EXPERIMENTAL INVESTIGATION OF CLOSED LOOP PULSATING HEAT PIPE WITH DIFFERENT FILLING RATIO OF WORKING FLUID

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APPROVAL

The Thesis Entitled "**Experimental Investigation of Closed Loop Pulsating Heat Pipe with Different Filling Ratio of Working Fluid**" Submitted by **Zubayer Hossain**, **Mehedi Hassan**, **Nasirul Haque Nishad**, **Rony Kumar Voumik** and **Kamrul Hasan** to the Department of Mechanical Engineering, Sonargaon University (SU), Dhaka, Bangladesh, has been accepted as satisfactory for science in Mechanical Engineering approved as to its style and contents.

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ABSTRACT

With the advancement of science and engineering the need for space restricted cooling devices has increased. Closed loop pulsating heat pipe (CLPHP) is a new promising technology for heat transfer of microelectronics. In it by simple mechanism heat is transferred effectively and efficiently. The combination of processes like bubble nucleation, collapse, formation of vapor plugs and slugs, agglomeration, dynamic instabilities and temperature/pressure perturbation, etc. leads to exceptional heat transfer capability of the heat pipe. The aim of this research paper is to better understand the operation of PHP through experimental investigations and obtain comparative results for different parameters. A series of experiments are conducted on a closed loop PHP (CLPHP) with 8 loops made of copper capillary tube of 2 mm inner diameter. The heat pipe structure is using normal, CLPHP. DI Water, Ethanol, Methanol and Acetone is taken as the working fluid. The operating characteristics are studied for the variation of heat input, filling ratio (FR) and orientation. The single filling ratios are 0%, 30%, 50%, 70% and mixed filling ratio are (50%-50%) based on its total volume. The orientations are 0° (vertical). This paper attempts to demonstrate the effect of variation of different parameters on the same structure as well as the variation of thermal performance depending on the presence of wire insert and fins on different structures. The experiment demonstrates the effect of filling ratio and inclination angle and structural variation on the performance, operational stability and heat transfer capability of ethanol as working fluid of CLPHP. Important insight of the operational characteristics of CLPHP is obtained and optimum performance of CLPHP using ethanol and Methanol is thus identified. Acetone and Methanol (Mixed) works best at 50% FR at wide range of heat inputs for all structures of CLPHP. The best performance is obtained with normal structures. The optimum performance of the device can be obtained at vertical position.

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NOMENCLATURE

R_{th}	=	Thermal resistance (°C/W)
T	=	Temperature difference (°C)
Q	=	Heat input (W)
p	=	Density (Kg/m3)
V	=	Specific Volume (m3/kg)
Cr	=	Specific Heat (kJ/kg-K)
s	=	Specific Entropy (kJ/kg-K)
e	=	Specific Enthalpy (kJ/kg)
μ	=	Dynamic Viscosity (cP)
L	=	Length(mm)
T_e	=	Evaporator Temperature (°C)
T_c	=	Condensation Temperature (°C)
S_o	=	Standard molar entropy
E	=	Specific enthalpy of vaporization
t_{freeze}	=	Freezing temperature
t_{boil}	=	Boiling temperature

ABBREVIATION

PHP	=	Pulsating Heat Pipe
CLPHP	=	Closed Loop Pulsating Heat Pipe
OLPHP	=	Open Loop Pulsating Heat Pipe
OEPHP	=	Open End Pulsating Heat Pipe
FR	=	Filling Ratio(%)
OD	=	Outer Diameter (mm)
ID	=	Inner Diameter (mm)
CV	=	Check Valve

CHAPTER l

INTRODUCTION

1.1 Introduction

With the advancement of modern society, the need for miniaturization and compactness has increased. Entire human civilization now opts for smart and convenient devices that are easy to carry and user friendly. Everyday latest versions of existing electronics are being launched and new designs are being fabricated. This has brought the present society face to face with problems of high power dissipation and heat density. Market demand for efficient microelectronics has posed the challenge of thermal management of increased power levels coupled with high heat fluxes.

To mitigate and solve this problem of power electronics, the use of liquid-vapor phase change cooling devices such as the heat pipes, have been introduced. Although the conventional heat pipes (e.g. mini or micro) are one of the proven technologies, the manufacturing of the complex, miniaturized wick structure/geometry of these heat pipes could become the most cost intensive factor. Another common limitation is the capillary limit, which occurs when the wick structure cannot return an adequate amount of liquid back to the evaporator.

With a view to overcome these difficulties researchers have come up with pulsating heat pipes (PHP) which work on the principle of oscillation of the working fluid and phase change phenomenon in a capillary tube. PHP is a meandering tube of capillary dimensions with many turns filled partially with a suitable working fluid with no wick structure. PHPs are passive two-phase thermal control devices first introduced by Akashi et al [1-4]. In this application, reduced diameter channels are used, which are directly influenced by the selected working fluid. The vapor plugs generated by the evaporation of the working fluid push the liquid slugs toward the condensation section and this motion causes flow oscillations that guide the device operation [5].

performance of a PHP depends upon many factors like the geometrical parameters of flow channel, the working fluid, the filling ratio, and number of turns, PHP configuration and the inclination angle [5]. Toe purposes of this investigation are to study the heat transfer characteristics of an CLPHP and evaluate several issues related to its performance.

There are many engineering practical situations where heat is being transferred under conditions of pulsating and reciprocating flow such as the operation of modem power producing facilities and industrial equipment used in metallurgy, aviation, chemical and food technology. Cavitation's in hydraulic pipelines, pressure surges and flow of blood are also some of familiar instance of such flows. The performance of this equipment in thermal engineering application is affected by the pulsating flow parameters (Al-Had-dad and Al-Binally, 1989). During the past few decades, numerous studies have been devoted to this pulsating flow and its associate heat transfer problems. A review of these studies with emphases on the onset of turbulence, velocity distribution and pipe flow as well as the heat transfer characteristics including axial heat transfer enhancement and convective heat transfer are presented in the following section.

1.2 Motivation

The Motivation for this research comes basically from the need of a compact thermal management system suitable for the present blossoming technology based civilization. The dearth of work on closed loop pulsating heat pipe has also championed the author's pursuit of the subject.

1.3 Objectives

This thesis work is designed with an aim with following objectives:

- To analysis the performance of the normal device.
- To compare the thermal resistances of the system with different and mixed working fluid with different filling ratios.
- To understand the different operational regimes of closed loop pulsating heat pipe.
- To determine optimum condition for the working fluid to be used in commercial basis in heat pipes for heat transfer and cooling applications.
- To determine the most efficient and beneficial working range of the device for which fluid and which inclination, maximum possible desired results can be obtained.

CHAPTER 2

LITERATURE REVIEW

2.1 Heat Pipe

Heat pipes are hollow metal pipes filled with a liquid coolant that moves heat by evaporating and condensing in an endless cycle. It combines the principles of both thermal conductivity and phase transition to efficiently manage the transfer of heat between two solid interfaces. As the lower end of the Heat pipe is exposed to heat, the coolant within it starts to evaporate, absorbing heat. As the coolant turns into vapor, it, and its heat load, convection within the heat pipe. The reduced molecular density forces the vaporized coolant upwards, where it is exposed to the cold end of the Heat pipe. The coolant then condenses back into a liquid state, releasing the latent heat. Since the rate of condensation increases with increased delta temperatures between the vapor and Heat pipe surface, the gaseous coolant automatically streams towards the coldest spot within the Heat pipe. As the coolant condenses, and its molecular density increases once more, gravitational forces pull the coolant towards the lower end of the Heat pipe. To aid this coolant cycle, improve its performance, and make it less dependent on the orientation of the Heat pipe towards earth gravitational center, modern Heat pipes feature inner walls with a fine, capillary structure. The capillary surfaces within the Heat pipe break the coolants surface tension, distributing it evenly throughout the structure. As soon as coolant evaporates on one end, the coolants surface tension automatically pulls in fresh coolant from the surrounding area. As a result of the self-organizing streams of the coolant in both phases, heat is actively convection through Heat pipes throughout the entire coolant cycle, at a rate unmatched by solid Heat spreaders and Heat sinks.

Heat pipes enable passive cooling solutions for high heat load and high temperature equipment, lacking moving parts and boasting extraordinary lifetimes as a result. The idea of heat pipe was given by Gaugler [6] from General Motors. However, the technology of that period presented no clear need for such a device and it lay dormant for two decades. Then it was in 1963 when a Los Alamos National Research Laboratory engineer named George Grover [6] demonstrated the first heat pipe. Heat pipe technology was borrowed from simple heat conducting pipes used by English bakers 100 years ago. Since 1963, heat pipes progressed and modem applications of this technology range from miniature heat pipes for cooling processors inside laptop computers, to groups of half inch diameter and five feet long pipes that will be used in NASA spacecraft, to pipes of two inches diameters (or more) which are used to cool injection molds used in plastic forming. The lengths of the pipes can vary from inches to 24 feet or more.

Starting in the 1980s Sony began incorporating heat pipes into the cooling schemes for some fits commercial electronic products in place of both forced convection and passive finned heat sinks. Initially they were used in tuners and amplifiers, soon spreading to other high heat flux electronics applications.

2.2 Closed Loop Pulsating Heat Pipe

Closed pulsating is a new addition to the heat pipe family. The principle difference is that CLPHP has no wick structure. This enable the working fluid to transfer heat by formation and collapse of vapor bubbles. The vapor formed at the evaporator is pushed towards the condenser in the form of discrete vapor bubbles among packets of fluid at the condenser. At the condenser the vapor gets condensed and releases the latent heat of vaporization and returns to the evaporator to complete the cycle. The entire essence of thermo-mechanical

Fig 2.1: model of CLPHP

physics lies in the closed (constant volume), two- phase, bubble-liquid slug system formed inside the tube-bundle due to the dominance of Surface tension forces.

These can be divided into 3 groups at least: (a) closed loop PHP (CLPHP); (b) CLPHP with check valves; (c) open loop PHP (OLPHP), also called closed end PHP (CEPHP). It is simple in structure with a coil of capillary tubes filled with certain working fluid in it and extended from the heat source to sink. Unlike a conventional heat pipe, PHP having no wick structure prevents the condensate from returning to the evaporator section. PHP works on the principle of fluid pressure oscillations created by means of differential pressure across vapor plugs from evaporator to condenser and back [7].

2.3 Operation features

A PHP is a complex heat transfer device with a strong thermo-hydraulic coupling governing its performance. It is essentially a non-equilibrium heat transfer device. The performance success of the device primarily depends on the continuous maintenance or sustenance of these non-equilibrium conditions within the system. The liquid and vapor slug transport results because of the pressure pulsations caused in the system. Since these pressure pulsations are fully thermally driven, because of the inherent constructions of the device, there is no external mechanical power source required for the fluid transport.

Consider a case when a PHP is kept isothermal throughout, say at room temperature. In this case, the liquid and vapor phases inside the device must exist in equilibrium at the saturated pressure corresponding to the fixed isothermal temperature. Referring to the pressure-enthalpy diagram, the thermodynamic state of all the liquid plugs, irrespective of their size and position, can be represented by point A. Similarly point B represents the thermodynamic state of all the vapor bubbles present in the PHP.

Suppose the temperature of the entire PHP structure is now quasi-statically increased to a new constant value. Then the system will again come to a new equilibrium temperature and corresponding saturation pressure, point A "and point B". In doing so, there will be some evaporation mass transfer from the liquid until equilibrium is reached again. A similar phenomenon will be observed if the system is quasi-statically cooled to a new equilibrium condition A" and B" (exaggerated representation for clarity).

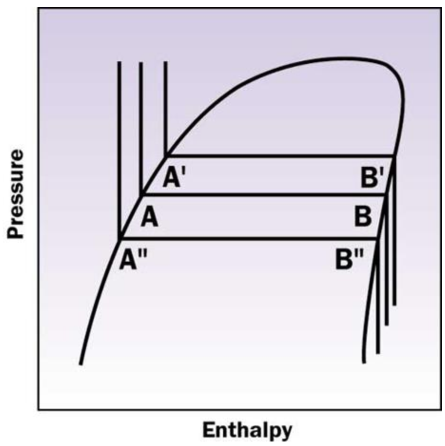

Fig 2.2: Pressure Vs Enthalpy Graph

In an actual working PHP, there exists a temperature gradient between the evaporator and the condenser section. Further, inherent perturbations are always present in real systems as a result of:

- Pressure fluctuations within the evaporator and condenser sections due to the local non uniform heat transfer always expected in real systems.
- Unsymmetrical liquid-vapor distributions causing uneven void fraction in the tubes.
- The presence of an approximately triangular or saw-tooth alternating component of pressure drop superimposed on the average pressure gradient in a capillary slug flow due to the presence of vapor bubbles.

The net effect of all these temperature gradients within the system is to cause non-equilibrium pressure condition which, as started earlier, is the primary driving force for thermo-fluidic transport. As shown m upper figure, heating at the evaporator continuously tries to push point.

2.4 Evolution of PHP

pulsating Heat Pipes (PHPs) has been the subject of research in an increasing number of laboratories m the recent times. PHPs were presented in 1971 by Smyrnov in a Russian Patent and in 2003 in a U.S. patent. PHP in the form as they are investigated today have been first proposed by Akachi [1-4] in the 1990.

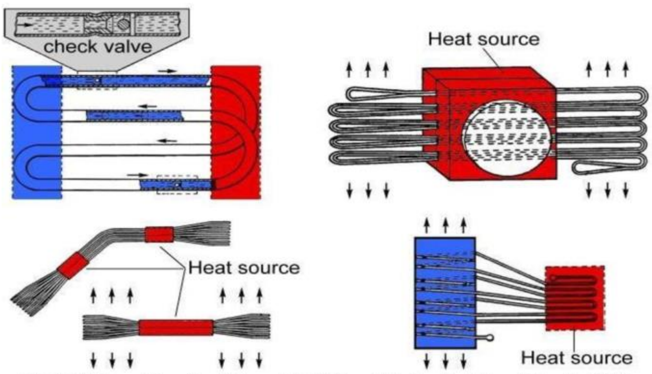

Figure 2.1 Loop Type Heat Pipe and its Different Design Variations [Akachi, 1990]

Fig 2.3: Earlier versions presented of heat pipe by Akachi

P. Charoensawan et.al [8]; has a work on effect to CLPHP thermal performance depends on various parameter like internal diameter of tube, number of turns, working fluid and inclination angle of the device and experimentally studied. The conclusion of P. Charoensawan's experimentation were, gravity has a great influence on the performance on the CLPHP, internal diameter must be specified with critical Bond number within the limit, the performance can be increased by increasing the ID and/or no. of meandering turns, the buoyancy forces effect bubble shape. Different fluids are beneficial under different operating conditions and the relative share of latent beat and sensible heat, flow behaviour.

Honghai Yang et.al. [9] presented a paper on experimental study on the operational limitation of closed loop pulsating heat pipes (CLPHPs). investigated, viz. vertical bottom flux on thermal performance and performance limitation were investigated. The CLPHPs were operated till performance limit characterized by serious evaporator overheating (dry- out) occurred. Rather high heat loads could be accommodated. An experimental study was pe formed on tow closed loop pulsating heat pipes (CLPHPs) to investigate the effects of inner diameter, filling ratio, operational orientation and heat toad on thermal performance and occurrence of performance limitation in the form of evaporator dry-out. In general, the CLPHPs obtain the best thermal performance and maximum performance limitation when they operate in the _vertical bottom heat mode with 50% filling ratio. As the inner diameter decreases, performance differences due to the different heat modes (i.e. the effect of gravity) become relatively small and even insignificant. The effect of inner diameter and inclination angles on operation limit of a closed loop oscillating heat pipes with check valves (CLOHP/CV) were studied in this paper. Copper tubes of ID 1.77 and 2.03 mm with 10 turn, with Rl23 was used as the working fluid. The inclination angles were 0°.

P. Meena et al. [10] were concluded that when the inner diameter changed from 1.77-2.03 mm the critical temperature increased. And when increase the inclination angles from O until to 90° the critical temperature increased.

S. Rittidech et.al. [11] a visualization study of the internal flow patterns of a closed-loop oscillating heat-pipe with check valves (CLOHP/CV) at normal operating condition for several evaporator lengths (Le), and ratio of check valves to number of turns (Rev) has been conducted. This article describes the effects of varying Le, and Rev on flow patterns. It was found that the internal flow patterns could be classified according to the Le and Rev as follows: At the high heat source when the Le decreases the main flow changes from the bubble flow with slug flow to disperse bubble flow. The Rev decreases the main flow changes from the disperse bubble flow with bubble flow to disperse bubble When the velocity of slug increases, the length of vapor bubbles rapidly decreases and the heat flus rapidly increases. The ratio of check valves to number of turns decreases the main flow changes from the dispersed bubble flow with bubble flow to disperse bubble flow for the high heat source.

P. Meena, et.al. [12] has aims to study the effect of evaporator section lengths and working fluids on operational limit of closed loop oscillating heat pipes with check valves (CLOHP/CV). It is experimentally concluded when the evaporator lengths increased the critical heat transfer flux decreased. There was working fluids change from R123 to Ethanol and water the critical heat flux decreased. The latent heat of vaporization affects the critical heat flux. The working fluid with the lower latent heat of vaporization exhibits a higher critical heat flux.

Stephane Lips Ahlem Bensalem et al. [13] various experiments were conducted on two full size pulsating heat pipes (PHP) which differed from their diameter, number of turns, and working fluid. The analysis of the experimental results for low heat fluxes the PHP performance is sensitive to the orientation and for high heat fluxes, it is independent from the orientation. The experiments were conducted at the scale of a single branch of a PHP. The test section was either adiabatic or heated. The adiabatic experiments brought to therefore the importance of dynamic contact angles in the flow and the dissymmetry between the

N. Panyoya et al. [14] the purpose of this research was to determine the effects of aspect r ratios (ratio of evaporator length to the inner diameter of tube) and number of meandering turns on performance limit of an inclined closed-loop oscillating heat pipe. The geometrical sizes, which were the variable parameters were the internal diameter, the evaporator section length of, the adiabatic and condenser section length of each set was equal to the evaporator length and the numbers of meandering turn and also variable inclination angles adjusted by 10°.The result indicated that the aspect ratio, the ratio of evaporator length by internal diameter and number of meandering turns significantly affect the maximum heat flux and inclination angle. The effects of aspect ratios and number of meandering turns on maximum heat flux of an inclined CLOHP have been thoroughly investigated in this study. In the case of aspect ratio, it can be seen that, the highest maximum heat flux occurs at inclination angle about 70-90° and lower value of aspect ratio. In the case of number of meandering turns, it can be seen that, when number of meandering turns increases, number of meandering turns does not affect to the maximum heat flux. In the case of inclination angle, it can be seen that,

when the inclination angle increases from 0-90°, the maximum heat flux increases with respect to increasing numbers of tubes. Moreover, the highest maximum heat flux occurs at vertical position to about 70°. P. Sakulchangsatjatai et al. [15] this research studies the effect of length ratios on heat transfer characteristic of Closed Loop Oscillating Heat Pipe with Non-Uniform Diameter (CLOHP/NUD) 1.e. inner diameter of capillary tube were alternated connection and bent into several numbers of turns and both ends were connected to form of loop. It was found that, the CLOHP/NUD transferred higher heat than the conventional Closed Loop Oscillating Heat Pipe (CLOHP) with the same heat transfer area because the working fluid flowed in only one direction. Working fluid moved to condenser
section in larger inner diameter and returned to evaporator section in smaller inner diameter. The heat transfer performance of CLOHP/NUD can be improved if one directional circulation of working fluid can be induced. The effects of length ratios and working fluids on the heat performance of CLOHP/NUD have been experimentally investigated and conclude heat flux increased when the length ratio decreased.

2.5 Heat Pipe Materials and Working Fluids

Heat pipes have an envelope, a wick, and a working fluid. Heat pipes are designed for very long tern operation with no maintenance, so the heat pipe wall and wick must be compatible with the working fluid. Some material/working fluids pairs that appear to be compatible are not. For example, water in an aluminium envelope will develop large amounts of non-condensable gas over a few hours or days, preventing normal operation of the heat pipe.
Since heat pipes were rediscovered by George Grover in 1963, extensive life tests have been conducted to determine compatible envelope/pairs, some going on for decades. In a heat pipe life test, heat pipes are operated for long periods of time, and monitored for problems such as non- condensable gas generation, material transport, and corrosion.
The most commonly used envelope (and wick)/fluid pairs include:

- Copper envelope/Water working fluid for electronics cooling. This is by far the most common type of heat pipe.
- Copper or Steel envelope/Refrigerant Rl34a working fluid for energy recovery in HVAC systems
- Aluminium envelope/Ammonia working fluid for Spacecraft Thermal Control

- Super alloy envelope/Alkali Metal (Cesium, Potassium, Sodium) working fluid for high temperature heat pipes, most commonly used for calibrating primary temperature measurement devices

Other pairs include stainless steel envelopes with nitrogen, oxygen, neon, hydrogen, or helium working fluids at temperatures below 100 K, copper/methanol heat pipes for electronics cooling when the heat pipe must operate below the water range, aluminium/ethane heat pipes for spacecraft thermal control in environments when ammonia can freeze, and refractory metal envelope/lithium working fluid for high temperature (above 1050 °C) applications.

CHAPTER 3
EXPERIMENTAL SETUP

3.1 General Aspect

An experimental facility has been designed, fabricated and installed to collected data for this research. The detailed description of experimental apparatus and experimental procedure are presented in this chapter.

Thus, apparatus used in this experiment are-

- Copper Capillary Tube
- Silicon Tube
- Normal structure
- Ethanol
- Methanol
- Acetone
- DI Water
- Test stand (Chassis)
- Variable power supply
- Nichrome Wire
- Thermocouple
- Digital Multi Meter
- Insulator
- Aluminium foil
- Glass wool
- Inserted wire (Copper)
- AC fan
- Watt Meter
- Digital Temperature Display Meter
- Electric Wire
- Selector Switch

3.2 View of experimental setup

Fig 3.1: Experimental setup

Fig 3.2: Copper Tube .

3.3 Pulsating Heat Pipe

A closed loop pulsating heat pipe or oscillating heat pipe consists of a metallic tube of capillary dimensions wound in a serpentine manner & joined end to end. It consists of sections. They are:

- Evaporator section
- Adiabatic section
- Condenser section

In this research was conducted taking normal structure, and structure with fin. Th fin was attached to the outer surface of the adiabatic section. This has increased the heat transfer surface.

For this experiment heat pipe with insert of copper wire was also done. The insert was tested and new result was found out. shows the different cross sections of the heat pipe with insert.

3.3.1 Evaporator Section

In the evaporator section of the heat pipe the working fluid absorbs heat from the heat source. It is located on the bottom section of the heat pipe. Heat is supplied to the heat pipe using Nichrome wire connected to a variable power supply. The Nichrome wire is wound around the pipes in the evaporator section on top of a layer of mica tape. The mica sheet prevents direct contact of copper tube with Nichrome wire to prevent any possibility of short circuit connection. The evaporator section is further enveloped by asbestos sheet to reduce heat loss to the environment.

3.3.2 Adiabatic Section

It is located between the evaporator section & condenser section. In here the liquid & vapor phases of the fluid flow in opposite directions and no significant heat transfer occurs between the fluid & surrounding medium. The part of the tube in adiabatic section is wound with aluminium foil, glass wool and finally covered with heat insulating tape to prevent heat transfer to the surrounding environment.

3.3.3 Condenser Section

It is the section of the heat pipe where heat is rejected from the working fluid to the surrounding. In this section, the working fluid condenses & rejects the same amount of heat which is absorbed from the evaporator section. In this experiment, this section is located on upper section of the heat pipe and a AC fan help dissipation of heat continually.

3.4 Working Fluid

Here is the use of Ethanol and Methanol as a working fluid.

3.4.1 Ethanol

Commonly referred to simply as alcohol or spirits, ethanol is also called ethyl alcohol, and drinking alcohol. It is the principal type of alcohol found in alcoholic beverages produced by the fermentation of sugars by yeasts. It is a neurotoxic psychoactive drug and one of the oldest recreational drugs used by humans. It can cause alcohol intoxication when consumed in sufficient quantity. Ethanol is used as a solvent, an antiseptic, a fuel and the active fluid in modem (post-mercury) thermometers. It is a volatile, flammable, colorless liquid with a strong chemical odor. Its structural formula CH3CH2OH, is often abbreviated as C2H5OH or C2H6O.

Fig 3.4: Ethanol

TABLE 3.1: PROPERTIES OF ETHANOL

SL	Parameters	Symbol	Quantity	Unit
1.	Freezing temperature	t_{freeze}	-114.1	°C
2.	Boiling temperature	t_{boil}	78.37	^{o}C

3.4.2 Methanol

Methanol, also known as methyl alcohol among others, is a chemical with the formula CH3 OH. Methanol acquired the name wood alcohol because it was once produced chiefly by the destructive distillation of wood. Today, methanol is mainly produced industrially by hydrogenation of carbon monoxide.

Fig 3.5: Methanol

TABLE 3.2: PROPERTIES OF METHANOL

Sl. No.	Parameters	Symbol	Quantity	Unit
1.	Freezing temperature	t_{freeze}	-97.6	°C
2.	Boiling temperature	t_{boil}	64.7	°C

3.4.3 Acetone

acetone (CH3COCH3), also called 2-propanone or dimethyl ketone, organic solvent of industrial and chemical significance, the simplest and most important of the aliphatic (fat-

Fig 3.6: Acetone

derived) ketones. Pure acetone is a colourless, somewhat aromatic, flammable, mobile liquid that boils at 56.2 °C (133 °F).

TABLE 3.3: PROPERTIES OF ACETONE

Sl No	Parameters	Symbols	Quantity	Unit
1.	Freezing temperature	t_{freeze}	-95	°C
2.	Boiling temperature	t_{boil}	56	°C

3.5 Test Stand

The test stand is a wood structure. The whole structure is supported by two columns which is

Fig 3.7: Test Wood stand

situated in a large wood base.

3.6 Heating Apparatus

Heating apparatus can be used to transfer heat. It is used for heat transfer from one place to another.

3.6.1 Variable power supply

Variable power supply provides variable voltage to run different types of operations or the operation that requires different voltage in times. We used voltages ranges from 20-60 volts

Fig 3.8: Variable power supply

by this power source. It is connected to the power supply unit to provide variable power (beat input) by varying voltage output

TABLE 3.4: VARIABLE POWER SUPPLY SPECIFICATION

Phase	3ϕ
Rated capacity	300 volt
Rated frequency	60Hz
Input voltage	220 volt

3.6.2 Nichrome wire

Nichrome wire is an al1oy typically made of 80 percent nickel and 20 percent chrome. Because of Nichrome wire's high internal resistance, it heats up rapidly when applying electricity and also cools rapidly when shut off or removed from a heat source. It maintains Its strength as the temperature rises and has a higher melting point than other wire. It does not oxidize or corrode, and is non-magnetic and highly flexible.

Fig 3.9: Nichrome Wire

3.6.3 Insulating Materials

Material that is used to stop the passage of electricity, heat, or sound from one conductor to another.

3.6.4 Aluminium Foil

For the insulation of pulsating heat pipe, aluminium foil tape is used. It resists flame, moisture, temperature extremes, UV exposure and most chemicals with a backing that withstands harsh environments. We used metal-backed foil tapes with the conformability to wrap tightly around virtually any shape or contour. With the used of this aluminium foil tape, we managed to resist heat loss during temperature measurement in different voltage was in run.

Fig 3.10: Aluminium Foil

3.6.5 Glass wool

Glass wool is an insulating material made from fibers of glass arranged using a binder into a texture similar to wool. The process traps many small pockets of air between the glass, and these small air pockets result m high thermal insulation properties. It reduces the beat loss and is used as insulator.

Fig 3.11: Glass wool

3.6.6 Heat insulating Foam Tape

This is used as a heat insulator and the adhesive side of the tape helps to bind the glass wool in the adiabatic section.

Fig 3.12: Foam tape

3.6.7 Copper Capillary Tube

Capillary Copper Tube stands for copper tubes have a very small diameter. It could be either straight or coil form, as the condition hard or annealed are both available.

Fig 3.13: Copper Capillary Tube

3.6.8 K-Type Thermocouple

A Type K thermocouple refers to any temperature sensor containing Chromel and Alumel conductors, that meets the output requirements as stated in ANSI/ASTM E230 or IEC 60584 for Type K thermocouples. This may be an immersion sensor, a surface sensor, wire or another style of sensor or cable.

Fig 3.14: K-type Capillary Tube

3.6.9 Silicon Tube

Silicone tubing is a tough elastomer. Flexible but strong, the tubing is a vital material across a broad range of industries. The material's resistance makes it suitable for lots of different devices, but some important properties make it particularly useful in healthcare and pharmaceutical sectors.

Fig 3.15: Silicon Tube

3.6.10 Selector Switch

Selector switch means a device by which the supply of electric current can be transferred from one or two or more circuits to others without the possibility of disconnection of the common supply line.

Fig 3.16: Selector Switch

3.6.11 Watt Meter

The wattmeter is an instrument for measuring the electric active power (or the average of the rate of flow of electrical energy) in watts of any given circuit. Electromagnetic watt meters are used for measurement of utility frequency and audio frequency power; other types are required for radio frequency measurements.

Fig 3.17: Watt Meter

3.6.12 Digital Multi Meter

A Multi meter is an electrical test tool that combines a basic digital multi-meter with a current sensor. Multi-meter measure current. Probes measure voltage.

Fig 3.18: Digital Multi Meter

3.6.13 Electric Wire

To conduct the flow of electricity, we used electric wire of negligible diameter. Heat sensing element that combines LM35 required electric wire to receive and supply data through electricity.

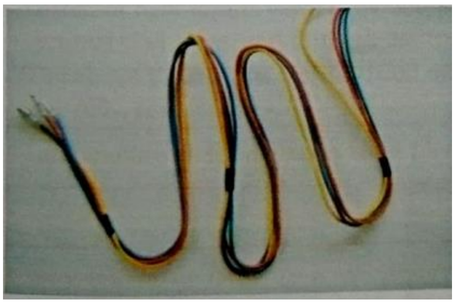

Fig 3.19: Electric Copper Wire

3.6.14 AC Fan

AC fans are the fans that are powered by alternating sinusoidal electric current. These fans are powered by positive current and the same amount of negative current. The general frequencies of AC voltage for the AC fans are 100 volts, 120 volts, 200 volts, 220 volts, 230 volts, and 240 volts at max.

Fig 3.20: AC Fan

CHAPTER 4
EXPERIMENTAL PROCEDURE

4.1 Mathematical Equations and Calculations

4.1.1 Calculation of filling ratio

Let,

V = Internal volume of the heat pipe

= 100 % Fill Ratio

A = Area of heat pipe

L = Total length of heat pipe

Now, $V = (\pi \times D^2 \times L)/4 \quad mm^3$

$= (3.1416 \times 2.35^2 \times 1143)/4 \quad mm^3$

$= 4957.61 \quad mm^3$

$= 4.96$ ml

As there is no additional container for working fluid in the test setup, the total internal volume of the pipe is considered to be the maximum capacity of the system. For example, to study the heat transfer characteristics 0.66ml and 2.31ml of working fluids were used which yielded 30%, 50% and 70% filling ratio respectively.

4.1.2 Calculation of Thermal Resistance

T_e = Average temperature of evaporation

T_c = Average temperature of condenser

$\Delta t = T_e - T_c$

Let, R_{th} = Thermal Resistance

$= \Delta t/Q$

$= (T_e - T_c)/Q \quad ^oC/W$

4.2 Experimental Setup

The setup consisted of the CLPHP, the controllable cooling fan, the AC power source. The CLPHP was made of three copper looped U-turns in the evaporator, condenser and adiabatic section (ID 2.35 mm). These copper tubes are connected with the help of flexible silicon tubes. Nichrome resistance wires, wrapped around the copper U-turns on the side of heater section. The U-turns of the other sides were rounded through a cooling fan. The adiabatic section was covered from all sides by glass wool. The height of the copper U-turns in the heater adiabatic and cooler zones was 30mm, 40mm and 60mm.

Two thermocouples (K-type) were used to measure the adiabatic tube temperature and two thermocouples each were pleased in the condenser section and on the evaporator U-tubes to measure the average temperature respectively. The CLPHP was mounted on a wooden frame. The filling could be done by first evacuating the set up and then administering the desired quantity of working fluid through a calibrated pipette and the metering valve. All experiments with this set up were done with a volumetric fluid filling ratio of 30%, 50% and 70%. In this experiment, absolute heating power of not more than 80W could be employed because of the presence of flexible silicon tube connectors that ruptured due to over heating over pressure. Even with these design limitations, important insights into the thermo-hydrodynamics of the device could be explored.

Fig 4.1: Experimental Setup

4.3 EXPERIMENTAL METHODS

The heat pipe is fabricated using a copper tube of 140 mm length and 2.35 mm inner dia and 2.80 mm outer diameter. Ni-Cr wire having length 3.10 ft was used to make a heater of 35 V and 80W capacity and heater was used for providing the required heat source at the evaporator

The evaporator and adiabatic sections of the heat pipe are insulated using insulator and glass wool to minimize the heat loss through these portions. Variac and multimeter were provided to control and measure the power input respectively. K- Type thermocouple wires were used as temperature sensors .

Thermocouples numbering the position of thermocouple 1 is 160 mm from the base, position of thermocouple 2, 3, 4, 5 and 6 are respectively 100, 240, 240, 100 and 160 mm from the base. A simple 8- channel digital temperature indicator is used to measure the temperature. Five copper fins of length 50mm, width 15 mm, and thickness 0.5mm were brazed on the condenser end. Experiments were conducted with dry run (i.e. without working fluid in the tube) and wet run (with working fluid inside). The heat pipe without working fluid essentially represents metallic conductor. Its performance is considered as the base for the evaluation of heat pipe (i.e. with working fluid in it). The heater is put "on" and the temperature rise was observed at regular intervals till the steady state is achieved, Experiments were repeated for different heat inputs with different fill ratios and various plots were drawn to study the performance of miniature heat pipe to optimize the fluid inventory.

Fig 4.2 : Experiment Copper Capillary tube

4.4 EXPERIMENTAL PROCEDURE

The test section consists of three parts, as mentioned earlier, evaporator, adiabatic and condenser sections.

In the experiment the heat transfer characteristics were measured for four different liquids distilled water, (50% Methanol + Acetone) and Ethanol. Also the characteristics were measured for dry run condition (without any liquid). So, two miniature heat pipes were fabricated. For dry run condition the heat pipe was sealed at bottom and top.

In case of the heat pipe where liquids were used the bottom was sealed and top was sealed by a silicon tube. Ni- Cr thermic wire was wound round the evaporator section Power to the heater was provided from line supply through a variac. A (AC) fan of 220 V were attached at the condenser section and a fan was directed towards the condenser area for forced convection to occur at this section. Six sets of thermocouple wires were fixed with the body by means of glue.

At first each thermocouple sets were fused together at the top point and it was ensured that except the top point, they do not touch at any other points. Then they were attached with the body. The other ends of the thermocouple wires were connected with the digital thermocouple reader by means of connecting wires. Thermocouples were placed at six points on the surface of the heat pipe, two at evaporator section, two at adiabatic sections and two at condenser section. Thermocouples at each section were placed at an interval of 20 mm. Experiments were conducted with dry run (without any working fluid in the tube) and wet run (with working fluid inside). The heat pipe without working fluid essentially represents metallic conductor.

Its performance is considered as the base for the evaluation of the heat pipe (with working fluid in it). The transient tests were conducted on the heat pipe, in which heater was put on and the temperature rise was observed at regular intervals till the steady state was achieved. After achievement of steady state the temperatures at the six points were noted by changing the positions of the selector switch.

This experiment was repeated for different heat inputs, different fill ratios and for different working fluids. Various plots were drawn to study the performance of the miniature heat pipe to optimize the fluid inventory. The different heat inputs were achieved by changing the output voltage from the variac.

Fill ratio means the percentage of the evaporator section volume that is filled by the working fluids. The fill ratios used in this experiment were 30%, 50% and 70% of the evaporator volume for all four different working fluids. All the temperature readings, at the six points on the heat pipe surface, were taken for all three working fluids for all the fill ratios after reaching steady state condition.

CHAPTER 5

Results and Discussions

5.1 Inclination Compare and Filling Ratio Compare

Experiment run for DI water 30%

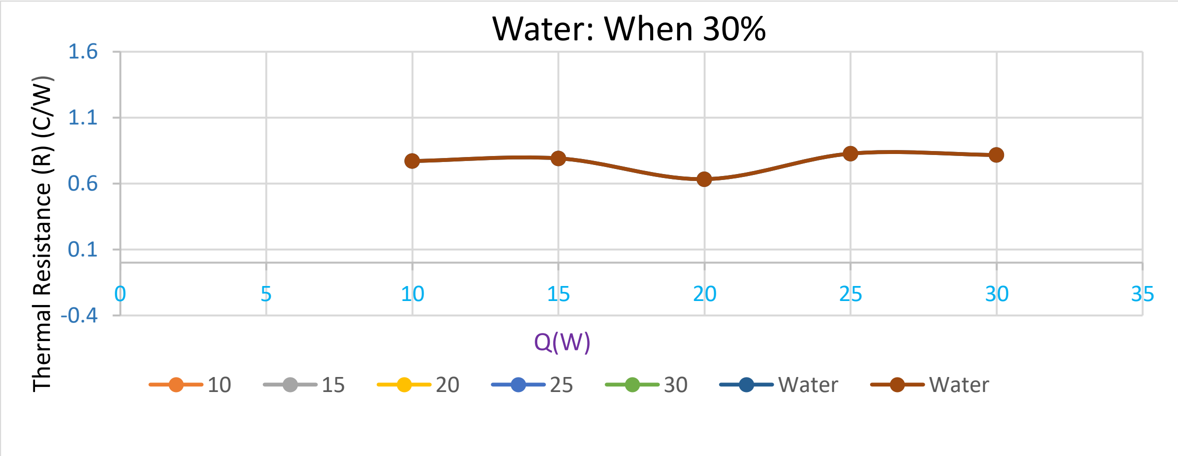

Fig 5.1: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 25 to 30W heat input all behaves same; because DI Water get dry out

Experiment run for DI water 50%

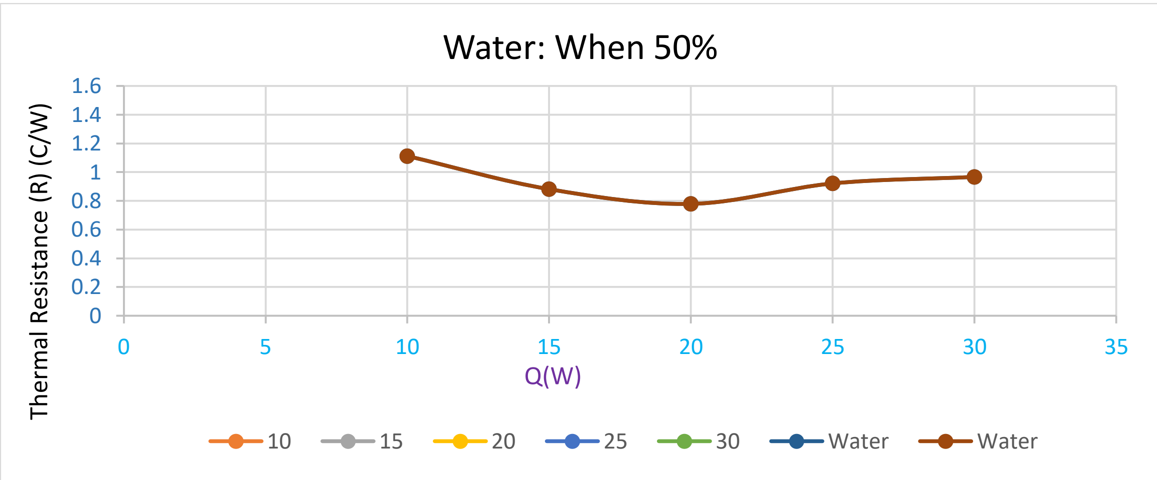

Fig 5.2: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 25 to 30W heat input all behaves same; because DI Water get dry out

Experiment run for DI water 70%

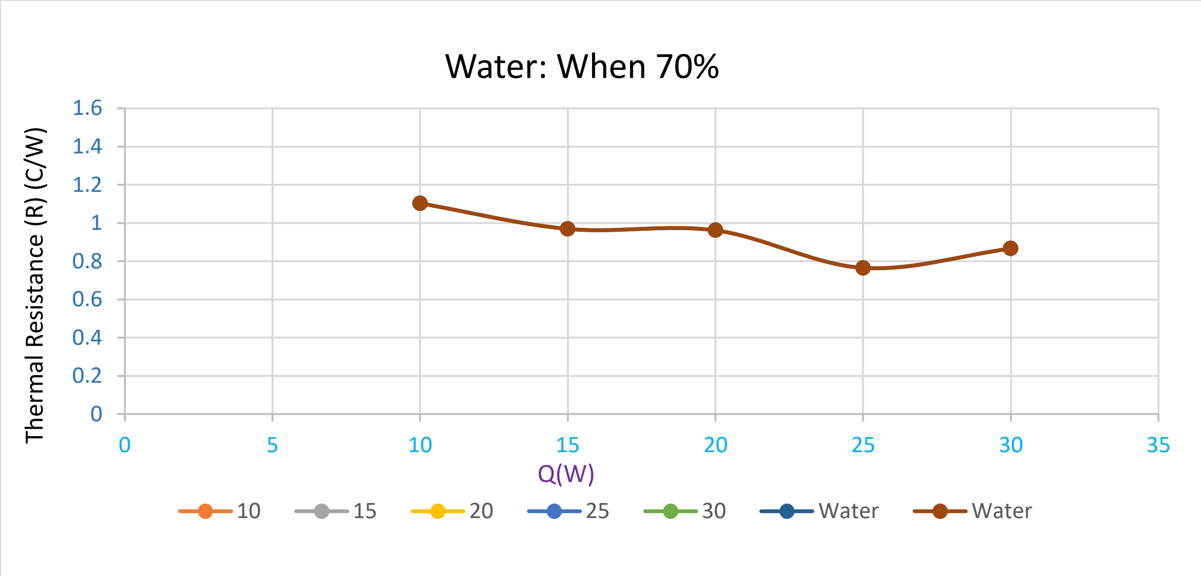

Fig 5.3: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 25 to 30W heat input all behaves same; because DI Water get dry out

Experiment run for (50% Acetone + 50% Methanol) 30%

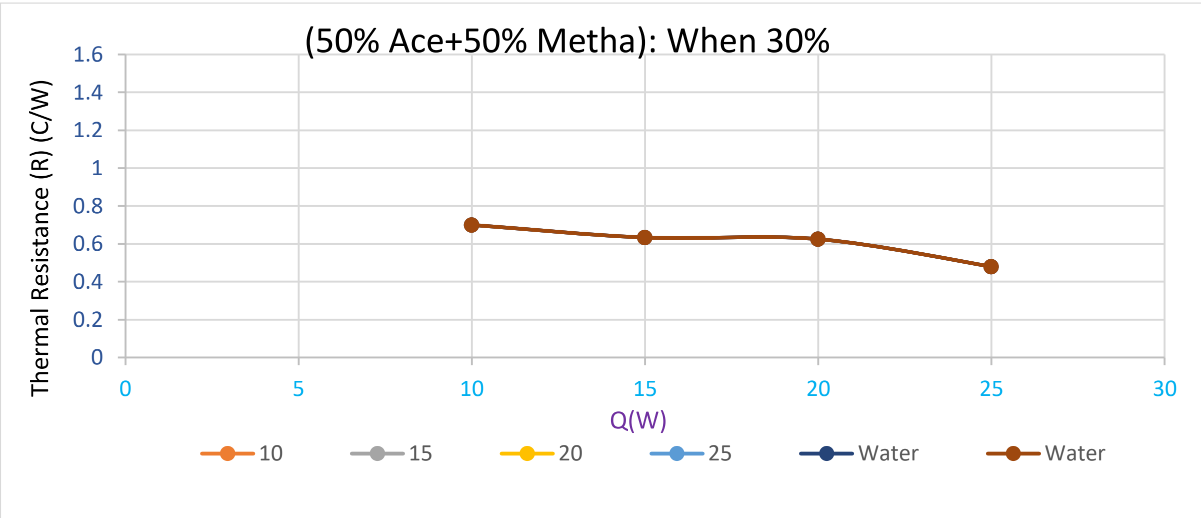

Fig 5.4: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 15 to 25W heat input all behaves same; because Acetone-Methanol mixture get dry out

Experiment run for (50% Acetone + 50% Methanol) 50%

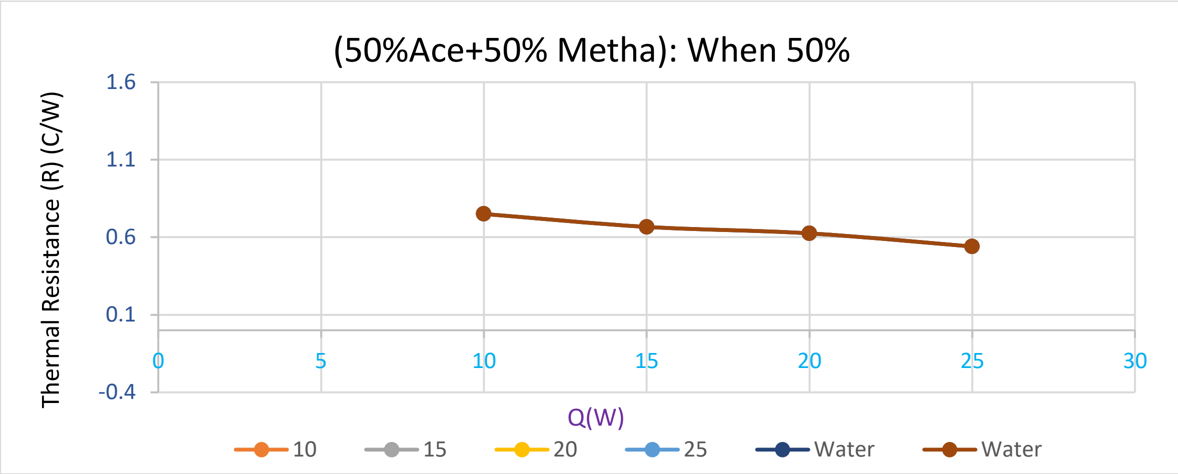

Fig 5.5: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 15 to 25W heat input all behaves same; because Acetone-Methanol mixture get dry out

Experiment run for (50% Acetone + 50% Methanol) 70%

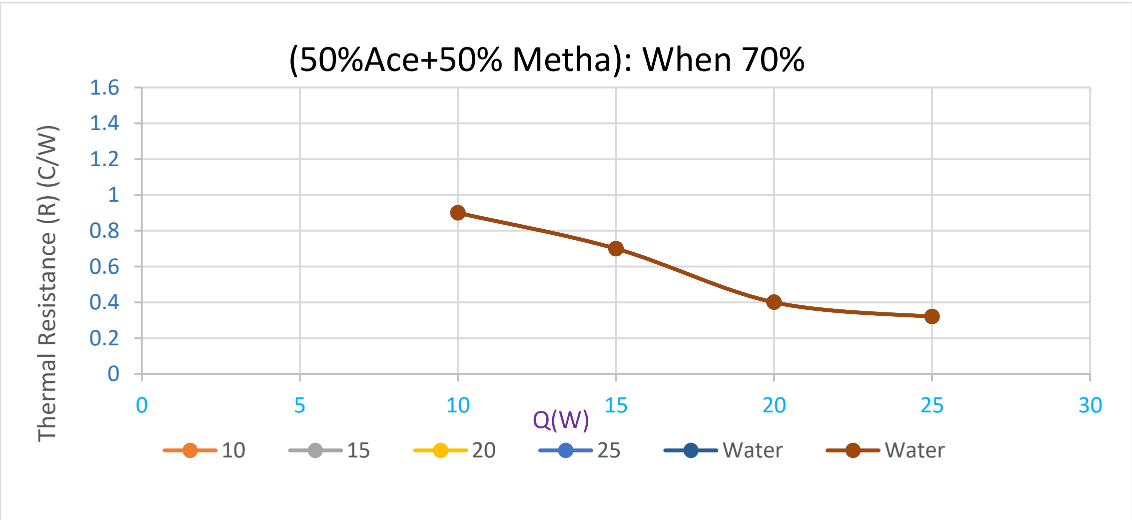

Fig 5.6: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 15 to 25W heat input all behaves same; because Acetone-Methanol mixture get dry out

Experiment run for Ethanol 30%

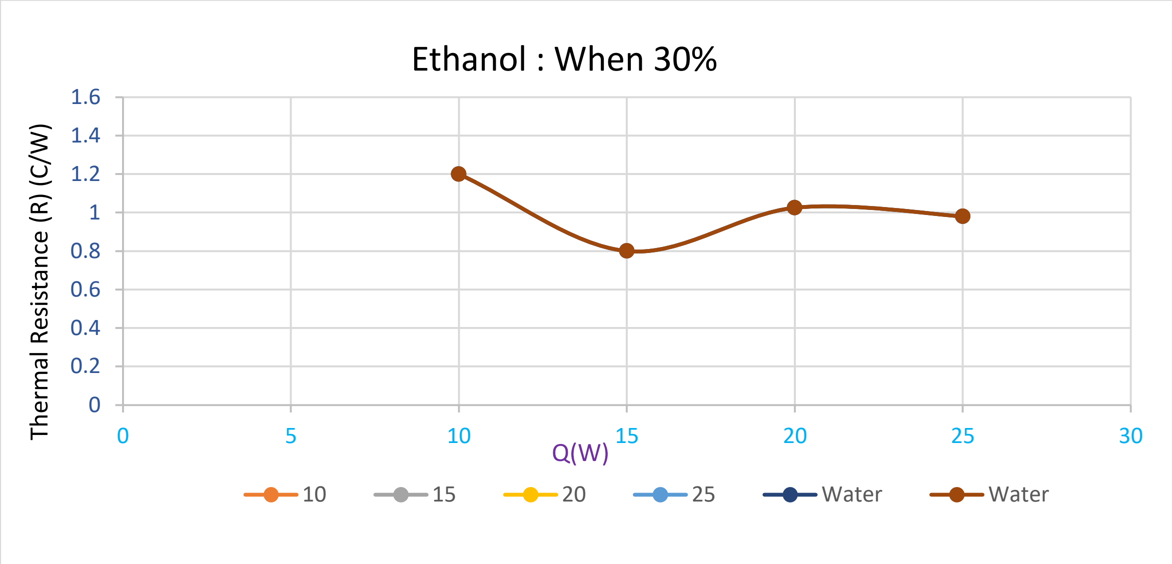

Fig 5.7: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 20 to 25W heat input all behaves same; because ethanol get dry out

Experiment run for Ethanol 50%

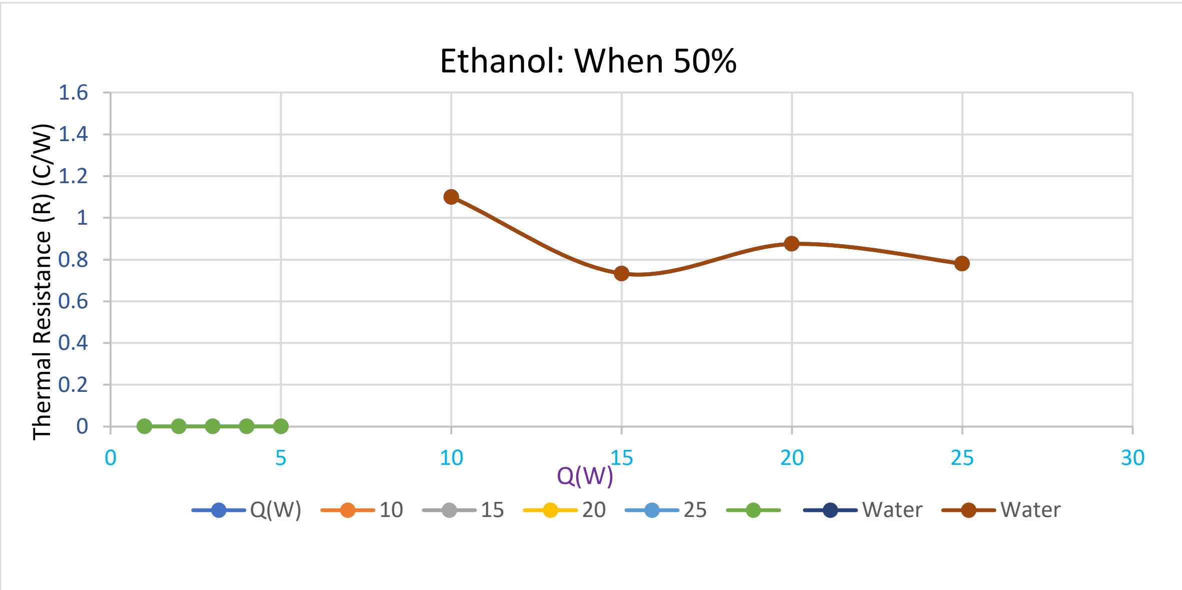

Fig 5.8: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 20 to 25W heat input all behaves same; because ethanol get dry out

Experiment run for Ethanol 70%

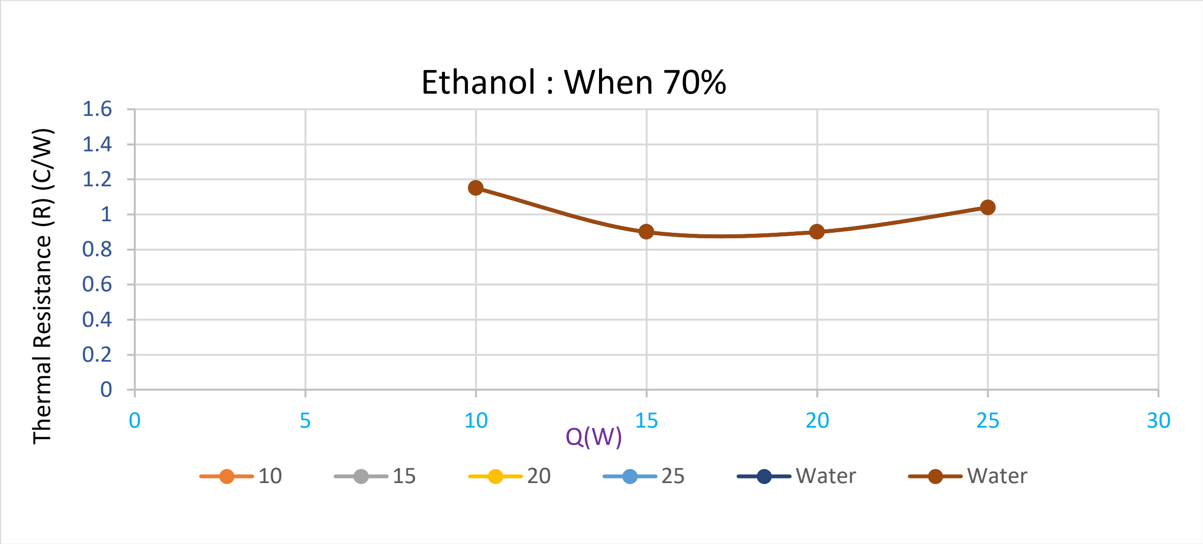

Fig 5.9: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of the pure working fluid are varying but in 20 to 25W heat input all behaves same; because ethanol get dry out

Experimental run for Dry Run

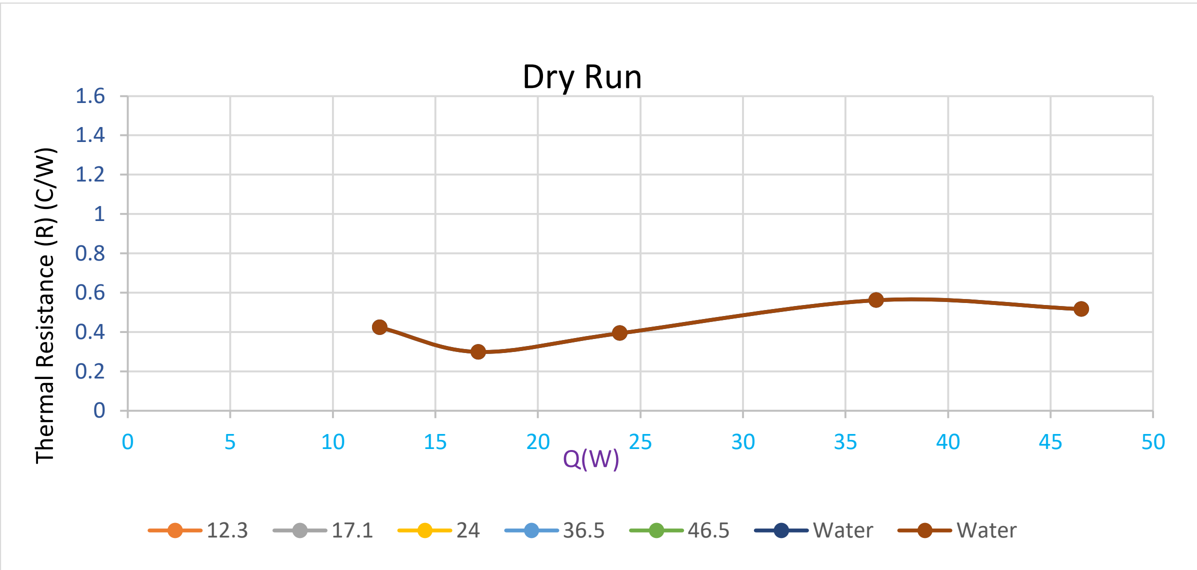

Fig 5.10: Thermal resistance v/s Heat input

In this graph it is clear that during starting at 10W to 15W the thermal resistance of dry run are varying but in 35 to 46W heat input all behaves same.

Evaporator Condenser Temperature v/s Heat input of Water With fill ratio 0%

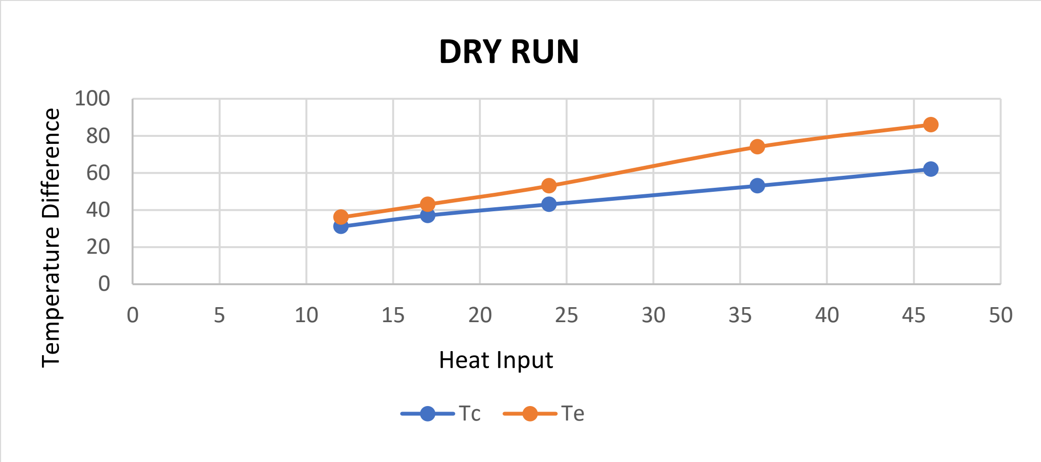

Fig 5.11: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat input of Water With fill ratio 30%

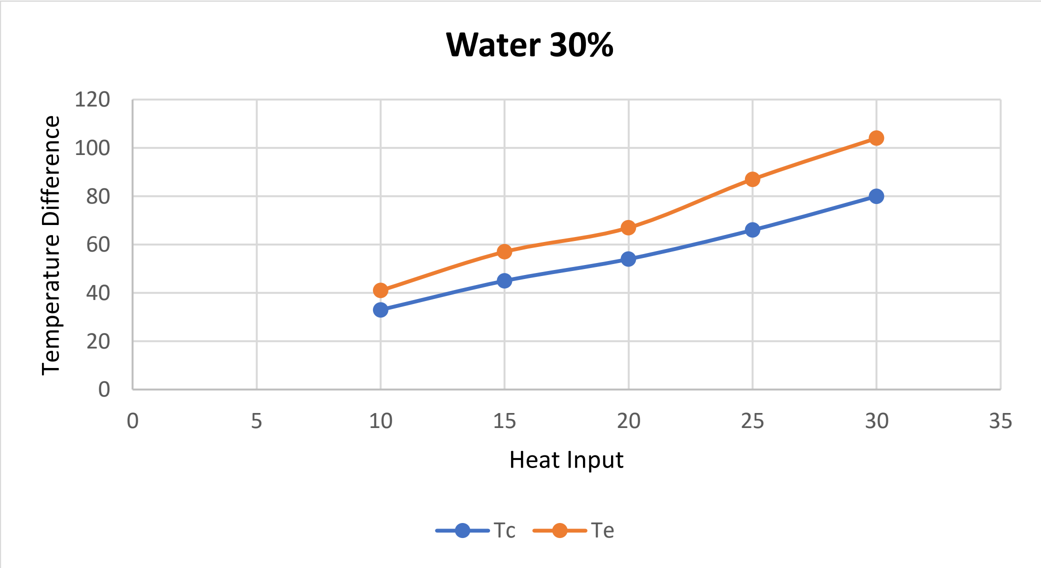

Fig 5.12: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of Water With fill ratio 50%

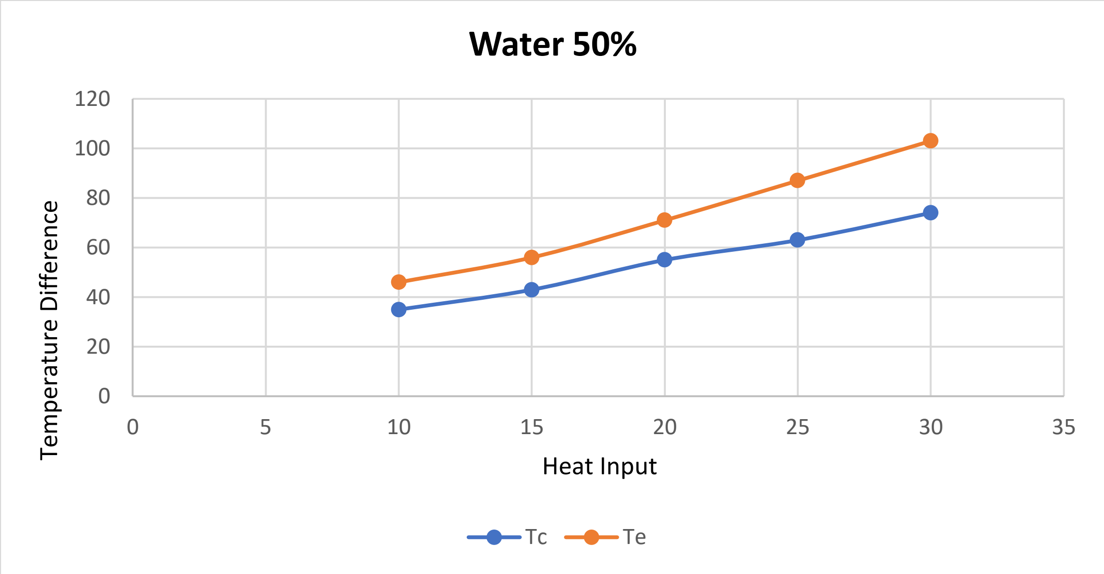

Fig 5.13: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of water With fill ratio 70%

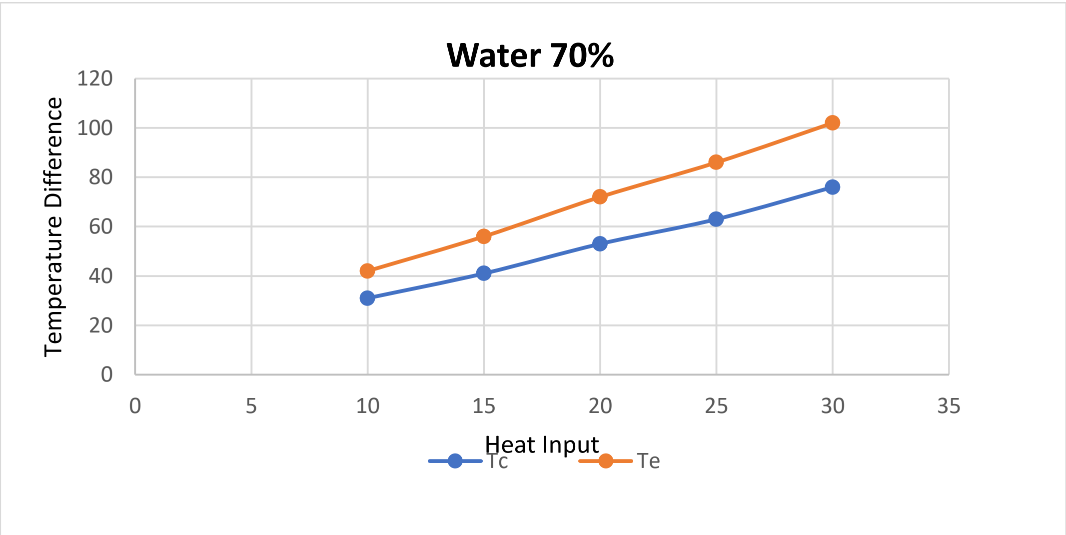

Fig 5.14: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of water With fill ratio 30%

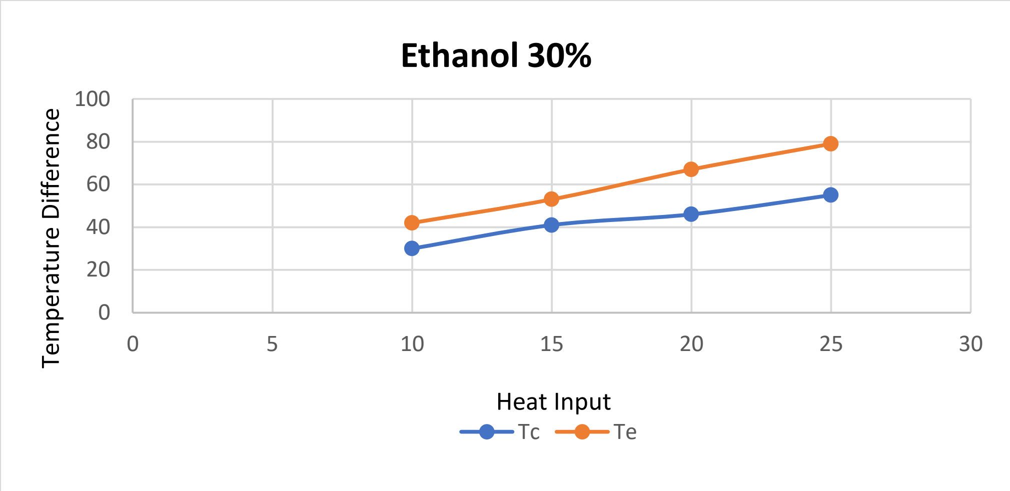

Fig 5.15: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of Ethanol With fill ratio 50%

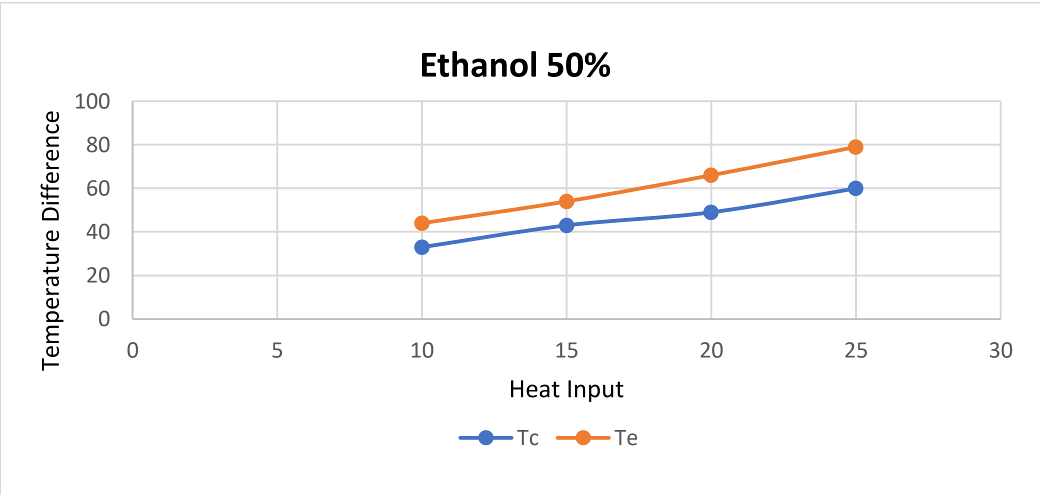

Fig 5.16: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of Ethanol With fill ratio 70%

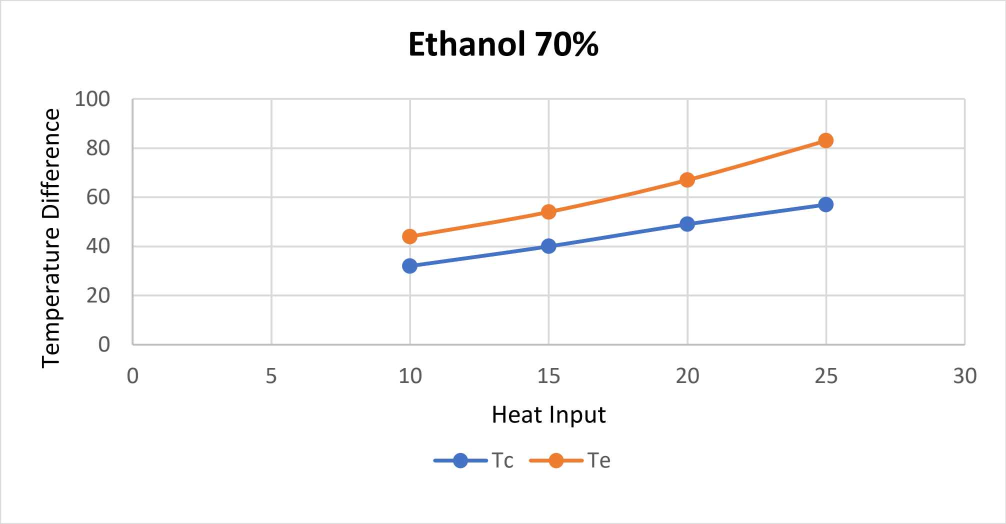

Fig 5.17: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of water With fill ratio 50% Acetone + 50% Methanol (30%)

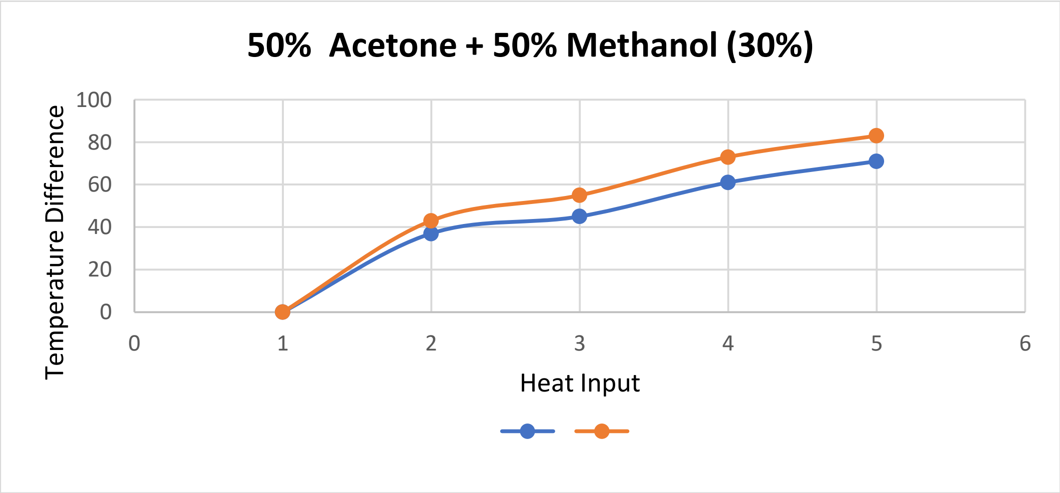

Fig 5.18: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of water With fill ratio 50% Acetone + 50% Methanol (50%)

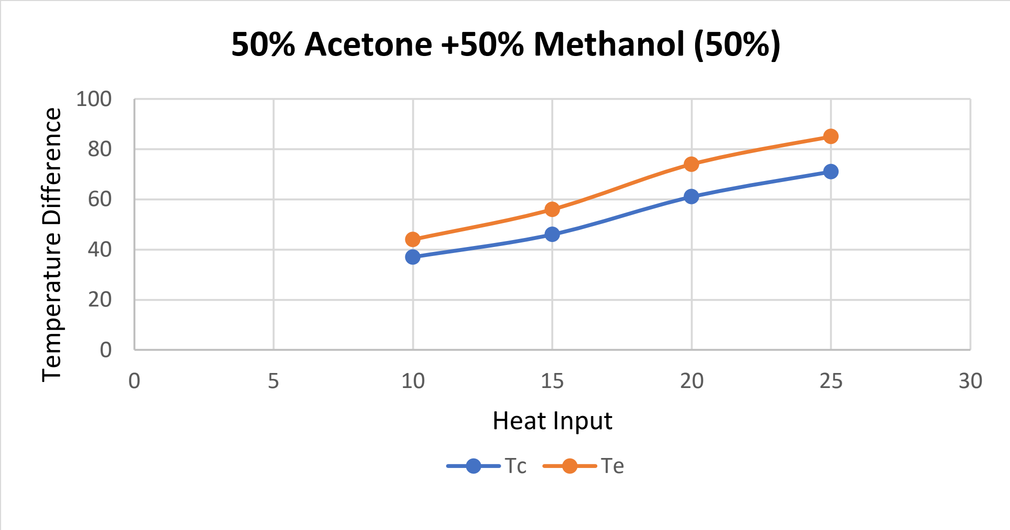

Fig 5.19: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

Evaporator Condenser Temperature v/s Heat Input of water with fill ratio 50% Acetone + 50% Methanol (70%)

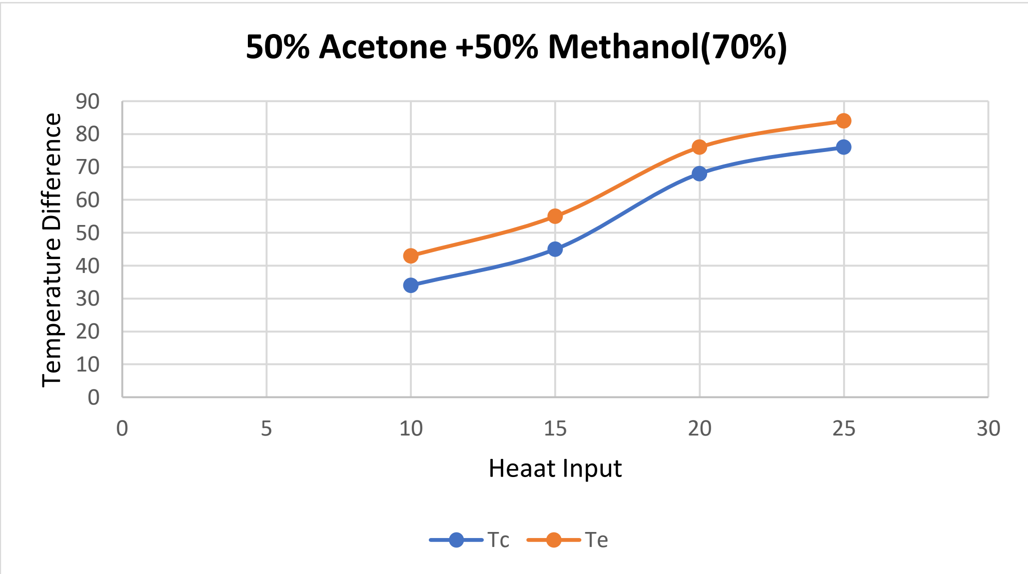

Fig 5.20: Temperature difference v/s Heat input

In this experiment it is clear that when heat is increasing the temperature of evaporator increase, at the same time the temperature of condenser also increase. But the rate of temperature of evaporation sector is higher than the temperature of condenser section.

CHAPTER 6
CONCLUSION

6.1 CONCLUSIONS

The following are the major conclusions of the study:

1. A comprehensive literature search on pulsating heat pipes was carried out. Since the device is a rather recent invention, there is in general, a dearth of literature on the topic. The then state of the art indicated that although these devices were already finding market niches in electronics cooling, the physical understanding of the device was at a very primary stage. No authoritative mathematical models were in existence.
2. Closed loop pulsating heat pipes are complex heat transfer systems with a very strong thermo-hydrodynamic coupling governing the thermal performance. The cooling philosophy of these devices draws inspiration from conventional heat pipes on one hand and single-phase forced flow liquid cooling on the other. Thus, the net heat transfer is a combination of sensible heat of the liquid plugs and the latent heat of the vapor bubbles. If the internal flow pattern remains predominantly in the slug flow regime, then it has been demonstrated that latent heat will not play a dominant role in the overall heat transfer. If there is a transition to annular flow under the imposed thermo-mechanical boundary conditions, then the dominance of latent heat increases. The most interesting (disturbing!) aspect is the fact that the best performing closed loop pulsating heat pipe no longer behaves as a pulsating device. Alternating tubes then have slug flow and annular flow and the bulk flow takes a fixed direction. Strictly speaking, the term pulsating 'heat pipes' then becomes a misnomer. The construction of the device is such that on a macro level, the heat transfer mechanism can be compared to an extended surface 'fin' system. Although such an analogy provides important insight on the mechanism of heat transfer, extrapolations cannot be authoritatively done unless more experimental data base is available.
3. The internal tube diameter is one of the parameters which essentially defines a CLPHP. The physical behaviour adheres to the 'pulsating' mode only under a certain range of diameters. The Eötvös number criterion i.e. Eö = ¢, for surface tension

dominated adiabatic slug flow, only provides a tentative design rule for the internal diameter of a CLPHP. Although the Eötvös number of water and ethanol was much below the prescribed maximum limit of Eö = ¢, gravity forces were definitely seen to affect the performance (refer Appendix-I). Similarly, while the Eö number of R-I23 is higher than the maximum suggested limit for the working temperature range, the CLPHP still worked quite effectively. This suggests that though at Eö > ¢ the tendency of slug flow diminishes as surface tension tends to reduce, a certain amount of liquid transport is still possible by the bubble pumping action in diabatic flows thereby providing substantial heat transfer. Beyond a certain maximum range of Eö, the device will function as an interconnected array of normal gravity assisted thermosyphons. Below a certain Eö, dissipative flow resistance will lead to a decrease in thermal performance.

4. The applied heat flux to the system is the most vital parameter for proper operation. Not only because of the fact that it provides the driving energy, it also catalyses two important phenomena, i.e., (i) flow pattern transition in the device and, (ii) affecting the two-phase flow instabilities and thereby the level of internal perturbations. A certain minimum heat flux is required to overcome the dissipative flow losses. Thereafter, an increase in the input heat flux leads to a series of changes in the internal flow patterns (from slug to zemi-annular and fully developed annular flow) which directly affects the heat transfer characteristics. The study strongly indicates that design of these devices should aim at thermo-mechanical boundary conditions which result in convective flow boiling conditions in the evaporator leading to higher local heat transfer coefficients. Also, in conjunction with a minimum number of turns, a minimum heat flux (in present experiments = 5-6 W/cm , based on tube ID) is required additionally to make the thermal performance nearly independent of the orientation.

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Appendix

Table1.1: Experimental data table for DI water (FR 30%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	40	42	36	39	37	28	41	33.29	0.77
15	55	60	50	51	45	45	57.5	45.63	0.79
20	59	75	62	63	52	55	67	54.33	0.63
25	82	93	71	69	60	72	87.5	66.82	0.83
30	101	108	85	75	75	85	104.5	80	0.82

Table1.2: Experimental data table for DI water (FR 50%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	44	49	40	43	35	34	46.5	35.38	1.11
15	55	58	48	51	45	40	56.5	43.28	0.88
20	70	73	62	68	55	55	71.5	55.92	0.78
25	89	85	75	81	60	66	87	63.97	0.92
30	102	104	85	90	68	80	103	74	0.97

Table1.3: Experimental data table for DI water (FR 70%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	40	45	34	39	30	31	42.5	31.47	1.10
15	52	60	49	52	40	41	56	41.46	0.97
20	68	77	55	62	48	57	72.5	53.27	0.96
25	82	90	71	80	63	69	86	66.87	0.77
30	101	103	88	91	70	82	102	76	0.87

Table1.4: Experimental data table for(50%Acetone+50% Methanol) (FR 30%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	43	43	41	41	37	35	43	36	0.7
15	55	55	47	50	45	46	55	45.5	0.63
20	73	74	61	64	59	63	73.5	61	0.63
25	83	84	72	74	67	76	83.5	71.5	0.48
	0	0	0	0	0	0	0	0	0

Table1.5: Experimental data table for(50%Acetone+50% Methanol) (FR 50%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	43	45	41	44	37	36	44	36.5	0.75
15	54	58	48	50	44	48	56	46	0.67
20	73	75	63	65	58	65	74	61.5	0.63
25	82	88	70	72	68	75	85	71.5	0.54

Table1.6: Experimental data table for(50%Acetone+50% Methanol) (FR 70%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	43	44	40	40	38	31	43.5	34.5	0.9
15	53	58	51	52	45	45	55.5	45	0.7
20	75	77	64	66	76	60	76	68	0.4
25	82	86	72	74	82	70	84	76	0.32

Table1.7: Experimental data table for Ethanol (FR 30%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	40	45	35	40	30	31	42.5	30.5	1.2
15	52	55	45	49	40	43	53.5	41.5	0.8
20	64	70	50	55	43	50	67	46.5	1.025
25	75	84	60	63	50	60	79.5	55	0.98

Table1.8: Experimental data table for Ethanol (FR 50%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	42	46	36	40	32	34	44	33	1.1
15	53	55	46	48	41	45	54	43	0.73
20	65	68	53	62	48	50	66.5	49	0.875
25	77	82	59	75	55	65	79.5	60	0.78

Table1.9: Experimental data table for Ethanol (FR 70%)

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
10	43	45	40	43	35	30	44	32.5	1.15
15	51	57	47	53	40	41	54	40.5	0.9
20	65	69	60	62	46	52	67	49	0.9
25	82	84	71	75	52	62	83	57	1.04

Table1.10: Experimental data Table for Dry Run

	Evaporation		Adiabatic		Condenser				
Q(W)	T2	T5	T1	T6	T3	T4	Eva(avg)	T con(Avg)	R
12.3	37	36	36	36	33	29	36.5	31.30	0.42
17.1	42	44	41	40	36	39	43	37.89	0.30
24	52	54	45	51	41	45	53	43.56	0.39
36.5	75	73	62	58	51	55	74	53.52	0.56
46.5	83	89	65	76	56	68	86	62	0.52

Table1.11:Evaporator Condenser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat input of Water With fill ratio 0%

QW	Tc	Te
12	31	36
17	37	43
24	43	53
36	53	74
46	62	86

Table1.12:Evaporator Condenser Temperature v/s Heat Input

Evaorator Condenser Temperature V/S Heat Input of water Fill Ratio 30%

QW	Tc	Te
10	33	41
15	45	57
20	54	67
25	66	87
30	80	104

Table1.13:Evaporator Condenser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat input of Water With fill ratio 70%

QW	Tc	Te
10	31	42
15	41	56
20	53	72
25	63	86
30	76	102

Table1.14:Evaporator Condenser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat input of Ethanol 30%

QW	Tc	Te
10	30	42
15	41	53
20	46	67
25	55	79

Table1.15:Evaporator Condenser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat Input of Ethanol 50%

QW	Tc	Te
10	33	44
15	43	54
20	49	66
25	60	79

Table1.16:Evaporator Conden ser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat input of Ethanol 70%

QW	Tc	Te
10	32	44
15	40	54
20	49	67
25	57	83

Table1.17:Evaporator Conden ser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat input of (50% Aceton + 50% Methanol) of 30%

QW	Tc	Te
10	37	43
15	45	55
20	61	73
25	71	83

Table1.18:Evaporator Conden ser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat input of (50% Aceton + 50% Methanol) with Fill Ratio 50%

QW	Tc	Te
10	37	44
15	46	56
20	61	74
25	71	85

Table1.19:Evaporator Conden ser Temperature v/s Heat Input

Evaporator Condenser Temperature v/s Heat Input of (50% Acetone + 50% Methanol) With Fill Ratio 70%

QW	Tc	Te
10	34	43
15	45	55
20	68	76
25	76	84