AN INVESTIGATION INTO THE PERFORMANCE OF A THERMOSYPHON HEAT EXCHANGER

Thesis submitted towards the partial fulfillment of the requirements for the award of degree of

MASTER OF ENGINEERING

IN

THERMAL ENGINEERING (Part-Time)

Submitted By:

Gurtej Singh

Roll No. 851483001

Under the guidance of

Dr. Kundan Lal

Assistant Professor, MED

Thapar University, Patiala

THAPAR INSTITUTE
OF ENGINEERING & TECHNOLOGY
(Deemed to be University)

MECHANICAL ENGINEERING DEPARTMENT

THAPAR INSTITUTE OF ENGINEERING & TECHNOLOGY

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CERTIFICATE

I hereby declare that the thesis entitled "**An Investigation into the Performance of a Thermosyphon Heat Exchanger**" is an authentic record of my work carried out as requirements for the award of the degree of Master of Engineering in Thermal Engineering (Part –Time)at Thapar Institute of Engineering and Technology, Patiala under the supervision of **Dr. Kundan Lal**, Assistant Professor, Mechanical Engineering Department, Thapar Institute of Engineering and Technology, Patiala during July, 2014 to April, 2019. No part of the matter embodied in this report has been submitted to any other university or institute for the award of any degree.

Date: 08/04/2019

Gurtej Singh
851483001

It is certified that the above statement made by the student is correct to the best of my/our knowledge and belief.

08/04/19

Dr. Kundan Lal
Mechanical Engineering Department
Thapar Institute of Engineering and Technology,
Patiala – 147004

Countersigned by

8/4/2019

Head, Mechanical Engineering Department
Thapar Institute of Engineering and Technology,
Patiala – 147004

Dean of Academic Affairs 8.4.19
Thapar Institute of Engineering and Technology,
Patiala – 147004

Dedication

I dedicate this thesis to my beloved mother Paramjit Kaur and my brother Rajinder Singh, who are an ever supporting and encouraging with their great patience. I also dedicate this to my family

I am want highly thankful to Yogender Pal, Lakhwinder Singh, and Darshan Singh my beloved friends for their constant encouragement and support to make it possible.

My little Nieces Noorpreet & Simreet, my sister in-law Jasvir

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Above all, I express my indebtedness to the "Sri Guru Granth Sahib Ji" for showering there blessing and kindness.

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ABSTRACT

In this investigation work was carried out to evaluate the thermal performance of Thermosyphon and its use as heat exchanger in air condition test rig for heat recovery. Thermosyphon was chosen after considering the critical parameters that affects its performance such as temperature range ($20^\circ C – 40^\circ C$), container material (Cu) and working fluid (methanol) etc. Thermosyphon was built with 50% charging for its use in vertical position inclination angle 90°). Testing of Thermosyphon at various heat loads (100 – 600 watt) was done. The results obtained showed that the temperature variation along its length was almost constant. Therefore, for the same geometry three thermosiphons were constructed and fitted inside the air conditioning test rig to investigate its performance for low temperature application ($20^\circ C – 40^\circ C$). Thermosyphon heat exchanger subjected to the temperature of the ambient air and temperature of the test chamber. After taking the overall temperature into account the experimental effectiveness was found to be 34.3 %. The effect of the variation of air face velocity (1- 6m/sec) on evaporator section of thermosyphon heat exchanger was also investigated. Air face leads to higher thermal heat transport the COP of the air conditioning test rig has which been also investigated to study the effect of thermosyphon heat exchanger on its performance. The COP observed was found to be improved to 4.73 from 1.54; however the achieved COP was below the theatrical COP (5.583). The study has helped to bring down the difference between actual and theoretical COP. Moreover, the effectiveness of thermosyphon heat exchanger using fins, increasing the no. of rows and decreasing the pitch to diameter ratio can be improved further.

Keywords: Thermosyphon, Methanol, COP, Energy Recovery, Effectiveness, Air Conditioning, Air Face Velocity

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NOMENCLATURE

COP	Coefficient of performance
TPHEx	Thermosyphon Heat Exchangers
HVAC & R	Heating Ventilation Air Conditioning and Refrigeration
PV	Pressure Volume
VCRS	Vapor Compression Refrigeration Cycle
HPHEx	Heat Pipe Heat Exchanger
KW	Kilowatt
$\dot{m}$	Mass flow rate
$\dot{v}$	Volume flow rate
Psi	Pound square inch
Ppm	Parts per million
amp	Ampere
V	Volts
Kg	Kilogram
CC	Centimeter cube
P	Pressure
Hz	Frequency
Cp	Specific heat
K	Temperature in Kelvin
kg/hr	Kilogram per hour
kWh	Kilowatt hour energy consumed
Kw	Power consumption
$v^{\cdot}$	Specific volume
R	Characteristics Gas constant
ε	Effectiveness
γ	Adiabatic Index
ϕ	Figure of Merit
^{o}	Degree
T	Temperature

μ	Dynamic Viscosity
ρ	Mass Density
ν	Specific Volume
ν	Kinematic Viscosity
σ	Surface Tension
τ	Time

Chapter- 1

Introduction

1.1 Introduction to Heat Exchanger

Systems involving to thermal loads are subjected to number of heat fluxes. In the modern era, due to rapid rise in technological gadgets and industrial processes. Heat transfer is becoming a thrust area for the thermal engineers. Undermining this fact, there is a significant rise in heat exchangers which are commonly employed in these processes and their products as well. Heat exchangers are available in configurations ranging from shell and tube heat exchangers, condensers, evaporators, plate heat exchangers, and heat pipe heat exchangers. [R.K.Shah et. al.]. The driving force for the use of thermosyphon heat exchangers (TPHEx) is to attain the overall compactness in the system owing to the weight limitations. Manufactures of the energy systems provide the base models in numerous configurations for the various applications. These systems are then converted to the full package by requiring the accessories for the heat exchangers. Manufactures relies on the vendors for the completions of the heat exchangers, which are not optimally designed. This change in configuration can alter.

r the effectiveness of the heat exchangers, even though they are designed for the particular configuration. Heat exchangers basically operate in latent heat transfer, sensible heat transfer and combination of these types. In the sensible heat transfer heat exchangers, one may look for the increase in surface area of the heat exchangers, to achieve the higher effectiveness as compared to the base system of the heat exchanger. Another alternative can be for these heat exchangers is to increase in mass flow rates of the fluid, that will lead to high pressure drop in the heat exchanger. These two methods of increasing the effectiveness are tedious as the first one requires more space and the latter needs serious design implications. Fluids which are involved in the process of heat transfer in heat exchangers are prone to the fouling which can dent the overall effectiveness of the heat exchangers. Thermosyphon heat exchanger (TPHEx) has emerged as a great alternative to many industrial applications for the recovery of the waste heat, in HVAC& R processes in both commercial and industrial sectors [A. Fahgri, 1995]. TPHEx's are also used as heat sinks in the many electronics components. As the development in the high performance chips is taking place it becomes very crucial to remove the heat from the chips to eliminate the risk of malfunctioning in the devices. The TPHEx's are also used in the space application due their excellent heat carrying capabilities.

The present study aims at the conducting a research on the TPHEx for the use heat recovery in the HVAC& R applications. In this work ambient air is directly used in the evaporator of the heat exchanger and fed as the supply air to the cooling coil of the air conditioning systems. In the condenser section of the TPHEx return air comes from the test chamber of test rig is used which is having a lower temperature as that of the ambient temperature. Heat exchangers, complete design involves various aspects of designing such as mechanical and thermal, etc. Thermal design problems of heat exchangers are basically of two types rating and sizing. In rating the thermal performance of the heat exchangers is evaluated for a given configuration, while in sizing new configuration is designed on the basis of specified thermal performance of a heat exchanger

1.2 Fundamental Studies on Thermosyphon

1.2.1 Introduction

This chapter presents the outline TPHExof the research work that had been carried out in the area of Thermosyphon heat exchangers (TPHEx) and fundamental studies on these from application point view. General empirical correlation for the thermosyphon studies had been carried to full fill the following objectives:

- To understand the basic functionality of thermosyphon.
- To understand the use of TPHEx as an energy recovery device.
- To identify the key characteristics of the TPHEx.

1.2.2 Fundamental Studies on Wick less Heat Pipe (Thermosyphon)

The Perkins tube was a device working on two phase flow of the fluid, demonstrated by Ludlow Patton Perkins in mid nineteenth century. The device developed by the Perkins is shown in the Figure1.1, which is nothing but a single phase thermosyphon or gravity assisted heat pipe working in closed loop. The air expansion device is a perfect example of this. In the early applications this tube was used in HVAC, boilers, to limit window fogging, cooing of internal combustion engines etc. The developments of modern thermosyphon started after Gaugler coined the idea of two phase closed device.

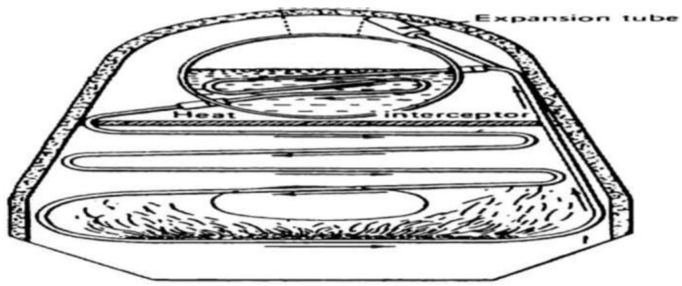


Figure: 1.1 Perkins boiler

The idea of heat pipes was first put forward by R.S. Gaugler in 1942 [David Reay et. al., 2005] and then later 1963 G.M. Grover invented its remarkable properties & serious development started on the application of heat pipes [David Reay et. al., 2005].Heat pipe is device which consist of closed container, working fluid and wick structure depending on its orientation can employed in the heat pipe. The heat is basically discretized into three main parts.

- Evaporator Section
- Condenser Section
- Adiabatic Section

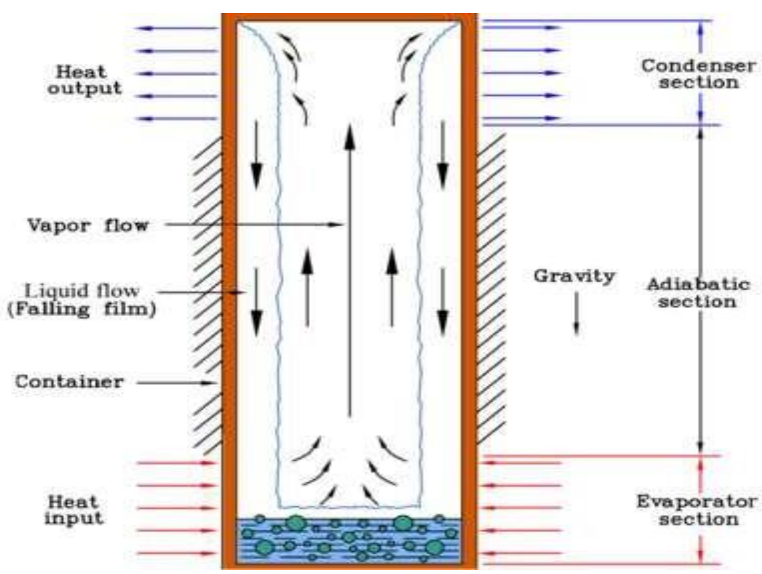


Figure: 1.2 Thermosyphon

As shown in figure 2.2 heat input is provided at the evaporator section of the heat pipe, where working fluid inside the container absorbs the heat and by using the latent heat of vaporization in get converted into vapor phase. Then these moves towards the low pressure region, means towards the condenser section of the pipe. The middle portion of the heat pipe acts as adiabatic section (no heat exchange with surroundings). In condenser section heat carried by vapor from the evaporator is lost, resulting in change of phase for the working fluid. The condensed liquid then comes back to evaporator with gravity force.

1.2.2 Components involved in a heat pipe

- The container sealed from both ends
- The working fluid (Charge)
- The wick or capillary structure

The function performed by container is to isolate the charge from the outside environment (surroundings). It must be leak proof, should also maintain the pressure differential across the walls of the container, and enable transport of thermal energy takes place to and fro from the working fluid.

1.2.3 Prime Requirements for Container

- Compatibility (Both with working fluid and External environment)
- Porosity
- Wettability
- Ease of fabrication including welding, machinability and ductility
- Thermal conductivity
- Strength to weight ratio

1.2.4 Working Fluid

In order to choose a charge the first and foremost consideration of the operating temperature range for which working fluid to be used. Within that specified range combination of various possible working fluids may exist. In order to use desired charge various limits can then be evaluated for that particular working fluid

Prime Requirements for Working Fluid:

- Compatibility of wick and wall materials is also must
- Good thermal stability at desired temperature.
- Wettability of wick and walls of the container
- High latent heat of vaporization
- High thermal conductance
- Low liquid and vapor viscosities for the given range
- High surface tension of the charge

1.2.5 Wick

The wick structure employed inside of heat pipe enables the liquid to return from the evaporator section from the condenser section. The objective of wick is to generate the capillary pressure inside the sealed container and then to distribute the liquid evenly around the entire evaporator section of the heat pipe. The most commonly employed wick structure is a wrapped screen wick.

1.2.6 Heat-pipe Types

In modern era due to extensive research in field of heat pipes lots of verity if found in heat pipes depending upon their shape, size and application for which heat pipe to be employed in the system. Commonly used in the various applications are listed below.

- Tubular heat pipes
- Annular heat pipes
- Baffle heat pipes
- Bent tubular heat pipes
- Flexible tubular heat pipes
- High performance heat pipe heat sinks
- Diode heat pipes
- Variable conductance heat pipe
- Pressure controlled heat pipes

  Hik™ plate heat pipes

- Vapor chambers etc.

1.3 Vapor Compression Refrigeration System (VCRS)

The above diagrams shows the working diagram and PV plot for the cycle vapor compression refrigeration cycle. In vapor compression refrigeration cycle (VCRS), a refrigerant is employed as the charge. Working fluid (charge) is a very crucial part of the whole system. In the working of VCRS four main processes are involved i.e. compression, condensation, throttling and evaporation. Using these processes refrigeration and air conditioning is achieved. Vapor compression refrigeration system is most commonly employed for this purpose. In the system of VCRS low pressure and low temperature vapor refrigerant goes into the compressor via inlet valve and it comes out as high pressure and high temperature refrigerant. Then it goes into the condenser part of system in which phase change of refrigerant takes place at constant pressure and temperature. In section of cycle heat is rejected into the surroundings. After this working fluid is throttled using a throttle valve, which results in low temperature refrigerant. After throttling low pressure and low temperature refrigerant enters into evaporator, where it absorbs heat and the cycle continues. The main parameter to evaluate the performance of the VCRS is the COP which is computed by the ratio of refrigerant effect produced and work input

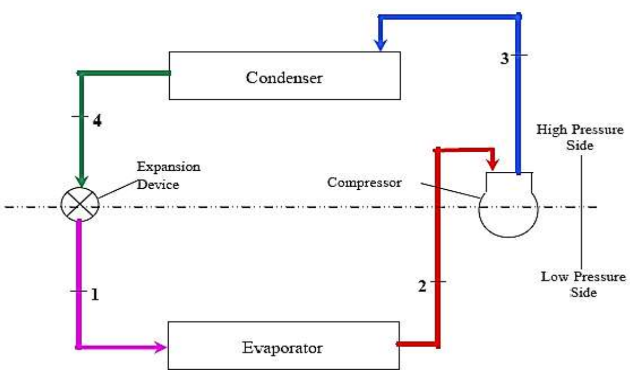


Figure: 1.3 Working diagram of VCRS

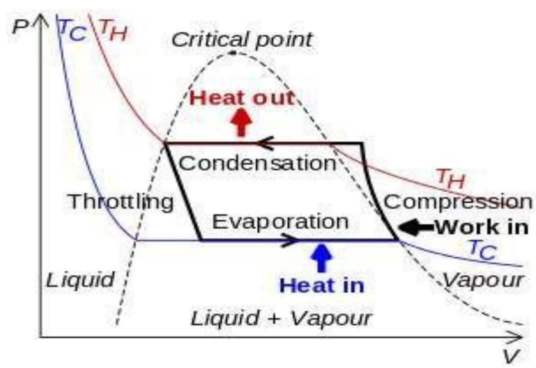


Figure: 1.4 PV plot for VCRS

The above diagrams shows the working diagram and PV plot for the cycle vapor compression refrigeration cycle. In vapor compression refrigeration cycle (VCRS), a refrigerant is employed as the charge. Working fluid (charge) is a very crucial part of the whole system. In the working of VCRS four main processes are involved i.e. compression, condensation, throttling and evaporation. Using these processes refrigeration and air conditioning is achieved. Vapor compression refrigeration system is most commonly employed for this purpose. In the system of VCRS low pressure and low temperature vapor refrigerant goes into the compressor via inlet valve and it comes out as high pressure and high temperature refrigerant. Then it goes into the condenser part of system in which phase change of refrigerant takes place at constant pressure and temperature. In section of cycle heat is rejected into the surroundings. After this working fluid is throttled using a throttle valve, which results in low temperature refrigerant. After throttling low pressure and low temperature refrigerant enters into evaporator, where it absorbs heat and the cycle continues. The main parameter to evaluate the performance of the VCRS is the COP which is computed by the ratio of refrigerant effect produced and work input.

Technology improves time by time and made the amendments in the refrigeration system all around the world. Various researches held to increase the performance of the system.This is achieving by the refrigeration and air conditioning. Vapor compression refrigeration is the system among all most used for this purpose.Demand for the refrigeration and air conditioning has been increasing day by day. Industry as well as in the homes there is need for the comfort environment for human.

1.3.1 Refrigerant

Refrigerant is the term which is commonly used for working fluid. It is used in refrigeration and air conditioning systems for heat exchange purposes. It is one of major parameters that effects performance of the system. The properties of refrigerant depends upon no. of factors such as boiling point, chemical stability etc. There are various types of refrigerants available viz. HFC's, HC's azeotropes etc. In earlier times CFC's were commonly used, but due to their severe environment ill effects these are being phasing out. HFC134a its chemical name is 1,1,1,2-Tetrafluoroethane most widely accepted as alternative refrigerant. The HFC 134a (R134a) have a little high greenhouse warming potential (GWP) compare to other refrigerant, in many countries R134a has been used as a long-run alternative refrigerant. HFC refrigerants are mostly used due to their high performance characteristics. In automobile air conditioners and domestic refrigerator application

1.3.2 Condenser

A condenser is a device used to condense a fluid from its gaseous state to its liquid state. In vapor compression refrigeration system condenser is used to condense refrigerant from vapor phase into liquid phase. It rejects heat to the environment and its inlet is outlet of the compressor from which high temperature and pressure refrigerant enters. It rejects heat to the environment through the surface which is either air cooled or water cooled. The heat rejection capacity of a condenser depends mainly on following factors:

- The temperature difference between the refrigerant and the cooling media
- The flow rate of the cooling media through the condenser
- The flow rate of the refrigerant through the condenser

Types of condensers are

- Finned-static condenser
- Finned-forced convection condenser
- Wire-static condenser
- Plate-static condenser

1.3.3. Evaporator

The liquid refrigerant enters the evaporator from the refrigerant flow control device is under low pressure and low temperature. Evaporator makes contact with the water and he medium from which heat has to be removed. Here liquid refrigerant changes phase to vapor phase. In this set up emersion coil type evaporator is used. Evaporator is dipped in 10.5 litre of water. Water is stirred with help of the agitator to ensure uniform heat transfer. The heat exchange rate within an evaporator is influenced by these factors:

- The temperature difference between the refrigerant and the water being cooled
- The flow rate of the water through the evaporator

1.3.4 Compressor

Compressor is an integral and important part of any refrigeration system. Its main work is to raise the pressure and temperature of the refrigerant across the pipe by giving some pumping power. There are different types of compressors are listed below:

- Reciprocating
- Rotary screw compressors,
- centrifugal compressors,
- scroll compressors
- Hermetically sealed compressor

The latter one is commonly used in refrigeration systems depending upon the requirements. Normally in hermetic and mainly semi-hermetic compressors the compressor and motor driving the compressor are integral part of compressor, and run inside the refrigerant system.

1.3.5 Refrigerant Flow Device

An expansion valve is a component in vapour compression refrigeration system which controls the quantity of refrigerant flow into the evaporator. It maintains the required pressure & temperature in the evaporator. There are different types of expansion devices which are commonly used. They are as follows:

- Capillary tube
- Automatic expansion valve
- Thermostatic expansion valve
- Manual expansion valve

Chapter - 2

Literature Review

Per Wallin (2012) conducted his study to deepen the knowledge of working fluid (charge) which are commonly employed in the heat pipes. Performance of heat pipe or thermosyphon is greatly affected by the working fluid which is filled inside the container.

Aloke Kumar Mozumder et al. (2011) performed their study to investigate the performance of heat for micro applications such electronic cooling system. In this work they used a 5 mm diameter pipe with 150mm length. They compared the performance of the pipe for different charges viz. acetone, water and methanol. They used four filling ratio for an individual working fluid. The thermal capacity of 10 watt was employed in the system and performance of evaporator and condenser was determined under various load conditions. They also evaluated the heat transfer coefficient on the evaporator section, which was dominating factor for micro applications. It was also found out that the filing ratio of 85% most for micro heat pipes. In this work a new correlation to obtain the heat transfer coefficient was also proposed.

A. K. Mujumdar et al. (2010) work was carried out to design and fabricate a micro heat pipe with 5 mm diameter and 150 mm length. 10 watt heat load was supplied under variable temperature condition for the different fluid such as water, methanol and acetone. Performance evaluated and quantified for overall heat transfer coefficient and thermal resistance for the micro heat pipe. For low temperature applications study was carried out. It was found out that acetone had the optimum filling ratio from these three charges in low temperature conditions. Acetone also had the maximum value for the overall heat transfer coefficient for the micro heat pipes under the given range of the temperature.

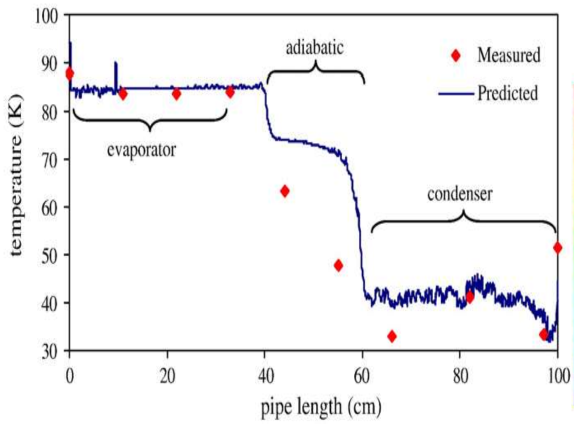
Chiang Wei et al. (2009) in the study Nano fluids were used to increase the overall performance for a conventional heat pipe with grooved structure. The mean diameter for the Nano particles was found out to be 10 nm. Nano particles helped in the reduction of thermal resistance for the pure water. When same volume of charge was compared for both Nano fluids and pure water, mean decrease in the value of thermal resistance was from 28% - 44%. The working fluid volume of 0.15 mL was used to compare both the charges. This work was carried out to optimize the performance of heat sink of typical Pentium–IV generation PC. Its current heat generation capacity increased to 130W. Silver nanoparticles was used in am aqueous solution for a 211 mm wide grooved miniature circular heat pipe.

L.M. Qiu et al. (2007) carried out their work on two - phase closed thermosyphon to evaluate the effect of the filling ratio. They formulated a model to study the effect of filling ratio in thermosyphon in vertical position. Liquid pool in the evaporator section was studied with flow patterns of natural convention and the effect of nucleate boiling. In this work two different geometries was construed with the nitrogen as charge to evaluate experimental results. A new correlation was proposed in this work to study the effect the work of filling ration ton the phase closed thermosyphon. Various variable were taken into consideration in the work such as heat input, operating pressure and geometrical various were also considered.

Zhongliang Liu et al. (2008) investigated the performance of two phase thermosyphon by observing condensation in condenser section and boiling phenomena in evaporator section. Various heat fluxes were used in combination for different working fluids. As heat pipe conductivity is all most constant along its length. Study was also carried out to obtain the variation of temperature along the length of the thermosyphon. Effect of the grooves on the evaporation surface to increase the value of boiling heat transfer.

Bandar Fadhl e. al. (2012) investigated the performance of gravity assisted heat pipe also called commonly known as thermosyphon using CFD modeling technique. Iwo ecofriendly refrigerants R404a and R134a were used in the pipe to evaluate its performance. The purpose of this work was to understand and simulate two phase heat transfer taking place inside the pipe. In the simulation both start up and steady state was taken into account to simulate the thermosyphon. Results obtained by simulation was compared with the experimental data. The comparison between these two was satisfactory, as reproduction in the simulation was nearly same. During the work both pool boing in the condenser section and liquid film condensation in the condenser section was successfully achieved.

MasoudRahimi et al. (2009) in this work analysis was carried out on a gas/liquid two-phase flow. Also the wark was done on the simultaneous evaporation and condensation phenomena which takes inside the thermosyphon was modeled using volume of fluid (VOF) technique in Ansis Fluent. The purpose of the model is understand interaction between phase changes that take place in the thermosyphon. Experimental work was also done in a thermosyphon were at various operating conditions. The temperature profile predicted using CFD simulationin a wickless heat pipe was compared with experimental data.After comparison a good agreement was found between these two.



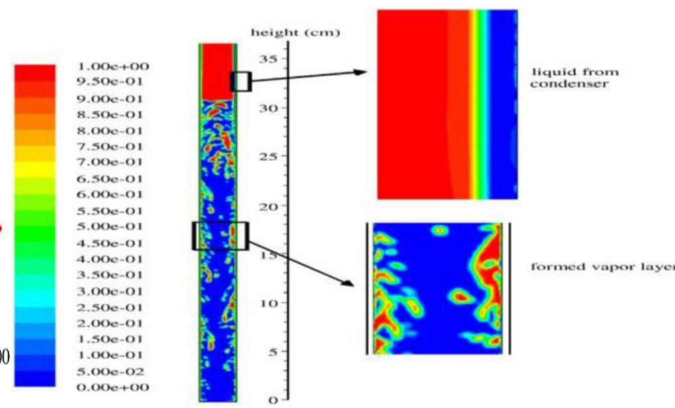


Figure:2.1 Temperature Distribution Figure: 2.2 Temperature Contours

Temperature profile was predicted using CFD for the wick less heat pipe or thermosyphon was in good agreement with experimental data. Therefore CFD can be used as good tool to model complex flow behaviors.

.Yau at el. (2014) studied the performance of heat pipe heat exchanger. In this heat exchanger R143a was chosen as working fluid, to analyze the performance of the charge five different filling ratios were used.

H. Mroue et al. (2015) carried out his work on the heat pipe heat exchanger to study the effect of multiple air passes and different air inlet temperature, mass flow rate to evaluate the performance to the heat exchanger. It was found out that air inlet temperature in evaporator section was directly proportional to effectiveness. But the mass flow was inversely proportional to this section. In this work maximum effectiveness of 29% was attained. A CFD modeling for the multiple air pass was also done successfully as shown in figure 3.1(c).

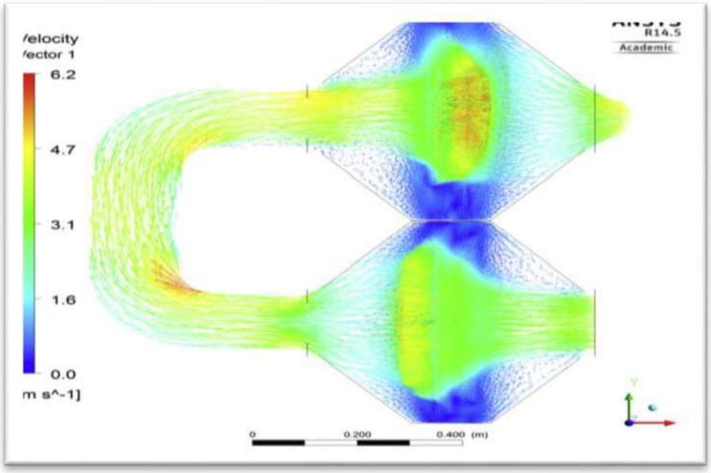


Figure: 2.3 Temperature Distribution in HPHEx for Multi Pass

Yau et al. (2014) carried out their work to investigate thermal performance of a heat pipe heat exchanger. In this study R134a was used as working fluid for the temperature range 27- 35. In the evaporator section of heat exchanger. The condenser side temperature was maintained at 24. It was found that effectiveness decreased due to pipe flooding. Optimum thermal performance can be achieved with charging. Slightly above the wick Saturation.

Longo A. G. et al. (2014) in this work a large bundle of copper tubes equipped with aluminum fins was used with equal diameter of evaporator and condenser. Study was carried to residue the effect of global warming by using HF01234 was used instead of convectional R134a was used. HF0134 can be considering viable alternative to R134a.

Babek Rashidain (2013) in this work a MATLAB program was formulated for heat pipe heat exchanger. It was observed that MATLAB can be considered to model heat pipe heat exchanger. The designed model was used for heat recovery and lesser air pollution and conservation of the environment. It was observed that heat pipe heat exchanger can be modeled due to their simplicity and flexibility good compactness and high efficiency.

R. Laubscher et al. (2013) investigated the performance heat pipe heat exchanger high temperature application for nuclear coolants it was found that radioactive condiment tritium can be replaced with down thermal at temperature of at 220. It is more changelings to design a system for 800 for the same purpose.

M.H. Saber et al. (2012) carried out their work to evaluate the performance evaporator section heat pipe heat exchanger compositional fluid dynamics (CFD). In this Work numerical stimulation achieved to perform to achieve optimum temperature fir the evaporator section. In the CFD fluent was used to investigate the performance and graphical representation work obtained and results work compare with the available data.

E Azad (2011) investigated the thermal performance of an air-air heat exchanger. The model was developed found out heat transfer capacity which gives the thermal performance of heat pipe heat exchanger fined work was also employed achieve more heat transfer rate. The arrangement of the heat exchanger, for both the sections evaporator and condenser respectively.

Engin Gedik et al.(2015) carried out their work evaluate the thermal performance of gravity heat pipe heat exchanger using to different fluid charges namely (R134a) and (R410a) they varied the flue gas temperature from 75 to 200 Effectiveness of these two working fluid varied between 35.6% to 57.7% after the quantifying the data it was found effectiveness (R134a) using six more copper pipe was 40% more when it was compared with (410a).

Jadhav S.T. et al. (2014)performed their work to investigate the dehumidification effect of the heat pipe heat exchanger. Study was took place at climatic condition in Pune city located near Mumbai in India. In their work they found COP of the system and also obtained the energy efficiency ratio (EER). They observed heat pipe heat exchanger can serve as a great alternative for dehumidifying in the climatic conditions.

Lian Zhang et al.(2013)investigated the performance of heat pipe heat exchanger for energy recovery by developing a CFD model in the Fluent. In their work they employed residual technique from 10 ~~5~~ high precision to determine and compare the convergence curve. The work carried out using fluent compared with already existing data. It was found satisfactory hence the developed model can used for the further study. The result obtained using simulation was of high accuracy hence can form the basis for the design of wick less heat pipe heat exchanger.

Y.H. Yau et al. (2011)carried out their work for the energy recovery in the air conditioning unit with the use of heat pipe heat exchanger. They used low temperature on the condenser side was maintained at 20 degree centigrade. The experiment was carried out for the two weeks. It was found out that within 1.5 year heat pipe heat exchanger can pay for itself.

Y.H. Yau et al. (2009) carried out the literature review on the heat pipe heat exchanger as well as thermosyphon for the heating ventilation air conditioning and refrigeration (HVAC&R). They found out during the review that these heat exchangers due to their high thermal transport capacity can serve as great alternative to reduce carbon foot print and lower the effect of global warming, will also help to conserve the energy by heat recovery even for low temperatures.

Zukauskaset.al. given the empirical correlation along with Nusselt number, Reynolds number, and Prandtl number for the flow over cylinder and also calculating the average heat transfer coefficient.

$Nu = C\,Re^m\,Pr^n\,(Pr/Pr_s)^{1/4}$ for $0.7 < Pr < 500$ and $1 < Re < 10$

Where, 'C' and m are constant

$n = 0.37$ for $Pr \leq 10$ and $n = 0.36$ for $Pr > 10$.

Table 2.1values of 'C' and m for Zakaukas equation

Sr. No.	Range of Reynolds number	C	m
1	1 – 40	0.75	0.4
2	40 – 100	0.51	0.5
3	$1000 - 2 \times 10^5$	0.26	0.6
4	$2 \times 10^5 - 10^6$	0.076	0.7

Zukauskas also purposed correlations for calculating average heat transfer coefficient in tube bundle having more than 20 tubes

$Nu = C\,Re^m\,Pr^n\,(Pr/Pr_i)^{1/4}$

For $0.7 < Pr < 500$ and $1 < Re < 10^6$ and $N_L \geq 20$

Table 2.2 for the values of C and m

Sr. No.	Range of Reynolds number	Staggered		Aligned	
1.	10 – 100	0.90	0.4	0.8	0.4
2.	100 – 1000	0.51	0.5	0.51	0.5
3.	$1000 - 2 \times 10^{5}$	$S_T/S_L < 2$ $0.35\ (S_T/S_L)^{1/5}$ $S_T/S_L > 2$: 0.4	0.6	$S_T/S_L > 0.7$ 0.27	0.63
4.	$2 \times 10^{5} - 10^{6}$	0.076	0.7	0.21	0.84

In tube bundle, the Reynolds number is defined as

$Re = v_{max} \times D_L / \nu$

If the number of the rows n the tube bundle is less than 20, a correction factor is required, which is empirically related as

$NuI_{Nl<20} = C_I\, Nu\, I_{Nl>20}$

Where, C_I given in Table 2.3

Correction Factor for C_I for $N_L < 20$ ($Re > 10^3$)

Table 2.3 Correction Factor for C_I for N_L

Sr. No.	N_L	1	2	3	4	5	7	10	13	16
1.	Aligned	0.70	0.80	0.86	0.90	0.92	0.95	0.97	0.98	0.99
2.	Staggered	0.64	0.76	0.84	0.89	0.92	0.95	0.97	0.98	0.99

Zukauskas has further given the value for pressure drop across the staggered tube arrange

ment can be computed by

$\Delta P = N_L ((\mu/\mu_s)^{0.14} (\rho \times v_{max}^2 / 2)\, f$

$F = \{C_2 + C_3 / (S_{T/}do\ - 1)^n\}Re^m$

The value of C_2 , C_3, n, m are given in the table

Table 2.4 Coefficients of pressure drop for tube bank

Sr. No.	Tube arrangement	C_2	C_3	n	m
1.	Staggered bank	1.0	0.470	1.08	0.16
2.	Aligned Bank	0.176	0.34 ($S_{T/}$do)	0.43 + 1.13/S_T	0.15

Churchill and Bernstein have proposed a correlation for Nusselt number which covers wide range of Prandtl number and Reynolds number under forced convection:

$$Nu = 0.33 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/2}}{\left[1+\left(\frac{0.4}{\mathrm{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

Rayleigh number, which is given by the product of Grashoff number (Gr) & Prandlt number (Pr). The transition will occurs at the critical value of Rayleigh number of given as $Ra_x \pm 10^9$.

$$Ra_x = \frac{g\beta(T_s - T_a)L^3\rho^2\,\mathrm{Pr}}{\mu^2}$$

Churchill & Chu also given the correlation for free vertical plane surface for the entire range of Rayleigh number.

$$Nu_L = \left\{0.825 + \frac{0.387\,Ra^{1/6}}{\left[1+\left(\frac{0.492}{\mathrm{Pr}}\right)^{9/16}\right]}8/27\right\}^2$$

For $10^{-1} < Ra_L <^{12} 10$

Nusselt number for the laminar flow can be found by the following relation.

$$Nu_L = 0.68 + \frac{0.670 Ra^{1/4}{}_L}{\left[1 + \left(0.492 / \mathrm{Pr}\right)^{9/16}\right]^{4/9}}$$

For $0 < Ra_L < 10^9$

Chapter - 3

Gaps in Study & Objectives

4.1 Introduction

Literature review shows that lots is being done on the thermal performance of heat pipe heat exchanger and their uses in refrigeration and air conditioning systems have taken a steep rise due to excellent heat transport capabilities of heat pipe. HFC refrigerant like R134a, R410A, are also used as charge in heat pipe to deepen the knowledge. Already available working fluids based on past success are also used in variety of applications.

4.2 Gaps in study

Following are the points were considered to investigate the performance thermosyphon heat exchanger.

1. For two-phase closed thermosyphon working on water as charge will have high effectiveness on high inlet air temperature and low mass flow rate of air for Air-Water HPHEx. Increase in no. air passes will lead to more effectiveness
2. To reduce the power consumption and increase the quality of intake air in convention air conditioners heat pipe heat exchanger will serve as a great alternative.
3. This will help in the energy recovery for an air conditioning unit with the use of heat pipe heat exchanger.
4. For Indian sub-continent literature on heat recovery using TPHX is hardly found so it is necessary to perform research on the TPHX to deepen the knowledge.

4.3 Research Objectives

After an extensive literature survey it has been decided to investigate the performance heat pipe heat exchanger for the waste heat recovery on air conditioning using the application of the heat pipe. The main research objectives for this work are listed below:

1. To study the temperature distribution in a thermosyphon under varying load conditions. (100 watt - 600 watt).
2. Effectiveness of heat exchanger using the temperature and air face velocity condition for an air conditioning testing unit.
3. COP comparison of air conditioning heat exchange testing unit with and without the use of Thermosyphon heat exchanger for low temperature application (20 - 40).

Chapter - 4
Design Considerations for Thermosyphon Heat Exchanger

4.1 Introduction

This chapter presents the design considerations of the heat pipe along with analytical model developed for the thermal performance evaluation of the heat pipe heat exchanger for the heat recovery in air conditioning system.

4.2 Design Considerations for a thermosyphon

Various design considerations for the design of thermosyphon are:

- Selection of working fluid
- Container
- Computation of heat transport limits

These above mentioned parameters are presented below, so that the heat pipe can be optimum ally designed for the required applications. The operating temperature range for the present investigation is 20 -40.

4.2.1 Selection of the working fluid

The first consideration in the identification of the working fluid is the operating vapor temperature range. Within the approximate temperature band, several possible working fluids may exist and a variety of characteristics must be examined in order to determine the most acceptable of these fluids for the application considered on the basis of compatibility with wick and wall materials, good thermal stability, wettability of wick and wall materials high latent heat High thermal conductivity low liquid and vapor viscosities and having high surface tension.

The useful range of commonly used working fluid in the heat pipes are listed below in the table. In this list fluids are covered from the very low temperature applications to high temperature applications are given in the ascending order. Methanol chosen as working fluid.

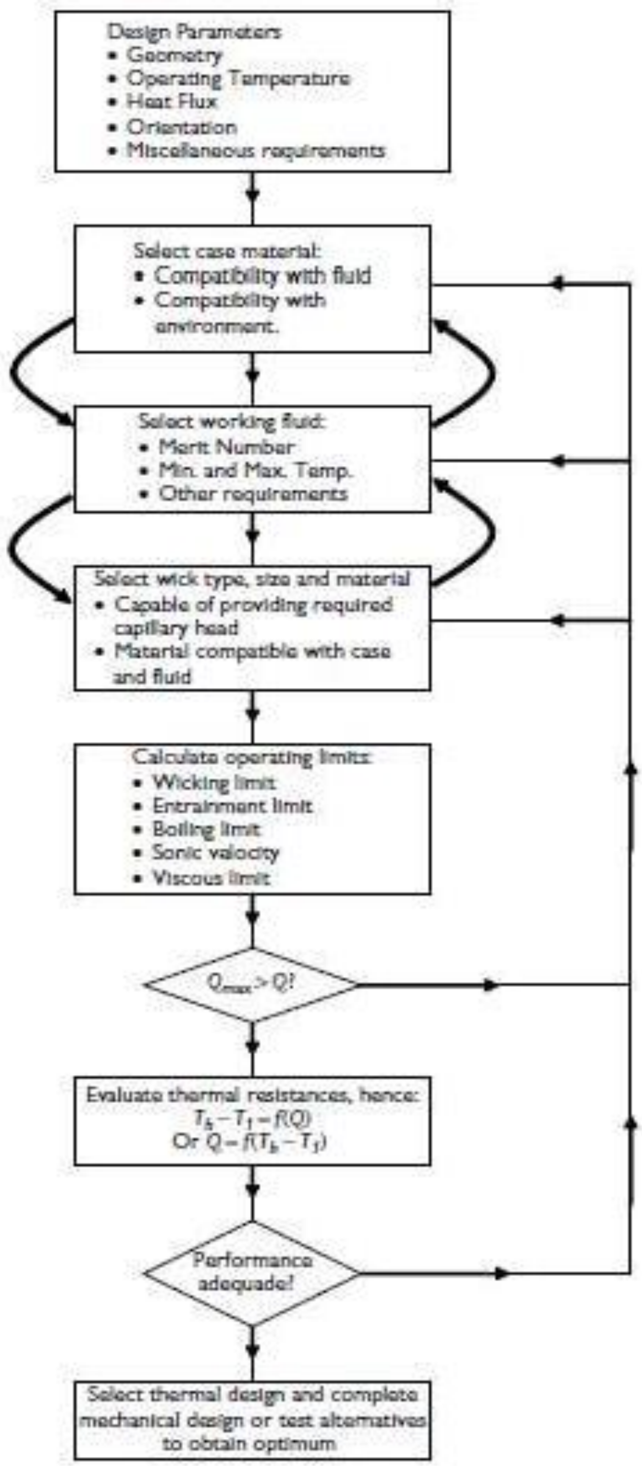


Fig. 4.1 Flow sheet for heat pipe design.

Figure: 4.1 Flow Chart for Heat Pipe Design [David Reay et. al. 2005]

Table: 4.1 Operating temperature range for various working fluids [www.thermopedia.com]

Medium	Melting Point (deg.)	Boiling point at atmos. Pressure (deg.)	Useful range
Helium	-271	-261	-271 to -269
Nitrogen	-210	-196	-203 to -160
Ammonia	-78	-33	-60 to -100
Pentane	-130	28	-20 to 120
Acetone	-95	57	0 to 120
Methanol	-98	64	10 to 130
Flutec PP2	-50	76	10 to 160
Ethanol	-112	78	0 to 130
Heptane	-90	98	0 to 150
Water	0	100	30 to 200
Toluene	-95	110	50 to 200
Flutec PP9	-70	160	0 to 225
Thermex[2]	12	257	150 to 350
Mercury	-39	361	250 to 650
Cesium	29	670	450 to 900
Potassium	62	774	500 to 1000
Sodium	98	892	600 to 1200
Lithium	179	1340	1000 to 1800
Silver	960	2212	1800 to 2300

4.2.2 Selection of Container

Selection of container is another important aspect for the designing of a Thermosyphon. Container must have high strength to weight ratio. Material of the container must not react with the working fluid. Container material should be inert to atmosphere. Thermal conductivity of container should be high. There from Copper is selected as container.

4.2.3 Compatibility of Container & Working Fluid

The purpose served by the container is to completely enclose the working fluid from its surroundings. It must be leak proof, should able to maintain the pressure difference across the walls as required, and must it to enable transfer the thermal energy from the working fluid without any interference.

Methanol

Methanol is also commonly known methyl alcohol, wood alcohol, wood naphtha or wood spirits etc. It is a chemical formula is given as CH_3OH. Thermophysical properties for temperatures ranging -50-150 ^oC are listed below.

Table: 4.2 Thermophysical properties of Methanol [www.thermopedia.com]

Temperature (^oC)	Latent Heat (kJ/kg)	Liquid Density (kg/m^3)	Vapor Density (kg/m^3)	Liquid Thermal Conductivity (W/m^{o}C)	Liquid Viscosity (cP)	Vapor Viscosity (10^2cP)	Vapor Pressure (bar)	Vapor Specific Heat (kJ/kg^{o}C)	Liquid Surface Tension (10^{-2} N/m)
-50	1194	844	0.01	0.210	1.700	0.72	0.01	1.20	3.26
-30	1187	834	0.01	0.208	1.300	0.78	0.02	1.27	2.95
-10	1182	819	0.04	0.206	0.945	0.85	0.04	1.34	2.63
10	1175	801	0.12	0.204	0.701	0.91	0.10	1.40	2.36
30	1155	782	0.31	0.203	0.521	0.98	0.25	1.47	2.18
50	1125	764	0.77	0.202	0.399	1.04	0.55	1.54	2.01
70	1085	746	1.47	0.201	0.314	1.11	1.31	1.61	1.85
90	1035	724	3.01	0.199	0.259	1.19	2.69	1.79	1.66
110	980	704	5.64	0.197	0.211	1.26	4.98	1.92	1.46
130	920	685	9.81	0.195	0.166	1.31	7.86	1.92	1.25
150	850	653	15.9	0.193	0.138	1.38	8.94	1.92	1.04

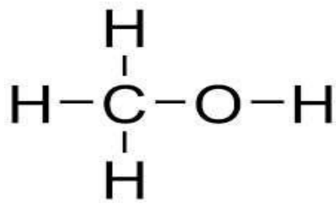
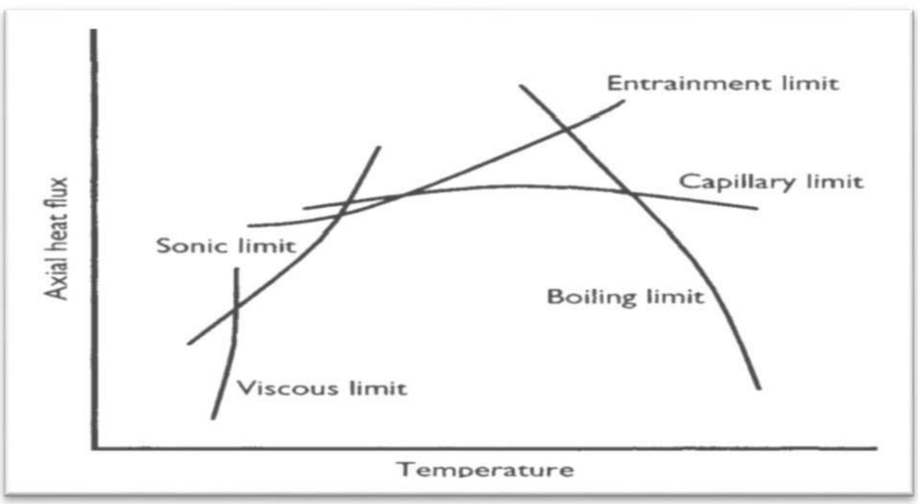


Figure: 4.2 Methanol Chemical formula
Figure: 4.3 Methanol Bottle

4.2.3 Heat Transport Limits for Thermosyphon

It is essential to choose correct material, for efficient transport of thermal energy between two systems. Figure 4.4 shows the various limitations for the transfer of heat using a heat pipe or Thermosyphon. In the study heat limits are excluded, and only Thermosyphon limits are taken into considerations. These limits are taken from [ESDU 80017 and David Reay et. al. 2005]



5.2Heat pipe limits [1-act.com]

4.2.4.1 Sonic limit

At a temperature above the vapor pressure limit, the vapor velocity can be comparable with sonic velocity and the vapor flow becomes "choked". The recommended maximum rate of heat transfer, to avoid choked flow conditions (i.e., sonic limit) is given by

$$Q'_s = 0.474 A_v h_{fg} \sqrt{(p_v \rho_v)}$$

4.2.4.2 Boiling Limit

The temperature drop across the wick structure in the evaporator region increases with evaporator heat flux. A point is reached when temperature difference exceeds the degree of superheat sustainable in relation to nucleate boiling conditions. The onset of Boiling within the wick structure interferes with liquid circulation. This eventually leads to "dry out", which in the case of constant heat flux heating can cause "Burn Out" of the evaporator containment.

In the event of nucleate boiling the relationship between bubble radius and pressure difference sustainable across the curved surface is given by

$$Q_b = 0.12 h_{fg} \sqrt{(p_v)} \left[g\sigma(\rho_l - \rho_v)^{1/4} \right]$$

4.2.4.3 Flooding Limit

This condition also refers to entrainment or flooding limit. The vapor velocity increases with temperature and may be sufficiently high to produce shear force effects on the liquid return flow from the condenser to the evaporator, which cause entrainment of the liquid by the vapor. The restraining force of liquid surface tension is a major parameter in determining the entrainment limit. Entrainment will cause a starvation of fluid flow from the condenser and eventual "dry out" of the evaporator. The entrainment limit is given by

$$Q_{man} = A_v h_{fg} f_1 f_2 f_3 \sqrt{\rho_v} \left[g\sigma(\rho_l - \rho_v) \right]^{1/4}$$

$f_1 \rightarrow$ Function of Bond No.

$$B_o = D_v \left[g\left(\frac{\rho_l - \rho_v}{\sigma} \right) \right]^{1/4}$$

$B_o = 1.0$ Then $f_1 = 4$

$B_o = 10.0$ Then $f_2 = 8$

4.2.4.4 Figure of Merit

In selecting the working fluid for a heat pipe or thermosyphon it is necessary to ensure that the device operates within the above defined limits. The choice of working fluid very much depends on the thermophysical properties of the fluid as well as the mode of operation of the device. A figure of merit (ϕ) may be used to establish the relative performance of a range of prospective working fluids.

$$\phi = \left[\frac{h_{fg} \lambda^3 \rho^2}{\eta} \right]^{1/4}$$

4.3 Heat Exchanger effectiveness criterion

To investigate the theoretical performance two criteria available to us:

Logarithmic Mean Temperature difference (LMTD)

No. of Transfer Units (NTU) method

Out of these as we already know the area of unit so NTU method can readily employed various correlations used in this method are taken from literature.

$\varepsilon_{h1} = 1\text{-exp } (\text{-NTU})_h$

$\varepsilon_{c1} = 1\text{-exp } (\text{-NTU})_C$

$(NTU)_h = \frac{(UA)_h}{C_h}$

$C_h = (\dot{m}C_p)_h$

$C_c = (\dot{m}C_p)_c$

For Evaporator

$$\varepsilon_{hn} = \frac{\left(\frac{1-\frac{C_h}{C_L}\varepsilon_{h1}}{1-\varepsilon_{h1}}\right)^n - 1}{\left(\frac{1-\frac{C_h}{C_L}\varepsilon_{h1}}{1-\varepsilon_{h1}}\right)^n - \frac{c_h}{c_L}}$$

For Condenser

$$E_{cn} = \frac{\left(\frac{1-\frac{C_c}{C_L}\varepsilon_{c1}}{1-\varepsilon_{c1}}\right)^n - 1}{\left(\frac{1-\frac{C_c}{C_L}\varepsilon_{c1}}{1-\varepsilon_{c1}}\right)^n - \frac{c_c}{c_L}}$$

For $C_h/C_L = 0$ and $C_c/C_L = 0$ the above equation reduced to

$$\varepsilon_{hn} = 1 - (1 - \varepsilon_{h1})^n$$

&

$$\varepsilon_{cn} = 1 - (1 - \varepsilon_{c1})^n$$

The overall of tiveness of heat exchanger ε_t is given by:

If $C_h > C_c$

$$\varepsilon_t = \left(\frac{1}{\varepsilon_{cn}} + \frac{C_c/C_h}{\varepsilon_{hn}}\right)^{-1}$$

If $C_c > C_h$

$$\varepsilon_t = \left(\frac{1}{\varepsilon_{hn}} + \frac{C_h/C_c}{\varepsilon_{cn}}\right)^{-1}$$

The outlet temperature of evaporator and condenser

$$T_{h,out} = T_{h,in} - \varepsilon_t\left(\frac{(\dot{m}Cp)_{min}}{(\dot{m}Cp)_h}\left(t_{h,in} - t_{c,in}\right)\right)$$

$$T_{c,out} = T_{c,in} + \varepsilon_t\left(\frac{(\dot{m}Cp)_{min}}{(\dot{m}Cp)_h}\left(t_{h,in} - t_{c,in}\right)\right)e$$

To evaluate the experimental effectiveness of the thermosyphon heat exchanger following formulas were taken from the literature

$$\varepsilon = \left(\frac{q_{actual}}{q_{max}}\right)$$

If $C_h > C_c$

$$\varepsilon_t = \left(\frac{t_{c,out} - t_{c,in}}{t_{h,in} - t_{c,in}}\right)$$

If $C_c > C_h$

$$\varepsilon_t = \left(\frac{t_{h,in} - t_{h,out}}{t_{h,in} - t_{c,in}}\right)$$

Chapter- 5

Experimental Setup and Methodology

5.1 Experimental Setup for Air Conditioning Rig with Thermosyphon

This section provides a detailed description of the components and their working for experimentation in this system. Following figures shows the setup for thermosyphon equipped with RTD sensors, non-return valves, pressure gauge indicating negative pressure inside the tube.

Figure: 5.1 Thermosyphon setup

Figure 6.1: Air conditioning test rig

In this experimentation above figure shows the test rig for actual vapor compression refrigeration system (Air Conditioning) where experimentation for heat recovery was performed. After the testing of thermosiphon's and getting the expected performance from them, these were fitted inside the test rig of a VCRS. Test chamber temperature of the test maintained at a constant temperature and blown over condenser section of heat exchanger and fresh air was introduced in the evaporator section of heat exchanger.

Table 5.1 Components of Air Conditioning Test Rig

Sr. No.	Components	Qty.	Specification's
1	Compressor	1	190Litre.
2	Expansion Device	1	Automatic
3	Condenser	1	Forced convection with fan
4	Evaporator	1	Coil Type
5	Filter	1	Silica Gel employed
6	Pressure Gauge	4	Low And High Pressure
7	Heating Element	1	250 W
8	Rotameter	1	1-45 LPH
9	Refrigerant	4500gms	Pure Hydrocarbon(R134a)
10	Voltmeter	1	0-320 volt
11	Amp Meter	1	0-20 Amp
12	Energy meter	1	Normal Reading
13	Digital Temperature Controller	1	0 – 300 ^{o}C
14	Digital panel meter	1	
15	Flexible Charging Line	1	
16	Vacuum Pump	1	

Compressor

Compressor is an integral and important part of any refrigeration system. Its main work is to raise the pressure and temperature of the refrigerant across the pipe by giving some pumping power. Hermetically sealed compressors, which are commonly used in vapor compression systems, depending upon the requirement. In this setup hermetically sealed compressor of 250capacity has been used.

Fig. 6.3 Compressor

Evaporator

The liquid refrigerant enters the evaporator from the refrigerant flow control device is under low pressure and low temperature. Evaporator makes contact with the water and he medium from which heat has to be removed. Here liquid refrigerant changes phase to vapor phase

Figure 6.4 Evaporator

Filter

Impurities present in the vapor compression refrigeration system can damage number of components of the system like compressor and expansion device can damage due to impurities. To obstruct any impurity present in the system and to avoid any kind of choking filter has been used in experimental setup at condenser outlet. Mesh size of filter varies from (5-5000) micrometers. Nanoparticles can pass through this filter easily.

Figure 6.5 Filter

Pressure Gauge

It is used to measure pressure of the hydrocarbon Refrigerant at respective points. Four pressure gauges are used in which one gauge is at compressor inlet, and other at outlet, one gauge after expansion device and other after evaporator outlet. Bourdon tube type refrigeration pressure gauges are used to measure the pressure. The bourdon pressure gauge works on the principle that a flattened tube tends to straighten or regain its original form in cross-section when pressurized. In this set up two types of gauges are used high pressure gauge and low pressure gauge.

Figure 6.6 Pressure Gauge

Refrigerant

In the experimental setup it has been decided to use R-134a refrigerant, due to reason of its wide commercial use and acceptability. R134a is commonly used refrigerant in domestic refrigerators and cars.

Figure 5.10 R134a Refrigerant Can

Rota meter

A rotameter consists of glass tube, with a float actually a shaped weight which is pushed up by the drag force of the flow and pulled down by gravity. For the flow rate of refrigerant glass tube rotameter has been used. Its position is vertical and its range is 2- 40LPH. The liquid refrigerant coming from the condenser passes through the rotameter and in this way moving element gives reading. The rotameter is manufactured by Zest engineering Delhi.

Figure 5.11 Rotameter

Voltmeter

A voltmeter is a device used to calculate electrical potential difference between two points in an electric circuit. Analog voltmeters move a pointer across a scale in proportion to the voltage of the circuit whereas digital voltmeter shows the voltage digitally. It draws only a smallest amount of current to operate.

Figure 5.12 Voltmeter

Ammeter

An ammeter is a measuring device used to measure the electric current in an electrical circuit. Electric currents are measured in amperes (A). In this set up digital ammeter has been used.

Figure 5.13 Ammeter

Digital Temperature Controller

Temperature controller is a device used to measure variation in temperature of space and can be adjusted to achieve a desired temperature. Digital temperature controller is used to cut off the heater supply when the temperature exceeds a particular set value in the evaporator. Itisconnected to a Relay device which is connected to heating element in which it sends signal to the relay switch which cuts off the supply as well as on the supply of the heating element. There is a sensing element which always dipped in the water in evaporator. With the help of digital temperature controller constant heat flux in the evaporator can be maintained.

Figure 5.14 Digital Temperature Controller Figure 5.15 Relay Switch

4.2.15 Temperature Gauge (Digital Panel Meter)

Temperature sensors are used to measure temperature at different points in the vapour compression refrigeration system. The temperature sensors used here is PT100 thermocouples. The temperature sensors are placed at 4 different points, namely, the inlet and outlet of compressor and the inlet and outlet of the expansion valve. The readings from all the temperature sensors are displayed on a digital panel meter. The temperature at the required point can be noted by operating the selector switch present on the digital panel meter.

Figure 5.16 Digital panel meter

Flexible Charging Line

Flexible line is used to charge the refrigerant into the system through compressor charging line. Line is connected to the gas cylinder and then connected to the valve from where refrigerant goes in the vapor compression system.

Figure 4.17 Flexible Charging line

Vacuum Pump

Vacuum compressor is a compressor similar to the compressor used in the vapor compression refrigeration system. It can be used to check for any leakage and is also used to do charging of the system with the air. Also before charging refrigerant the vacuum is produced in the system to remove moisture or any air.

Figure 4.18 Vacuum Pump

Chapter - 6

Results and Discussion

In this part of the dissertation analysis of various results had been carried out and based on that some discussion were carried out. This chapter is divided into two sub chapters. In the first part analysis of single heat pipe based on its performance parameters filling ratio, heat load and inclination angle etc. was done. In the second part of this chapter analysis of thermosyphon heat exchanger was carried for an air conditioning test rig. In this part performance parameters for thermosyphon heat exchanger viz. cold air inlet temperature for condenser section, air face velocity for the evaporator section were taken into consideration. Later in this chapter effect of the thermosyphon heat exchanger was studied on the C.O.P of the system

6.1 For Single Thermosyphon

Effect of variable heat input 100–600 watt and constant coolant flow rate 5.5 kg/hr. in the condenser section of the thermosyphon. The filling ratio for the thermosyphon was chosen in the range from 30% - 60 %. The inclination angle was not varied and kept constant at 90°.

Variable Parameters:

- Filling Ratio
- Heat Input

Constant Parameters:

Inclination Angle

Coolant Flow Rate

Effect of Filling Ratio

Firstly the effect of filling ratio was studied in the thermosyphon. Thermosyphon was charged with 30 % filling ratio (filled from evaporator side). Then RTD sensors was placed over the thermosyphon on respective places as shown in graph 7.1.1. Adiabatic section was well insulated and only one sensor was applied in this section to see the variation of temperature along the length. Two sensors each for evaporator and condenser section was study temperature drop along the length of the thermosyphon. A heat load of using electric heater with heat input of 145 watt was used. In the condenser section of thermosyphon a water jacket was employed, having a constant coolant flow rate of 5.5 kg/hr. This procedure then again repeated for the 40%, 50% and 60% filling ratio.

After following the above mentioned procedure various data was collected. The relationship between length and temperature were established. Following graphs are plotted.

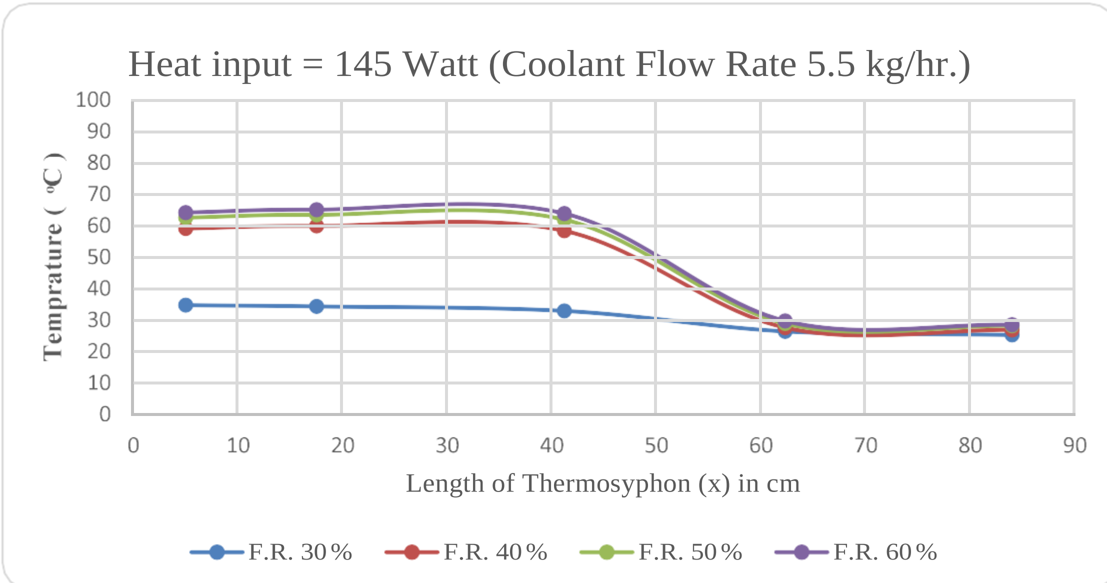


Figure: 6.1 Temperature distribution at 145 watt

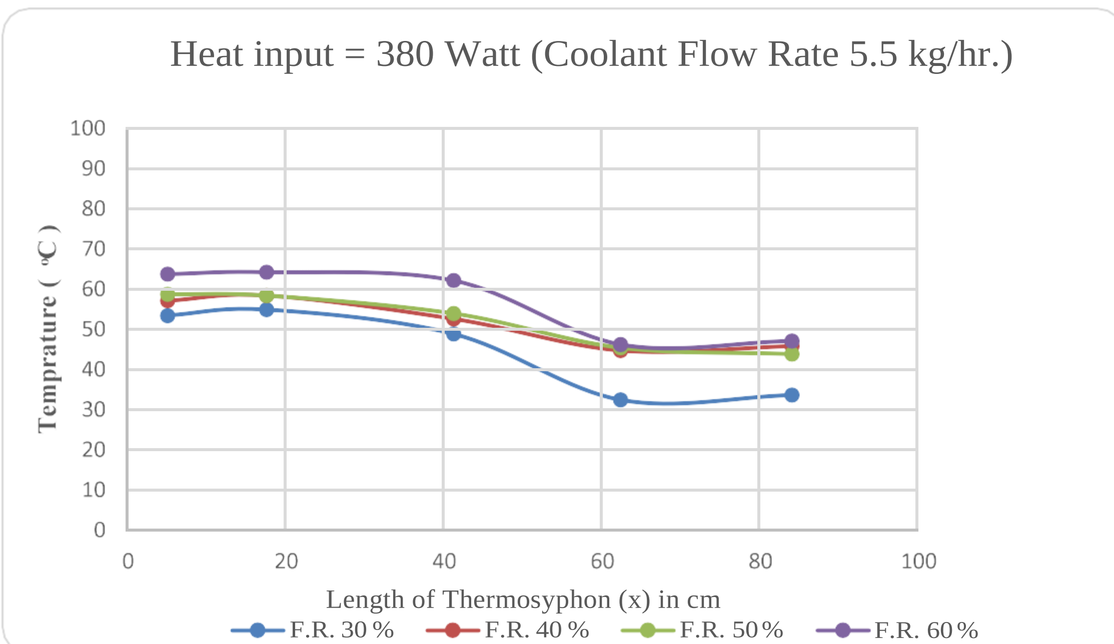


Figure: 6.2 Temperature distribution at 380 watt

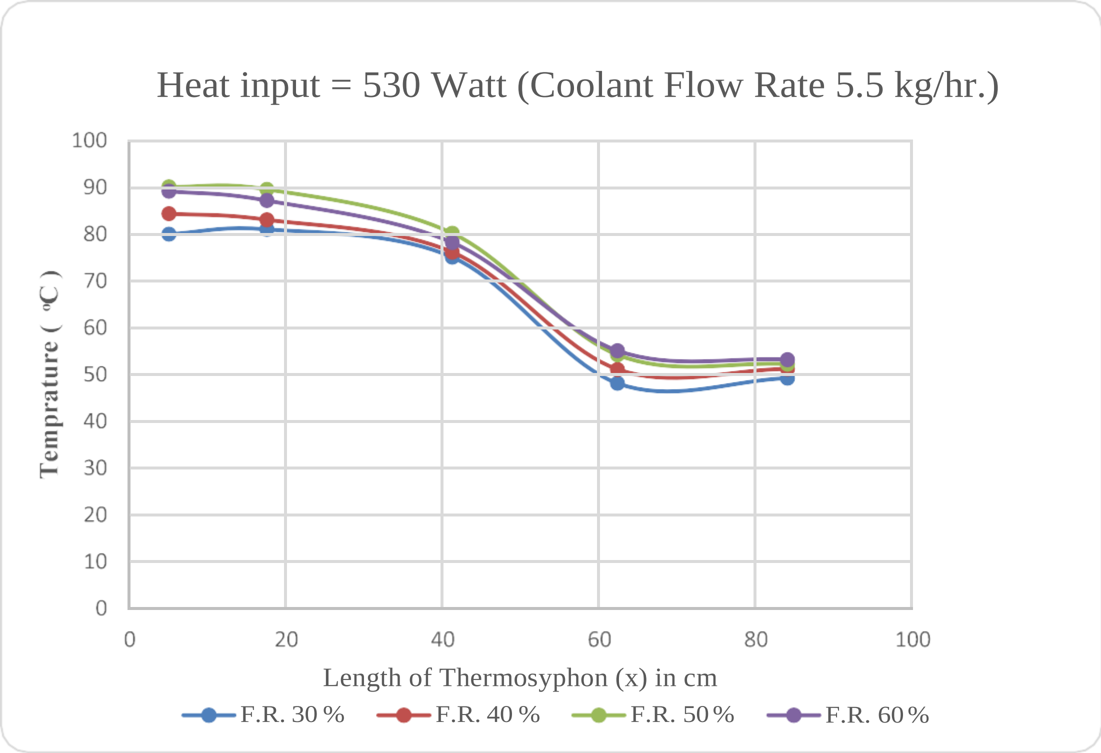


Figure: 6.3 Temperature distribution at 530 watt

Figure 7.1 (a) - (c) shows the variation in the temperature along the length of the thermosyphon. In figure (a) at low filling ratio at low value of heat load supplied, due this rise in the temperature for that curve is minimum which is visible from the figure (a). But as we increase the heat load from 145 W to 530 W, the value for the maximum filing ratio 60% chosen in work, temperature drop occurs. Hence high filling ratio is not recommended in case of vertical thermosyphon. In intermediate heat flux of 380 watt shown in the Figure (b), due low vapor pool variation at 30% filling ratio is quite significant.

Effect of Heat Load

In this part of the work effect of heat load on the single thermosyphon was studied. Plots were obtained to compare all heat loads for a single filling ratio. As was in case of filling ratio, both temperature and length of thermosyphon was taken on Y and X axis respectively.

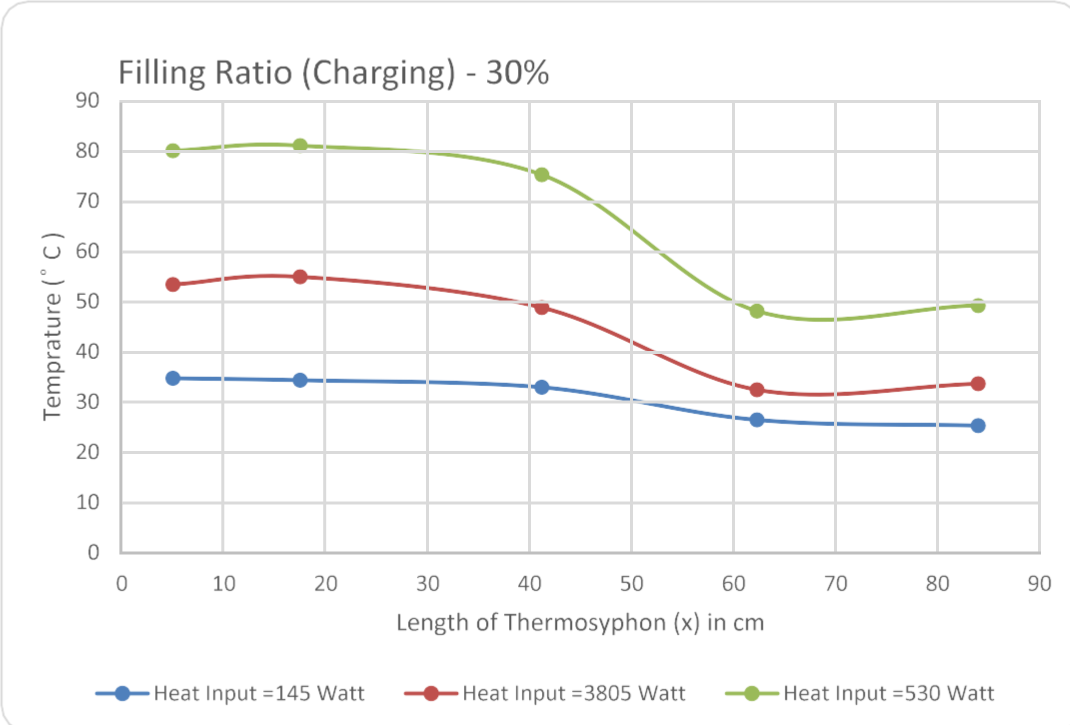


Figure: 6.4 Heat input comparison at Filling Ratio 30%

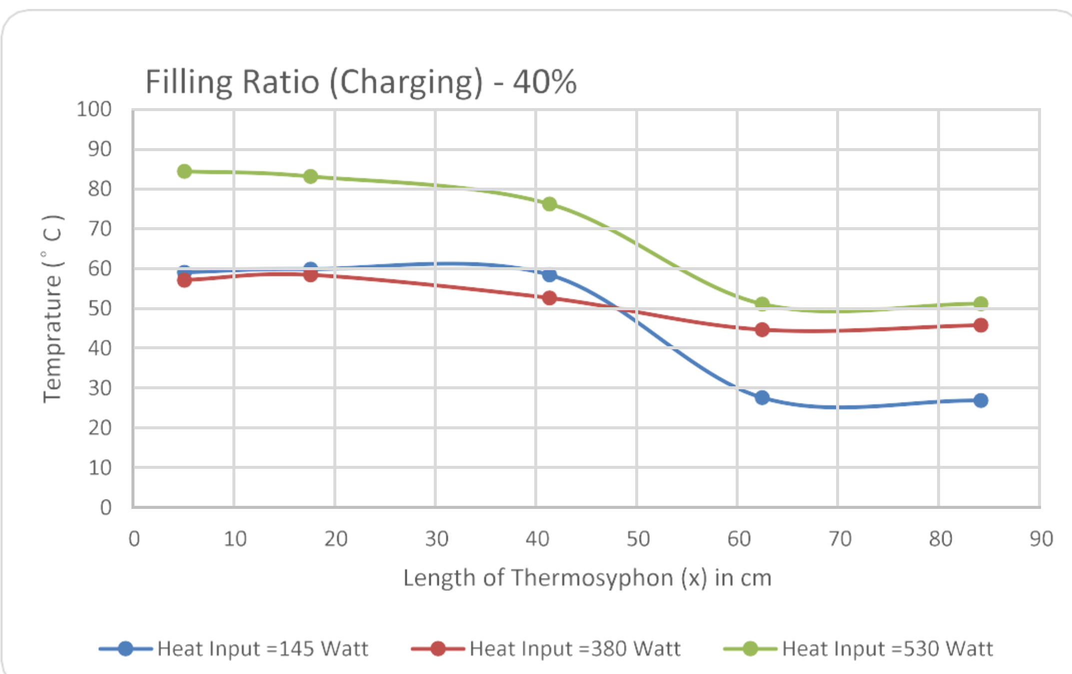


Figure: 6.5 Heat input comparison at Filling Ratio 40%

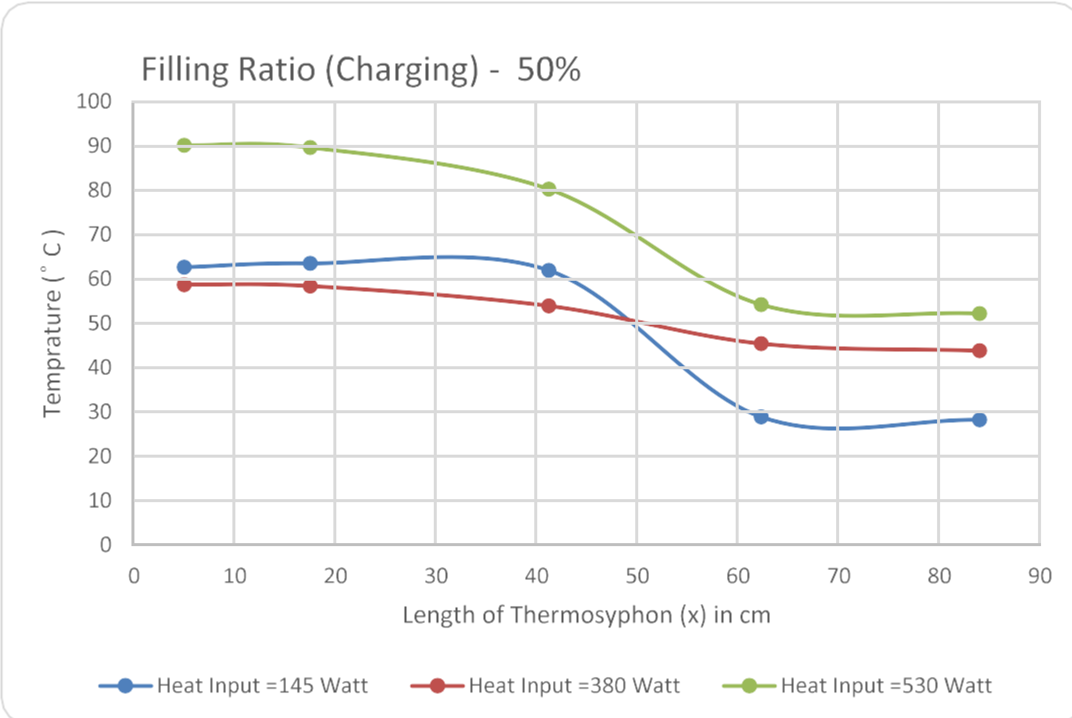


Figure: 6.6 Heat input comparison at Filling Ratio 50%

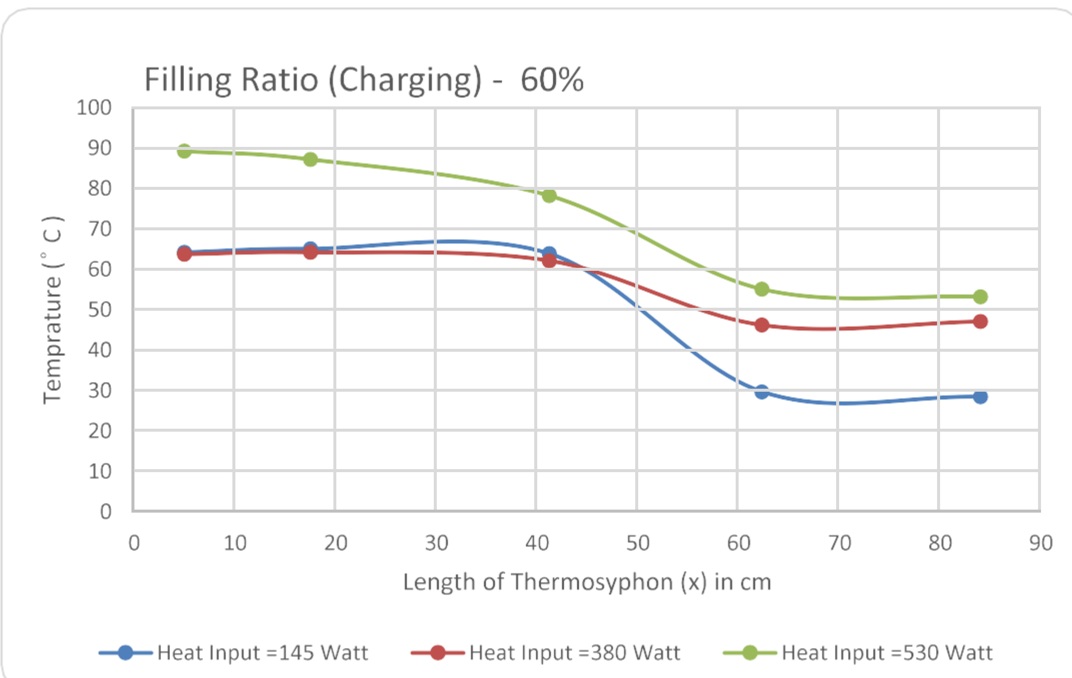


Figure: 6.7 Heat input comparison at Filling Ratio 60%

7.2 For Thermosyphon Heat Exchanger

In this study, performance thermosyphon heat exchanger was investigated by performing the tests under low temperature conditions. The one row inline three copperpipe thermosyphon heat exchanger was constructed and fitted inside the air conditioning test rig. There was a counter flow heat exchange arrangement, in which fresh air from the atmosphere and cold air from the test chamber were blown of over the heat exchanger respectively, having constant mass flow rate. This work was done to evaluate the performance of thermosyphon heat exchanger as heat recovery device for low temperature applications in the case of HVAC & R. The temperature range was selected between 20 – 40°C. In this work temperature at the inlet of condenser section of heat exchanger was consistently varied, to study the variation of effectiveness with respect to condenser inlet temperature. Air face velocity coming out from the test chamber, was fairly constant with minor variations. Air face velocity was considered an important parameter, so in this work effect of air face velocity on the evaporator section was also studied.

- Mean temperature of air to evaporator inlet $T_{E,\,in}$ = 38.2 °C
- Mean temperature of air to evaporator outlet $T_{E,\,out}$ = 31.3 C
- Mean temperature of air to condenser inlet $T_{C,\,in}$ = 18.1 C
- Mean temperature of air to condenser outlet $T_{C,\,out}$ = 24.7 C

Thermal heat transport air was very high as compared to single thermosyphon heat conduction in axial direction. Therefore in this investigation only overall Thermal heat transport air is taken in consideration.

To determine the thermal transport carrying capacity of both section of the thermosyphon general formulas were taken from the literature:

$$Q = mc_p \Delta T$$

$$Q = \rho u A c_p \Delta T \qquad [\ m = \rho u A\ ]$$

For evaporator section:

$$Q_E = \rho u A c_p (t_{E,in} - t_{E,out})$$

$$Q_E = 1.148 \times 2.3 \times (0.35 \times 0.13) \times 1006.1 \times (38.2 - 31.3)$$

$$Q_E = 834.01 watt$$

For condenser section:

$$Q_C = \rho u A c_p (t_{C,in} - t_{C,out})$$

$$Q_C = 1.198 \times 2.3 \times (0.25 \times 0.13) \times 1006.1 \times (24.7 - 18.1)$$

$$Q_C = 594.6 watt$$

The effectiveness of heat exchanger

$$\varepsilon = \frac{Q_{act}}{Q_{\max}} = \frac{t_{E,in} - t_{E,out}}{t_{E,in} - t_{C,out}}$$

$$\frac{38.2 - 31.3}{38.2 - 18.1} = \frac{6.9}{20.1} = 0.343$$

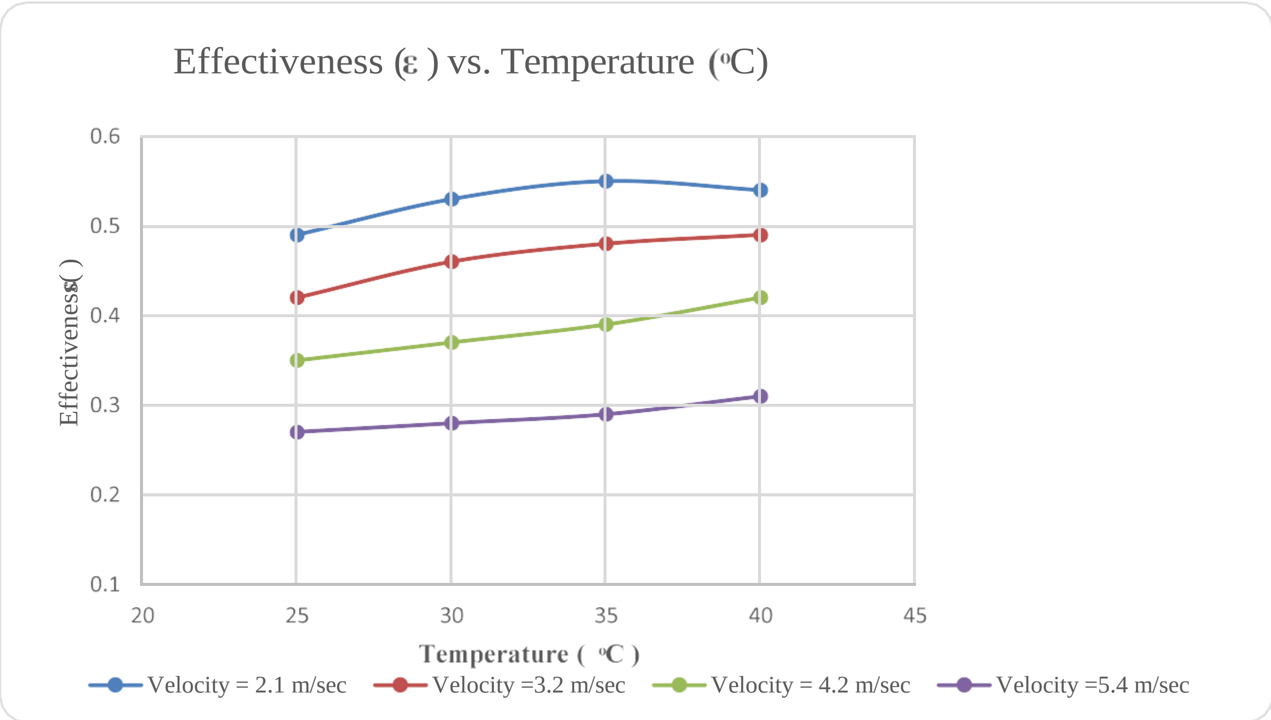


Figure: 7.8 Effectiveness vs Temperature

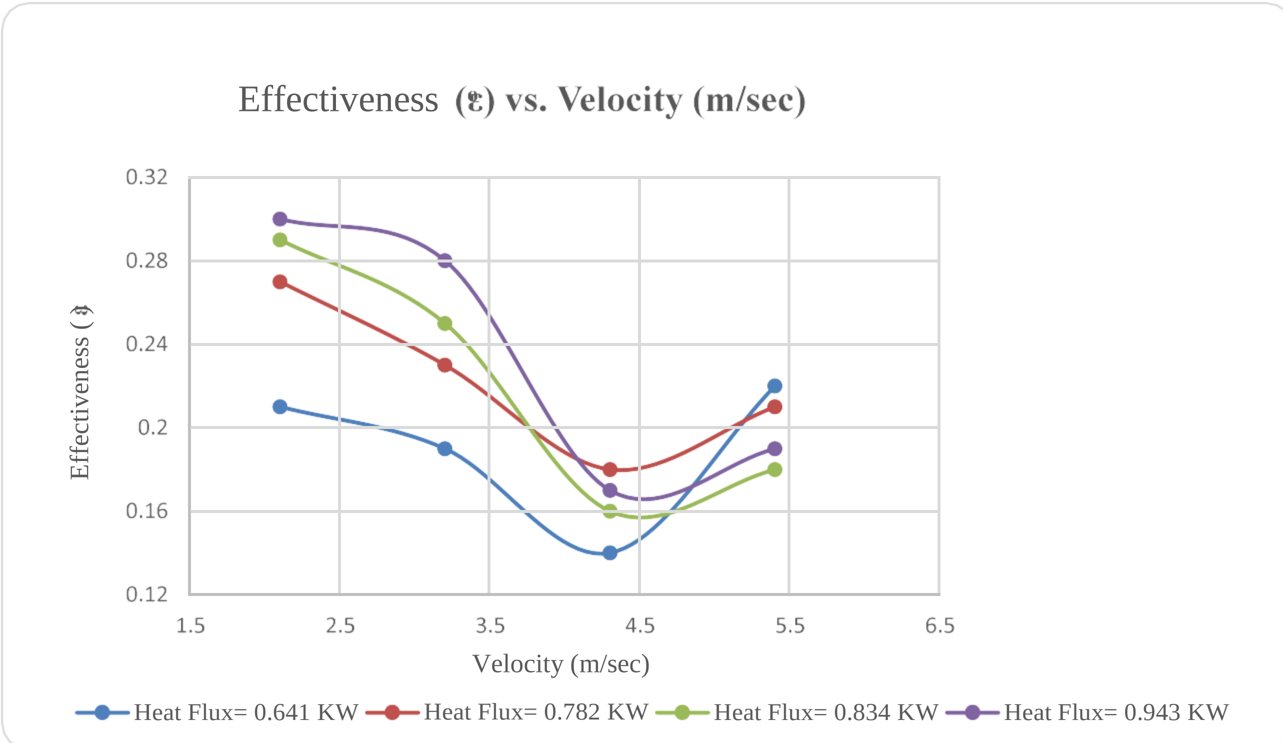


Figure: 7.9 Effectiveness vs Velocity

8.3 Coefficient of Performance (C.O.P.)

C.O.P. is defined as the ratio of refrigeration effect and work supply to the system. In this case, actual refrigeration effect is equal to of power required by heater submerged in water it means heat absorb by the evaporator coil is equal to heat released by heater submerged in water, so C.O.P. is the ratio of power required by heater submerged in water and power consumed by the compressor. Power required by heater submerged in water and power consumed by the compressor both are measured by Energy meter.

$$\mathrm{C.O.P} = \frac{\text{Refrigeration Effect}}{\text{Power Supplied}}$$

Following plots shows the variation in COP for actual, theoretical and thermosiphon's with respect to condenser and evaporator temperature respectively.

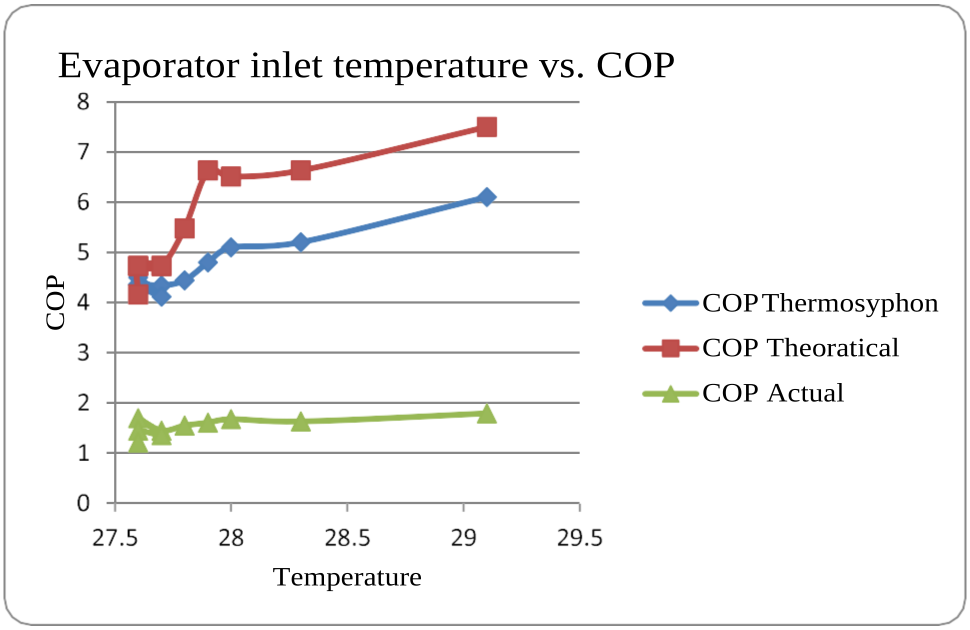


Figure: 7.10 Evaporator inlet temperature vs COP

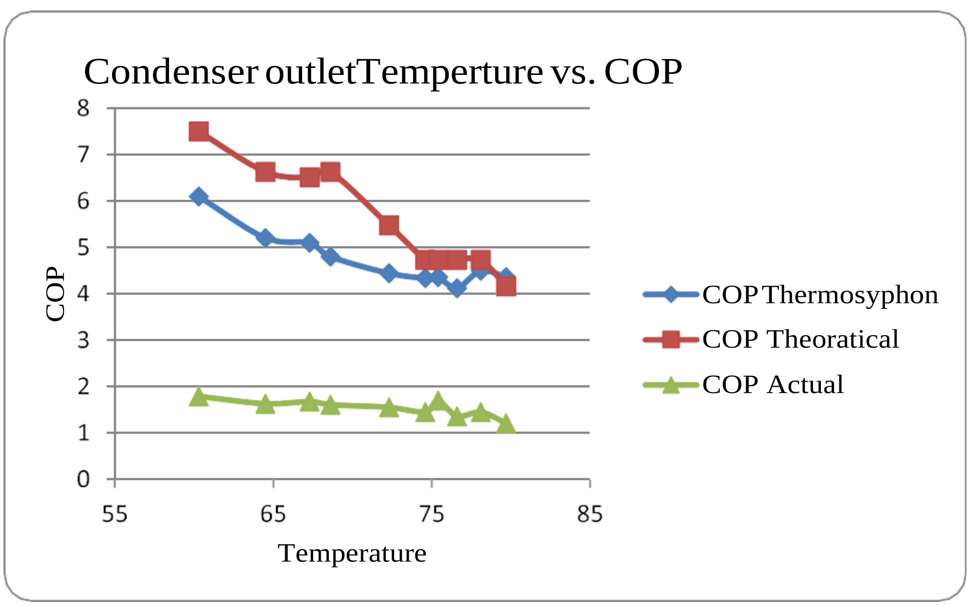


Figure: 7.11 Condenser outlet Temperature vs COP

Chapter - 7

Numerical Simulation

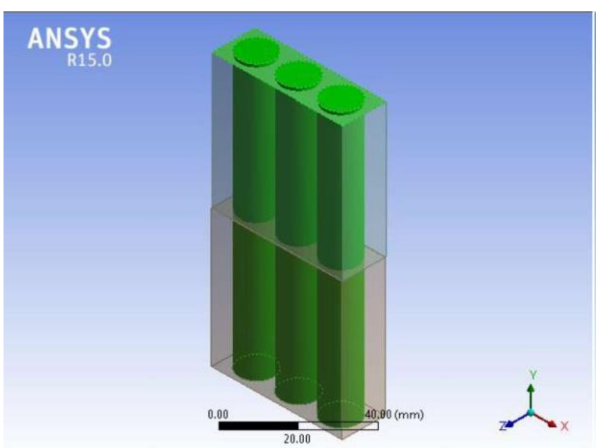
8.1 Introduction

In this work numerical simulation of heat pipe heat exchanger has been performed. Heat pipe was assumed as solid rod of constant conductivity. In this analysis was carried out on copper water Thermosyphon to investigate the effect of inlet air temperature and air mass flow rate was studied The temperature was varied from 20 deg. to 40 deg. with 5 deg. Step Inlet air mass flow rate from evaporator was varied from 0.03 kg/sec to 0.09 kg/sec with 0.03 kg/sec step The mesh dependency and grid independence test was performed in order to reduce the error in the work. The results obtained was compared with the available literature. The temperature contours for condenser and evaporator was plotted.

8.2 Geometric modeling

For Geometry 1

- Length of Thermosyphon 700 mm Diameter of HP- 30mm
- No of HP's – 3 Length of Evaporator- 350 mm
- Length of Condenser – 350 mmPlane - XZ



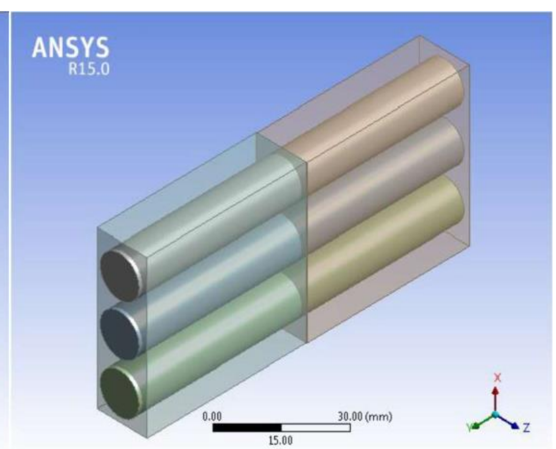


Figure: 7.1 Geometry 1 Isometric view

Figure 7.2 Geometry 1 Side view

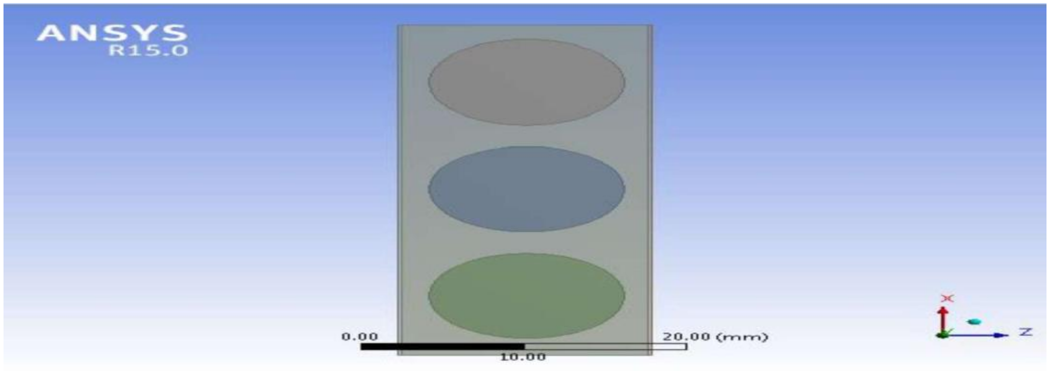
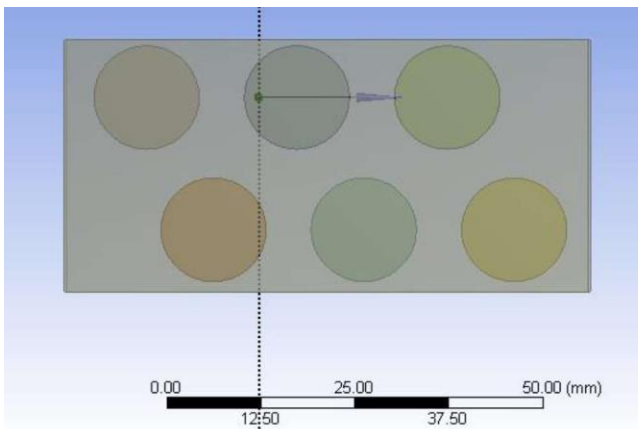


Figure: 7.3 Geometry 1 Top view

For Geometry 2



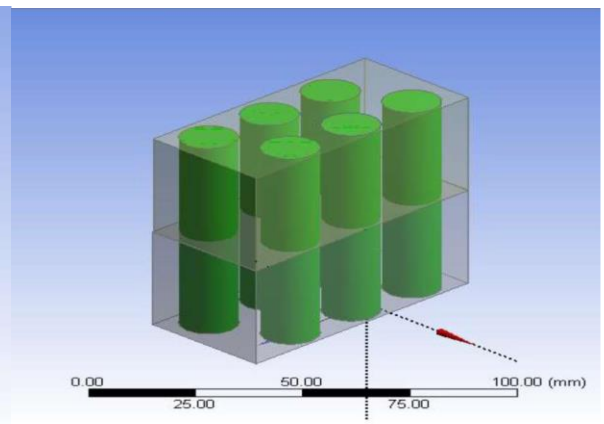


Figure: 7.4 Geometry 2 Top view

Figure: 7.5 Geometry 2 Isometric view

- Length of Heat Pipe (HP)- 700 mmDiameter of HP- 30mm
- No of thermosyphon ~~6~~Length of Evaporator- 350 mm
- Length of Condenser – 350mm Plane - XZ

8.3 Meshing:Meshing is very important parameter in order to accurately simulate a given system.

Steps of Grid Generation:

- Geometry was imported from the design modular of Ansis (Fluent)
- Click on the mesh button to view all the messing options.
- Click on auto generate the mesh

- Inflation was added to capture the in turbulent flow near the wall

Then the sizing and assembly parameters were also varied to get the required mesh

Table: 7.1 Meshing parameters for geometry 1

Method	No. of cells	Type of cells	Skewness	Time pre iter.
Auto Mesh	4704	Tetra+hexa	Max. 0.46	10-12 sec
Cut shell	12892	Sqr+ hexa	Max. 0.94	15-20 sec
Tetrahedron	62194	Tetra +Hexa	Max. 0.91	18-21 sec

Table: 7.2 Meshing parameters for geometry 2

Method	No. of cells	Type of cells	Skewness	Time per iter.
Auto Mesh	12117	Hexa+Tetra	Max. 0.46	15-20 sec
Cut shell	82136	Sqr + Hexa	Max. 0.936	25-32 sec
Tetrahedron	614227	Tetra+hexa	Max. 0.923	40-59 sec

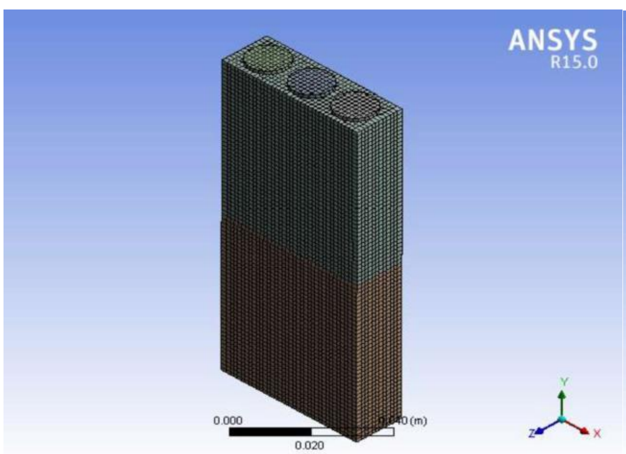


Figure: 7.6Meshing for Geometry 1

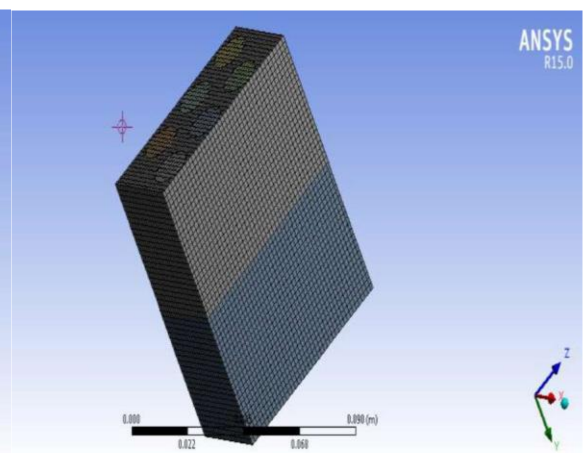


Figure: 7.7 Meshing for Geometry 2

8.4 Solver

Following steps were taken to perform the simulation in the Fluent

Go to general to check the mesh, display all the named section, generate a report on the quality of meshing (orthogonal meshing value will be between 0-1 (try to achieve near 1) for good mesh

Select the appropriate model according to geometry.Choose the material for the bodyCheck the cell zone boundary conditionApply the suitable boundary conditions.In the solution methods select the suitable spatial discretization.Select the initialization method give the value of no. of iteration.Run the solver

Boundary conditions

A. For Geometry 1

- Inlet Temperature of Cold Fluid- 18deg Inlet Temperature of Hot Fluid - 35 deg
- Velocity of Cold Inlet – 3.6 m/sec Velocity of Hot Inlet- 2.6 m/sec

B. For Geometry 2

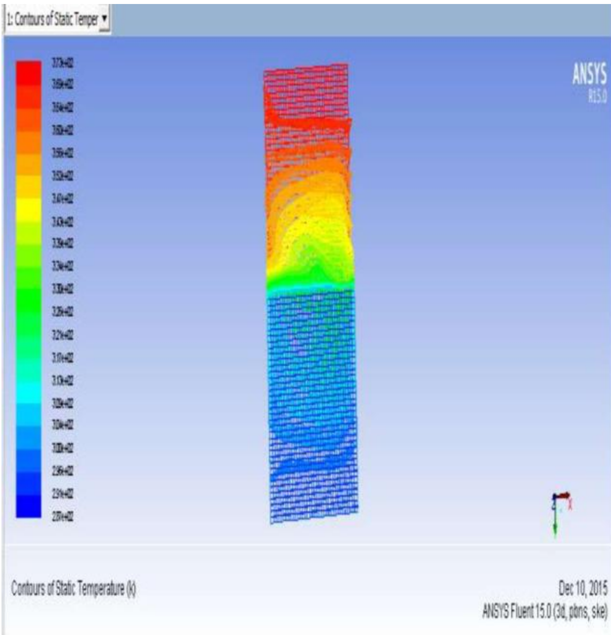
- Inlet Temperature of Cold Fluid- 18deg Inlet Temperature of Hot Fluid - 35deg
- Velocity of Cold Inlet – 3.6 m/s Velocity of Hot Inlet- 2.6 m/ sec

Parameter Varied:

- Hot fluid inlet temperature - 20 - 40deg Velocity of hot fluid inlet-2.6 m/sec – 4.6 m/sec
- Cold Fluid inlet Temperature18deg (constant) Cold fluid inlet Velocity - 3.6 m/sec (constant)

8.5 Results

After applying the solver to the designed model, solution is converged for the given boundary conditions. Various parameters such as static temperature, turbulent kinetic energy, velocity etc. can be studied. Their temperature, velocity and kinetic energy contours plotted as shown in figure The variation of these parameters along the length was also plotted shown by figure For Geometry



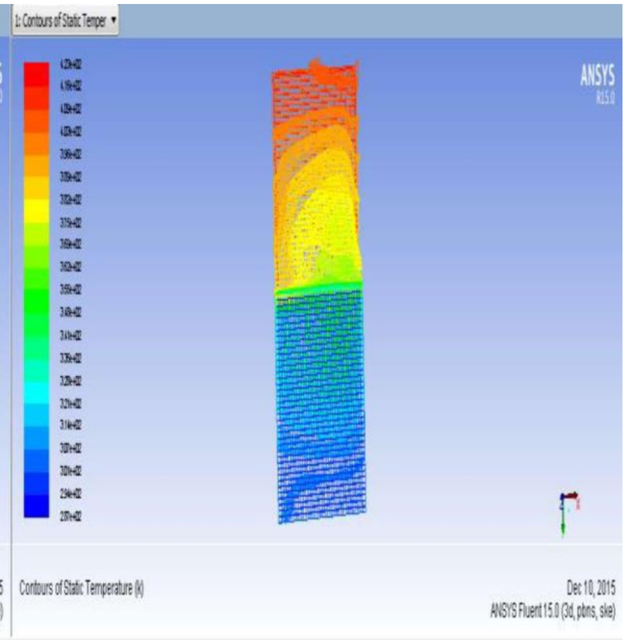
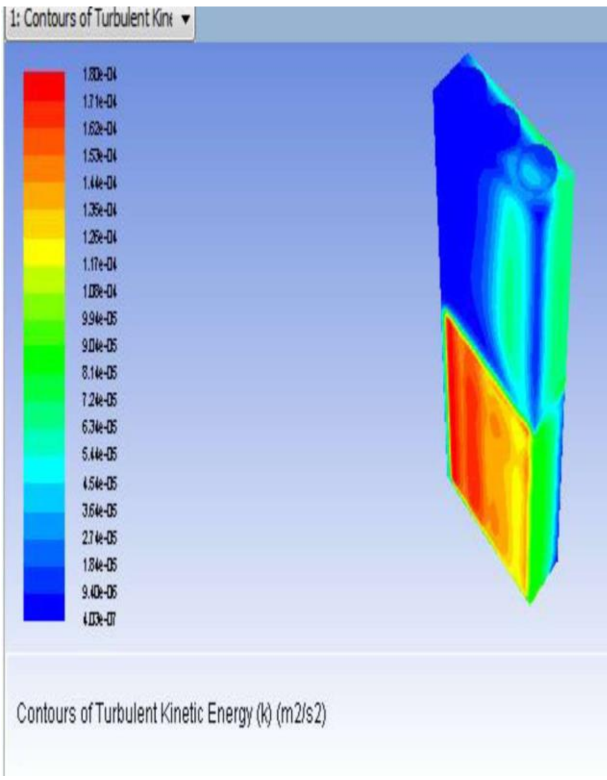


Figure: 7.8 Temperature Contours

Figure: 7.9 Velocity variations



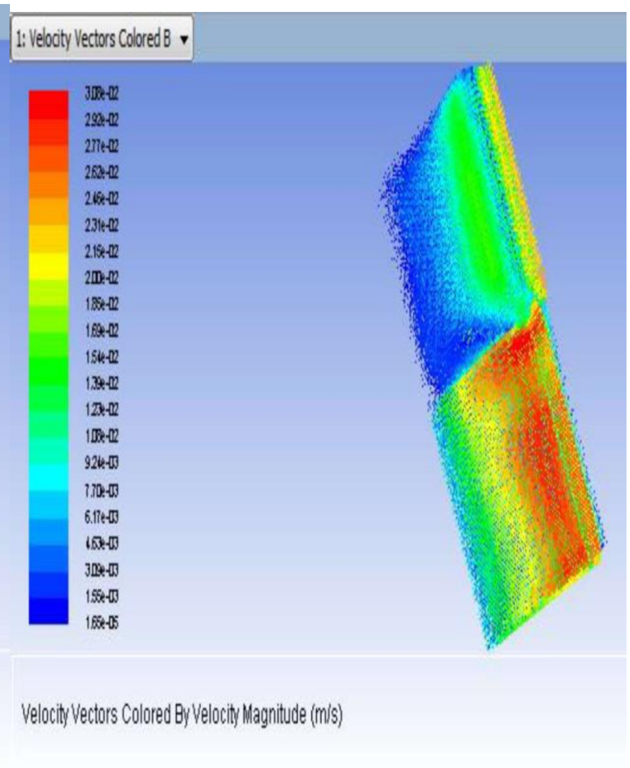
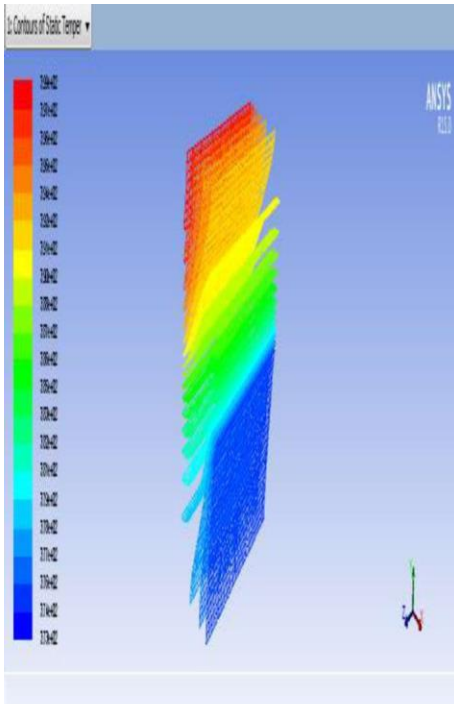


Figure: 7.10 Temperature Contours

Figure: 7.11 Velocity variations

For Geometry 2:



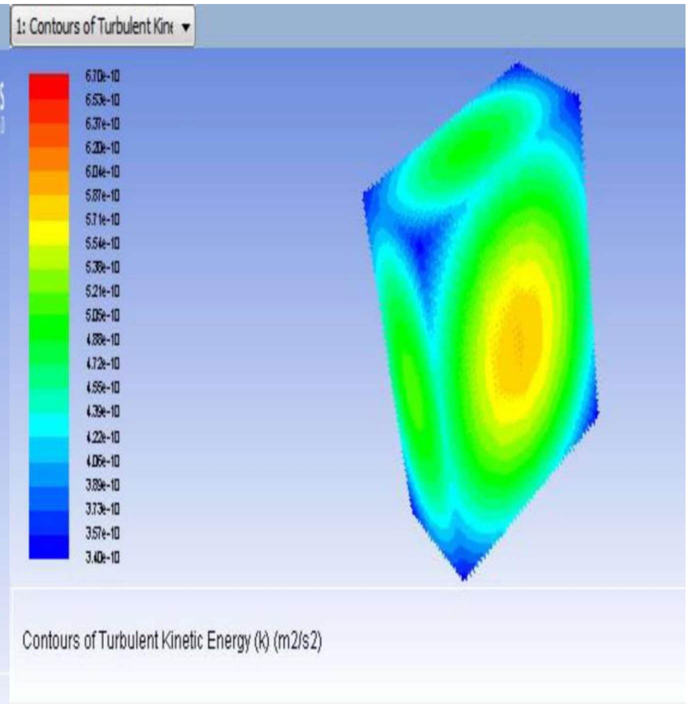


Figure: 7.12 Temperature Contours Figure: 7.13 Turbulent Kinetic Energy Contours

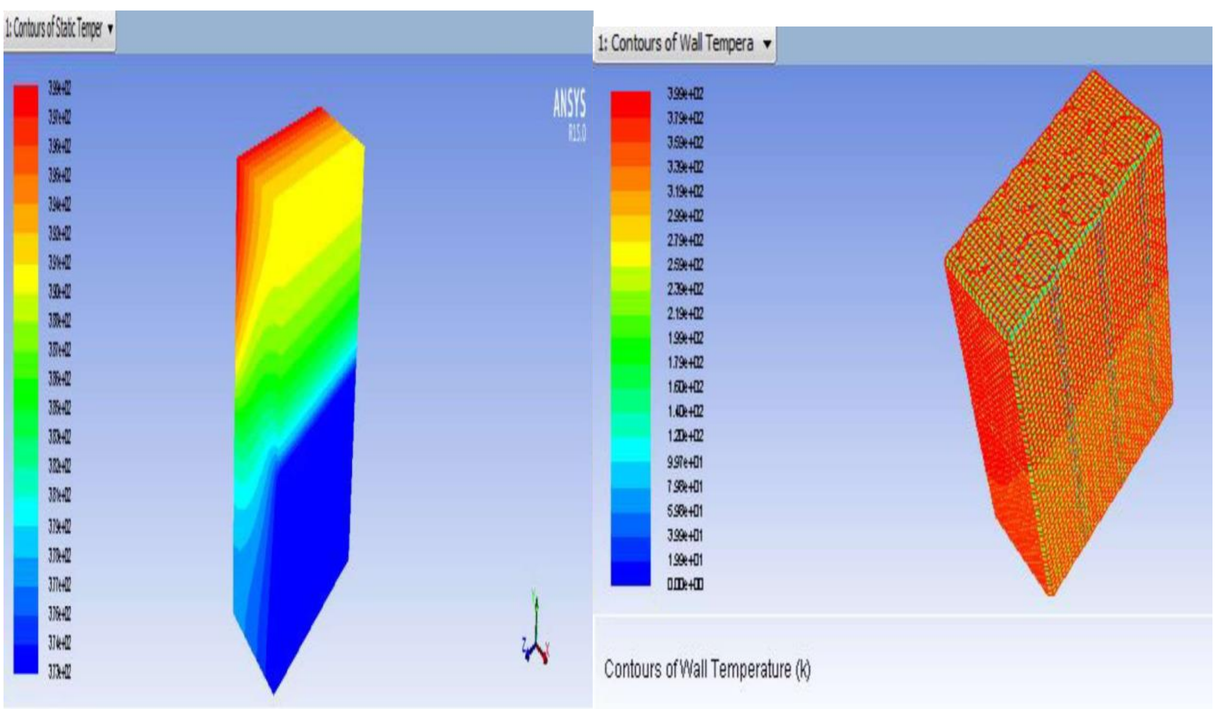
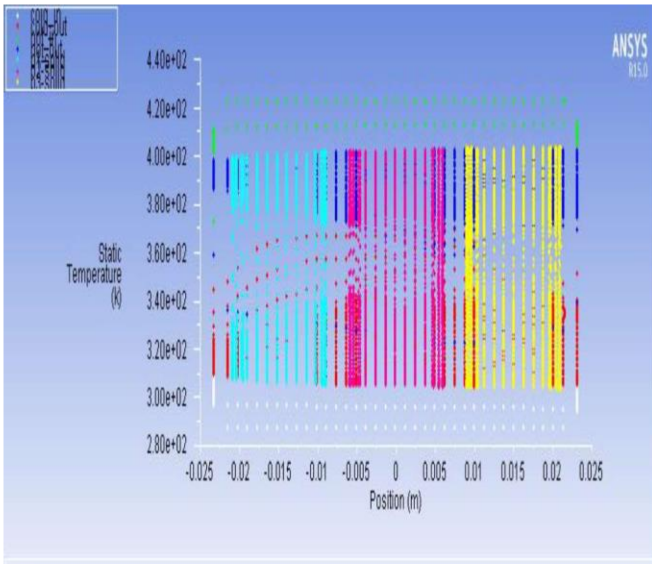


Figure: 7.14 Static wall temperature variation along the length of thermosyphon

Plots for Geometry 1 and 2 for Temperature, Turbulence versus variation along the length of pipe - :

For Geometry 1:



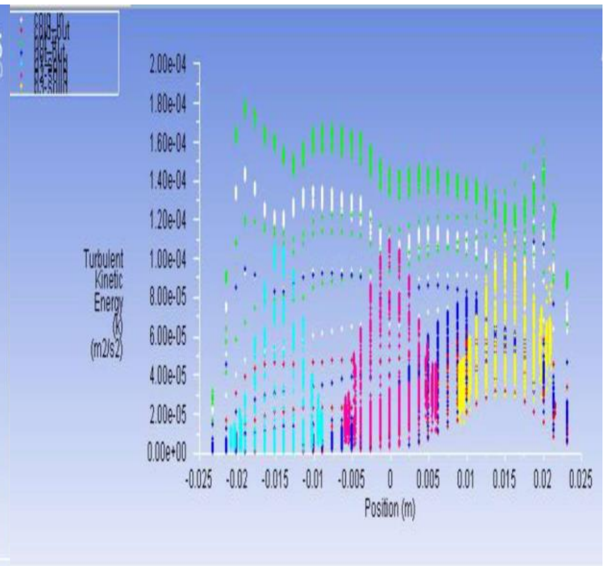
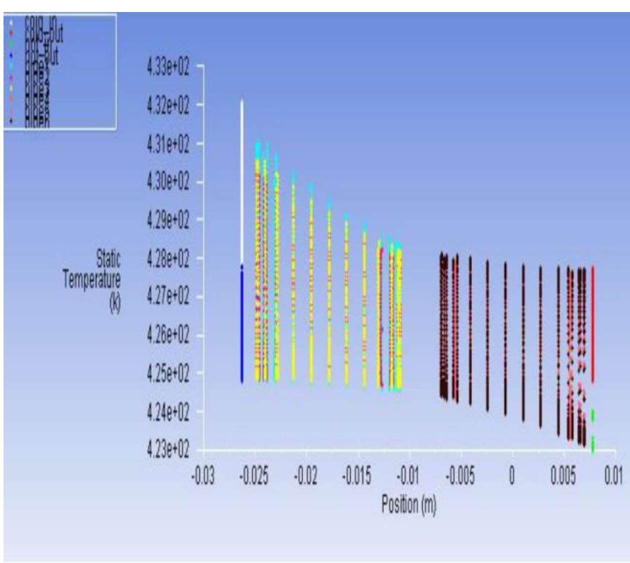


Figure:7.15 Temperature vs length

Figure: 7.16 Turbulent Kinetic Energy vs length

For Geometry 2:



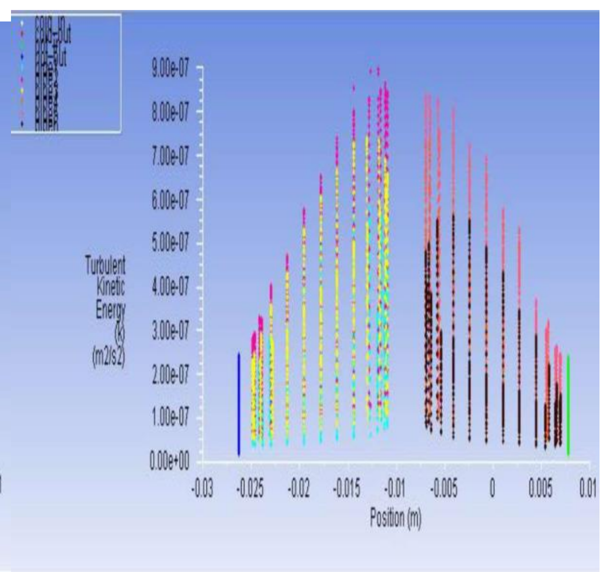


Figure: 7.17 Temperature vs length

Figure: 7.18 Turbulent Kinetic Energy vs length

Chapter- 8

Conclusions& Future Scope

In this work thermal performance air conditioning unit equipped with thermosyphon heat exchanger experimental setup was studied. Thermosyphon was charged with methanol and developed in air conditioning laboratory. Results obtained with the use of thermosyphon heat exchanger shown good improvement. Following are the outcomes of the experimental work that was carried out:

1. Thermosyphon testing at different filling ratio (30% - 60) gives us the notion that for vertical wickless heat pipe the charging should equal to half of the thermosyphon means 50% charging is optimum for 90° inclination.

2. At higher heat load (100 watt–600 watt) thermosyphon shown better thermal performance. The countercurrent limit for the thermosyphon was crossed as the heat flux was increased.

3. The overall experimental effectiveness of the thermosyphon heat exchanger was found to be 0.34 or 34% for the mean values of temperature used in the air conditioning test rig. The low value of effectiveness can be attributed to only one row of copper pipes with high to pitch to diameter ratio and no use of fins in the heat exchanger.

4. Coefficient of performance (COP) for the vapour compression system is found to be improved (4.43 with Thermosyphon) by precooling of air using thermosyphon heat exchanger. This improvement was found as the value of air face velocity varied between (1 m/sec –6 m/sec). Thus it was observed that higher value of the effectiveness of the heat exchanger can lead to greater improvement in the C.O.P. of the system.

5. System worked without any abnormality with deployment of heat exchanger

So from above comparison we can see that heat pipe heat exchanger can serve as a great alternative in order to improve the COP at no extra cost. However it is suggested that to use to perform more research.

By using the numerical simulation betterwayscould be suggested to improve the performance of a thermosyphon heat exchanger. Further studies on the velocity profile and temperature profile of a thermosyphon was made and it can be further used for the modeling a wick less heat pipe using a software package such as CFD.Higher heat transfer rate was achieved at higher inlet temperatures and low air face velocity. Effectiveness was found to be proportional to the air inlet temperature and inversely proportional to the inlet air face velocity.K- Epsilon gives the fast convergence over the k-omega for 2nd Geometry, as no. elements was increased time for meshing increased for both the geometries.

8.2 Future Scope

The results in this work shows that thermosyphon heat exchanger have huge potential to enhance thermal performance air conditioning units. But due to operating condition limits (condenser above evaporator) heat pipe gains more attention. Moreover, challenges with the use heat pipe is compatibility between wick, container and the working fluid. More work needs to carry out on the thermal performance improvement of the heat pipes.As in this work was also seen thermosyphon or heat pipe greatly effects the heat transport between the systems. Experimental needs to be carried on the heat transport limits for the heat pipes hamper the performance of the heat pipe. Working fluids used in the container can be studied in deep with various mixtures and materials.

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Table 1 Heat input = 145 Watt (Coolant Flow Rate 5.5 kg/hr.)						
F. R.	Length of Thermosyphon (x) in cm / Temperature (^{o}C)	5	17.5	41.2	62.3	84
30%	Temperature (^{o}C)	34.7	34.3	32.9	26.4	25.3
40%	Temperature (^{o})	59.1	59.9	58.4	27.6	26.9
50%	Temperature (^{o})	62.6	63.5	61.9	28.9	28.2
60%	Temperature (^{o})	64.2	65.1	63.9	29.7	28.5

Table 2 Heat input = 380 Watt (Coolant Fow Rate 5.5 kg/hr.)						
F. R.	Length of Thermosyphon (x) in cm / Temperature (^{o}C)	5	17.5	41.2	62.3	84
30%	Temperature (^{o}C)	53.4	54.9	48.8	32.4	33.6
40%	Temperature (^{o})	57.1	58.4	52.6	44.7	45.8
50%	Temperature (^{o})	58.7	58.4	53.9	45.4	43.8
60%	Temperature (^{o})	63.7	64.2	62.1	46.2	47.1

Table 3 Heat input = 530 Watt (Coolant Fow Rate 5.5 kg/hr.)

F. R.	Length of Thermosyphon (x) in cm / Temperature (^{o}C)	5	17.5	41.2	62.3	84
30%	Temperature (^{o})	80	81	75.1	48.1	49.2
40%	Temperature (^{o})	84.4	83.1	76.2	51.1	51.2
50%	Temperature (^{o})	90.1	89.6	80.2	54.2	52.2
60%	Temperature (^{o})	89.2	87.2	78.2	55.1	53.2

F. R.	Heat input (Watts)	Length of Thermos-yphon (x) in cm / Temperature (^{o}C)	5	17.5	41.2	62.3	84
30%	145	Temperature (^{o}C)	34.7	34.3	32.9	26.4	25.3
	380	Temperature (^{o}C)	53.4	54.9	48.8	32.4	33.6
	530	Temperature (^{o}C)	85.4	86.1	80.2	52.3	53.1
40%	145	Temperature (^{o}C)	59.1	59.9	58.4	27.6	26.9
	380	Temperature (^{o}C)	57.1	58.4	52.6	44.7	45.8
	530	Temperature (^{o}C)	84.4	58.4	52.6	44.7	45.8
50%	145	Temperature (^{o}C)	62.6	63.5	61.9	28.9	28.2
	380	Temperature (^{o}C)	58.7	58.4	53.9	45.4	43.8
	530	Temperature (^{o}C)	58.7	58.4	53.9	45.4	43.8
60%	145	Temperature (^{o}C)	64.2	65.1	63.9	29.7	28.5
	380	Temperature (^{o}C)	63.7	64.2	62.1	46.2	47.1
	530	Temperature (^{o}C)	63.7	64.2	62.1	46.2	47.1

Heat Flux (Kw)	Velocity (m/sec) Effectiveness / Effectiveness	2.1	3.2	4.3	5.4
0.641	Effectiveness	0.21	0.19	0.14	0.22
0.782	Effectiveness	0.27	0.23	0.18	0.21
0.834	Effectiveness	0.29	0.25	0.16	0.18
0.943	Effectiveness	0.3	0.28	0.17	0.19

Velocity (m/sec)	Temperature (℃) Effectiveness	20	25	30	35
2.1	Effectiveness	0.21	0.25	0.27	0.29
3.2	Effectiveness	0.27	0.28	0.29	0.31
4.3	Effectiveness	0.35	0.37	0.39	0.42
5.4	Effectiveness	0.42	0.46	0.48	0.49
6.1	Effectiveness	0.49	0.53	0.55	0.54

T1	T2	T3	T4	T5	T6	P1	P2	V	Q	RH1	RH2
°C	°C	°C	°C	°C	°C	bar	bar	m/s	kW	%	%
29.1	60.3	36.2	17.7	26.5	24.8	0.22	8.31	14.82	0.4	41	48.9
28.3	64.5	36.5	18.4	25.5	23.8	0.17	8.28	10.73	0.39	42.3	50.6
28	67.3	36.5	17.3	25.1	23.3	0.28	8.35	11.03	0.42	43.4	51.8
27.9	68.6	36.8	17.4	24.9	23.2	0.23	8.4	10.73	0.41	43.8	52.2
27.8	72.3	37	17	24.8	23.1	0.31	8.45	10.86	0.43	43.7	52
27.7	74.6	36.8	17.2	24.6	23	0.26	8.43	10.93	0.42	43.8	52.2
27.6	75.4	37	16.9	24.5	22.8	0.31	8.48	10.98	0.43	44.3	52.8
27.7	76.6	37.3	17	24.5	22.8	0.27	8.48	10.83	0.42	44.3	52.3
27.6	78.1	37.1	17.4	24.5	22.9	0.27	8.43	10.84	0.42	44.5	52.7
27.6	79.7	37.3	16.9	24.5	22.8	0.34	8.53	10.76	0.43	44.1	52.3

AN INVESTIGATION INTO THE PERMORMANCE OF A THEROMSYPHON HEAT EXCHANGER

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