s^2$ for other fluids. Impingement protection is always required for gases which are corrosive or abrasive, saturated vapors and two phase mixtures. Shell or bundle entrance or exit areas should be designed such that a ρv^2 of 5950 $kg/m \cdot s^2$ is not exceeded.

12. Material of Construction

In general, shell and tube exchangers are made of metal, but for specialist applications (e.g., involving strong acids or pharmaceuticals), other materials such as graphite, plastic and glass may be used.

1.7. DIMENSIONLESS NUMBERS

In the following, some of the most important dimensionless numbers which have been used in this project will be introduced.

1.7.1. Reynolds Number

Re is the ratio of inertial force to viscous force in a fluid. It is defined as:

$$Re = \frac{\rho V d}{\mu} = \frac{V d}{\nu}$$

where,

ρ = Density of fluid, kg/m^3

V = Velocity of flow, m/s

L = Characteristic length, m

μ = Dynamic viscosity, kg/m·s

ν = Kinematic viscosity, m^2/s

The characteristic length is dependent upon the geometry of flow (Internal or external flow, geometry of pipe), for instance, for a closed tube or duct we have internal flow with the hydraulic diameter, D_h as the L. D_h is defined as the ratio of 4 times the area over the periphery which is the diameter of a circular pipe or the length of a square in a square duct as characteristic length.

$$D_h = \frac{4A}{P}$$

For a circular pipe:

$$D_h = \frac{4A}{P} = \frac{4\pi r^2}{2\pi r} = 2r = d$$

For a square shaped cross section duct:

$$D_h = \frac{4A}{P} = \frac{4a^2}{4a} = a$$

The lower Re shows more laminarity for the flow and the higher Re shows more turbulence for the flow which has different rules and correlations. Between these two regions, there is a transition region. If we have developing flows or entry region the definitions of friction factor will be different.

1.7.2. Nusselt Number

Nu is the ratio of convective heat transfer over conductive heat transfer. So the bigger the Nu, the more convection there is compared to conduction. That means the fluid has more capacity to transfer heat through convection which is the dominant heat transfer mechanism in liquids and gases than the conduction which is more dominant in solid materials.

$$Nu = \frac{hD}{k}$$

1.7.3. Prandtl Number

Prandtl number is another dimensionless number which is the ratio of viscous diffusion rate over thermal diffusion rate.

$$Pr = \frac{\nu}{\alpha} = \frac{C_p \mu}{k}$$

where,

ν = Kinematic viscosity, m^2/s

α = Thermal diffusivity, m^2/s

μ = Dynamic viscosity, kg/m·s

c_p = Specific heat, J/kg·˚C

k = Thermal conductivity, W/m·˚C

Higher Prandtl number shows that momentum diffusivity dominates and lower Prandtl number shows that thermal diffusivity dominates. For instance, Mercury is a metal with high thermal conductivity which has dominant conductivity so Pr number is small for mercury.

For fluids with almost $Pr = 1$ such as gases, thermal and velocity boundary layers coincide with each other.

1.8. FLOW REGIMES

1.8.1. Laminar Flow

There are various definitions for flow regimes and range of Reynolds numbers. For instance in an internal flow, $Re < 2100\text{-}2300$ accounts for laminar flow. $2100\text{-}2300 < Re < 4000$ depicts transition region and $Re > 4000$ outlines turbulent regime. The differences between these regions are in friction factor and relative roughness of pipes in turbulent flows which will have effects on pressure drop.

We know that when the viscosity increases (as the only variable while the other parameters kept constant) pressure drop increases. The criteria of this research are the transition from laminar to turbulent forced flow in a horizontal circular tube when the flows are fully developed.

For internal Flow of a circular pipe:

$Re < 2300$	Laminar Flow
$2300 < Re < 4000$	Transitional Flow
$Re > 4000$	Turbulent Flow

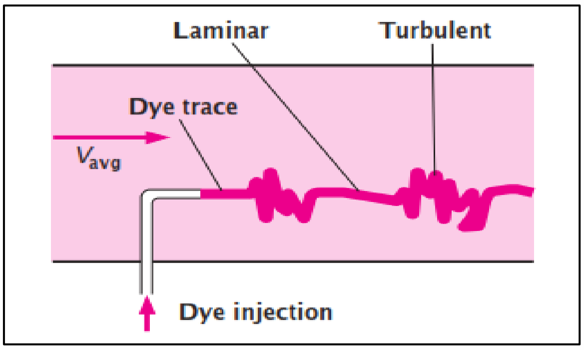


Figure 1.18. Trace of dye in laminar and turbulent flows

If the tube remains smooth and flow disturbance can be avoided, laminar flow can be achieved up to $Re = 100{,}000$ which is very hard to maintain in practice.

The characteristics of a laminar flow are as follows:

- Flow in layers parallel to boundary
- Low Re number
- From mixing point of view, it has small molecular diffusion
- Velocity profile is parabolic
- Shear stress is lower
- Solution to the flow is analytical

1.8.2. Turbulent Flow

The characteristics of a turbulent flow are as follows:

- Higher Re number
- Resulted by the intricate interaction between the inertia terms and viscous terms existing in the momentum equations
- Irregularity and randomness
- Difficult full deterministic approach so these flows are explained using statistics
- Always chaotic (All chaotic flows are not necessarily turbulent since a turbulent flow is diffusive as well)
- Diffusivity causes fast mixing and increased rates of momentum, heat, and mass transfer
- Rotational with 3 dimensional vortices
- Dissipation of the flow which is conversion of kinetic energy to heat because of viscous shear stresses

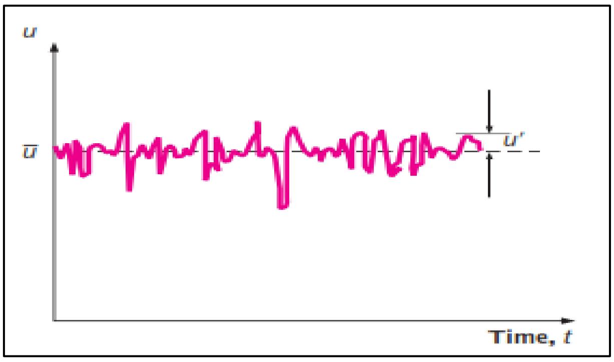


Figure 1.19. Fluctuation of velocities in a turbulent flow considering the mean velocity

The above figure shows the fluctuations in turbulent flow for the u as the velocity component with t as the time at a specified location.

1.8.3. Transitional Flow

It appears that the transition from laminar to turbulent flows is also dependent on the fluctuations in the flow, pipe vibrations and roughness of the pipe as it can affect the degree of disturbance of the flow by surface roughness. In the transitional flow region, the flow switches between laminar and turbulent randomly.

When one applies external disturbance, it can be observed that there are uneven fluctuations. Sporadic laminar and turbulent flow arise, i.e. phases take place that have characteristics of laminar flow and phases come up which have the attributes of turbulent flow.

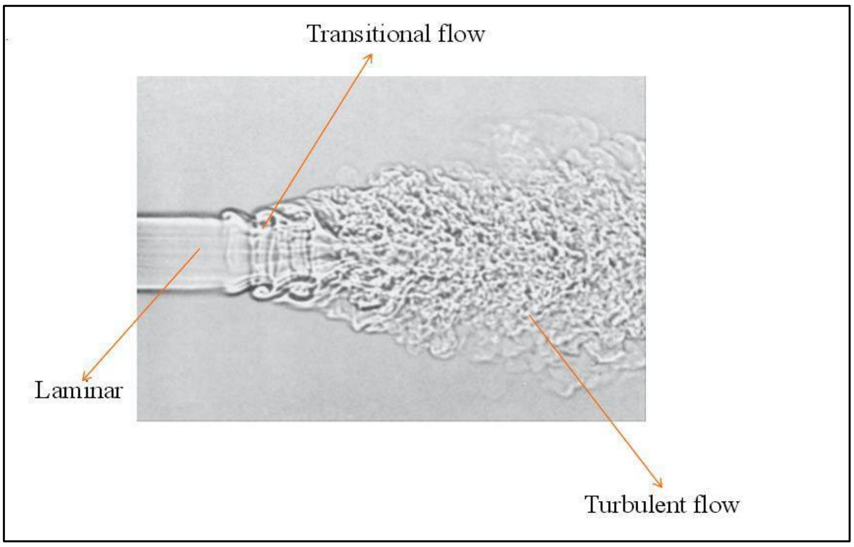


Figure 1.20. Areas of laminar, transitional and turbulent flow shown on a subsonic open jet

In a laminar flow, the velocity signal depicts a constant trend while in a turbulent flow, a time variation of the local velocity exists and one can observe the velocity fluctuations around the mean value.

We should note that the Re>2300 criterion is for circular pipes and is not the same for various pipe cross sections such as rectangular or flows surrounding a blade. Even for circular pipes 2300 is not necessarily absolute since this is dependant also on whether there is any disturbance present. This means that if the pipe is smooth and the experiment is done carefully we may reach higher degrees of Re in laminar region without the transition. On the other hand, if Re is less than 2300 no matter the disturbances exist the flow will be laminar.

In addition to what explained above, in transition region, for a fixed Reynolds number, friction factor increases with increasing Prandtl number.

1.9. DIFFERENT MECHANISMS FOR ENHANCEMENT OF HEAT TRANSFER

Generally, there are different mechanisms suggested for the anomalous enhancement of heat transfer like Brownian motion, interfacial layer theory, electrical double layer theory, aggregation and diffusion etc. which will subsequently be discussed.

1.9.1. Brownian motion

Brownian motion is a transport phenomenon in which small particles suspended in a stationary fluid move randomly, first observed by a Scottish botanist, Robert Brown in 1827 and this theory was developed by Einstein’s works, later on.

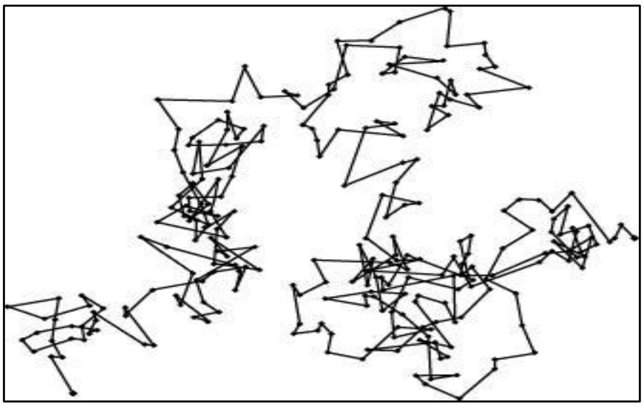


Figure 1.21. A schematic view of Brownian random movements of an assumed particle

Einstein's result (Einstein, 1905) for the diffusion coefficient D of a spherical particle which is a significant property of the motion of particles in the medium is defined as:

$$D = \frac{k_B T}{6\pi\mu r}$$

where,

k_B = Boltzmann constant, $m^2 \cdot kg/s^2 \cdot {}^\circ C$

T = Temperature, ˚C

μ = Viscosity of the medium, kg/m·s

r = Radius of the particle, m

It can be understood from above equation, the smaller the size of the particle the more diffusion that particle has. Diffusion of particles also increases with decline of viscosity and increase in temperature. That is why Brownian motion is effective on small scale systems especially in higher temperatures with low viscid fluids and as a general rule, viscosity decreases when temperature increases which leads to increase in diffusion.

1.9.2. Interfacial layer theory (Kapitza resistance)

The Kapitza resistance is a thermal boundary resistance that is a measure of an interface's resistance to thermal flow and is generated when thermal energy carrier at an interface is scattered like what happens to phonons and electrons. The kind of carrier scattered is dependent to the materials controlling what happens in the interfaces. When there are nanoparticle-base fluid interfaces, for instance in liquid-solid interfaces the Kapitza resistance is assured to decline consequently the overall thermal resistance of the nanofluid (or generally the system) will reduce.

1.9.3. Aggregation and diffusion

Aggregation and diffusion can be simply explained when nanoparticle chains formed together and make a linear assembly right after they are suspended in the base fluid. This chain assembly is assumed to escalate the heat propagation with faster thermal diffusion due to the providing a more rapid heat transfer path through the nanofluid.

1.9.4. Electrical double layer (EDL) theory

The mechanism explained by EDL theory suggests a way to boost the heat transfer of molecules by a slight shift in the strength of intermolecular interaction forces that effectively alters the mean free path of the nanoparticles.

1.10. NANOFLUIDS

1.10.1. Introduction to nanofluids

Nanofluids are dilute liquid suspended nanoparticles which have only one critical dimension smaller than ~100 nm. Much research work has been made in the past decade to this new type of material because of its high rated properties and behaviour associated with heat transfer, mass transfer. The thermal behaviour of nanofluids could provide a basis for a huge innovation for heat transfer, which is a major importance to number of industrial sectors including transportation, power generation, micro-manufacturing, thermal therapy for cancer treatment, chemical and metallurgical sectors, as well as heating, cooling, ventilation and air-conditioning. Nanofluids are also important for the production of nano structured materials, for the engineering of complex fluids, as well as for cleaning oil from surfaces due to their excellent wetting and spreading behaviour.

The concept of nanofluid is not new as in 1857 Michael Faraday first reported the study on the synthesis and colours of colloidal gold, but it was possible to put it into practice only after the tremendous development of nanotechnologies during the last decade [1]. Over the past decade, nanofluids, which are liquids containing suspensions of nanoparticles have been reported to possess substantially higher thermal conductivity than anticipated from the effective medium theories. This makes them very attractive for the usage as heat transfer fluids in many applications. For example, nanofluids would be useful as coolants in automobile and electronics industries. However, the reported high thermal conductivity sometimes cannot be reproduced, and the potential mechanisms leading to the enhancement are still under scrutiny [2].

Nature is full of nanofluids, like blood, a complex biological nanofluid where different nanoparticles (at molecular level) accomplish different functions, and functional components actively respond to their local environment. According to the types of liquids (organic and inorganic) and kinds of nanoparticles, one can get different types of nanofluids like process extraction nanofluids, environmental (pollution-controlling nanofluids), bio-, and pharmaceutical nanofluids. A new class of polymer nanofluids, drag-reducing nanofluids, aim at enhanced heat transfer, as well as, flow friction reduction. A wide range of active self-assembly mechanisms for nanoscale structures start from a suspension of nanoparticles in fluid. Addition of nanoparticles in liquid remarkably enhances energy transport process of the base liquid [3].

By suspending nanophase particles in heating or cooling fluids, the heat transfer performance of the fluid can be significantly improved. The main reasons may be listed as follows:

1. The suspended nanoparticles increase the surface area and the heat capacity of the fluid.
2. The suspended nanoparticles increase the effective (or apparent) thermal conductivity of the fluid.
3. The interaction and collision among particles, fluid and the flow passage surface are intensified.
4. The mixing fluctuation and turbulence of the fluid are intensified.
5. The dispersion of nanoparticles flattens the transverse temperature gradient of the fluid.

Various methods that have been done to enhance heat transfer, such as modifying surface roughness as turbulence promoter, flowing fluids through micro channels, and using nanofluids. In the past 20 years many researchers have been studying the properties of nanofluids, and it’s expected to be the next generation of heat transfer technology due to the better thermal performances compared to that of traditional heat transfer fluid [1]. Nanofluid can be defined as a fluid in which solid particles with the sizes under 100 nm are suspended and dispersed uniformly in a fluid. The base fluid used the same as traditional heat transfer fluids, e.g., water, oil, and ethylene glycol.

Many researchers observed the phenomenon of higher thermal conductivity of various nanofluids compared to that of the base fluids. However, differences between the results were observed, i.e., some showed that the increase of thermal conductivity of nanofluids is an anomaly that cannot be predicted by the existing conventional equation [4, 5]; while some others showed that the increase is not an anomaly and can be predicted by using the existing conventional equation [6].

The twenty-first century is an era of technological development and has already seen many changes in almost every industry. The introduction of nanoscience and technology is based on the famous phrase "There's Plenty of Room at the Bottom" by the Nobel Prize-winning physicist Richard Feynman in 1959. Feynman proposed this concept using a set of conventional-sized robot arms to construct a replica of themselves but one-tenth the original size then using that new set of arms to manufacture an even smaller set until the molecular scale is reached.

1.10.2. Use of nanofluids

The main goal or idea of using nanofluids is to attain highest possible thermal properties at the smallest possible concentrations (preferably<1% by volume) by uniform dispersion and stable suspension of nanoparticles (preferably<10 nm) in hot fluids. A nanofluid is a mixture of water and suspended metallic nanoparticles. Since the thermal conductivity of metallic solids are typically orders of magnitude higher than that of fluids it is expected that a solid/fluid mixture will have higher effective thermal conductivity compared to the base fluid. Nano fluids are extremely stable and exhibit no significant settling under static conditions, even after weeks or months.

1.10.3. Heat conduction mechanisms in nanofluid

Nanofluid is nothing but fluid particles which are less than even a micron (nearly 10^{-9} times smaller) in diameter and highly reactive and efficient material which can be used to increase factor like rate of reaction, thermal conductivity of any metal or material, they are that much reactive and strong. [8] presented four possible methods in nanofluids which may contribute to thermal conduction:

1. Brownian motion of nanoparticles
2. Liquid layering at the liquid/particle interface
3. Ballistic nature of heat transport in nanoparticles
4. Nano particle clustering in nanofluids

The Brownian motion of nanoparticles is too slow to transfer heat through a nanofluid. This mechanism works well only when the particle clustering has both the positive and negative effects of thermal conductivity which is obtained indirectly through convection.

1.10.4. Types of nanofluids

There are different types of nanofluids basically:

1. Al_2O_3 + water
2. SiC + water
3. CuO + water
4. TiO_2 + water
5. CH_3CH_2OH + water

Out of these, we are going to use Al_2O_3 + water and SiC + water as our nanofluid in heat exchanger.

1.10.5. Heat Transfer Enhancement of nanofluids

Low thermal conductivity of process fluid hinders high compactness and effectiveness of heat exchangers, although a variety of techniques is applied to enhance heat transfer. Improvement of the thermal properties of energy transmission fluids may become a trick of augmenting heat transfer. An innovative way of improving the thermal conductivities of fluids is to suspend small solid particles in the fluids. Various types of powders such as metallic, non-metallic and polymeric particles can be added into fluids to form slurries. The thermal conductivities of fluids with suspended particles are expected to be higher than that of common fluids. An industrial application test was carried out by [7] and [9], in which the effect of particle volumetric loading, size, and flow rate on the slurry pressure drop and heat transfer behaviour was investigated. In conventional cases, the suspended particles are of lm or even mm dimensions. Such large particles may cause some severe problems such as abrasion and clogging. Therefore, fluids with suspended large particles have little practical application in heat transfer enhancement.

Application of nanoparticles provides an effective way of improving heat transfer characteristics of fluids [10]. Particles <100 nm in diameter exhibit properties different from those of conventional solids. Compared with micron-sized particles, nanophase powders have much larger relative surface areas and a great potential for heat transfer enhancement. Some researchers tried to suspend nanoparticles into fluids to form high effective heat transfer fluids. [1] is the first who used the term nanofluids to refer to the fluids with suspended nanoparticles. Some preliminary experimental results [10] showed that increase in thermal conductivity of approximately 60% can be obtained for the nanofluid consisting of water and 5 vol. % CuO nanoparticles.

1.11. SURFACTANTS

The word “Surfactant” is a contraction of the three words “Surface Active Agents.” Surfactants are materials that lower the surface tension (or interfacial tension) between two liquids or between a liquid and a solid. In the general sense, any material that affects the interfacial surface tension, can be considered a surfactant, but in the practical sense, surfactants may act as wetting agents, emulsifiers, foaming agents, and dispersants.

Surfactants play an important role as cleaning, wetting, dispersing, emulsifying, foaming and anti-foaming agents in many practical applications and products, including: paints, emulsions adhesives, inks, biocides (sanitizers), shampoos, toothpastes, fire-fighting (foams), detergents, insecticides, deinking of recycled papers, ski waxes, spermicides (nonoxynol-9).

In the bulk aqueous phase, surfactants form masses, such as micelles, where the hydrophobic tails form the core and the hydrophilic heads are immersed in the surrounding liquid. Other types of structures can also be formed, such as spherical micelles or lipid bilayers. The shape of the molecules depends on the balance in size between hydrophilic head and hydrophobic tail. A measure of this is the HLB, Hydrophilic-lipophilic balance. Higher HLB surfactants (>10) are hydrophilic (“water loving”) and form O/W (Oil-in-water) emulsions. Lipophilic surfactants possess low HLB values (1-10) and form W/O (water-in-oil) emulsions. Dish detergents, surfactants for emulsion polymerization, and the following example (SLS = Sodium Lauryl Sulfate) are high HLB surfactants.

The dynamics of surfactant adsorption is of great importance for practical applications such as in emulsifying or coating processes as well as foaming, where bubbles or drops are rapidly generated and need to be stabilized.

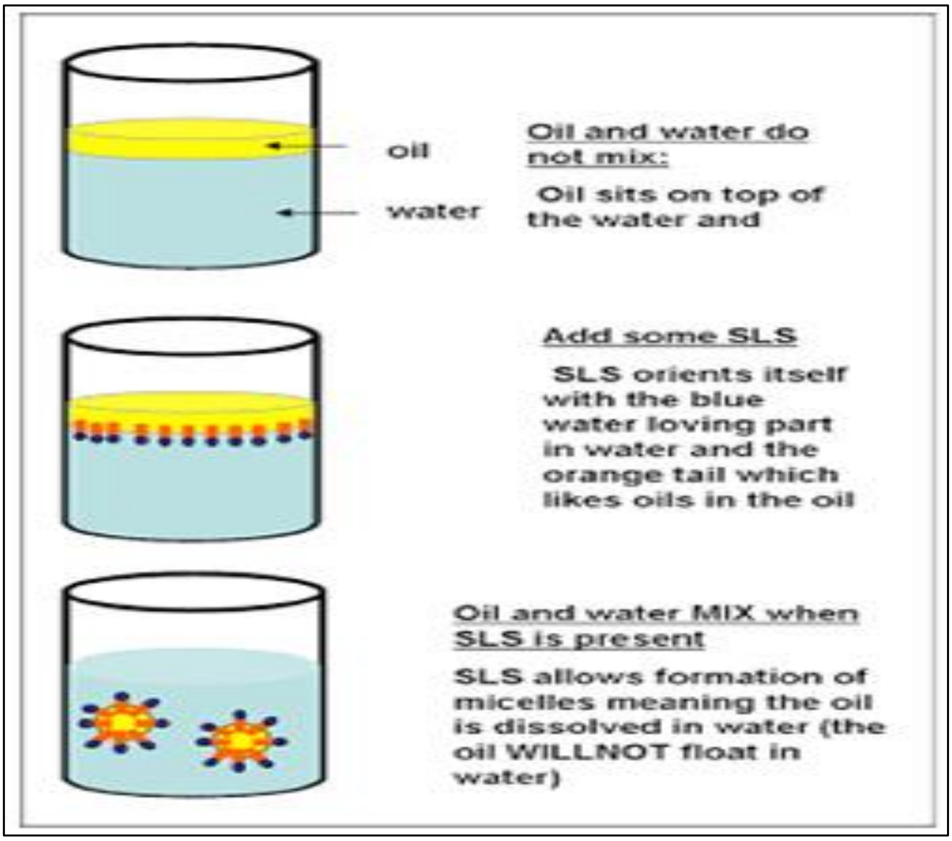


Figure 1.22. Example of surfactant

As the interface is created, the adsorption is limited by the diffusion of the surfactant to the interface, which can result in the kinetics being limited. These energy barriers can be due to steric or electrostatic repulsions; steric repulsions form the basis of how dispersants function. Surface rheology of surfactant layers, are important to the stability of foams and emulsions.

Most surfactants' "tails" are fairly similar, consisting of a hydrocarbon chain, which can be branched, linear, or aromatic. Fluoro surfactants have fluorocarbon chains. Siloxane surfactants have siloxane chains. Recent advances in surfactant technology has seen the development of mixed chains or/and complex structures.

There are 4 types of surfactants with a brief review of each as follows. These classifications are based upon the composition of the polarity of the head group:

1. Non-ionic
2. Anionic
3. Cationic
4. Amphoteric

A non-ionic surfactant has no charge groups in its head. The head of an ionic surfactant carries a net charge. If the charge is negative, the surfactant is more specifically called anionic; if the charge is positive, it is called cationic. If a surfactant contains a head with two oppositely charged groups, it is termed zwitter-ionic.

1.11.1. Non-Ionic Surfactant

Many long chain alcohols exhibit some surfactant properties. Some examples of non-ionic surfactants include:

Table 1.3. Examples of non-ionic surfactants

Trade name	Structure/Name	Applications
Triton™ X-100	Polyoxyethylene glycol octylphenol ethers $C_8H_{17}–(C_6H_4)–(O\text{-}C_2H_4)_n–OH$ (n=1-25)	Wetting agent - coatings
Nonoxynol-9	Polyoxyethylene glycol alkylphenol ethers $C_9H_{19}–(C_6H_4)–(O\text{-}C_2H_4)_n–OH$ (n=1-25)	Spermicide
Polysorbate	Polyoxyethylene glycol sorbitan alkyl esters	Food ingredient
Span®	Sorbitan alkyl esters	Polishes, cleaners, fragrance carriers
Poloxamers, Tergitol™, Antarox®	Block copolymers of polyethylene glycol and polypropylene glycol	Various

1.11.2. Anionic Surfactant

Anionic surfactants contain anionic functional groups at their head, such as sulfonate, phosphate, sulfate and carboxylates. Alkyl sulfates include ammonium lauryl sulfate, sodium lauryl and the related alkyl-ether sulfates sodium laureth sulfate, also known as sodium lauryl ether sulfate (SLES), and sodium myreth sulfate. These are the most common surfactants and comprise the alkyl carboxylates (soaps), such as sodium stearate. The stearates comprise >50% of the global usage of surfactants. Many of these find utilization in emulsion polymerization. Other anionic surfactants include dioctyl sodium sulfosuccinate (DOSS), perfluorooctanesulfonate (PFOS), linear alkylbenzene sulfonates (LABs) and perfluorobutanesulfonate, as well as alkyl-aryl ether phosphates. More specialized species include sodium lauroyl sarcosinate and carboxylate-based fluorosurfactants such as perfluorononanoate, perfluorooctanoate (PFOA or PFO).

Table 1.4. Examples of anionic surfactants

Trade name	Structure/Name	Applications
PENTEX® 99	Dioctyl sodium sulfosuccinate (DOSS)	Wetting agent – coatings, toothpaste
PFOS	Perfluorooctanesulfonate (PFOS)	Scotchguard™, Skydrol™
Calsoft®	Linear alkylbenzene sulfonates	Laundry detergents, dishwasher detergents
Texapon®	Sodium lauryl ether sulphate	Shampoos, bath products

1.11.3. Cationic Surfactant

Cationic surfactants are comprised of a positively charged head. Most of cationic surfactants find use as anti-microbials, anti-fungals, etc. in HI & I (Benzalkonium chloride (BAC), Cetylpyridinium chloride (CPC), Benzethonium chloride (BZT). The cationic nature of the surfactants is not typically consistent with the world of non-ionic and anionic charges, and they disrupt cell membranes of bacteria and viruses. Permanently charged quaternary ammonium cations include Alkyltrimethylammonium salts like cetyl trimethylammonium bromide (CTAB) and cetyl trimethylammonium chloride (CTAC).

1.11.4. Amphoteric Surfactant

Zwitterionic (amphoteric) surfactants have both cationic and anionic centers attached to the same molecule. The anionic part can be variable and include sulfonates, as in the sultaines CHAPS (3-[(3-Cholamidopropyl) dimethylammonio]-1-propanesulfonate). Betaines such as cocamidopropyl betaine have a carboxylate with the ammonium. The cationic part is based on primary, secondary, or tertiary amines or quaternary ammonium cations. Zwitterionic surfactants are often sensitive to pH and will behave as anionic or cationic based on pH. Fast dry ("coacervation") latex traffic paints are based on this concept, with a drop in pH triggering the latex in the paint to coagulate.

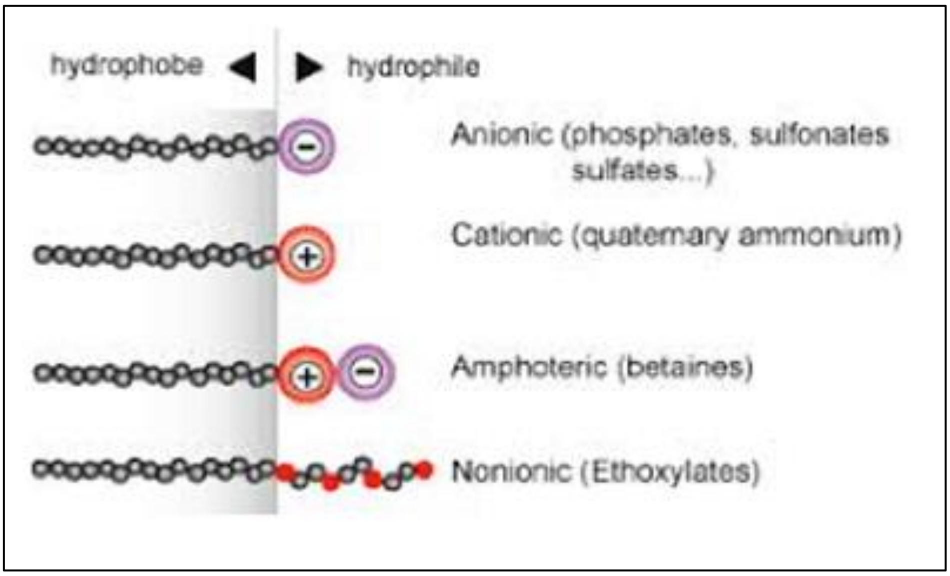


Figure 1.23. Representation of 4 types of surfactants

Some commonly used surfactants are given below:

1. Polysorbates (Tween™)
2. Sodium dodecyl sulfate (sodium lauryl sulfate)
3. Lauryl dimethyl amine oxide
4. Cetyltrimethylammonium bromide (CTAB)
5. Sodium dodecyl benzene sulfonate (SDBS)
6. Polyethoxylated alcohols
7. Polyoxyethylene sorbitan
8. Octoxynol (Triton X100™)
9. N, N - dimethyldodecylamine-N-oxide
10. Hexadecyltrimethylammonium bromide (HTAB)
11. Polyoxyl 10 lauryl ether
12. Brij 721™
13. Bile salts (sodium deoxycholate, sodium cholate)
14. Polyoxyl castor oil (Cremophor™)

CHAPTER

2

LITERATURE REVIEW

We have carried out literature review of 52 research/review papers, out of which selected ones are mentioned here.

Chavda et al., 2014 [11] have carried out an experimental study on the effect of nanofluid on heat transfer characteristics of double pipe heat exchanger with the effect of Al_2O_3 nanofluid. An experimental investigation is carried out to determine the effect of various concentrations of Al_2O_3 nano-dispersion mixed in water as base fluid on heat transfer characteristics of double pipe heat exchanger for parallel flow and counter flow arrangement. The volume concentrations of Al_2O_3 nanofluid prepared are 0.001% to 0.01%. The conclusion derived for the study is that overall heat transfer coefficient increases with increase in volume concentration of Al_2O_3 nano-dispersion compared to water up to the volume concentration of 0.008% and then decreases.

Kirubhadurai et al., 2014 [12] have carried out an experimental study in order to find out the effects of silicon nitride nanofluid, having nanoparticles concentration of 0.1% by volume on heat transfer, mass flow rate, effectiveness, LMTD of a shell and tube heat exchanger.

Mehta et al., 2012 [13] have performed an experimental and numerical study on Al_2O_3/water nanofluid flowing through the double pipe counter flow heat exchangers under laminar flow conditions. They have found that the heat transfer performance of both the double pipes increases with increase in the hot and cold volume flow rates as well as the particle concentrations and nanofluid inlet temperature compared to pure water.

Arvind, 2015 [14] has investigated the heat transfer of a fluid containing nanoparticles of aluminum nitride with a diameter of about 20 nm, with the water volume fraction (0.1-0.3) percent in a horizontal double pipe counter flow heat exchanger under turbulent flow conditions. They have found that heat transfer of nanofluid in comparison with the heat transfer of fluid is slightly higher than 9%.

Velagapudi et al., 2008 [15] have analysed the empirical correlations associated with the previous research papers and gave the correlations for thermal conductivity, density, Nusselt number, Reynolds's number and viscosity of the nanofluid.

Wang et al., 2008 [16] studied the fluid flow and heat transfer characteristics of nanofluids in forced and free convection flows and potential applications of nanofluids. They gave a tabular form of models for effective viscosity and also gave a tabular form for models of effective heat transfer coefficient for both laminar as well as turbulent flows.

Mukherjee et al., 2013 [17] have given the various methods for preparation of nanofluids and its stability mechanisms and effects of surfactants on nanofluids.

Wong et al., 2010 [18] studied the various applications of nanofluids. The fields include heat transfer applications, automotive applications, electronics applications, biomedical applications etc.

Albadr et al., 2013 [19] made experimental study on the forced convective heat transfer and flow characteristics of a nanofluid consisting of water and different volume concentrations of Al_2O_3 nanofluid (0.3-2)% flowing in a horizontal shell and tube heat exchanger counter flow under turbulent flow conditions.

Huminic et al., 2012 [20] presented an overview of the recent investigations in the study of the thermo physical characteristics of nanofluids and their role in heat transfer enhancement from heat exchangers. General correlations for the effective thermal conductivity, viscosity and Nusselt number of nanofluids are presented.

Asirvatham, 2015 [21] studied the applications of nanofluids by focusing on the previous research papers and concluded the importance of nanofluids in various fields by providing some experimental results.

Tiwari, 2015 [22] investigated the thermal performance of a shell and tube heat exchanger using nanofluids and found that the effectiveness was increased by a considerable amount, while the convective and overall heat transfer coefficient increased even further with the addition of 3% Al_2O_3 nanoparticles in water based fluid.

Aghayari et al., 2014 [23] investigated the enhancement of heat transfer coefficient and Nusselt number of a nanofluid containing nanoparticles (γ-Al_2O_3) with a particle size of 20 nm and volume fraction of 0.1%-0.3% (V/V). Effects of temperature and concentration of nanoparticles on Nusselt number changes and heat transfer coefficient in a double pipe heat exchanger with counter turbulent flow are investigated.

Dhamecha, 2014 [24] reported the theoretical study on the heat transfer and flow characteristics of nanofluids consisting of water flowing in a horizontal shell and tube heat exchanger. The Al_2O_3 nanoparticles are used in the present study. The result show that the heat transfer coefficient of nanofluid is slightly higher than that of the base liquid at same mass flow rate and at same temperature.

Kumar et al., 2014 [25] showed the review of various research papers in nanofluids and concluded that nanofluids have great potential for heat transfer enhancement and are highly suited for application in heat transfer processes.

The experimental and analytical studies by Wang et al., 1999 [26], Li et al., 1999 [27], Wang et al., 2003 [28], Koo et al., 2004 [29] showed that nanofluids have a higher thermal conductivity than that of pure fluids and therefore has great affinity for heat transfer enhancement.

Experimentally, enhancement of laminar flow convection coefficient of Al_2O_3-water nanofluids under constant wall temperature in heating process is much higher than that predicted by single phase heat transfer correlation used in conjunction with the nanofluids properties, as in Heris et al., 2007 [30]. It was also concluded that the heat transfer enhancement of nanofluids is not merely due to the thermal conductivity increase of nanofluids which means other factors may contribute to this phenomenon.

Anoop et al., 2009 [31] carried out convective heat transfer experiments with Al_2O_3-water nanofluids in the developing region of pipe flow with constant heat flux to evaluate the effect of particle size on convective heat transfer coefficient. In their work, two particle sizes (45 nm and 150 nm) were used and it was observed that the nanofluid with 45 nm particles showed higher heat transfer coefficient than that with 150 nm particles.

Sudarmadji et al., 2014 [32] carried out research to investigate the convective heat transfer and pressure drop of nanofluid, using alumina-water nanofluid under laminar flow regime. The test section was using 1.1 m long and 5 mm inner diameter tube of a double-pipe heat exchanger with constant wall temperatures. The hot nanofluid was flowing inside tube, while the cold water flowed outside. The volume concentration of the nanoparticles varied from 0.15%, 0.25% and 0.5%. Experiment showed that the convection heat transfer increased remarkably with the increase of the nanoparticles concentration under various values of Reynolds number. The Nusselt number increased about 40.5% compared to pure water under 0.5% vol. concentration. The pressure drop of nanofluid increased slightly with increasing volume concentration. However, compared with using pure water the difference of the pressure drop is insignificant, so that the use of nanofluid has little penalty on pressure drop.

Sekhar et al., 2015 [33] studied the specific heat capacity and viscosity properties of water-based nanofluids containing alumina nanoparticles of 47 nm average particle diameter at low concentrations. Nanofluids were prepared with deionised water as base fluid at room temperature by adding nanoparticles at low volume concentration in the range of 0.01%-1% to measure viscosity. The effect of temperature on viscosity of the nanofluid was determined based on the experiments conducted in the temperature range of 25˚C to 45˚C. The results indicated a nonlinear increase of viscosity with particle concentration due to aggregation of particles. The estimated specific heat capacity of the nanofluid decreased with increase of particle concentration due to increase in thermal diffusivity. Generalised regression equations for estimating the viscosity and specific heat capacity of nanofluids for a particular range of particle concentration, particle diameter and temperature were established.

Senthilraja et al., 2015 [34] investigated the thermal conductivity of Al_2O_3/water, CuO/water and Al_2O_3-CuO/water hybrid nanofluid experimentally. The two step method was adopted to prepare the hybrid nanofluid. Three different volume concentrations of nanofluids (0.05, 0.1 & 0.2%) were prepared by dispersing Al_2O_3 and CuO nanoparticles in water. The properties of single and hybrid nanofluids were measured by varying the temperature from 200˚C to 600˚C. The obtained results demonstrated that the thermal conductivity of nanofluids are the function of volume concentration and temperature. Also the experimental results showed that a maximum of 9.8 % enhancement of thermal conductivity was observed for 0.2% particle volume concentration. The experimental thermal conductivity values were compared with the theoretical thermal conductivity values.

Boda et al., 2017 [35] summarized the important published articles on the enhancement of the convection heat transfer with nanofluids. Over the last 2-3 decades, there has been intensive research into the behaviour of substances that contain extremely small particles. Nanotechnology is the science and engineering of working at the nanoscale, where the individual particles are 1-100 nm in size. It’s hard to imagine the size of nanoparticles, but there are about 2, 54, 00, 000 nm in an inch. Nanofluids which are less than even a micron (nearly 10^{-9} times smaller) in diameter, highly reactive and efficient material which can be used to increase factor like rate of heat transfer, thermal conductivity of any metal or material, they are that much reactive and strong. The thermal conductivity increases with decreasing the grain size of the material. As the thermal conductivity increases the heat transfer rate increases.

Farajollahi et al., 2010 [36] measured heat transfer characteristics of γ-Al_2O_3/water and TiO_2/water nanofluids in a shell and tube heat exchanger under turbulent flow condition. The effects of Peclet number, volume concentration of suspended nanoparticles, and particle type on the heat characteristics were investigated. Based on the results, adding of naoparticles to the base fluid causes the significant enhancement of heat transfer characteristics. For both nanofluids, two different optimum nanoparticle concentrations existed. Comparison of the heat transfer behaviour of two nanofluids indicated that at a certain Peclet number, heat transfer characteristics of TiO_2/water nanofluid at its optimum nanoparticle concentration were greater than those of γ-Al_2O_3/water nanofluid while γ-Al_2O_3/water nanofluid possessed better heat transfer behaviour at higher nanoparticle concentrations.

Yang et al., 2005 [37] measured the convective heat transfer coefficients of several nanoparticle-in-liquid dispersions (nanofluids) under laminar flow in a horizontal tube heat exchanger. The nanoparticles used in this research were graphitic in nature, with aspect ratios significantly different from one (l/d ~ 0.02). The graphite nanoparticles increased the static thermal conductivities of the fluid significantly at low weight fraction loadings. However, the experimental heat transfer coefficients showed lower increases than predicted by either the conventional heat transfer correlations for homogeneous fluids, or the correlations developed from the particle suspensions with aspect ratios close to one. New correlations on heat transfer need to be developed for nanofluid systems.

Lee et al., 2011 [38] reported that nanofluids are nanotechnology-based colloidal dispersions engineered by stably suspending nanoparticles. Transmission electron microscopy and scanning electron microscope images were acquired to characterize the shape and size of SiC nanoparticles, because the properties of the nanofluids depend on the morphologies of nanoparticles. The dispersion behaviour for SiC/deionized water (DIW) nanofluids were investigated under different pH values and characterized with the zeta potential values. The isoelectric point of SiC/DIW nanofluid was identified in terms of colloidal stability. Then their viscosity and thermal conductivity were investigated as a function of volume fraction to evaluate SiC/DIW nanofluids' potential to function as more effective working fluids in heat transfer applications.

Chavda, 2015 [39] carried out an experimental evaluation to determine the effect of various concentration of CuO nanoparticles mixed in water on heat transfer characteristics of double pipe heat exchanger for parallel flow and counter flow arrangement. The volume concentrations of CuO nanofluid prepared by two step method were 0.002 %, 0.003% and 0.004 %. The conclusion derived for the study was that overall heat transfer coefficient increases with increase in volume concentration of CuO nanoparticle compared to water which have been validated by theoretical prediction also.

Timofeeva et al., 2010 [40] investigated the effect of average particle sizes on basic macroscopic properties and heat transfer performance of α-SiC/water nanofluids. The average particle sizes, calculated from the specific surface area of nanoparticles, were varied from 16 to 90 nm. Nanofluids with larger particles of the same material and volume concentration provided higher thermal conductivity and lower viscosity increases than those with smaller particles because of the smaller solid/liquid interfacial area of larger particles. It was also demonstrated that the viscosity of water-based nanofluids can be significantly decreased by pH of the suspension independently from the thermal conductivity. Heat transfer coefficients were measured and compared to the performance of base fluids as well as to nanofluids reported in the literature. Criteria for evaluation of the heat transfer performance of nanofluids were discussed and optimum directions in nanofluid development were suggested.

Singh et al., 2009 [41] investigated thermal conductivity and mechanical effects of silicon carbide nanoparticles uniformly dispersed in water. Mean size of SiC particles was 170 nm with a polydispersity of 30% as determined from small-angle x-ray scattering and dynamic light scattering techniques. Room temperature viscosity of the nanofluids ranged from 2 to 3 cP for nominal nanoparticle loadings 4-7 vol. %. On a normalized basis with water, viscosity of the nanofluids did not significantly change with the test temperature up to 85°C. Optical microscopy of diluted nanofluid showed no agglomeration of the nanoparticles. Thermal conductivity of the fluid was measured as a function of the nominal nanoparticle loading ranging from 1-7 vol. %. Enhancement in thermal conductivity was approximately 28% over that of water at 7 vol. % particle loadings under ambient conditions. Enhancements in thermal conductivities for the nanofluids with varying nanoparticle loadings were maintained at test temperatures up to 70°C. Results of thermal conductivity were rationalized based on the existing theories of heat transfer in fluids. Implications of using this nanofluid for engineering cooling applications were discussed.

Yu et al., 2009 [42] performed heat transfer experiments with a water-based nanofluid containing 170 nm silicon carbide particles at a 3.7% volume concentration and having potential commercial viability. Heat transfer coefficients for the nanofluid were presented for Reynolds numbers ranging from 3300 to 13,000 and were compared to the base fluid water on the bases of constant Reynolds number, constant velocity, and constant pumping power. Results were also compared to predictions from standard liquid correlations and a recently altered nanofluid correlation. The slip mechanisms of Brownian diffusion and thermophoresis postulated in the altered correlation were investigated in a series of heating and cooling experiments.

Timofeeva et al., 2011 [43] presented experimental data for the thermal conductivity, viscosity, and turbulent flow heat transfer coefficient of nanofluids with SiC particles suspended in ethylene glycol EG/water H_2O mixture with a 50/50 volume ratio. The results were compared to the analogous suspensions in water for four sizes of SiC particles 16–90 nm. It was demonstrated that the heat transfer efficiency is a function of both the average particle size and the system temperature. The results showed that adding SiC nanoparticles to an EG/H_2O mixture can significantly improve the cooling efficiency while water-based nanofluids are typically less efficient than the base fluids. This was one of the few times that substantial nanofluid heat transfer enhancement was reported in the literature based on a realistic comparison basis of constant velocity or pumping power. The trends important for engineering efficient heat transfer nanofluids were summarized.

The review of existing literature reveal the facts related to the application of nanofluid in heat exchangers for the enhancement of heat transfer rate. They can be summarized as follows. Nanofluids are relatively practiced recently to enhance the heat transfer rate. There are different types of nanofluids which can be used to enhance the heat transfer rate and many researchers have carried out numerical and experimental analysis on the application of nanofluid in the enhancement of heat transfer rate under different conditions. Extensive experiments have been carried out on double pipe heat exchanger to determine the effect of different type of nanofluids (having nanoparticles of Al_2O_3 and SiC) and concentration of nanoparticles in nanofluid during parallel and counter flow condition.

A few studies were reported on the application of Al_2O_3/SiC nanofluid in a heat exchanger (shell and tube). Therefore, this study reported experimental studies of utilization of nanofluid Al_2O_3/SiC in a heat exchanger (shell and tube).

As one can notice, there are plenty of articles that reported heat transfer enhancement. Then the literature survey was done for the range of Reynolds number from turbulent flow to laminar region, as transition flow.

The deterioration of ethylene glycol-based titania nanofluids is very interesting. It is believed to be related with the high viscosity of the base liquid. There seems to be a correlation between the rheological behaviour and convective heat transfer behaviour. For instance, for water-based carbon nanotube nanofluids, an extreme rise in the convective heat transfer coefficient happens at a flow rate where shear viscosity is almost near to the lowest. However, from researcher's experience, working with titania nanofluids could cause clogging of pipes in some experiments.

It was needed to gather data and literature that could be applicable to the setup. So based on the previous research which was done on this setup in the vertical heat exchanger and in turbulent flow, it was decided to use laminar flow in the horizontal heat exchanger.

So based on the recommendations, looking for different articles was reasonable to use nanofluids in heat exchangers with laminar flow (preferably in the horizontal mode) which had Ethylene Glycol partially or totally as the base fluid. Then, because of the unavailability of nanoparticles with the small size to be dispersed in Ethylene Glycol as base fluid, water as the base fluid was chosen. The literature was written based on the mentioned criteria and following:

- Authors
- Base fluids (wherever available)
- Nanoparticles' material, size and concentration (wherever available)
- Flow arrangements such as flow geometry, horizontal/vertical mode, nanofluid cooling/heating method
- Preparation such as usage of pH adjustment or surfactants, mixing/sonication

- Particle characterization which includes the method used to measure particle size distribution, measurement of distribution from powder or suspension, probability of agglomeration, clusterization or sedimentation (wherever available)
- Methods for evaluation of thermal properties such as thermal conductivity, viscosity, specific heat and density (wherever available)
- Range of measured Re number and Nu number
- Results based upon changes in heat transfer coefficient or Nu number compared to the base fluid, pressure drop or combination of these results
- Error estimation and repetition of tests (wherever available)

CHAPTER

3

EXPERIMENTAL WORK

3.1. OBJECTIVES

Nanofluid Al_2O_3/SiC was used in a heat exchanger (shell and tube) for the observation of behaviour of heat transfer with flow rate and concentration of nanoparticles at ambient temperature. The main objective of this project was to enhance the thermal conductivity, heat transfer coefficient and other heat transfer related parameters by using nanofluid+water instead of simple water. Major aim was to enhance the heat transfer related parameters up to desired level with minimal use of nanofluid. The application of this project is widely in chemical based industries, hence, maximum enhancement of properties with minimal use of nanofluid is the most feasible method.

3.2. MATERIALS

The Nano Labs India supplied the nanoparticles used for this project. The average particle size of nanofluid Al_2O_3 and SiC was 30-50 nm and 40-60 nm in diameter respectively. The Avantor Performance Materials India Limited supplied the surfactant Sodium Dodecyl Benzene Sulfonate (SDBS) which was used as a mixing agent.

3.3. EXPERIMENTAL SET-UP

The apparatus used for this project is shown in Figure 1. Experimental observations have been performed for heat transfer due to convection with laminar flow using nanofluid Al_2O_3/SiC with a concentration range of 0.1-0.3 vol. %. Experimental set-up consists of a test section (straight steel), outer tube, pump and reservoir tank. The tube side dimensions are d_o = 12.7 mm and d_i = 10 mm and the shell side dimensions are d_o = 125 mm and d_i = 120 mm. Length of tube is 500mm. The total number of tubes are 14. The variation of flow of fluid is from 1-5 LPM. Heated fluid was re-circulated to keep constant temperature at the inlet of the test section. The material of construction is SS-304.

Figure 3.1. Experimental set-up

3.4. PROCEDURE FOR MAKING NANOFLUID

((Al_2O_3/SiC + Surfactant (generally Triton X-100)) + Water) with 30 min magnetic stirring and 60 min sonication. Above Surfactant can also be used for MWCNT nanoparticles and also for CuO. With the help of the surfactant the mixture will be stable for 3 days easily. In place of ultra-sonication, we can also use magnetic stirring at high RPM values.

3.5. NANOFLUID PHYSICAL PROPERTIES

The thermo-physical properties considered for Water (Base fluid), Al_2O_3 and SiC at set temperature, i.e., 50˚C are:

Table 3.1. Thermo-physical properties for Water, Al_2O_3, SiC

Properties	**Density (kg/m^3)**	**Specific heat (J/kg·K)**	**Thermal Conductivity (W/m·K)**
Water	988.02	4182	0.6435
Al_2O_3 nanoparticles	3890	880	35
SiC nanoparticles	3216	610	15

Table 3.2. Effective thermo-physical properties for Water+Al_2O_3

Sample	**Density (kg/m^3)**	**Specific heat (J/kg·K)**	**Thermal Conductivity (W/m·K)**	**Viscosity (kg/m·s)**
Water	988.02	4182	0.6435	0.000547
Water+0.1% Al_2O_3	990.893	4169.1663	0.6453	0.0005484
Water+0.2% Al_2O_3	993.812	4156.204	0.6472	0.000549
Water+0.3% Al_2O_3	996.6998	4143.4542	0.649	0.000551

Table 3.3. Effective thermo-physical properties for Water+SiC

Sample	**Density (kg/m^3)**	**Specific heat (J/kg·K)**	**Thermal Conductivity (W/m·K)**	**Viscosity (kg/m·s)**
Water	988.02	4182	0.6435	0.000547
Water+0.1% SiC	990.225	4170.518	0.6452	0.0005484
Water+0.2% SiC	992.467	4158.897	0.6469	0.000549
Water+0.3% SiC	994.6838	4147.5475	0.6486	0.000551

3.6. DATA PROCESSING

The nanofluid presented equation are calculated by using the Pak and Cho [44] correlations, which are defined as follows:

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

where,

ρ_{nf} = Density of the nanofluid, kg/m^3

$\emptyset$ = Particles volume concentration

ρ_f = Density of the base fluid, kg/m^3

ρ_p = Density of the nanoparticles, kg/m^3

The specific heat is calculated from Xuan and Roetzel [45] as following:

$$C_{p_{nf}} = \frac{(1 - \emptyset)\rho_{bf}\, C_{p_{bf}} + \emptyset_{\rho_p}\, C_{p_p}}{\rho_{nf}}$$

where,

C_{pnf} = Heat capacity of the nanofluid, J/kg·˚C

C_{pbf} = Heat capacity of the base fluid, J/kg·˚C

C_{pp} = Heat capacity of the nanoparticles, J/kg·˚C

The viscosity of the nanofluid was calculated by well-known Einstein's equation which is applicable in volume fractions less than 5.0 vol. % suggested by Drew and Passman [46] and is defined as follows:

$$\mu_{nf} = (1 + 2.5\emptyset)\, \mu_{bf}$$

where,

μ_{nf} = Nanofluid viscosity, kg/m·s

μ_{bf} = Water viscosity, kg/m·s

An alternative formula for calculating the thermal conductivity was introduced by Maxwell [47], which is expressed in the following form:

$$K_{eff} = \frac{K_p + 2K_{bf} + 2(K_p - K_{bf})\emptyset_p}{K_p + 2K_{bf} - (K_p - K_{bf})\emptyset_p} K_{bf}$$

where,

K_p = Thermal conductivity of the nanoparticle, W/m·˚C

K_{bf} = Thermal conductivity of the water, W/m·˚C

ϕ_p = Volume fraction

The theoretical proof was validated using Li and Xuan [49] correlation, which is expressed in the following form:

$$Nu = 0.4328\ (1 + 11.285\emptyset^{0.754} Pe^{0.218}) Re^{0.333} Pr^{0.4}$$

where,

ϕ = Volume fraction

Pe = Peclet number

Re = Reynolds number

Pr = Prandtl number

The properties of the nanofluid shown in the above equations are evaluated from water and nanoparticles at set temperature of 50˚C.

3.7. RESULTS AND DISCUSSION

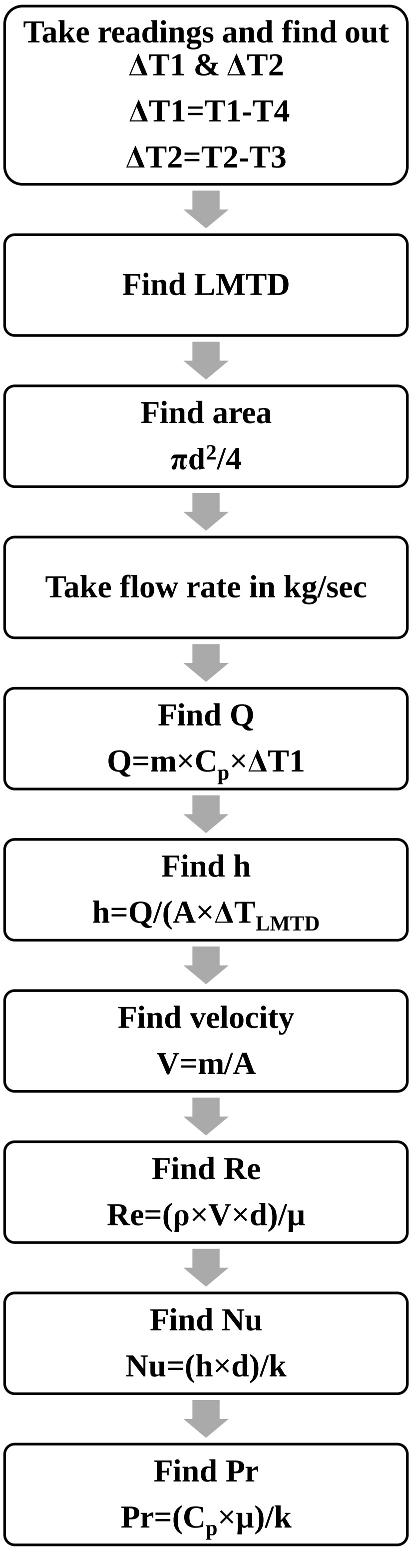


Figure 3.2. Steps for calculation

The experimental analysis is carried out in shell and tube heat exchanger using tap water and a mixture of tap water and nanoparticles. Flow rate was varied from 1-5 LPM and volume concentration was varied from 0.1-0.3%. Set temperature was 50˚C and ambient temperature was measured to be 25˚C. The measured temperatures of hot water inlet, hot water outlet, cold water inlet and cold water outlet are presented in table 3.4 to 3.38. From table 3.9 onwards, we can clearly observe that with the increase in volume concentration of nanoparticles and LPM, the steady state is achieved earlier compared to that of water.

Water

Table 3.4. 1 LPM (T1=47.7681, T2=42.7090, T3=37.8564, T4=41.897)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min	100 min	105 min
T1	43.3	43.8	44.1	44.6	45.2	45.8	46.5	47	47.7	48.3	48.7	49.1	49.5	49.5	49.5	49.6	49.6	49.7	49.7	49.7	50.0	50.0
T2	33.4	36.5	37.8	38.7	39.8	40.6	41.4	42.1	42.9	43.5	44.1	44.7	45.0	45.1	45.4	45.4	45.5	45.5	45.5	45.5	45.6	45.6
T3	29.3	30.5	31.5	33.2	34.9	36.0	37.2	38.2	39.0	39.6	40.4	40.9	41.3	41.5	42.0	42.2	42.3	42.4	42.4	42.4	42.5	42.5
T4	32.0	35.8	37.2	38.4	39.2	40.3	40.9	41.7	42.4	43.1	43.7	44.5	45.0	45.2	45.5	45.5	45.6	45.6	45.6	45.6	45.7	45.7

Table 3.5˙ 2 LPM (T1=48.7857, T2=45.7714, T3=41.0667, T4=43.0429)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min	100 min
T1	44.6	46.8	46.8	46.8	47.1	47.5	48.0	48.5	49.0	49.5	49.9	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0
T2	39.5	41.4	41.7	42.3	42.8	43.4	44.0	44.8	45.5	46.2	47.0	47.4	47.8	48.2	48.4	48.4	48.4	48.5	48.5	48.5	48.5
T3	29.7	33.7	35.4	37.0	37.9	39.3	40.1	41.0	41.7	42.4	42.8	43.5	43.7	44.0	44.1	44.2	44.3	44.4	44.4	44.4	44.4
T4	34.1	37.9	38.8	39.6	40.3	41.3	42.0	42.8	43.4	44.0	44.6	45.1	45.3	45.4	45.5	45.5	45.6	45.6	45.7	45.7	45.7

Table 3.6. 3 LPM (T1=47.0833, T2=44.7652, T3=40.1043, T4=41.5696)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min	100 min	105 min	110 min	115 min
T1	42.0	42.1	42.2	42.5	43.0	43.9	44.5	45.3	46.0	46.7	47.2	47.9	48.6	48.9	49.4	49.8	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0
T2	37.6	38.3	38.9	39.6	40.5	41.4	42.4	43.1	44.0	44.5	45.3	45.8	46.4	47.1	47.6	47.9	48.2	48.4	48.4	48.5	48.5	48.6	48.6	48.6
T3	25.7	29.0	31.6	33.9	35.5	36.8	38.1	39.1	40.0	40.6	41.4	42.0	42.5	43.0	43.6	44.0	44.4	44.8	44.9	45.2	45.3	45.5	45.5	45.5
T4	29.6	32.2	34.0	35.6	37.0	38.2	39.4	40.3	41.1	41.8	42.6	43.2	43.7	44.3	44.8	45.2	45.6	45.9	46.2	46.2	46.4	46.4	46.4	46.4

Table 3.7. 4 LPM (T1=46.64, T2=44.845, T3=40.28, T4=41.465)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min
T1	41.0	41.7	41.7	42.2	43.0	43.6	44.6	45.6	46.5	47.3	48.0	48.5	49.3	49.8	50.0	50.0	50.0	50.0	50.0	50.0
T2	36.5	38.4	39.2	40.2	41.0	42.0	43.5	44.5	45.3	46.0	46.4	47.0	47.6	48.1	48.6	48.8	48.9	48.9	49.0	49.0
T3	23.8	30.4	33.1	35.1	36.7	38.0	39.3	40.2	41.1	41.8	42.5	43.1	43.6	44.2	44.7	45.1	45.5	45.6	45.9	45.9
T4	28.3	32.4	34.5	36.3	37.8	39.0	40.0	41.0	41.9	42.8	43.4	44.1	44.6	45.1	45.6	46.2	46.4	46.5	46.7	46.7

Table 3.8. 5 LPM (T1=47.9526, T2=46.385, T3=42.6579, T4=43.5053)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min
T1	44.0	44.1	44.2	44.4	45.0	46.2	46.8	47.7	48.9	49.8	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0
T2	39.0	40.6	41.5	42.0	43.2	44.0	44.9	45.5	46.3	47.2	48.0	48.8	49.3	49.5	49.6	49.6	49.6	49.7	49.7	49.7
T3	30.0	32.3	35.2	37.6	39.1	40.6	41.5	42.8	44.0	44.8	45.6	46.2	46.6	47.0	47.2	47.4	47.4	47.6	47.6	47.6
T4	32.4	34.2	37.0	38.6	40.1	41.3	42.4	43.5	44.6	45.6	46.4	46.8	47.1	47.4	47.6	47.8	47.8	48.0	48.0	48.0

Water+0.1% Al_2O_3

Table 3.9. 1 LPM (T1=47.55, T2=42.145, T3=37.705, T4=42.09)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min
T1	42.9	43.3	44.2	44.6	45.1	45.5	46.2	47.3	47.6	48.3	48.7	49.2	49.4	49.7	49.7	49.8	49.8	49.9	50.0	50.0
T2	33.0	35.5	37.2	38.2	39.4	40.3	41.4	42.0	42.4	43.2	44.0	44.1	45.1	45.1	45.3	45.3	45.3	45.4	45.4	45.4
T3	29.5	30.8	31.4	32.3	33.0	33.9	34.8	35.9	36.5	37.6	38.6	39.7	40.6	41.4	41.4	41.9	41.9	42.3	42.3	42.3
T4	31.8	33.0	34.1	35.3	36.2	37.9	41.0	41.8	42.6	43.4	44.1	44.8	45.9	46.6	47.3	47.3	47.3	47.8	47.8	47.8

Table 3.10. 2 LPM (T1=47.95, T2=44.49, T3=39.9172, T4=42.6072)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	44.0	44.7	45.3	45.9	46.3	46.9	47.5	48.0	48.4	48.9	49.1	49.3	49.4	49.5	49.7	49.9	49.9	50.0	50.0
T2	38.9	39.8	40.5	41.1	41.9	42.6	43.4	43.9	45.0	45.9	46.3	46.7	47.0	47.0	47.0	47.1	47.1	47.1	47.1
T3	29.0	29.8	30.9	32.0	32.9	33.9	34.8	35.9	36.8	38.0	38.9	39.8	41.5	42.8	43.2	43.4	43.4	43.4	43.4
T4	34.0	34.9	35.0	35.9	36.5	37.6	38.8	40.0	40.9	41.4	42.1	42.2	43.3	43.3	43.7	43.8	43.8	43.8	43.8

Table 3.11. 3 LPM (T1=44.4, T2=41.6526, T3=37.2006, T4=39.17)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	42.0	42.5	42.7	43.2	43.8	44.5	45.2	45.8	46.4	47.0	47.6	48.4	48.8	49.2	49.6	49.8	49.8	50.0	50.0
T2	37.2	38.8	39.6	40.4	40.9	41.7	42.5	43.3	43.5	43.8	44.8	46.6	47.4	47.9	48.0	48.1	48.1	48.2	48.2
T3	26.2	29.8	31.8	32.2	34.9	36.4	37.0	37.6	39.0	40.2	41.2	42.8	43.6	44.0	44.3	44.7	44.9	45.0	45.0
T4	30.3	33.0	34.9	35.4	36.4	37.0	38.8	40.0	41.2	42.5	43.6	44.8	45.8	46.6	46.8	47.2	47.3	47.6	47.6

Table 3.12. 4 LPM (T1=46.4684, T2=44.3, T3=39.7199, T4=41.5884)

℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	40.2	40.8	41.6	42.4	43.0	43.8	44.9	46.0	46.8	47.3	48.2	48.7	49.4	49.8	50.0	50.0	50.0	50.0	50.0
T2	35.2	37.3	38.6	39.2	40.4	41.6	43.0	44.2	45.0	45.6	46.4	46.9	47.5	47.9	48.3	48.5	48.7	48.7	48.7
T3	25.4	29.8	31.8	34.7	35.4	36.0	37.4	38.4	40.0	40.8	41.8	42.6	43.6	44.8	45.2	45.3	45.5	45.5	45.5
T4	30.2	32.0	35.4	36.9	37.6	38.0	39.0	40.5	40.9	42.0	42.7	43.9	45.5	46.0	46.5	46.6	46.9	47.2	47.2

Table 3.13. 5 LPM (T1=47.553, T2=45.5, T3=41.321, T4=43.203)

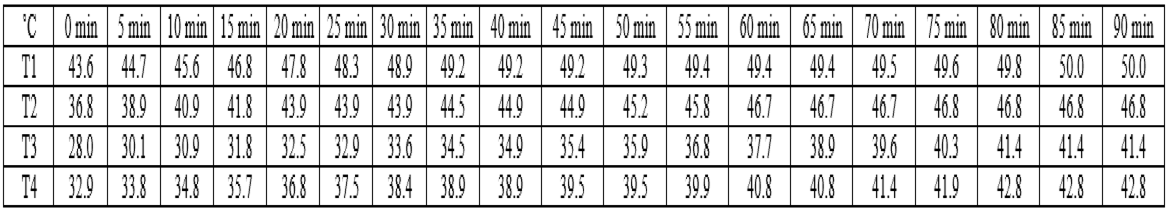
℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min
T1	42.0	43.0	44.2	45.0	46.2	46.8	47.4	48.0	48.4	48.8	49.2	49.6	49.8	50.0	50.0	50.0	50.0
T2	37.5	39.0	39.8	42.0	43.6	45.0	45.8	46.4	47.0	47.4	48.0	48.2	48.4	48.6	48.8	49.0	49.0
T3	28.0	29.2	31.0	32.6	33.8	35.0	36.2	37.0	38.2	39.0	40.2	41.2	41.6	42.0	43.6	44.0	44.0
T4	31.0	32.4	34.0	35.5	36.1	37.0	37.8	38.4	39.8	40.4	41.6	42.0	43.0	43.8	45.0	45.4	45.4

Water+0.2% Al_2O_3

Table 3.14. 1 LPM (T1=46.76, T2=41.135, T3=36.895, T4=41.52)

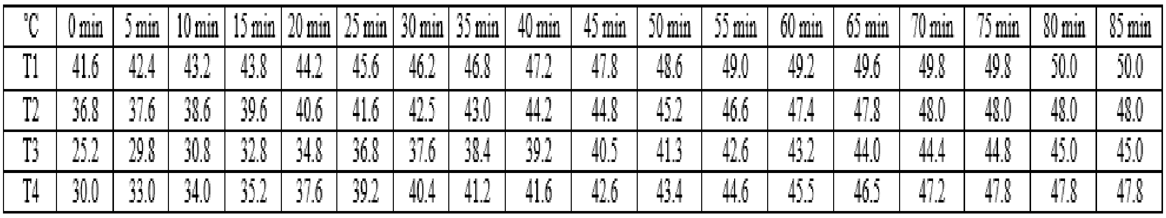
℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min
T1	41.0	41.9	42.8	43.3	44.0	44.6	45.3	46.0	46.4	47.0	47.8	48.4	49.1	49.5	49.5	49.5	49.7	49.8	50.0	50.0
T2	32.9	33.8	34.4	35.3	36.0	37.1	38.8	39.9	41.7	42.8	42.8	43.0	43.4	44.0	44.9	45.6	46.2	46.7	46.7	46.7
T3	28.5	29.2	29.9	30.7	31.4	32.3	32.9	33.4	33.9	34.8	35.4	36.7	37.6	38.6	39.8	41.4	41.9	41.9	41.9	41.9
T4	31.0	31.9	33.2	34.8	35.9	36.7	37.8	38.7	39.9	40.8	41.9	42.8	43.9	44.4	44.8	46.9	48.2	48.5	48.5	48.5

Table 3.15. 2 LPM (T1=48.35, T2=44.352, T3=39.99, T4=43.25)



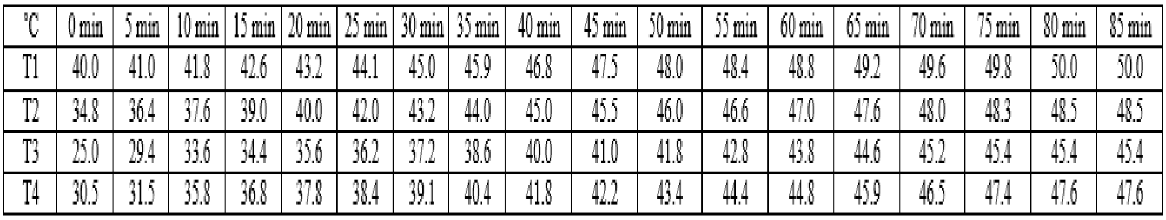
°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	43.6	44.7	45.6	46.8	47.8	48.3	48.9	49.2	49.2	49.2	49.3	49.4	49.4	49.4	49.5	49.6	49.8	50.0	50.0
T2	36.8	38.9	40.9	41.8	43.9	43.9	43.9	44.5	44.9	44.9	45.2	45.8	46.7	46.7	46.7	46.8	46.8	46.8	46.8
T3	28.0	30.1	30.9	31.8	32.5	32.9	33.6	34.5	34.9	35.4	35.9	36.8	37.7	38.9	39.6	40.3	41.4	41.4	41.4
T4	32.9	33.8	34.8	35.7	36.8	37.5	38.4	38.9	38.9	39.5	39.5	39.9	40.8	40.8	41.4	41.9	42.8	42.8	42.8

Table 3.16. 3 LPM (T1=46.933, T2=43.794, T3=39.581, T4=42.013)



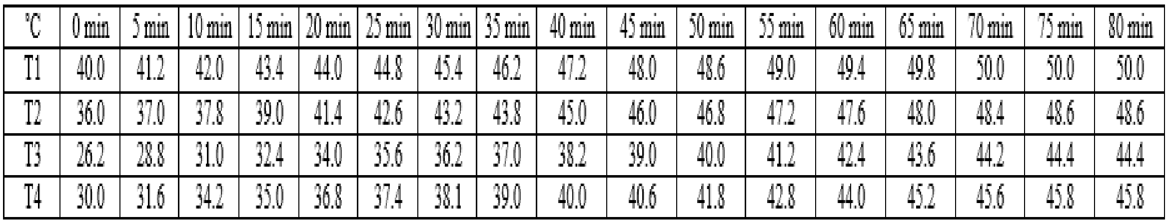
°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min
T1	41.6	42.4	43.2	43.8	44.2	45.6	46.2	46.8	47.2	47.8	48.6	49.0	49.2	49.6	49.8	49.8	50.0	50.0
T2	36.8	37.6	38.6	39.6	40.6	41.6	42.5	43.0	44.2	44.8	45.2	46.6	47.4	47.8	48.0	48.0	48.0	48.0
T3	25.2	29.8	30.8	32.8	34.8	36.8	37.6	38.4	39.2	40.5	41.3	42.6	43.2	44.0	44.4	44.8	45.0	45.0
T4	30.0	33.0	34.0	35.2	37.6	39.2	40.4	41.2	41.6	42.6	43.4	44.6	45.5	46.5	47.2	47.8	47.8	47.8

Table 3.17. 4 LPM (T1=46.2056, T2=43.7778, T3=39.5578, T4=41.5856)



°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min
T1	40.0	41.0	41.8	42.6	43.2	44.1	45.0	45.9	46.8	47.5	48.0	48.4	48.8	49.2	49.6	49.8	50.0	50.0
T2	34.8	36.4	37.6	39.0	40.0	42.0	43.2	44.0	45.0	45.5	46.0	46.6	47.0	47.6	48.0	48.3	48.5	48.5
T3	25.0	29.4	33.6	34.4	35.6	36.2	37.2	38.6	40.0	41.0	41.8	42.8	43.8	44.6	45.2	45.4	45.4	45.4
T4	30.5	31.5	35.8	36.8	37.8	38.4	39.1	40.4	41.8	42.2	43.4	44.4	44.8	45.9	46.5	47.4	47.6	47.6

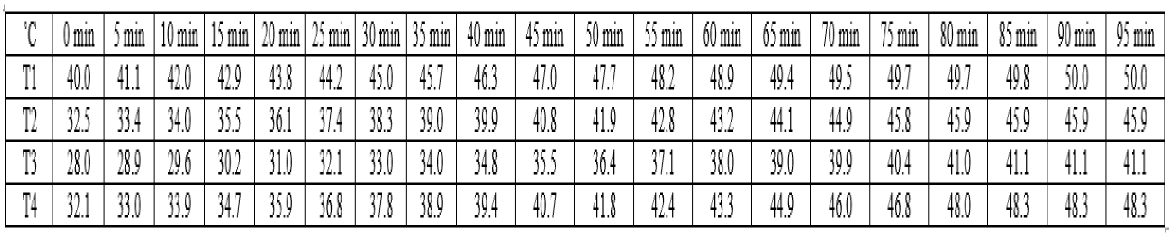
Table 3.18. 5 LPM (T1=43.833, T2=41.5, T3=37.5647, T4=39.733)



°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min
T1	40.0	41.2	42.0	43.4	44.0	44.8	45.4	46.2	47.2	48.0	48.6	49.0	49.4	49.8	50.0	50.0	50.0
T2	36.0	37.0	37.8	39.0	41.4	42.6	43.2	43.8	45.0	46.0	46.8	47.2	47.6	48.0	48.4	48.6	48.6
T3	26.2	28.8	31.0	32.4	34.0	35.6	36.2	37.0	38.2	39.0	40.0	41.2	42.4	43.6	44.2	44.4	44.4
T4	30.0	31.6	34.2	35.0	36.8	37.4	38.1	39.0	40.0	40.6	41.8	42.8	44.0	45.2	45.6	45.8	45.8

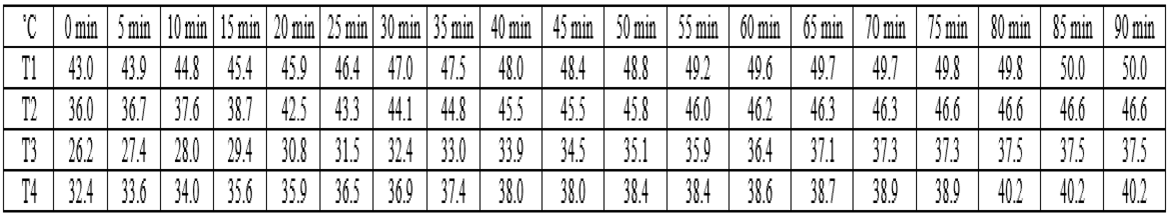
Water+0.3% Al_2O_3

Table 3.19. 1 LPM (T1=46.525, T2=40.66, T3=36.81, T4=41.595)



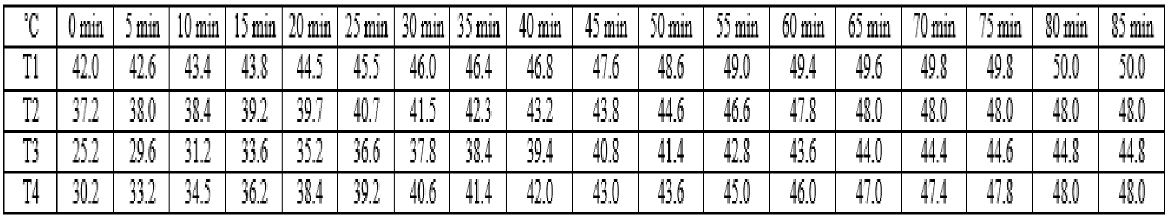
˚C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min	95 min
T1	40.0	41.1	42.0	42.9	43.8	44.2	45.0	45.7	46.3	47.0	47.7	48.2	48.9	49.4	49.5	49.7	49.7	49.8	50.0	50.0
T2	32.5	33.4	34.0	35.5	36.1	37.4	38.3	39.0	39.9	40.8	41.9	42.8	43.2	44.1	44.9	45.8	45.9	45.9	45.9	45.9
T3	28.0	28.9	29.6	30.2	31.0	32.1	33.0	34.0	34.8	35.5	36.4	37.1	38.0	39.0	39.9	40.4	41.0	41.1	41.1	41.1
T4	32.1	33.0	33.9	34.7	35.9	36.8	37.8	38.9	39.4	40.7	41.8	42.4	43.3	44.9	46.0	46.8	48.0	48.3	48.3	48.3

Table 3.20. 2 LPM (T1=48.18, T2=43.77, T3=39.69, T4=43.48)



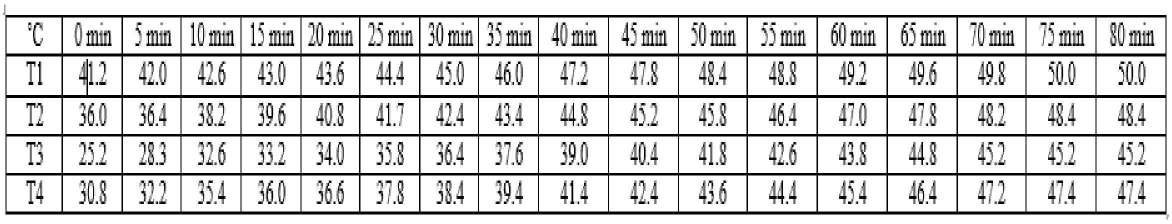
˚C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	43.0	43.9	44.8	45.4	45.9	46.4	47.0	47.5	48.0	48.4	48.8	49.2	49.6	49.7	49.7	49.8	49.8	50.0	50.0
T2	36.0	36.7	37.6	38.7	42.5	43.3	44.1	44.8	45.5	45.5	45.8	46.0	46.2	46.3	46.3	46.6	46.6	46.6	46.6
T3	26.2	27.4	28.0	29.4	30.8	31.5	32.4	33.0	33.9	34.5	35.1	35.9	36.4	37.1	37.3	37.3	37.5	37.5	37.5
T4	32.4	33.6	34.0	35.6	35.9	36.5	36.9	37.4	38.0	38.0	38.4	38.4	38.6	38.7	38.9	38.9	40.2	40.2	40.2

Table 3.21. 3 LPM (T1=46.933, T2=43.25, T3=39.322, T4=42.283)



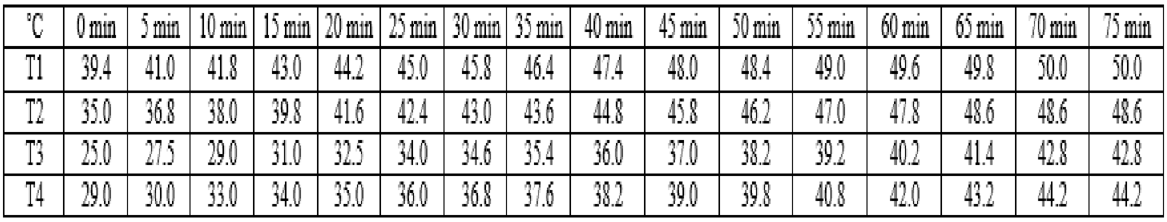
˚C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min
T1	42.0	42.6	43.4	43.8	44.5	45.5	46.0	46.4	46.8	47.6	48.6	49.0	49.4	49.6	49.8	49.8	50.0	50.0
T2	37.2	38.0	38.4	39.2	39.7	40.7	41.5	42.3	43.2	43.8	44.6	46.6	47.8	48.0	48.0	48.0	48.0	48.0
T3	25.2	29.6	31.2	33.6	35.2	36.6	37.8	38.4	39.4	40.8	41.4	42.8	43.6	44.0	44.4	44.6	44.8	44.8
T4	30.2	33.2	34.5	36.2	38.4	39.2	40.6	41.4	42.0	43.0	43.6	45.0	46.0	47.0	47.4	47.8	48.0	48.0

Table 3.22. 4 LPM (T1=46.388, T2=43.582, T3=39.5804, T4=41.998)



˚C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min
T1	41.2	42.0	42.6	43.0	43.6	44.4	45.0	46.0	47.2	47.8	48.4	48.8	49.2	49.6	49.8	50.0	50.0
T2	36.0	36.4	38.2	39.6	40.8	41.7	42.4	43.4	44.8	45.2	45.8	46.4	47.0	47.8	48.2	48.4	48.4
T3	25.2	28.3	32.6	33.2	34.0	35.8	36.4	37.6	39.0	40.4	41.8	42.6	43.8	44.8	45.2	45.2	45.2
T4	30.8	32.2	35.4	36.0	36.6	37.8	38.4	39.4	41.4	42.4	43.6	44.4	45.4	46.4	47.2	47.4	47.4

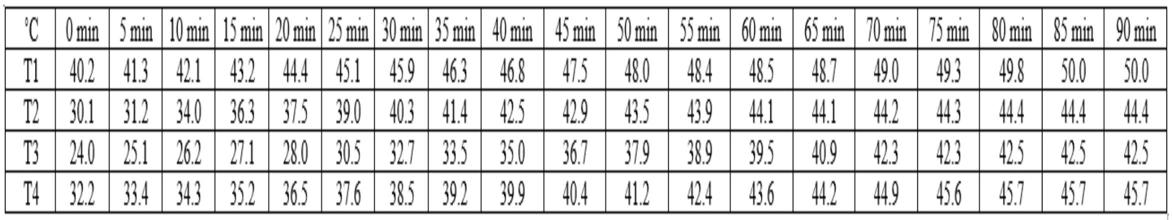
Table 3.23. 5 LPM (T1=46.175, T2=43.6, T3=39.98, T4=42.315)



℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min
T1	39.4	41.0	41.8	43.0	44.2	45.0	45.8	46.4	47.4	48.0	48.4	49.0	49.6	49.8	50.0	50.0
T2	35.0	36.8	38.0	39.8	41.6	42.4	43.0	43.6	44.8	45.8	46.2	47.0	47.8	48.6	48.6	48.6
T3	25.0	27.5	29.0	31.0	32.5	34.0	34.6	35.4	36.0	37.0	38.2	39.2	40.2	41.4	42.8	42.8
T4	29.0	30.0	33.0	34.0	35.0	36.0	36.8	37.6	38.2	39.0	39.8	40.8	42.0	43.2	44.2	44.2

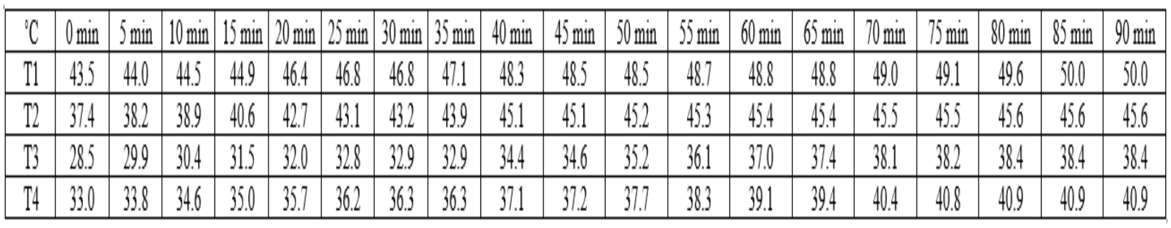
Water+0.1% SiC

Table 3.24. 1 LPM (T1=46.42, T2=40.65, T3=35.94, T4=41.1)



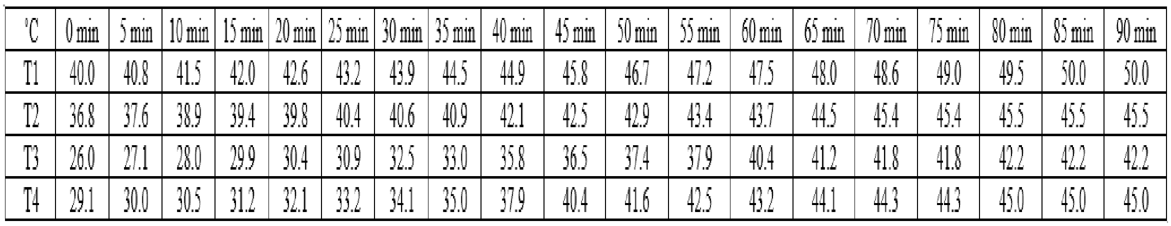
℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	40.2	41.3	42.1	43.2	44.4	45.1	45.9	46.3	46.8	47.5	48.0	48.4	48.5	48.7	49.0	49.3	49.8	50.0	50.0
T2	30.1	31.2	34.0	36.3	37.5	39.0	40.3	41.4	42.5	42.9	43.5	43.9	44.1	44.1	44.2	44.3	44.4	44.4	44.4
T3	24.0	25.1	26.2	27.1	28.0	30.5	32.7	33.5	35.0	36.7	37.9	38.9	39.5	40.9	42.3	42.3	42.5	42.5	42.5
T4	32.2	33.4	34.3	35.2	36.5	37.6	38.5	39.2	39.9	40.4	41.2	42.4	43.6	44.2	44.9	45.6	45.7	45.7	45.7

Table 3.25. 2 LPM (T1=47.43, T2=43.54, T3=39.195, T4=42.21)



℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	43.5	44.0	44.5	44.9	46.4	46.8	46.8	47.1	48.3	48.5	48.5	48.7	48.8	48.8	49.0	49.1	49.6	50.0	50.0
T2	37.4	38.2	38.9	40.6	42.7	43.1	43.2	43.9	45.1	45.1	45.2	45.3	45.4	45.4	45.5	45.5	45.6	45.6	45.6
T3	28.5	29.9	30.4	31.5	32.0	32.8	32.9	32.9	34.4	34.6	35.2	36.1	37.0	37.4	38.1	38.2	38.4	38.4	38.4
T4	33.0	33.8	34.6	35.0	35.7	36.2	36.3	36.3	37.1	37.2	37.7	38.3	39.1	39.4	40.4	40.8	40.9	40.9	40.9

Table 3.26. 3 LPM (T1=45.22, T2=42.14, T3=37.36, T4=40.08)



℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	40.0	40.8	41.5	42.0	42.6	43.2	43.9	44.5	44.9	45.8	46.7	47.2	47.5	48.0	48.6	49.0	49.5	50.0	50.0
T2	36.8	37.6	38.9	39.4	39.8	40.4	40.6	40.9	42.1	42.5	42.9	43.4	43.7	44.5	45.4	45.4	45.5	45.5	45.5
T3	26.0	27.1	28.0	29.9	30.4	30.9	32.5	33.0	35.8	36.5	37.4	37.9	40.4	41.2	41.8	41.8	42.2	42.2	42.2
T4	29.1	30.0	30.5	31.2	32.1	33.2	34.1	35.0	37.9	40.4	41.6	42.5	43.2	44.1	44.3	44.3	45.0	45.0	45.0

Table 3.27. 4 LPM (T1=46.5, T2=44.078, T3=39.543, T4=41.71)

°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min
T1	40.8	41.4	42.2	42.8	43.6	44.4	45.2	46.2	47.0	47.4	48.2	49.0	49.4	49.6	49.8	50.0	50.0	50.0
T2	35.0	36.4	38.8	39.6	41.4	42.2	43.0	44.0	45.2	45.8	46.4	47.0	47.6	47.9	48.0	48.2	48.5	48.5
T3	25.2	29.2	31.0	32.6	33.8	35.0	36.2	37.4	38.8	39.8	41.2	42.2	43.0	44.2	45.0	45.2	45.3	45.3
T4	30.6	31.8	33.6	34.2	36.4	37.2	38.4	39.4	40.4	41.8	42.6	43.5	44.8	45.7	46.6	47.2	47.5	47.5

Table 3.28. 5 LPM (T1=47.0353, T2=44.7058, T3=40.7268, T4=42.7953)

°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min
T1	40.0	42.2	43.6	44.4	45.0	45.8	46.6	47.2	48.0	48.8	49.2	49.4	49.8	49.8	50.0	50.0	50.0
T2	35.2	38.0	39.2	40.8	42.4	43.4	44.1	44.9	46.0	47.0	47.6	48.0	48.4	48.6	48.8	48.8	48.8
T3	25.4	26.8	29.0	30.2	31.2	32.2	33.0	33.8	34.7	35.8	36.8	37.8	39.0	40.4	41.4	41.6	41.6
T4	29.6	30.6	31.0	32.8	33.2	34.1	34.9	35.8	36.5	37.4	38.2	39.4	40.8	41.8	42.6	42.8	42.8

Water+0.2% SiC

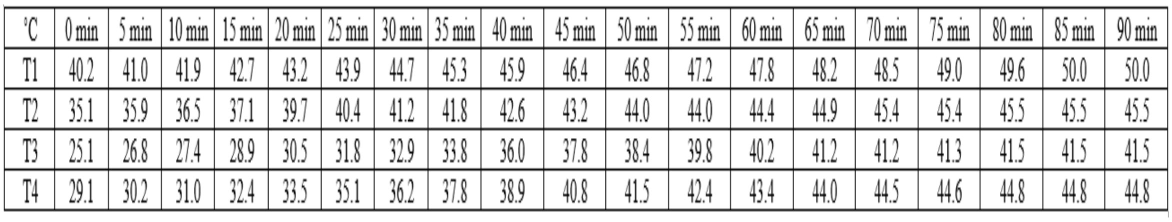
Table 3.29. 1 LPM (T1=45.57, T2=39.48, T3=34.65, T4=40.41)

°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	39.2	39.9	40.8	41.9	42.8	43.7	44.6	45.8	45.8	46.6	47.3	47.9	48.2	48.3	48.5	49.0	49.5	50.0	50.0
T2	27.1	28.3	29.5	32.8	35.7	36.9	37.7	38.4	40.2	41.9	42.4	44.7	44.7	44.9	44.9	45.0	45.0	45.1	45.1
T3	26.0	26.9	27.9	28.8	29.5	30.8	31.7	33.2	34.3	35.8	36.5	37.2	37.4	37.4	38.1	38.2	38.4	38.4	38.4
T4	32.0	33.2	34.6	35.3	37.2	39.8	40.4	40.9	41.3	41.4	41.4	41.8	41.9	42.0	42.1	42.2	42.3	42.3	42.3

Table 3.30. 2 LPM (T1=46.63, T2=42.39, T3=38.23, T4=41.71)

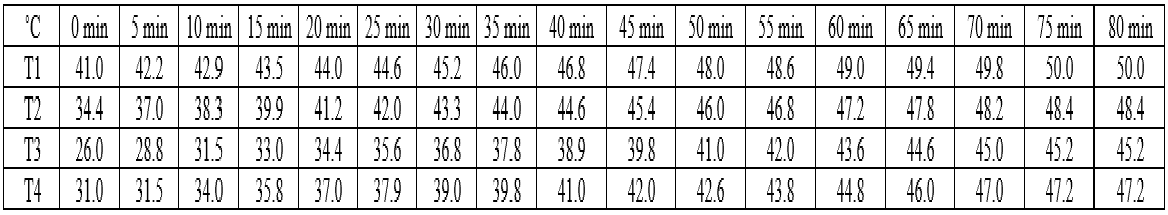
°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	41.0	41.8	42.4	42.9	44.8	45.2	45.8	46.1	47.0	47.6	48.0	48.2	48.4	48.7	48.8	49.0	49.5	50.0	50.0
T2	35.0	35.8	36.5	37.8	40.2	41.0	41.6	41.9	44.8	44.8	45.0	45.0	45.1	45.1	45.1	45.2	45.2	45.2	45.2
T3	27.0	27.8	28.6	29.7	32.8	33.4	34.0	35.2	36.0	36.7	37.1	37.5	37.8	37.8	37.9	37.9	38.0	38.0	38.0
T4	33.0	34.1	34.7	35.4	36.3	36.9	37.3	38.4	38.9	39.8	40.0	40.0	40.2	40.2	40.3	40.4	40.5	40.5	40.5

Table 3.31. 3 LPM (T1=45.67, T2=42.174, T3=37.61, T4=40.89)



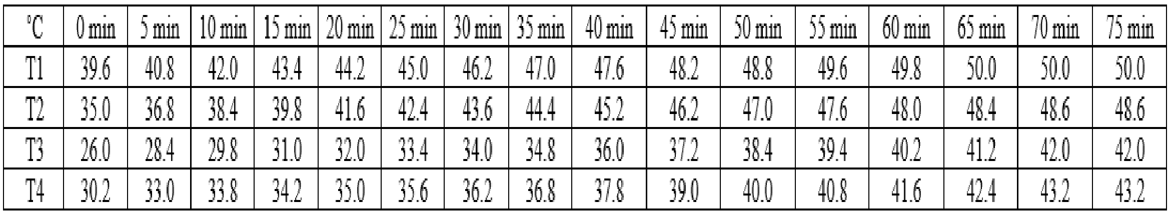
℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	40.2	41.0	41.9	42.7	43.2	43.9	44.7	45.3	45.9	46.4	46.8	47.2	47.8	48.2	48.5	49.0	49.6	50.0	50.0
T2	35.1	35.9	36.5	37.1	39.7	40.4	41.2	41.8	42.6	43.2	44.0	44.0	44.4	44.9	45.4	45.4	45.5	45.5	45.5
T3	25.1	26.8	27.4	28.9	30.5	31.8	32.9	33.8	36.0	37.8	38.4	39.8	40.2	41.2	41.2	41.3	41.5	41.5	41.5
T4	29.1	30.2	31.0	32.4	33.5	35.1	36.2	37.8	38.9	40.8	41.5	42.4	43.4	44.0	44.5	44.6	44.8	44.8	44.8

Table 3.32. 4 LPM (T1=46.3765, T2=43.7, T3=39.5777, T4=41.365)



℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min
T1	41.0	42.2	42.9	43.5	44.0	44.6	45.2	46.0	46.8	47.4	48.0	48.6	49.0	49.4	49.8	50.0	50.0
T2	34.4	37.0	38.3	39.9	41.2	42.0	43.3	44.0	44.6	45.4	46.0	46.8	47.2	47.8	48.2	48.4	48.4
T3	26.0	28.8	31.5	33.0	34.4	35.6	36.8	37.8	38.9	39.8	41.0	42.0	43.6	44.6	45.0	45.2	45.2
T4	31.0	31.5	34.0	35.8	37.0	37.9	39.0	39.8	41.0	42.0	42.6	43.8	44.8	46.0	47.0	47.2	47.2

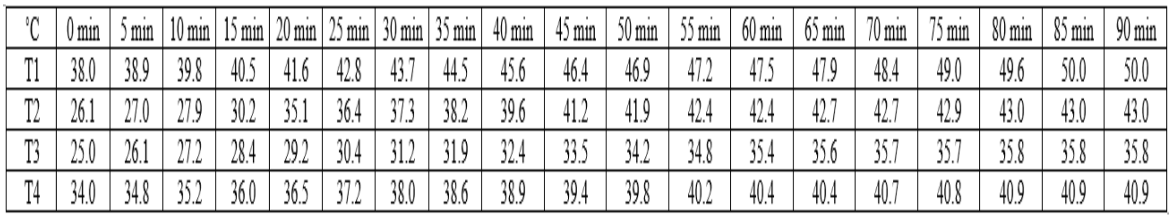
Table 3.33. 5 LPM (T1=46.3875, T2=43.85, T3=40.104, T4=42.4075)



℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min
T1	39.6	40.8	42.0	43.4	44.2	45.0	46.2	47.0	47.6	48.2	48.8	49.6	49.8	50.0	50.0	50.0
T2	35.0	36.8	38.4	39.8	41.6	42.4	43.6	44.4	45.2	46.2	47.0	47.6	48.0	48.4	48.6	48.6
T3	26.0	28.4	29.8	31.0	32.0	33.4	34.0	34.8	36.0	37.2	38.4	39.4	40.2	41.2	42.0	42.0
T4	30.2	33.0	33.8	34.2	35.0	35.6	36.2	36.8	37.8	39.0	40.0	40.8	41.6	42.4	43.2	43.2

Water+0.3% SiC

Table 3.34. 1 LPM (T1=44.64, T2=38.05, T3=33.56, T4=39.85)



℃	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	38.0	38.9	39.8	40.5	41.6	42.8	43.7	44.5	45.6	46.4	46.9	47.2	47.5	47.9	48.4	49.0	49.6	50.0	50.0
T2	26.1	27.0	27.9	30.2	35.1	36.4	37.3	38.2	39.6	41.2	41.9	42.4	42.4	42.7	42.7	42.9	43.0	43.0	43.0
T3	25.0	26.1	27.2	28.4	29.2	30.4	31.2	31.9	32.4	33.5	34.2	34.8	35.4	35.6	35.7	35.7	35.8	35.8	35.8
T4	34.0	34.8	35.2	36.0	36.5	37.2	38.0	38.6	38.9	39.4	39.8	40.2	40.4	40.4	40.7	40.8	40.9	40.9	40.9

Table 3.35. 2 LPM (T1=44.88, T2=40.21, T3=36.23, T4=40.26)

°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	40.0	40.7	41.4	42.2	43.1	43.4	43.8	44.1	46.0	46.4	46.7	47.0	47.3	48.1	48.6	49.1	49.6	50.0	50.0
T2	32.8	33.5	34.4	35.2	38.5	38.9	39.4	39.7	41.7	42.0	42.2	42.3	42.9	43.3	43.3	43.4	43.5	43.5	43.5
T3	26.0	26.7	27.4	27.9	31.0	31.6	32.4	32.9	33.1	33.6	34.2	34.7	34.8	35.4	35.8	35.9	36.0	36.0	36.0
T4	32.0	32.9	33.5	34.0	34.8	35.4	35.9	36.4	36.5	36.9	37.6	38.0	38.4	38.9	39.2	39.3	39.4	39.4	39.4

Table 3.36. 3 LPM (T1=45.44, T2=41.54, T3=37.46, T4=40.96)

°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min	85 min	90 min
T1	40.2	41.0	41.8	42.4	43.0	43.5	43.9	44.4	46.2	46.8	47.5	47.5	47.7	48.3	48.8	49.0	49.5	50.0	50.0
T2	36.0	36.8	37.5	38.2	39.2	39.7	40.1	40.6	41.8	42.5	43.3	43.4	43.8	44.2	44.2	44.4	44.5	44.5	44.5
T3	25.1	25.8	26.4	28.1	29.8	30.5	31.2	31.8	32.4	33.8	34.4	35.5	36.4	36.9	37.9	39.5	39.6	39.8	39.8
T4	29.0	29.9	30.5	31.2	32.8	33.5	34.0	34.8	36.0	37.5	38.4	40.5	40.9	41.5	41.8	42.0	42.3	42.3	42.3

Table 3.37. 4 LPM (T1=47.2045, T2=44.1812, T3=40.2243, T4=42.9885)

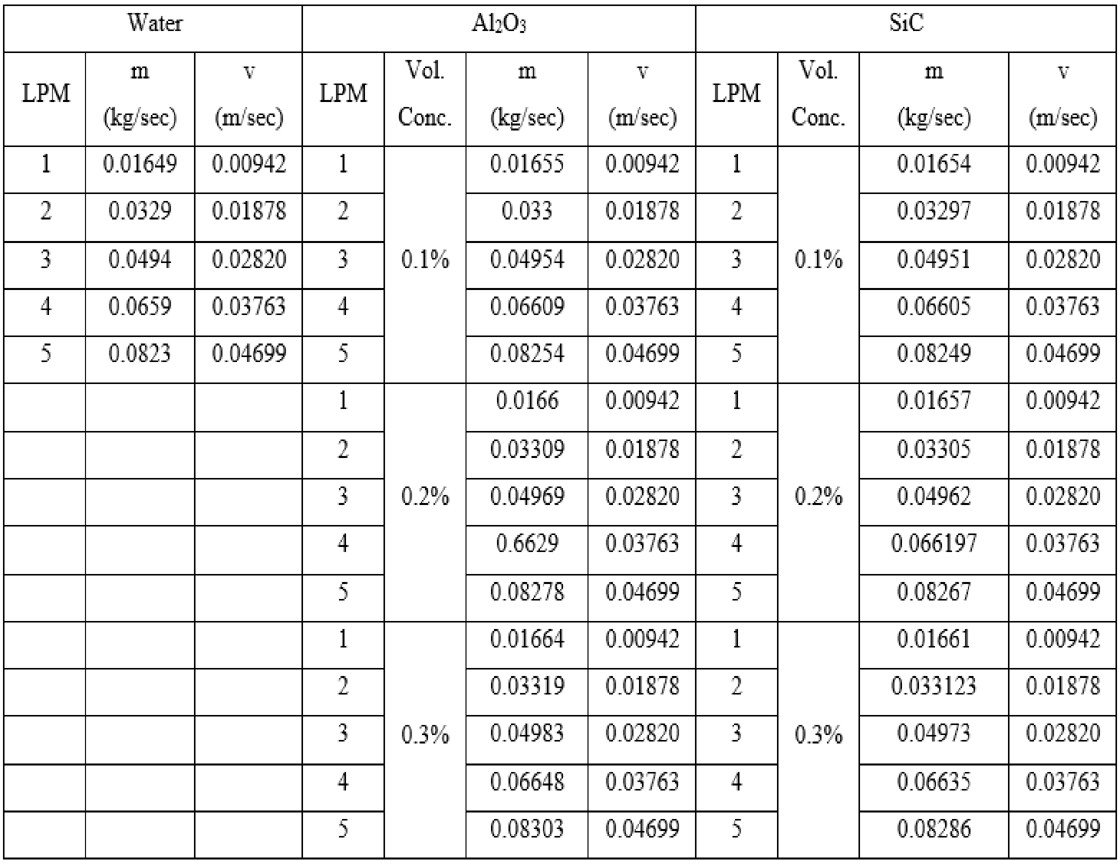
°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min	80 min
T1	41.2	42.0	42.6	43.0	43.6	44.4	45.0	46.0	47.2	47.8	48.4	48.8	49.2	49.6	49.8	50.0	50.0
T2	36.0	36.4	38.2	39.6	40.8	41.7	42.4	43.4	44.8	45.2	45.8	46.4	47.0	47.8	48.2	48.4	48.4
T3	25.2	28.3	32.6	33.2	34.0	35.8	36.4	37.6	39.0	40.4	41.8	42.6	43.8	44.8	45.2	45.2	45.2
T4	30.8	32.2	35.4	36.0	36.6	37.8	38.4	39.4	41.4	42.4	43.6	44.4	45.4	46.4	47.2	47.4	47.4

Table 3.38. 5 LPM (T1=46.53125, T2=43.625, T3=40.165, T4=42.84865)

°C	0 min	5 min	10 min	15 min	20 min	25 min	30 min	35 min	40 min	45 min	50 min	55 min	60 min	65 min	70 min	75 min
T1	40.0	41.6	43.0	43.8	44.5	45.1	45.9	46.8	47.6	48.4	49.0	49.4	49.6	49.8	50.0	50.0
T2	34.2	36.0	38.6	40.8	41.7	42.3	43.0	43.9	44.7	44.8	46.8	47.6	48.0	48.2	48.2	48.2
T3	25.0	28.0	31.0	32.8	33.7	34.3	35.3	36.2	37.1	38.0	39.0	39.9	40.8	41.4	41.6	41.6
T4	30.0	34.0	34.6	35.6	36.4	37.0	37.8	38.6	39.5	40.1	40.9	41.6	42.4	43.0	43.2	43.2

Table 3.39 shows calculated results of mass flow rate and velocity for different volume concentrations and LPM. Table 3.40 shows calculated results of heat transfer rate, heat trasnfer coefficient, Reynolds number, Nusselt number and Prandtl number. It is found that heat transfer rate and heat transfer coefficient increases with increase in LPM and volume concentration. The lowest was noted for 1 LPM water and the highest was noted for Water+0.3% SiC. The same pattern was also noted for Reynolds number and Nusselt number.

Table 3.39. Experimental volume concentration, mass flow rate and velocity values



Water			Al_2O_3				SiC			
LPM	m (kg/sec)	v (m/sec)	LPM	Vol. Conc.	m (kg/sec)	v (m/sec)	LPM	Vol. Conc.	m (kg/sec)	v (m/sec)
1	0.01649	0.00942	1	0.1%	0.01655	0.00942	1	0.1%	0.01654	0.00942
2	0.0329	0.01878	2		0.033	0.01878	2		0.03297	0.01878
3	0.0494	0.02820	3		0.04954	0.02820	3		0.04951	0.02820
4	0.0659	0.03763	4		0.06609	0.03763	4		0.06605	0.03763
5	0.0823	0.04699	5		0.08254	0.04699	5		0.08249	0.04699
			1	0.2%	0.0166	0.00942	1	0.2%	0.01657	0.00942
			2		0.03309	0.01878	2		0.03305	0.01878
			3		0.04969	0.02820	3		0.04962	0.02820
			4		0.6629	0.03763	4		0.066197	0.03763
			5		0.08278	0.04699	5		0.08267	0.04699
			1	0.3%	0.01664	0.00942	1	0.3%	0.01661	0.00942
			2		0.03319	0.01878	2		0.033123	0.01878
			3		0.04983	0.02820	3		0.04973	0.02820
			4		0.06648	0.03763	4		0.06635	0.03763
			5		0.08303	0.04699	5		0.08286	0.04699

Table 3.40. Experimental heat transfer rate, heat transfer coefficient, Reynolds number, Nusselt number, Prandtl number at different LPM

Fluid	LPM	Vol. Con.	Q	h	Re	Nu	Pr
Water	1	N/A	348.882	36818.8	216.089	726.649	3.5549
	2		414.731	44938.1	430.802	886.89	
	3		478.898	53231.7	646.891	1050.57	
	4		494.69	57381	863.21	1132.46	
	5		539.534	74664.8	1077.92	1473.57	
Al_2O_3	1	0.1%	372.943	42655.6	216.164	839.495	3.5431
	2		476.035	54277.5	430.952	1068.22	
	3		567.45	66271.2	647.116	1304.27	
	4		597.481	71285.2	863.51	1402.95	
	5		706.485	93473.3	1078.3	1839.63	
	1	0.2%	388.086	46362.1	216.564	909.763	3.5256
	2		549.84	65699.4	431.749	1289.22	
	3		648.272	80248.8	648.313	1574.72	
	4		668.895	85433.3	865.107	1676.46	
	5		802.67	112725	1080.29	2212	
	1	0.3%	404.348	52226.5	216.405	1022	3.5176
	2		606.428	78060.9	431.432	1527.54	
	3		760.372	100252	647.837	1961.79	
	4		772.881	103993	864.471	2034.99	
	5		885.821	133665	1079.5	2615.63	
SiC	1	0.1%	398.017	44829.2	216.019	882.41	3.5448
	2		534.883	63272.4	430.661	1245.44	
	3		635.966	72366.5	646.68	1424.45	
	4		667.171	80746.1	862.928	1589.39	
	5		801.409	110054	1077.57	2166.28	
	1	0.2%	419.68	47417.1	216.271	930.897	3.5295
	2		582.795	72589.2	431.165	1425.08	
	3		721.45	87131.5	647.436	1710.57	

	4		736.857	96053.3	863.936	1885.73	
	5		872.434	127449	1078.83	2502.09	
	1	0.3%	453.99	55217.2	2195.967	1081.19	3.5234
	2		641.561	84327.3	430.559	1651.18	
	3		804.404	106106	464.526	2077.62	
	4		831.981	114897	862.723	2249.76	
	5		998.779	157826	1077.31	3090.33	

Figure 3.3 and 3.4 shows the effect of nanoparticles concentration on Nusselt number with Reynolds number. From these figures, it can be concluded that changes in concentration of nanoparticles increases the Reynolds number hence Nusselt number. It was due to increase in heat transfer and Reynolds number on addition of nanoparticles in the fluid which improved thermo-physical properties of the mixture. Similar results were reported by Singh et al. (2016) [48]. Figure 3.4 revealed that heat transfer enhancement was more in comparison to Al_2O_3 for same volume concentration.

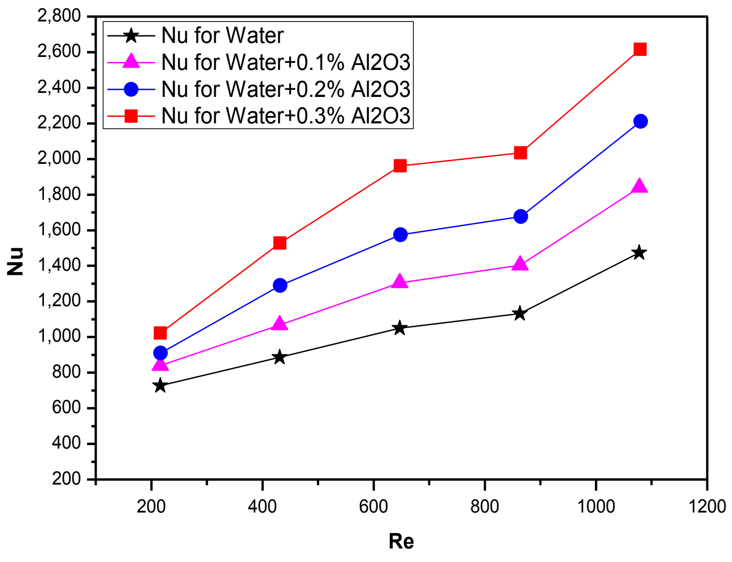


Figure 3.3. Experimental Re vs Nu for Water+0.1-0.3% Al_2O_3

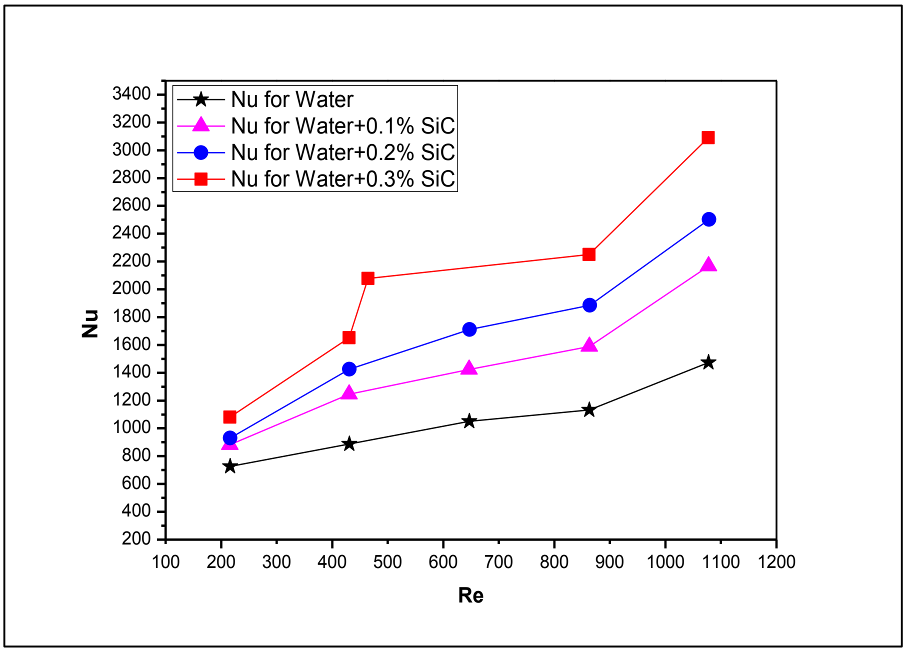


Figure 3.4. Experimental Re vs Nu for Water+0.1-0.3% SiC

Figure 3.5 clearly shows that with the increase in flow rate from 1-5 LPM, the heat transfer rate also increases. This eventually increases the heat transfer coefficient which further increases Nusselt number.

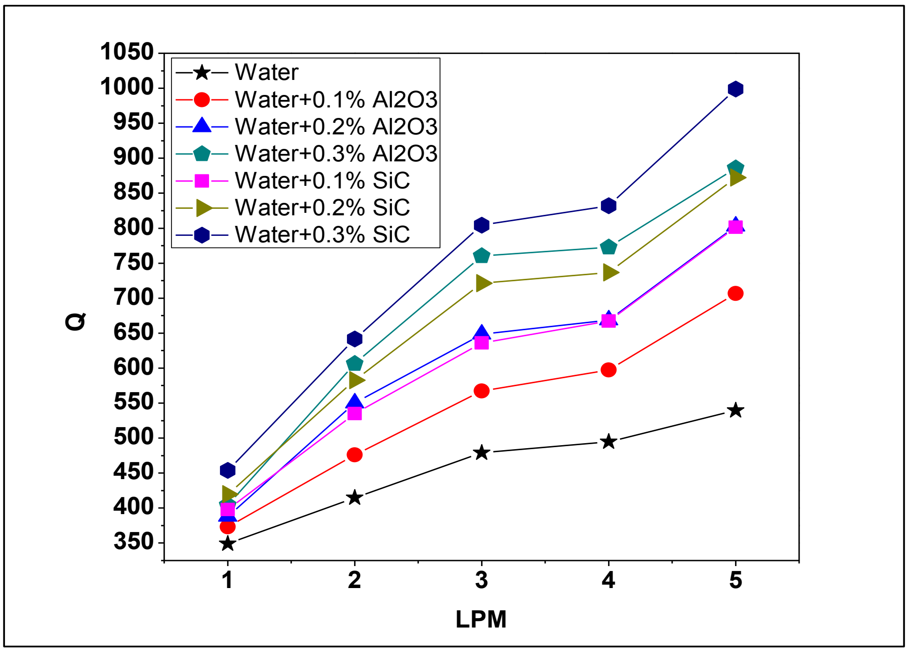


Figure 3.5. Experimental LPM vs Q for Water+0.1-0.3% Al_2O_3/0.1-0.3% SiC

Table 3.41 presents data for theoretical Reynolds number, Nusselt number and Prandtl number at different LPM. We used Li and Xuan [49] correlation to verify our data theoretically. We obtained similar pattern for the Reynolds number and Nusselt number; the highest being for Water+0.3% SiC and the lowest being for Water+0.1% Al_2O_3 as shown in figure 3.5.

Table 3.41. Theoretical Reynolds number, Nusselt number, Prandtl number at different LPM

Fluid	LPM	Vol. Con.	Re	Nu	Pr
Water	1	N/A	216.089	N/A	3.5549
	2		430.802		
	3		646.891		
	4		863.21		
	5		1077.92		
Al_2O_3	1	0.1%	216.164	4.28966	3.5431
	2		430.952	5.39765	
	3		647.116	5.53588	
	4		863.51	6.80326	
	5		1078.3	7.82557	
	1	0.2%	216.564	4.29464	3.5256
	2		431.749	5.40392	
	3		648.313	6.1873	
	4		865.107	6.81116	
	5		1080.29	7.33408	
	1	0.3%	216.405	4.30041	3.5176
	2		431.432	5.41117	
	3		647.837	6.1956	
	4		864.471	6.8203	

	5		1079.5	7.34393	
SiC	1	0.1%	216.019	4.28973	3.5448
	2		430.661	5.39774	
	3		646.68	6.18002	
	4		862.928	6.80336	
	5		1077.57	7.3257	
	1	0.2%	216.271	4.29468	3.5295
	2		431.165	5.40396	
	3		647.436	6.18735	
	4		863.936	6.81122	
	5		1078.83	7.33414	
	1	0.3%	215.967	4.30054	3.5234
	2		430.559	5.41135	
	3		464.526	6.1958	
	4		862.723	6.82052	
	5		1077.31	7.34417	

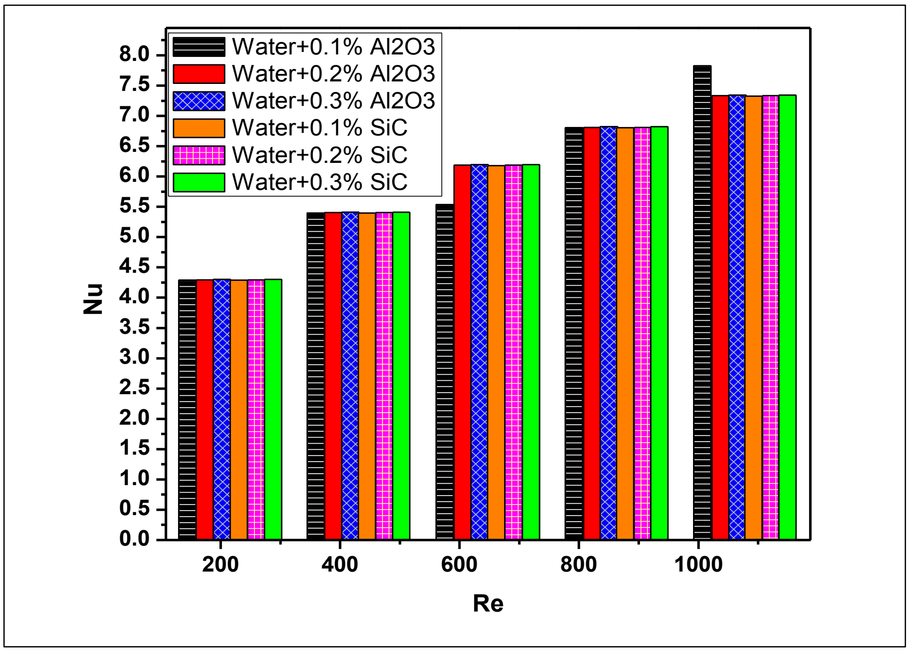


Figure 3.6. Theoretical Re vs Nu for Water+0.1-0.3% Al_2O_3/0.1-0.3% SiC

3.8. CONCLUSION

- Heat transfer rate increases with increasing concentration of nanoparticle.
- Heat transfer rate is directly proportional to the Reynolds number and Peclet number of nanofluid.
- The fine grade of nanoparticles increases the heat transfer rate but it’s having poor stability.
- Clustering and collision of nanoparticles is main factor to affect the heat transfer rate of nanofluid.
- Concentration of nanoparticles increases the pressure drop of nanofluid.
- Spherical shaped nanoparticles increases the heat transfer rate of nanofluid compared with other shaped nanoparticles.

CHAPTER

4

CODING FOR PROJECT

4.1. PROGRAM FOR FINDING LMTD, Area, Q, h, Re, Nu

```
#include<iostream>
#include<conio.h>
#include<math.h>
using namespace std;
int main()
{
  float t1,t2,t3,t4,n,d,m,cp,p,v,d1,u,d2,k,d3;
  cout<<"Enter the value for T1\n";
  cin>>t1;
  cout<<"Enter the value for T2\n";
  cin>>t2;
  cout<<"Enter the value for T3\n";
  cin>>t3;
  cout<<"Enter the value for T4\n";
  cin>>t4;
  float dt1=t1-t4;
  float dt2=t2-t3;
  float dt11=dt1/dt2;
  float dt22=log(dt11);
  float lmtd=(dt1-dt2)/dt22;
  cout<<"\n value of LMTD="<<lmtd<<endl;
  cout<<"\n value of dt1="<<dt1<<endl;
  cout<<"\n value of dt2="<<dt2<<endl;
```

```
cout<<"\nEnter the value for n\n";
cin>>n;
cout<<"Enter the value for d\n";
cin>>d;
float a=n*0.785*d*d;
cout<<"\n value of A="<<a<<endl;
cout<<"\nEnter the value for m\n";
cin>>m;
cout<<"Enter the value for Cp\n";
cin>>cp;
float dt=t1-t2;
float q=m*cp*dt;
cout<<"\n value of Q="<<q<<endl;
float h1=a*lmtd;
float h=q/h1;
cout<<"\n value of h="<<h<<endl;
cout<<"\nFor Re\n";
cout<<"Enter the value for p\n";
cin>>p;
cout<<"Enter the value for v\n";
cin>>v;
cout<<"Enter the value for d\n";
cin>>d1;
cout<<"Enter the value for u\n";
cin>>u;
float r1=p*v*d1;
float r=r1/u;
cout<<"\n value of Re="<<r<<endl;
cout<<"\nFor Nu\n";
cout<<"Enter the value for h\n";
cin>>d2;
```

```
  cout<<"Enter the value for d\n";
  cin>>d3;
  cout<<"Enter the value for k\n";
  cin>>k;
  float Nu=(d2*d3)/k;
  cout<<"\n value of Nu="<<Nu<<endl;
  getch();
  return 0;
}
```

4.2. PROGRAM FOR FINDING EFFECTIVE THERMAL CONDUCTIVITY USING HAMILTON AND CROSSER MODEL

```
#include<iostream>
#include<conio.h>
#include<math.h>
using namespace std;
int main()
{
  float f, n, kp, kbf, b, p;
 cout<<"Enter the value for kp\n";
  cin>>kp;
  cout<<"Enter the value for kbf\n";
  cin>>kbf;
  cout<<"Enter the value for b\n";
  cin>>b;
  cout<<"Enter the value for p\n";
  cin>>p;
 float d1=kp+(n-1)*kbf;
  float d2=(n-1)*p*(kp-kbf);
 float d3=kp+(n-1)*kbf+p*(kp-kbf);
 f=(d1-d2)/d3;
 cout<<"\n value of f="<<f<<endl;
 getch();
  return 0;
}
```

4.3. PROGRAM FOR FINDING EFFECTIVE THERMAL CONDUCTIVITY USING WASP MODEL

```
#include<iostream>
#include<conio.h>
#include<math.h>
using namespace std;
int main()
{
  float f, kp, kbf, b, p;
   cout<<"Enter the value for kp\n";
  cin>>kp;
  cout<<"Enter the value for kbf\n";
  cin>>kbf;
  cout<<"Enter the value for b\n";
  cin>>b;
  cout<<"Enter the value for p\n";
  cin>>p;
   float d1=kp+(2*kbf);
  float d2=2*(kp-kbf)*p;
   float d3=kp+(2*kbf)+(kp-kbf)*p;
   f=(d1-d2)/d3;
   cout<<"\n value of f="<<f<<endl;
   getch();
  return 0;
}
```

4.4. PROGRAM FOR FINDING EFFECTIVE THERMAL CONDUCTIVITY USING MAXWELL MODEL

```
#include<iostream>
#include<conio.h>
#include<math.h>
using namespace std;
int main()
{
  float f, kp, kbf, b, p;
   cout<<"Enter the value for kp\n";
  cin>>kp;
  cout<<"Enter the value for kbf\n";
  cin>>kbf;
  cout<<"Enter the value for b\n";
  cin>>b;
  cout<<"Enter the value for p\n";
  cin>>p;
   float d1=kp+(2*kbf);
  float d2=2*(kp-kbf)*p;
   float d3=kp+(2*kbf)-(kp-kbf)*p;
   f=(d1+d2)/d3;
   cout<<"\n value of f="<<f<<endl;
   getch();
  return 0;
}
```

4.5. PROGRAM FOR FINDING EFFECTIVE THERMAL CONDUCTIVITY USING YU AND CHOI MODEL

```
#include<iostream>
#include<conio.h>
#include<math.h>
using namespace std;
int main()
{
  float f, f1, f2, kp, kbf, b, p;
   cout<<"Enter the value for kp\n";
  cin>>kp;
  cout<<"Enter the value for kbf\n";
  cin>>kbf;
  cout<<"Enter the value for b\n";
  cin>>b;
  cout<<"Enter the value for p\n";
  cin>>p;
   float d1=kp+(2*kbf);
  float d2=2*(kp-kbf)*f2*p;
  f1=1+b;
  f2=pow(f1,3);
  float d3=kbf;
   float d4=kp+(2*kbf)-(kp-kbf)*f2;
   f=(d1+d2)*d3/d4;
   cout<<"\n value of f="<<f<<endl;
   getch();
  return 0;
}
```

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