STUDY AND EVALUATION OF Cu-H_2O BASED HEAT PIPE UNDER FORCED CONVECTION

A Thesis
Submitted in partial fulfillment of
requirements for the degree of

Master of Engineering
in
Thermal Engineering

by

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July, 2019

CERTIFICATE

I hereby declare that the dissertation entitled "**STUDY AND EVALUATION OF Cu-H_2O BASED HEAT PIPE UNDER FORCED CONVECTION**" is an authentic record of my work carried out as requirements for the award of the degree of Master of Engineering in Thermal Engineering at Thapar Institute of Engineering and Technology, Patiala under the supervision of **Mr. Sumeet Sharma** (Associate Professor, Mechanical Engineering Department) and **Dr. D. Gangacharyulu** (Professor, Department of Chemical Engineering). No part of the matter embodied in this report has been submitted to any other university or institute for the award of any degree.

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

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DECLARATION

I, **Chetan Sharma,** hereby declare that the thesis, entitled **"STUDY AND EVALUATION OF Cu-H_2O BASED HEAT PIPE UNDER FORCED CONVECTION"**, submitted to the Thapar Institute of Engineering and Technology in partial fulfilment of the requirements for the award of the **Master of Engineering** in **Thermal Engineering** submitted in Mechanical Engineering Department, Thapar Institute of Engineering and Technology, is a record of original and independent research work done by me during the period 2017–2019, under the supervision and guidance of Mr. Sumeet Sharma, Associate Professor, Mechanical Engineering Department and Dr. D. Gangacharyulu, Professor, Department of Chemical Engineering, Thapar Institute of Engineering and Technology. The work contained in this thesis has not been previously submitted to meet the requirements for a degree or diploma at this or any other higher education institution.

(Chetan Sharma)
Regd. No 801783003

Dedicated to

My parents

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(Chetan Sharma)

Abstract

Heat pipes are characterized as superconductors of heat as they are best known for excellent heat dissipation properties with minimum heat loss. Moving from conventional to advance heat pipes, a lot of research and experimentation has been done in the 21st century due to modernization and miniaturization of equipment. The operation of various engineering systems leads to generation of heat and hampering the performance of the system, thus heat pipe came to rescue by lofting the heat depending upon its capacity. Due to this reason its application varies from tiny mobile phone to giant satellites making it one of the most promising device in era of heat transfer. Gravitational and capillary forces play a very significant role in deciding the overall performance of a heat pipe. At every different tilt angle, the resultant of these two forces varies accordingly, leading to the change in performance of a heat pipe. So, there is a need to study the performance of the heat pipe at different tilt angles.

This study depicts the effects of different mass flow rates of water on the condenser side heat transfer coefficient at different tilt angles under forced convection in copper-water heat pipe. The results are expressed both via experimental as well as empirical method, which are found to be in fair agreement with maximum and minimum deviations of 18.55% and 0.04%. The present study also aims at discovering the tilt angle of the same heat pipe at which highest heat transfer coefficient of condenser section is offered. It was observed that maximum heat transfer coefficient across condenser section of heat pipe under forced convection was obtained at 25° tilt angle under air velocity of 3.36 m/s having 73.127 W/(m^{2}K) value whereas minimum value of heat transfer coefficient was observed at 0° tilt angle under air velocity of 1.35 m/s having value as 36.367 W/(m^2.K). Many different working fluids can be used inside the heat pipe. In recent times, nanoparticles are also being employed in heat pipes to improve their performance as they decrease the thermal resistance leading to high performance of heat pipe. Beyond 25° tilt the performance of heat pipe decreases due to flooding of evaporator section of heat pipe.

Keywords: Heat pipe; Thermal conductivity; Thermal resistance; Forced convection; Tilt angle; Condenser heat transfer coefficient; Capillary limit.

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Nomenclature

A_o	Area of condenser section, m^2
A_f	Area of fins
F	Frictional coefficient
G	Acceleration due to gravity, m/s^2
h	Heat transfer coefficient, $W/(m^2\text{-}K)$
K	Thermal conductivity, W/(m-K)
K	Wick structure permeability
L	Length, m
Nu	Nusselt number
R	Radius of heat pipe, m
Re	Reynolds number
t	Thickness, m
T	Temperature, ^oC
ΔT	Temperature difference, ^oC
Q	Heat input, W

Greek Symbols

Σ	Surface tension, N/m
Θ	Angle of inclined manometer, degree
M	Viscosity, $N\text{-}s/m^2$
ρ	Density, kg/m^3
h_{fg}	Latent heat of vaporization, J/kg
η	fin effectiveness
η_o	overall fin surface efficiency
ν	kinematic viscosity, m^2/s

Subscripts

a	Adiabatic
b	Bulk
c	Condenser
e	Evaporator
i	Inner
o	Outer
sat	Saturation
v	Vapour
w	Wall

Chapter 1
1.1. Introduction

Progression of heat, which is frequently unwished-for, is common contemporary, when we work on electronic equipment there is generation of heat, which needs to be dissipated. So there is need of a device which needs to lift the heat and makes the system work efficiently. Heat pipe is one of the best solution which is used widely to solve heat dissipation problem. Jacob Perkins was the first one to invent heat pipe in 1936 [1] and Richard Gaugler of General Motors patented the first capillary driven heat pipe in 1945 which was further individually rediscovered at Los Alamos National Laboratory by George Grover in 1963 [2]. Heat pipes are also known as superconductor of heat as they are best in heat dissipating abilities with least possible heat losses. Heat pipe consist of sealed container with wick lined on internal walls. Heat pipe is termed as the maximal and most cost-effective heat conductance mechanical unit in the era of heat dissipation.

Basically the heat pipe carries three major regions viz., evaporator section, adiabatic section and a condenser region encapsulated within the wick lining. Wick lining is basically acting as a natural path of arrival of condensate from condenser section back to evaporator zone by using the principle of capillary action [1]. As the pressure difference inside the heat pipe is very low which is equivalent to 10^{-6} torr, therefore it's boiling point reduces which act as a boon for the heat pipe. With a slight rise in temperature the working fluid starts boiling and thus it evaporates in the evaporator section and thus due to pressure difference it reaches the condenser end where it releases its latent heat of condensation. The wick results in the return of condensate back to evaporator section via capillary action and the cycle continues.

Heat pipe is relatable to thermosyphon in a few aspects. The crucial difference is that heat pipe consists of wick which helps in returning of condensate back to evaporator section but thermosyphon does not have it, but one more main characteristic nature of heat pipe is that it can work against the force of gravitation whereas thermosyphon fails to do so. Therefore, in case of conventional thermosyphon oriented vertically, the evaporator section should be placed at the bottom because as the working fluid evaporates and reaches the condenser section it will automatically reach to the evaporator zone under the action of gravity. Here gravity will assist the capillary forces but due to the vertical position of thermosyphon flooding of evaporator takes place which ultimately decreases its overall performance. The heat pipe container material, the working fluid and wick material are three essential components which are used in the manufacturing of heat pipe. The strength to weight ratio of container shall be maximum in order to attain the desired output.

Heat pipes are termed as superconductors of as they possess exceptional heat conduction abilities along with avoiding heat loss by maintaining uniform temperature across its container due to its compact and simple design. Heat pipes are extensively used for their compact, light and reliability aspects making it one of the most economical product in heat handling aspects. Few practical applications of heat pipe in modern industry are discussed below [3]

Die casting and injection molding: We use heat pipes in the die casting process during casting procedures to avoid temperature variation thus preventing cracks due to residual stresses in the final product, as displayed in figure 1.1.

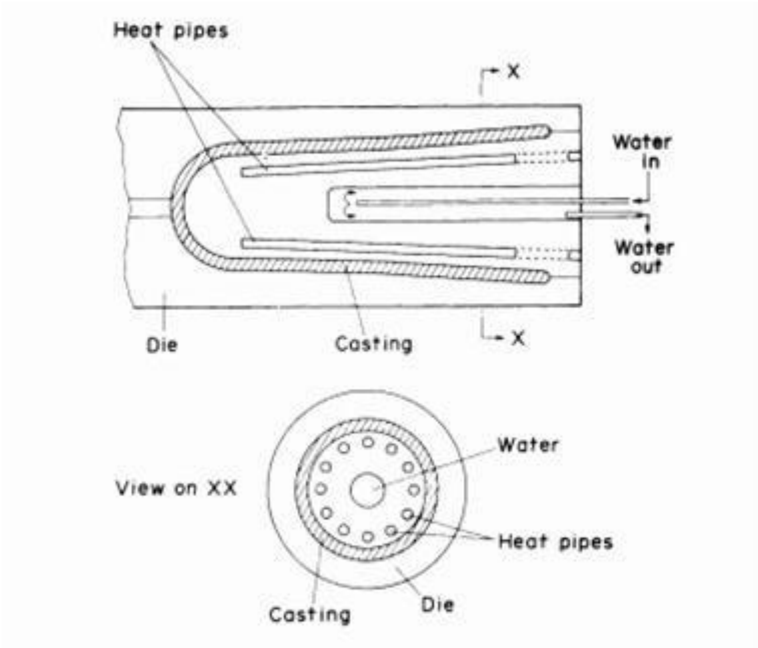

Figure 1.1 The use of heat pipe to reduce die wall temperature gradients [1]

Improving efficiency of IC-engines: The heat from exhaust gases is used by heat pipe to raise the temperature of air in inlet-manifold in IC engines, thus improving the efficiency.

De-icing of roads: In cold countries during harsh weather heat pipe is used to melt the frozen crust of ice by placing it few meters under the ground level thus helps in melting hard layer of ice as demonstrated in figure 1.2. [4]

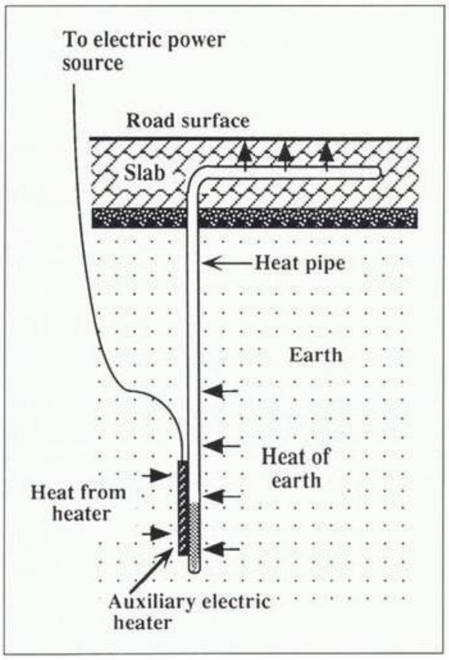

Figure 1.2. De-icing in cold countries [4]

Isothermalisation of satellites: The temperature difference across the sun facing side and opposite side of a satellite is balanced by positioning a heat pipe in Satellite interior thus preventing damage to the critical components of space craft. The below figure 1.3. shows the temperature flattening in satellites due to the presence of heat pipe.

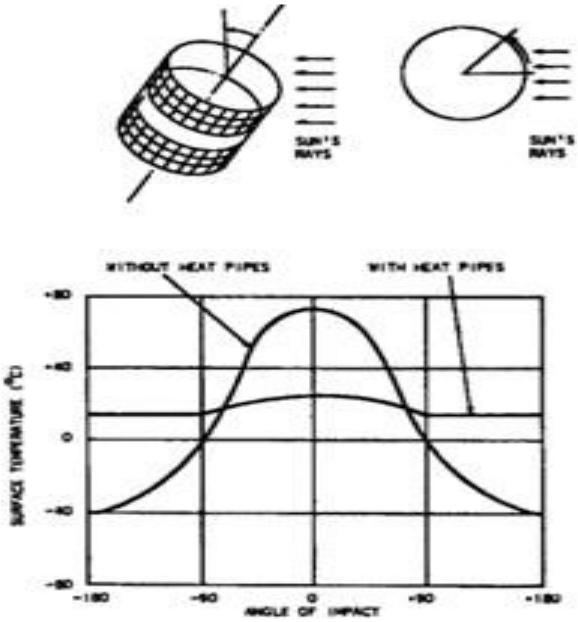

Figure 1.3. Satellites isothermalisation (Dornier Review) [1]

Management of heat in electronic components: Heat generated in the processors of laptop and super-computers is dissipated by using heat pipes as shown in figure 1.4. Nowadays it is also being used in high end mobile phones. [5]

Figure 1.4. Heat pipes used in laptops to dissipate heat from motherboard [5]

1.2. Basic Operating Sections of Heat Pipe

The heat pipe can be explained in three main parts which is shown in figure 1.5. [6]

- Evaporator section: Absorbs heat from target areas
- Adiabatic section: Prevents heat loss during working fluid transfer
- Condenser section: Dissipates heat from working fluid to ambient heat sink.

In between the container, which can be made up of different materials depending on the application, there exist a wick lined on the internal side of the wall. The heat pipe is a hollow sealed vacuum cavity, which act as a boon for heat transfer. Since the pressure inside the hollow cavity is, very less, which enables the working fluid to undergo the process of boiling with each degree change in the heat input, applied at the evaporator region. When certain tilt angle is given to the heat pipe its performance increases depending on inclination. The reason is that at a certain tilt angle both gravity forces and capillary forces assist the working fluid to return to the evaporator region quickly which increases the performance of the heat pipe. However, after certain tilt angle the performance decreases due to flooding of evaporator section.

Heat pipe working on nanofluids show better results in terms of performance because thermal resistance is reduced which results in more dissipation of heat. The problem with the use of nanofluid heat pipe after some time is agglomeration, which on a long run deteriorates its performance and efficiency.

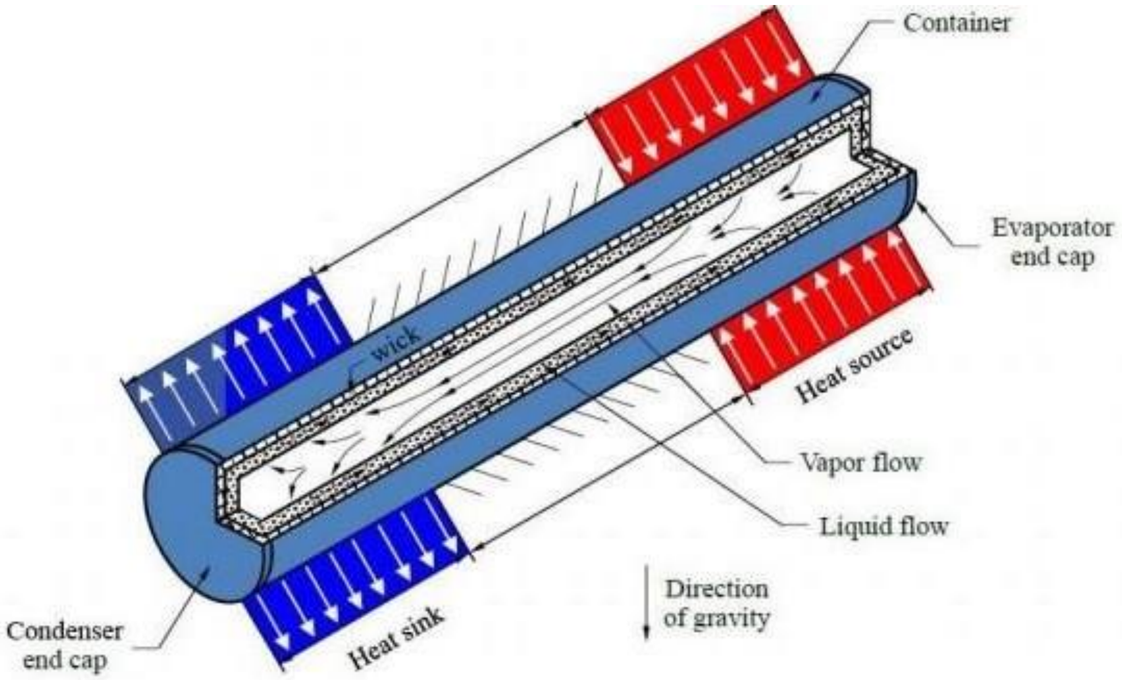

Figure 1.5. Simplified Diagram of typical Heat Pipe [6]

1.3. Construction Segments of Heat Pipes

The three elemental segment of a typical heat pipe are illustrated in Figure 1.6. A heat pipe usually has the following main constructional features or segments [4].

- The container encapsulating three major sections, viz., evaporator, adiabatic and condenser
- The working fluid inside heat pipe
- The wick

Proper selection of the above three components yields better result. Container should have good strength to weight ratio. The working fluid should have high thermal conductivity to dissipate more heat. Wicks are responsible for the capillary action, which allows the condensate to return from the condenser section to evaporator region. Sometimes capillary forces are assisted by the gravity forces by virtue of tilt angle of the heat pipe.

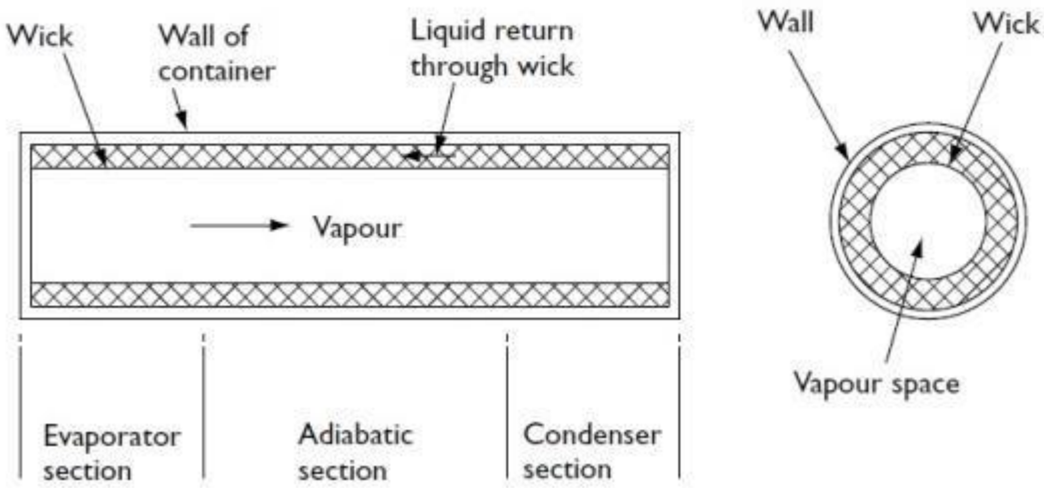

Figure 1.6. Illustration of The main sections of a typical heat pipe

Brief discussion about the ingredients of heat pipe are as follows.

1. Container

The container's primary aim is to maintain working fluid at a negative gauge pressure thus it must be vacuum sealed thus reducing the boiling point of working fluid(water). The material should also have high thermal diffusivity so as to efficiently dissipate the heat of working fluid. It should have least weight without comprising strength. In my experiment we have used copper material due to high heat diffusivity. We also used fins to raise the surface area so that efficiency of whole setup can be improved [7]. The length of the container depends on the area of application and can be varied accordingly.

The container's vital role is to separate the operating fluid from the environment. It must be leak-proof and allow the heat to dissipate from the working fluid.

Choice of container material is vital process, which depends on below mentioned factors. [1]

(i) It should be compatible with both the operating fluid and the external environment.

(ii) It should possess good strength and must have light weight.

(iii) It should have significant value of thermal conductivity.

(iv) There should be easiness in fabrication such as weldability, machinability and ductility.

(v) Porous nature would be appreciated.

(vi) Wettability.

(vii) The Structure or container should withstand the vibrations and acceleration caused during operation.

2. Working Fluid

The operating fluid acts as the transporter of heat from evaporator region to condenser region. It absorbs heat in evaporator region thus changing phase from liquid to vapor phase and releasing this heat in condenser section thus changing back to liquid. From here, it reaches back to evaporator via wick through capillary action. The sort of working fluid may rely on the temperature range of the application region. It can be ammonia, water, pentane etc. We have used water as the range of temperature in our experiments is between 22°C-70°C.

Other main key factors which are crucial while selecting the working fluids are [6]

- Significant compatibility with the wick and wall materials.
- It should have great thermal stability.
- Wick and wall material wettability is considered to be the utmost priority.
- It should possess high latent heat.
- It should have high thermal conductivity.
- Low viscosity of fluid and vapor.
- It should have high surface tension.

Table 1.1 tells about the brief idea to choose working fluid and is given below.

Table 1.1. Working fluids of heat pipes [1]

Medium	Melting point (°C)	Boiling point (°C)	Useful range (°C)
Ammonia	-78	-33	-60 to 100
Pentane	-130	28	-20 to 120
Acetone	-95	57	0 to 120
Methanol	-98	64	10 to 130
Ethanol	-112	78	0 to 130
Heptane	-90	98	0 to 150
Water	0	100	30 to 200
Toluene	-95	110	50 to 200
Mercury	-39	361	250 to 650
Lithium	179	1340	1000 to 1800
Silver	960	2212	1800 to 2300

Other point to be noted is that mercury is not used in heat pipe due to its opposite nature to capillary action. [1]

The operating fluid must not consist dissolved gases and other kind of liquids. It is known that techniques such as freezing and distillation purify the working fluid. Temperature also have a significant impact on some of the working fluids as they are receptive to operating temperature. If such conduct is traced, it is necessary to select the secure temperature band. In addition, elevated heat fluxes generate vigorous boiling action in wicks that can lead to erosion. Compatibility plays a vital role because the working fluid cannot respond to the container wall and the wick lining. The selection of the two liquids should be based on the compatibility and solubility of the gas in the working fluid in variable conductance heat pipes or VCHP.

3. Wick

The main purpose of wick is that it transports condensate back from condenser region to evaporator section via capillary action. In structure, it should be porous, which stimulates capillary action of working fluid. Different surface profiles of wick are being used to maximize capillary effect. A few examples are shown in figure 1.7.

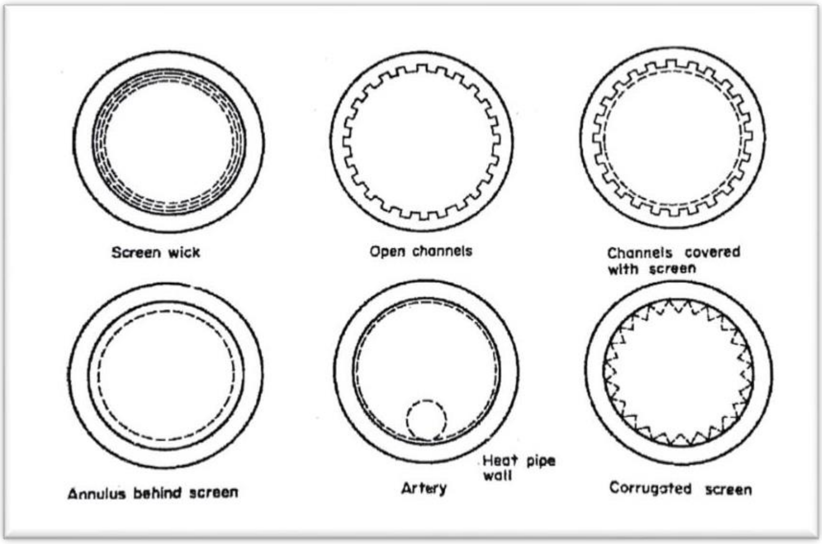

Figure 1.7. Various profiles of wicks [1]

1.4. Water and Steel Affinity

Water is usually an optimal heat pipe working fluid because it has a large value of latent heat. Moreover, it comparatively requires low inventory and low cost because of the ease of availability. It is consistent with a broad range of container materials; copper is the most prominent.

Steel corrosion in the lack of oxygen is nearly negligible. Therefore, within a closed system of steel heat pipe having working fluid as water, the corrosion occurs until all the container free oxygen is absorbed. When there is a deficiency of oxygen, low alloy steels in neutral alternatives create a hydrated magnetite layer by decomposing the ferrous hydroxide. Further chemical reaction of iron with water, occurs.

$$3\ FeOH_2 \rightarrow Fe_3O_4 + 2\ H_2O + H_2 \qquad (1)$$

$FeOH_2$ transformation leads to the Fe_3O_4 protective layer, particularly with mild carbon steels. This Fe_3O_4 layer functions as a good corrosion resistant to boiler steels in power station boilers during high water temperature. [8]

1.5. Assembly of Heat Pipe:

Heat pipe assembly is of utmost importance, as it will lead to proper testing and deduction of results [8]. Following are the steps, which are followed while assembly operation is carried out:

(i) Container material selection is the utmost priority.

(ii) Wick material is then selected.

(iii) Fabrication of wick along with end caps.

(iv) Container, wick and end caps cleaning.

(v) The metal components are outgassed.

(vi) Wick's insertion.

(vii) Welding of end cap.

(viii) Leak inspection of welds.

(ix) Selection of desired operating fluids.

(x) Purification of the operating fluid

(xi) Degassing of the operating fluid.

(xii) The heat pipe is evacuated and filled.

(xiii) Sealing of the heat pipe.

1.6. Heat Pipe Life Test Factors

If heat pipe is to be used in some field of work, then it should pass some of the below mentioned test factors which in long run enable the heat pipe to work smoothly and efficiently and are shown in figure 1.8. Its reliability will increase and thus can be used for desired time of application of the setup. Purity, corrosion, external environment, and vibration factors mostly decide the life of heat pipe. Heat pipes are economical which makes it most extensively used in the area of heat transfer.

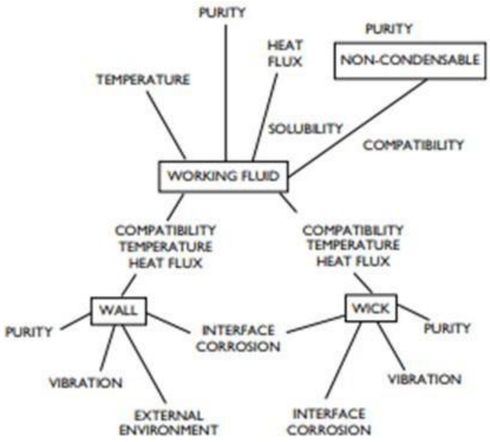

Figure 1.8. Heat pipe life test factors [8]

1.7. Working of a heat pipe

Heat pipe operation is such that, at heat source on evaporator section, Heat pipe working fluid evaporates and it travels through sealed internal passage heading to the condenser section due to pressure difference. At condenser end, it releases its latent heat of condensation and then travels back to evaporator zone through wick with the help of capillary action. This cycle continues as explained. Further, we have annular fins, which improve rate of heat dissipation thus further improving efficiency. [6]

1.8. Types of Heat Pipes

There are many kinds of heat pipes based on different purposes and few are discussed here.

Variable Conductance Heat Pipe: V.C.H.P. comprised of a pool of non-condensable inert gas which is generally attached to the condenser. V.C.H.P. operates in wider ranges of heat fluxes and temperature gradients.

The variable conductance heat pipe (VCHP), often called a gas-controlled or gas-charged heat pipe in a particular version, which holds a distinctive feature that distinguish it from rest of the heat pipe. It has a distinctive ability to keep a device mounted on the evaporator segment, which is nearly at a constant temperature. Its applications range from thermal control of parts and systems on satellites to accurate temperature calibration tasks and standard electronic temperature control systems. Figure 1.9. shows the cold reservoir variable conductance heat pipe.

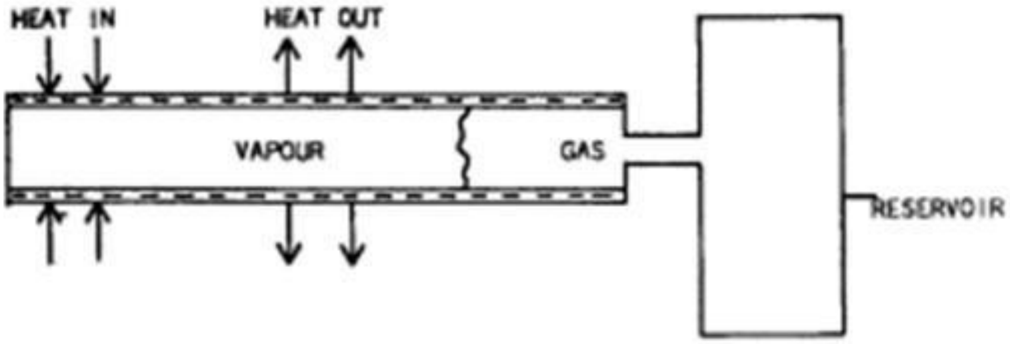

Figure 1.9. Cold reservoir variable conductance heat pipe [1]

Thermosyphons: Thermosyphon thermal dissipation is almost comparable to heat pipe dissipation, but it lacks the wick structure. Because of the above, the condenser section is placed above the evaporator region in order to return the condensate with effect of gravity which is clearly illustrated in figure 1.10.

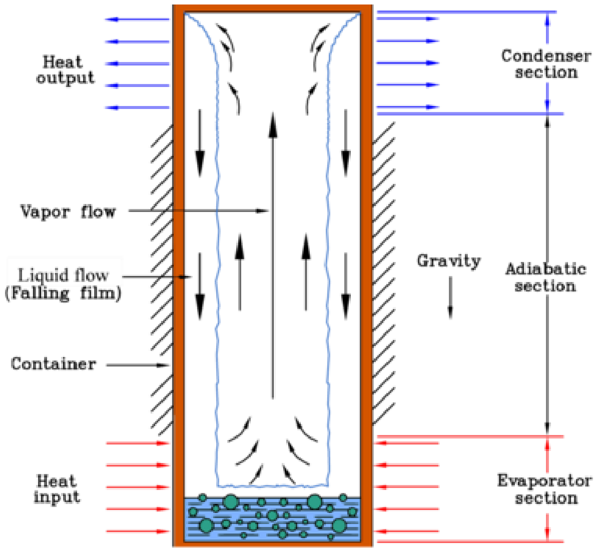

Figure 1.10. Thermosyphon [6]

Micro Heat Pipe (M.H.P): M.H.P. utilizes sharp corners intrinsic in its non-circular layout instead of wick because it offers capillary pressure to the operating fluid. Working fluid circulation mainly depends upon sharpness along with amount of sharp angled corners. [9]

Loop Heat Pipe: It is also known as capillary pump loop (C.P.L.). It comprises of evaporator, a compensation chamber, a condenser, vapor, and fluid transfer lines. The compensation chamber stores extra liquid which act as an alert against any kind of evaporator dry out. Figure 1.11 depicts the operating principle of loop heat pipe. [10]

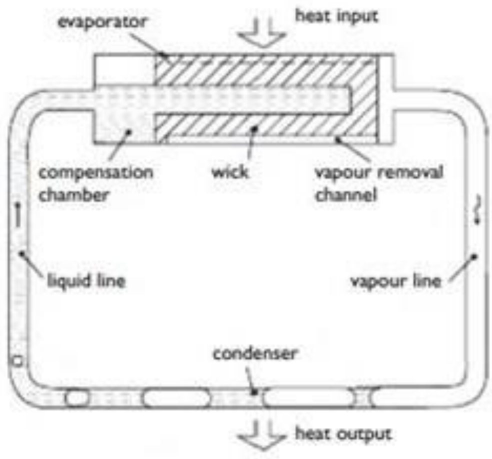

Figure 1.11. Operating principle of loop heat pipe [10]

Rotating Heat Pipes: The rotating heat pipes are two-phase thermosyphon where condensate is returned back to the evaporator section by the aid of centrifugal force. The rotating heat pipe is made up of closed hollow shaft, frequently having a minor inner taper across its axial length and encapsulating a set quantity of working fluid. The rotating heat pipe, similar to the conventional capillary driven heat pipe, is comprised of three regions, namely the evaporator zone, the adiabatic zone and finally the condenser zone. The rotation of heat pipe about the axis results in a centrifugal acceleration. The associated force will force the condensed working fluid or condensate to flow back to the evaporator portion along the wall. [11]

In the rotating heat pipes, high condensing coefficient is maintained due to the efficient removal of the condensate from the cooled liquid surface through centrifugal action. An illustration of rotating heat pipe is depicted in figure 1.12.

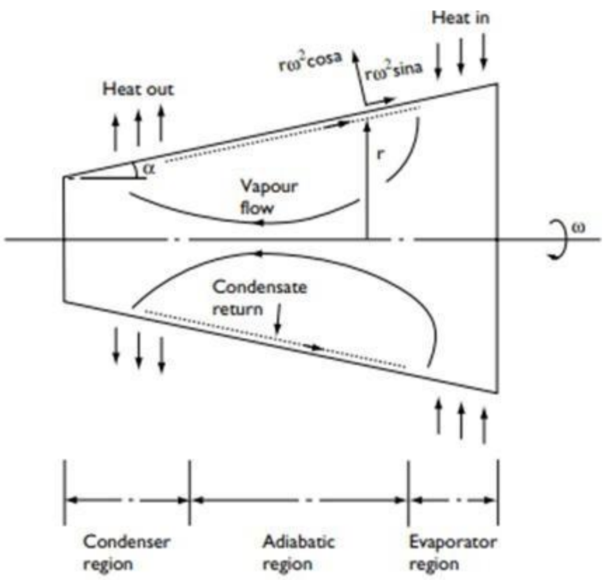

Figure 1.12. Rotating heat pipe [11]

Oscillating Heat Pipe: Pulsating, or oscillating, heat pipes comprises of a capillary diameter tube, evacuated and then partially filled with the operating fluid. When one end of the capillary tube, the evaporator end, is heated, the working fluid evaporates and raises the vapor pressure causing the bubbles to develop in the evaporator region. This pushes the liquid to the end of the low temperature, which is termed as the condenser end. The cooling of the condenser section outcomes in reduced vapor pressure and bubble condensation in the requisite heat pipe area. Both the development and the collapse of bubbles in the evaporator region and condenser region, leads to oscillating motion within the tube. [12]

Closed loop pulsating heat pipes or (CLPHPs) performs proficient than open loop devices due to the fluid circulation which is superposed upon the oscillations inside the loop. It was investigated that further performance in improvement of CLPHP can be increased by using check valves within the loop; however, owing to the inherently tiny nature of the device, the installation of required valves is both hard and expensive. Therefore, a closed loop device denied of a check valve is the utmost feasible implementation of the pulsating heat pipe. Figure 1.13 shows the simplified representation of pulsating heat pipe [13].

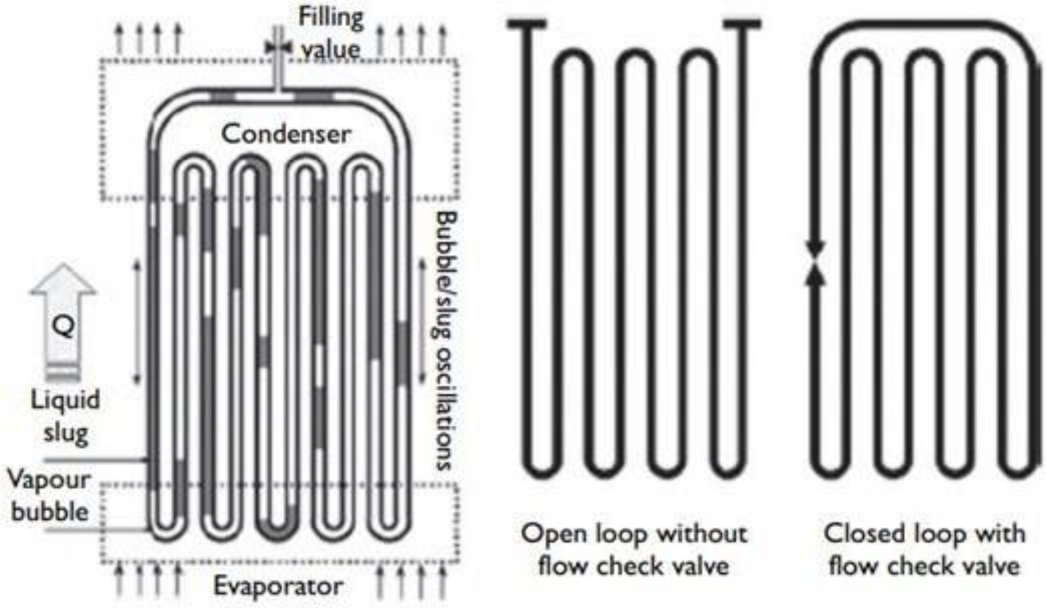

Figure 1.13: Schematic representation of pulsating heat pipe

1.9. Limitations

In order to work effectively heat pipe has to operate inside the dome region as illustrated in Figure 1.14. So it must be taken care of while designing a heat pipe.

Various limitation to heat pipe are discussed below [6]

Vapor Pressure or Viscous Limit

The minimum pressure at the condenser end of the pipe can be very small at low temperature range of operation of the working fluid, particularly when starting the heat pipe. The vapor pressure drop between the extreme end of the evaporator and the end of the condenser, represents a restriction in operation

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"

Entrainment Limit

The vapor velocity rises with temperature and can be considerably large to generate shear force which affects the liquid return flow from the end of the condenser to the end of the evaporator, causing entrainment of the liquid by the vapor.

Boiling Limit

The temperature drop along the wick structure in the evaporator zone escalates with the evaporator heat flux. An important point is reached, when the difference in temperature exceeds the degree of superheat sustainable in relation to nucleate boiling conditions. Boiling inside the wick structure interferes with the circulation of liquid. This eventually leads to dry out, which can trigger evaporator containment burning in the event of continuous heat flux heating.

Capillary Limit

The capillary limit is the maximum pumping pressure the heat pipe can hold and still return the condensate through capillary forces. The driving pressure for liquid circulation within the heat pipe is given by the capillary force established within the wick structure.

In order to function properly the parameters of heat pipe should be under the dome region as shown in figure 1.14.

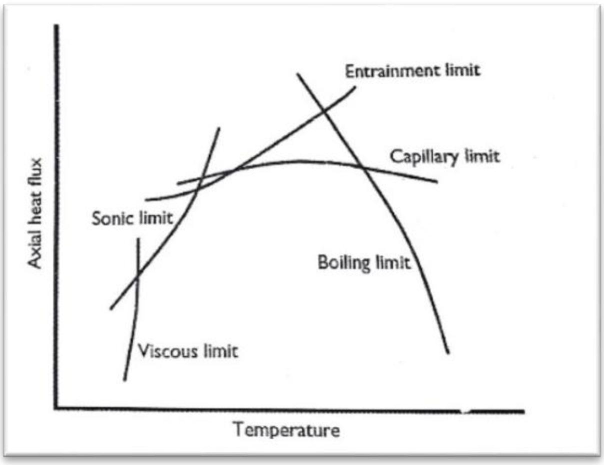

Figure 1.14. Limitations of heat pipe [1]

The above figure 1.14 depicts the condition in which heat pipe is to work effectively otherwise the working fluid may dry out. In the graph, a region is spotted which is under the dome created by the above said limitations and is considered the best region for the efficient working of heat pipe. In this region, the heat pipe will give its maximum performance thus leading to higher efficiency of the device, which ultimately yields better heat dissipation.

1.10. Selection of Fins

To boost heat transmission, extended surface or fins are used. Due to increase surface area of cross section and decrease in thermal resistance where convection process occurs. The fin material should possess high thermal conductivity and must minimize temperature variation from base to top. There are various types of fin including Pin fin, Straight fin, annular fin etc. Aluminum material is widely used for fabrication of fins due to its low weight and cost. Fin length is important criteria in heat dissipation process. It was noticed that the length of fin is directly proportional to the heat transfer rate. However, temperature drop across the fin follow exponentially path so that is the reason it reaches the surrounding temperature at some length. Beyond this length, it does not contribute in any heat transfer. Therefore, design of extra-long fin is not justified as it leads to wastage of material, increase in size and excessive weight and ultimately increases the cost criteria [14]. Figure 1.15 depicts the schematic diagram of proper length of fin used for heat dissipation.

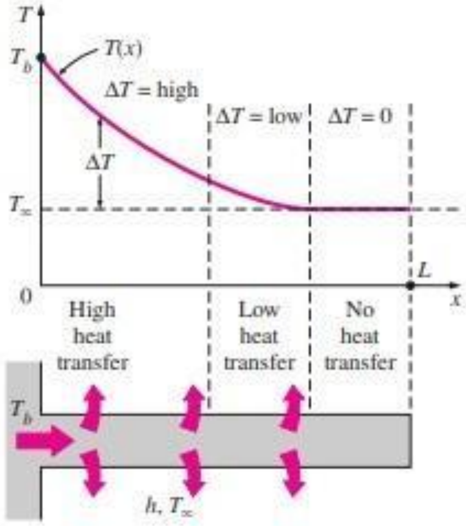

Fig.1.15 Proper length of fin showing heat dissipation [14]

1.11. Fabrication of Fins

In this study, cylindrical annular fins were used which are made up of aluminum alloys 6063. Because of its high thermal conductivity, it makes it one of the most widely used materials for manufacturing of fins and easily available [15]. It provides good resistance to corrosion and have high surface quality. A6063 is an alloy that is made up of two elements, magnesium and silicon. It also possesses good mechanical properties and high heat dissipation property. Chemical composition of A6063 is given in the Table 1.2. The 3D view of cylinder with annular fin as shown in Figure 1.17

Table 1.2. Chemical composition of fin material (A6063)

Material	Si	Fe	Cu	Mn	Mg	Cr	Ti	Other
Max%	0.6	0.35	0.10	0.10	0.9	0.10	0.10	0.15
Min%	0.2	0.35	0.10	0.10	0.45	0.10	0.10	0.05

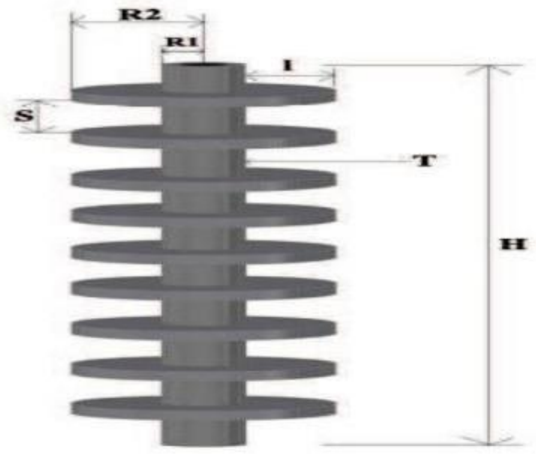

Fig1.16: View of cylinder with annular fins (REF)

1.12. Specifications of Fins

Various parameters on which heat transfer depends are length, spacing between the fins, number of fins and fin diameter. The optimum fin spacing for maximum heat dissipation ranges between 5 and 6 mm [15]. Thickness of fins is considered important parameter and it should be kept as small as possible to avoid conduction but in this present study, fins of uniform thickness of 1 mm were used because beyond this level it was very difficult to manufacture due to mechanical constraints. Keeping the above said consideration in view, the condenser section of heat pipe was provided with 9 annular aluminum fins of 32 mm diameter and at a pitch of 6 mm. Specification of annular fins used in experimentation is illustrated in Table 1.3

Table 1.3: Annular fins specification along with their notation

S. No.	Notation	Parameter	Value
1	R1	Fin base radius	6 mm
2	R2	Fin tip radius	16 mm
3	L	Length of fins	10 mm
4	S	Spacing between two fins	6 mm
5	H	Height of condenser length	65 mm
6	N	No. of fins	9 mm

Closure

As far as heat pipe is concerned its applications in tiny mobile phones to giant satellites makes it one of the most used heat dissipating devices in the era of heat transfer. Further advancement from the conventional to advanced heat pipes such as closed loop heat pipes and oscillating heat pipes etc. marks the revolutionary spark in heat dissipating area.

Chapter 2
Literature Review

The literature review is carried out to explore the latest developments in heat pipe technology and to choose an area to carry out the research work for the dissertation work. The outcomes are listed below.

Nuntaphan et al. [16] found that the enhancement of fin efficiency is done by replacing solid wire fin with oscillating heat pipe under forced convection. The device was tested in a wind tunnel, which acts as a duct, by exchanging heat between hot water flowing through the tube and the air stream flowing across the extended surface. Heat transfer between tube and solid wire fin was by conduction only but with oscillating heat pipe, it was due to conduction and convection. Thus, heat transfer increased. Findings were that effectiveness of heat exchanger increases by 10 % and fin efficiency was higher than 5 %.

Noie et al. [17] examined the impact of tilt angle and filling ratio on thermal performance of the two phase closed thermosyphon. The copper thermosyphon having distilled water as working fluid has outer diameter of 16 mm, inner diameter 14.5 mm and also length 1000 mm was taken into consideration. Filling ratio is described as ratio of volume of working fluid to the volume of evaporator section. Filling ratio of 15 %, 22 %, 30 % was taken into consideration. They achieved that condensation heat transfer coefficient and heat transfer rate increase as filling ratio increases. The maximum heat transfer coefficient of condensation end for the filling ratio = 22% and the filling ratio = 30% occurs at 30 ° inclination and at 45 ° for the filling ratio = 15%.

Teng et al. [18] examined the thermal efficiency of heat pipe with alumina nanofluid and considered alumina nanofluid, which was generated by direct synthesis method having three different concentrations 0.5%, 1% and 3% by weight. The experiment was performed on heat pipe, which was a straight copper tube having inner diameter 8 mm and corresponding length 600 mm. The optimum thermal efficiency value increased by 16.8% relative to the base fluid at a concentration of 1 weight %.

Idrus et al. [19] reported that under his investigation, a copper water heat pipe of length 300 mm and outer diameter 10 mm was taken into consideration with K-type thermocouple attached over the surface of heat pipe. At distinct heat inputs and tilt angle, performance of heat pipe was investigated and analyzed. The heat pipe having diameter 10 mm performs with good thermal performance at heat input of 70 W – 80 W and at tilt angle ranging between 30° to 60°.

Kang et al. [20] examined thermal performance of heat pipe with silver nanofluid having 35 nm diameter. The length of heat pipe is 200 mm and with outer diameter of 6 mm. Nanoparticles concentration ranges from 1 mg/l to 100 mg/l. A heat sink was connected to condenser region of heat pipe. Condenser section was cooled by water supplied. When the concentration and particle size of silver nanoparticle was increased, the thermal resistance decreases. With larger silver nano particles dispersed in working fluid, the temperature rise in the heat pipe wall was lower than that for heat pipes filled with pure water under distinct heat loads.

Plawsky et al. [21] reported heat pipe comprises of sealed container with wick present on the inner walls. Heat pipe is made up of three sections that is evaporator region, adiabatic region and condenser region. As the heat source is applied on the evaporator section, the working fluid evaporates and travels to the condenser region where it releases its latent heat and condensate travels back with the help of capillary action through wick lined on the internal wall of container. Wicks assist in returning the condensate back to heated end. Effective wick structure is vital key to better performance of heat pipe. Among various technologies available in today's era, heat pipes are characterized as best efficient heat dissipating device. It can easily transfer heat through long distances. It is primarily used for cooling electronic components as they can remove heat from restricted quantities to the atmosphere.

Jose et al. [22] stated that Loop heat pipes are used extensively because of their high efficiency and compact size. Nanofluid has proved to be of great importance for thermal performance of the heat pipe. Loop heat pipe operates on multiphase fluid flow cycle. Loop heat pipe comprises of following advantages: 1. High heat flux capability 2. Transfer of energy to great distances 3. Ability to operate over wide range of environments 4. Entrainment possibility is very low. Loop heat pipes are used in thermoelectric generators, LED street lights, for heat dissipation, solar water heaters, space application. Thermal management of electronic components, cryogenics, photovoltaic, cooling of electronic components. It was observed that thermal resistance of loop heat pipe decreases with the increase in nanoparticle concentration and moreover thermal efficiency of the loop heat pipe can be improvised by making use of nanofluid.

Huminic et al. [23] carried out a thermosyphon heat pipe, which was taken into consideration, and comparison of heat dissipation rate of thermosyphon was made with nanofluid and DI-water. The iron oxide nano particle were achieved by laser pyrolysis technique having concentration of 0 %, 2 % and 5.3 %. Thermosyphon are passive heat transfer devices having high value of thermal conductivity. The thermosyphon heat pipe is divided into three zones, which are evaporator (located near the heat source), condenser section (located at the heat sink), adiabatic (which is between the above two). Thermosyphon is similar to the heat pipe, but they are termed as wickless heat pipe. Heat input at the evaporator section vaporizes the operating fluid and because of pressure difference vapors travels through sealed passage to the condenser region where it condenses and releases latent heat of condensation. In thermosyphon, evaporator is placed vertically below the condenser, thus gravity will assure the condensate return to the evaporator. Comparing with the water thermosyphon heat pipe, there is substantial increase of heat transfer rate observed in case of thermosyphon heat pipe filled with various concentration level of iron oxide nanoparticle. The existence of 2 % iron oxide nanoparticle increases the heat transfer rate by 19 % and 5.3 % iron oxide nanoparticles increases the heat dissipation rate by 22.2 %. It was also noted that the growing concentration of nanoparticles reduces the thermal resistance of the thermosyphon heat pipe and thus improves efficiency.

Han et al. [24] reported that pulsating heat pipes are considered one of the most widely used heat transfer devices due to its wide advantages such as simple structure, low cost, excellent heat dissipation capability, and high flexibility. Pulsating heat pipes (P.H.P.) are also coined as oscillating heat pipe and it was proposed by Akachi in 1990. It consists of a lengthy capillary tube, bent in many turns. Its manufacturing is quite easy, as it comprises of no wick structure. Low cost of P.H.P. is because of its small diameter as it results in cost saving. Geometrical parameters that influence P.H.P. are inner diameter, shape of cross section and channel configuration and number of turns. P.H.P. is found to have wide variety of application; may it be solar water heater. Arab et al developed a new solar water heater, which was combined with two P.H.P.'s. For each P.H.P., the condenser section was put in water tank and evaporator zone was placed in collector. The adiabatic section's length varied with position. It was established that the highest efficiency was reported as 53.79 %. Other applications include electronic cooling, heat recovery devices, aerospace isothermalisation and electron cooling.

Chan et al. [25] studied that surface tension is considered the important property for selection of working fluid. The greater the surface tension, the greater the capillary effect which in the wick's presence must be chemically stable. According to temperature ranges, heat pipes are categorized into four categories

1. The temperature of more than 700 K for working fluids such as liquid metal.

2. 550-700 K Long organic carbon chain liquids such as naphthalene and biphenyl, etc.
3. 200-550 K organic fluid, including methanol, ethanol and acetone, water, ammonia and brief carbon chain.
4. 1-200 K noble gases including oxygen, nitrogen

Hudakorn et al. [26] examined the consequence of orientation of oscillating heat pipe on its overall performance. Heat pipe which was used in the experiment was constructed with Pyrex glass tube possessing inner width 1mm and evaporator length 50 mm respectively. The numbers of turns in the heat pipe are 10. All the three regions of heat pipe are of equal length. The operating fluid used in the heat pipe was R123 with FR 50 %. It was derived from the outcome that at horizontal position of the heat pipe the dry out of evaporator follows due to insufficient condensed liquid film. When the inclination angle was raised from horizontal to vertical position, performance limit arises because of flooding of evaporator region. The author then carried another set of experiment known as quantitative study in which oscillating heat pipes were built of copper tubes rather than Pyrex glass having inner diameters comprising of 0.66 mm, 1.06 mm and 2.03 mm. The number of turns were kept same but evaporator section length was made different i.e.50 mm, 100 mm and 150 mm. It was found that for all other different tilt angles, critical heat flux varies inversely to evaporator length and directly proprtional to internal diameter of heat pipe.

Grooten et al. [27] used large length and small-diameter thermosyphons having R-134a as the working fluids to conduct experiments to analyses the consequences of saturation temperature, filling ratio, and inclination angle on the limiting operational heat flux. It was examined that the thermosyphon performs well under tilt angle of 83°.It was concluded that by increasing the saturation temperature or by decreasing the inclination angle, reduction in operation limiting heat flux. was noticed. Inclination angles greater than 83° is not recommended for efficient working of heat pipe.

Riehl et al. [28] studied the application of water-copper nanofluid in the open loop pulsating heat pipe. This heat pipe was used with water-copper nanofluid with a mass of Cu nanoparticles incorporation of 5 %. The author noticed improvisations in the overall device operation while using nanofluid with lower temperatures. Due to more formation of bubbles, pulsations that are excessively intensive were noticed at the time of P.H.P. operation, resulting in the more presence of vapor in the channels. As a result, higher thermal conductance was noticed when compared with pulsating heat pipe operation having working fluid as pure water.

Senthilkumar et al. [29] scrutinized the behavior of the heat pipe at various orientations regulated on aqueous solution of n-pentanol. A comparison check was conducted between heat pipes operating with water and with n-pentanol as working fluids at different tilt angles. The result excellent performance of heat pipe while working on n-pentanol fluid due the fact that the aqueous solution has positive gradient of surface tension with respect to the temperature. There are some disadvantages of using water as operating fluid in spite of the fact that it is most extensively used operating fluid in order to eradicate these limitations, the working fluid water can be replaced with the dilute aqueous solution of n-pentanol. Due to high capillary limit and boiling limit of using n-pentanol which makes heat pipe suitable for large heat load applications.

Moraveji et al. [30] inspected the thermal performance of the heat pipe by using aluminum oxide Nano particle having dia 35 nm. A heat made of cooper with varied length was taken having sintered wick structure. Tested concentrations of Nano fluids were 0 %, 1 % and 3 % by weight. The conclusion obtained with the heat pipe charged with pure water were compared. The results show that with increase in heat input the wall temperature increases, the wall temperature of heat pipe with Nano fluid was low as compare to the heat pipe charged with pure water. The results show that the thermal resistance of the Nano fluid heat pipe is smaller than that of the water heat pipe.

Ghanbarpour et al. [31] examined the outcome of increase and then decrease of heat input on the performance of the heat pipe. A copper heat pipe was taken having 6.35 mm outer diameter and with length of 200 mm. It comprises of screen mesh wick structure and having alumina nanoparticle with various mass concentration of 5 % and 10 % respectively. The outcome concluded that with 5 %, concentration, heat pipe performance was enhanced with variable heat input and when 10 % concentration was used performance of heat pipe was ramshackled.

2.1. Closure

In the previous literature work, a lot of experimentation was carried out on various heat pipes but calculation of heat transfer coefficient of condenser section was missing with same specification of heat pipe as in setup under forced convection with same fan speed. Therefore, this is the driving force to find the desired results for heat pipe under forced convection for better heat dissipation techniques.

2.2. Gaps Identified

Different research in the heat pipe area has been carried out but performance of heat pipe at different inclinations under forced convection with same specifications of heat pipe were missing in literature. Calculation of experimental heat transfer coefficient through condenser section having annular fins with various fan speed in a rectangular duct of specific dimensions were also missing in the literature work

2.3. Objectives

1. To find out the convective heat transfer coefficient of condenser region of the heat pipe at different inclinations under forced convection.
2. To compare the condenser convective heat transfer coefficient obtained under forced convection for both experimental and empirical cases.
3. To find out pressure difference across condenser end through single column inclined manometer having hexane (liquid form) as the fluid because of its low density.

Chapter 3
Apparatus Details

In the apparatus, there is copper water heat pipe provided with an arrangement for providing tilt to heat pipe. The set of experiments were carried out on copper water heat pipe giving tilt under forced convection.

3.1 Purpose of Experiment

The purpose of doing the experiment was to mainly find the heat transfer coefficient of condenser section of heat pipe at different tilt angles. At a particular angle, we get maximum heat transfer coefficient across condenser section, thus there will be a fair scope of placing the heat pipe with same specifications at same tilt angle in order to have the maximum heat dissipation. Moreover, the pressure difference across condenser section end measured through inclined single column manometer filled with hexane in liquid form enables us to predict the losses across the condenser section of heat pipe. With this study, an optimum angle can tell the advancement in heat dissipation through condenser section having annular fins under forced convection.

1. The heat transfer coefficient across the condenser region of heat pipe under forced convection is studied. A stainless-steel duct was constructed to enclose the heat pipe.
2. A honey comb mesh was placed just next to fan at a distance of 26 cm from fan. The function of the mesh is that it will straighten the air passing through duct and flow across the condenser section of the heat pipe
3. The fan is set to provide for three air speeds, which are 1.35 m/s, 2.34 m/s and 3.36 m/s, respectively.
4. The air speeds are measured with the help of vane probe type anemometer by placing anemometer at the exit section of duct. The velocity of is measured by taking the grid measurement.
5. At the inlet of the duct, the fan cools the condenser section of heat pipe.
6. Duct is shaped rectangular with cross section of 15 cm × 25 cm. In order to traverse the tilt, angle up to 45 degrees for heat pipe, the height of the duct should be made more than 180 mm but also it is hinged from some another point beyond 18 cm. Therefore, ideal height of duct is kept at 25 cm.

Table 3.1 gives the specifications of heat pipe along with their notations.

Table 3.1. Specifications of Heat pipe along with their notations

S. No.	Notations	Parameter	Value
1	k_s	Thermal conductivity of solid wick material	50 W/mK
2	k_p	Thermal conductivity of heat pipe material	400 W/mK
3	A_v	Cross-section area of vapor core region	1.9625×10^{-5} m

4	A_w	cross-section area of wick	8.635×10^{-6} m^2
5	T_P	Wall thickness of Heat pipe	0.9×10^{-3} m
6	T_w	Thickness of wick wire	0.0406×10^{-3} m
7	K_{wick}	Wick effective thermal conductivity	1 W/mk^2
8	L_e	Length of evaporator section	60×10^{-3} m
9	L_a	Length of adiabatic section	55×10^{-3} m
10	L_c	Length of Condenser section	65×10^{-3} m
11	D_v	Vapor core diameter	5×10^{-3} m
12	$R_{h,v}$	Heat pipe hydraulic radius	2.5×10^{-3} m
13	R_i	Heat pipe inside radius	3×10^{-3} m
14	R_o	Heat pipe outside radius	3.9×10^{-3} m

The below figure 3.1. represents the schematic diagram of the heat pipe apparatus used for experimentation purposes.

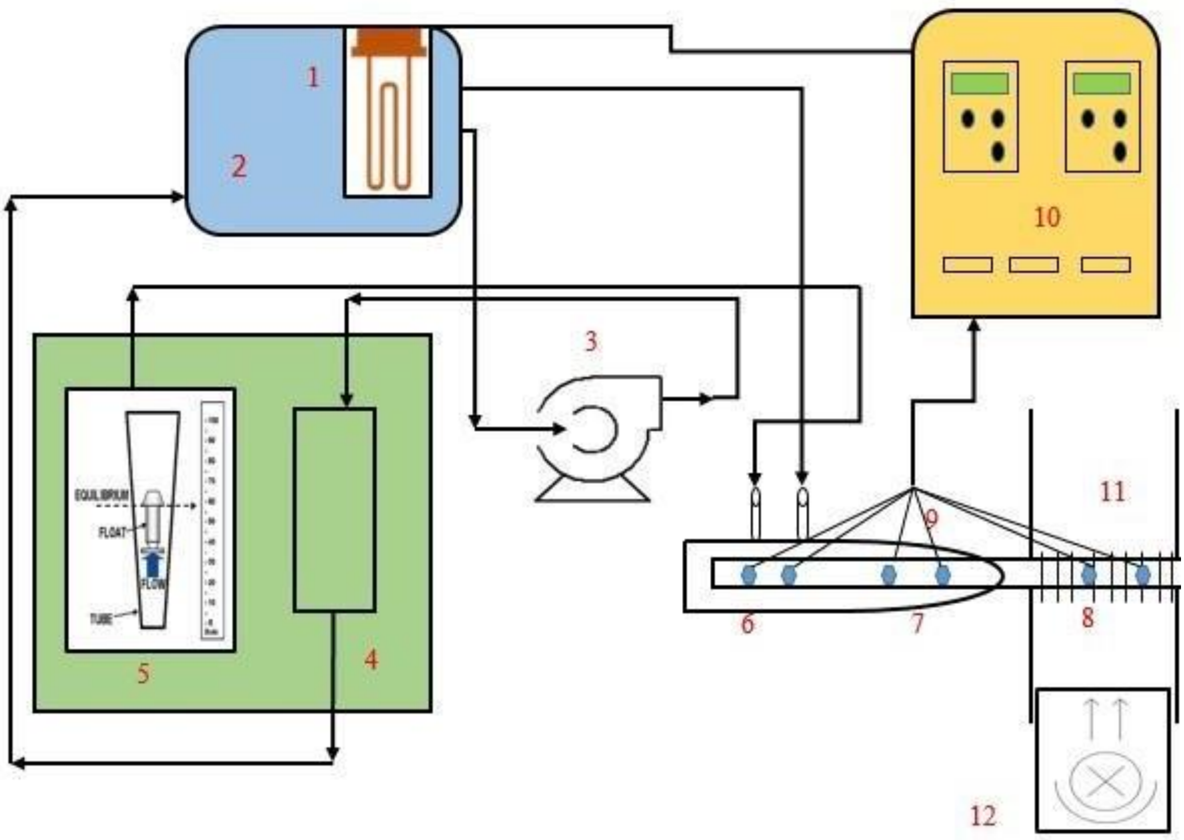

Figure 3.1: Schematic diagram of experimental facility

Below Table 3.1 depicts the part description of experimental test rig.

Sr.No.	Description
1	Heating element
2	Water tank
3	Pump
4	By-pass valve
5	Flow meter
6	Evaporator section of heat pipe
7	Adiabatic section of heat pipe
8	Condenser section of heat pipe having annular fins
9	Pt-100 RTD sensors
10	Data logger
11	Stainless steel duct
12	Fan

Below images depicts the apparatus details which was clicked at various angles and sides to have a clear image of the experiment rig.

3.2. Apparatus Details

Figure 3.2. Front glimpse of heat pipe setup

Figure 3.3. Fan mounted on a duct having a regulator

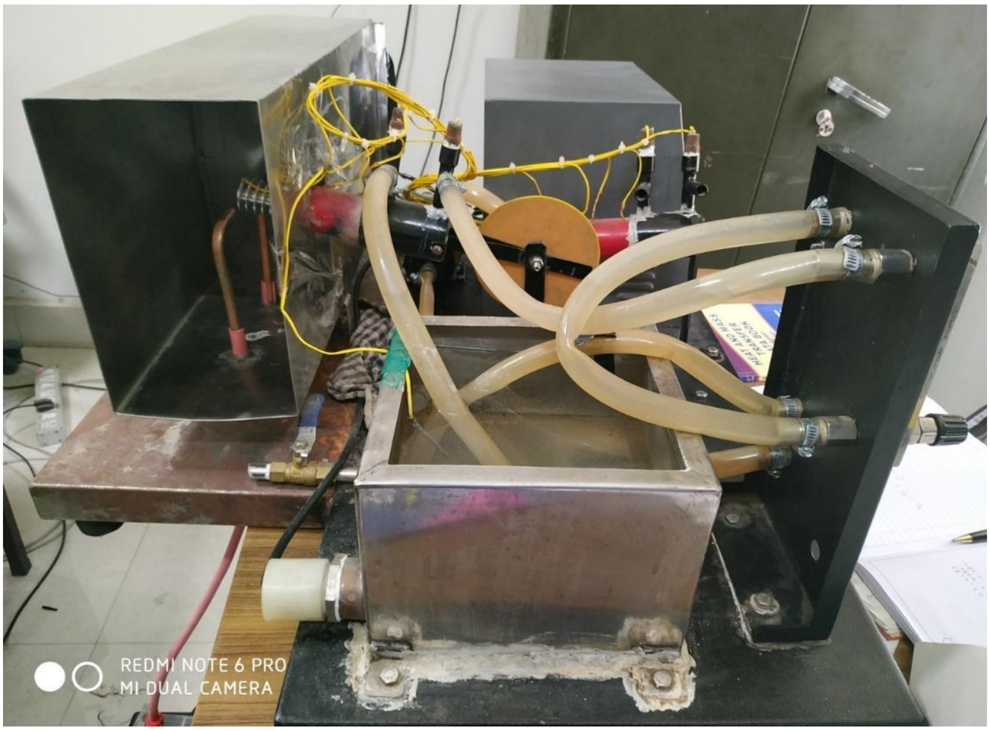

Figure 3.4. Side view depicting water tank with heating coil

Figure 3.5. Front side Depicting Rotameter and By-pass valve

Figure 3.6. Pressure pipes through the base of the duct connected to single column inclined manometer

Figure 3.7. Top view of the apparatus

Figure 3.8. Stainless steel duct

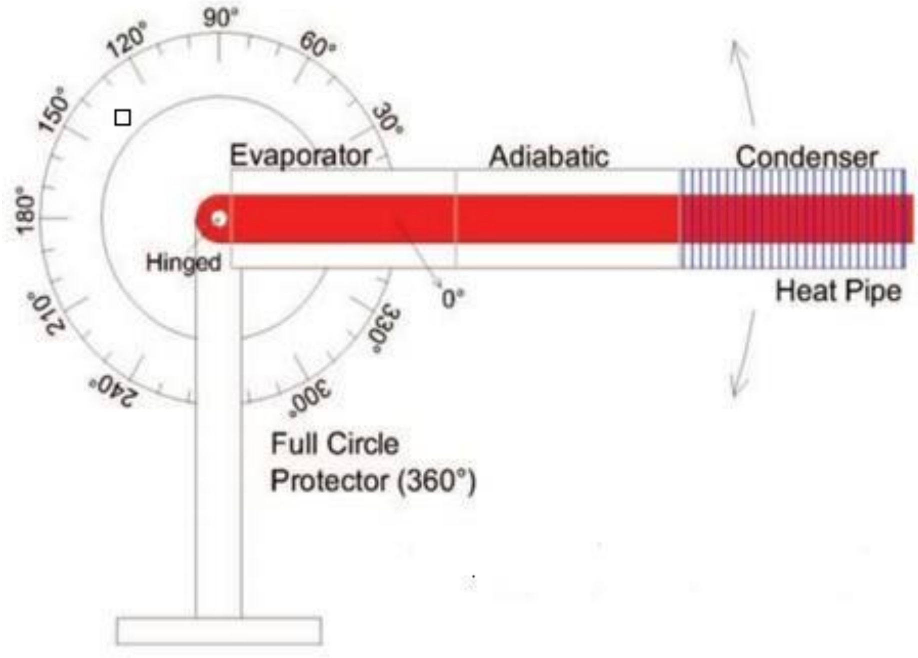

Figure 3.9. Tilting mechanism [32]

3.3. Steps taken to conduct the Experiment:

1. Before starting the experiment switch off the ceiling fan and close the window to control the turbulence in air at our own level as a precautionary step.
2. Switch on the fan mounted on the duct and allow the flow of air through the inlet of the duct towards the other end of the duct.
3. With the help of vane anemometer, measure the airflow velocity at the exit of the duct with the help of grid geometry.
4. Note down the velocity at one airflow, then with the help of regulator adjust two more velocities, and measure the same by placing vane anemometer at the exit section of duct.
5. Set process value and set value in the data logger as per the experiment requirement.
6. Set the tilt angle of heat pipe at which reading of condenser side heat transfer coefficient is taken.
7. Measure ambient temperature of room.
8. Take the initial reading in the single inclined manometer so as get access of the deflection in hexane liquid to predict pressure difference.
9. Now turn on the apparatus from the data logger by pressing fan speed button, heating coil button, and centrifugal pump button.

10. Adjust the probe of the Rotameter at desired mass flow rate of water.
11. Start taking the readings at an interval of 5 minutes.
12. After switching on all parameters take the initial readings of temperature through thermocouples from data logger at 0 minutes
13. Take subsequent readings at an interval of 5 minutes till steady state is achieved
14. After steady state is achieved note down the deflection in the single column inclined manometer and take the reading. The pressure difference across the condenser region of heat pipe is calculated by using expression $P= \rho g l * \sin\theta$, where θ is inclination angle of the single column inclined manometer.
15. Note down all the six temperatures readings from data logger and then calculate heat transfer coefficient across condenser section of the heat pipe by using Churchill and Bernstein equation. The heat transfer coefficient calculated through this process will be termed as empirical heat transfer coefficient of condenser region of heat pipe.
16. Similarly calculate the experimental value of heat transfer coefficient by using relations, which are discussed in methodology.
17. Make comparison between empirical and experimental value and simultaneously plot the graph between the two.

Chapter 4

Methodology

The heat pipe condenser side heat transfer coefficient was obtained both experimentally as well as empirically. The empirical correlations were deduced by Churchill and Bernstein under forced convection across cylindrical surfaces. [33] Following methodology was adopted to determine heat transfer coefficient across condenser end.

4.1 Experimental method

To examine the performance of heat pipe under forced convection, number of tests are to be conducted under steady state.

The inclination of heat pipe was varied from 0^0 to 30^0 -tilt angle.

The mass flow rate was also varied by adjusting the globe valve of rotameter mounted on the apparatus.

$$Q_{input} = \dot{m} c_p (T_1 - T_2)$$

$\dot{m}$= mass flow rate of water (kg/s)

c_p= Specific heat of water

T_1 =Temperat entrance of the heat pipe in evaporator section

T_2=Temperature at exit of the heat pipe in evaporator section

The heat dissipated across the condenser section of heat pipe having annular fin is given by

$$Q_{dissipated} = h_c (A_o + A_f \eta_f)(T_c - T_a) \quad [34]$$

Whereas overall fin surface efficiency is expressed by

$$\eta_f = 1 - \frac{A_f}{A_c}(1 - \eta) \quad [34]$$

whereas fin effectiveness is given by

$$\eta = \frac{tanh\phi}{\phi} \quad [34]$$

with the following parameters where ϕ is given by following relation

$$\phi = mL(R^*)^{\exp(0.13mL - 1.3863)} \quad [34]$$

$$R^* = \sqrt{\frac{d_{f,o}}{d_o}} \quad [34]$$

$$m = \sqrt{\frac{2h}{k_f t_f}} \quad [34]$$

Heat loss=Heat gained

$$\dot{m} c_p (T_1 - T_2) = hc(A_o + A_f \eta_f)(T_c - T_a)$$

Thus, h_c can be calculated by using the above equations.

With the help of above equations, we can determine the value of experimental heat transfer coefficient of heat pipe under forced convection

4.2 Empirical method

The empirical correlations were deduced by Churchill and Bernstein for cylindrical surfaces under forced convection. Firstly, we calculate the following parameter to find out the external heat transfer coefficient of condenser section of heat pipe.

Bulk mean temperature, T_f (°c)

T_f = (Average wall temperature of condenser+ Ambient temperature)/2 [34]

With the help of Bulk mean temperature, we will note down the thermo physical properties of air

a. Kinematic viscosity
b. Density
c. Specific heat
d. Thermal conductivity
e. Prandtl number

Nusselt Number is defined for various values of Reynold's number using Churchill and Bernstein equation for cylindrical pipes under forced convection.

$$Nu_d = 0.3 + \frac{0.62Re^{1/2}Pr^{1/3}}{[1+(0.4/Pr)^{2/3}]^{1/4}}\left[1+\left(\frac{Re}{282{,}000}\right)^{5/8}\right]^{4/5}$$

for $10^2 < Re_d < 10^7$; $Pe_d > 0.2$ [34]

$$Nu_d = 0.3 + \frac{0.62Re_d^{1/2}Pr^{1/3}}{[1+(0.4/Pr)^{2/3}]^{1/4}}\left[1+\left(\frac{Re_d}{282{,}000}\right)^{1/2}\right]$$

for $20{,}000 < Re_d < 400{,}000$; $Pe_d > 0.2$ [34]

$Nu = \frac{h\,d}{k_{fluid}}$ [14]

Thus, convective heat transfer coefficient value will be determined empirically using equations in a flow across cylinder under forced convection.

Chapter 5

Results and Discussion

This chapter discuss about the results obtained after performing set of 84 experiments on Cu-H_2O heat pipe at different tilt angles under forced convection. The experimental results were then compared with empirical relations given by Churchill and Bernstein. During experimentation, mass flow rate of water was varied from 80ml/min to 200ml/min. In this experiment, three air velocities are taken which are 1.35 m/s, 2.34 m/s, and 3.36 m/s respectively. The air velocities are measured with the help of vane probe anemometer.

5.1. Condenser heat transfer coefficient at different mass flow rate, tilt angle and at different speeds

The empirical correlations established by Churchill and Bernstein for forced convection over cylindrical surfaces were used to calculate the heat transfer coefficient of condenser section having annular fins on Cu-H_2O heat pipe. The experimental values were obtained by using the expressions as discussed in methodology. The heat transfer coefficient in forced convection in this experiment depends mainly on air velocity, mass flow rate of water, tilt angle of heat pipe and ambient temperature. Higher the air velocity more quickly it will cool the hot region by taking off the heat. In this experimentation the minimum air velocities achieved was 1.35 m/s and maximum was 3.36 m/s. Variation of mass flow rate was kept from 80 ml/min to 200 ml/min having subsequent increment of 40 ml/min. The results have been showing graphically depicted, the variation of heat transfer coefficient obtained experimentally and empirically at different mass flow rate keeping air velocity constant for a particular set over the condenser section of experiment.

The heat pipe experimentation was carried out under various air velocities and different tilt angles by adjusting mass flow rate with the help of rotameter mounted on apparatus. The experimentation gives the experimental and empirical values of condenser side heat transfer coefficient obtained under forced convection.

Followings are the graphs obtained experimentally and empirically

5.1.1 At tilt angle 0° and air velocity 1.35 m/sec

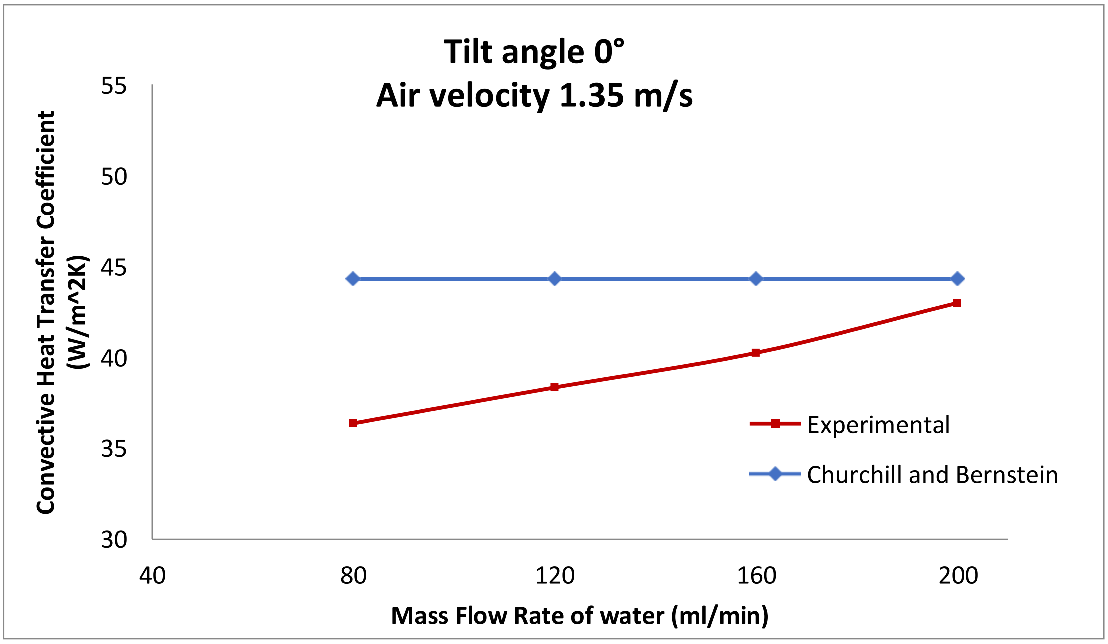

Figure 5.1. h at 0° tilt angle and 1.35 m/s air velocity

Table 5.1. Obtained values of h at 0° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	36.367	44.311
120	38.336	44.312
160	40.239	44.318
200	42.983	44.31

The graphical comparisons under forced convection heat transfer coefficients for experimental results and also for empirical correlations obtained through Churchill and Bernstein correlations were plotted in figure 5.1. The x-axis represents the mass flow rate of water in ml/min whereas the y-axis represents the convective heat transfer coefficient of condenser region in W/m^{2}K obtained both experimentally and empirically. From the above-depicted graph 5.1, the following results were concluded: -

- Heat transfer coefficient increases with increased mass flow rate of heating fluid.

- At 0° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 42.983 W/m^2K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 36.367 W/m^2K.
- At 0° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein correlations were observed to be within the range 17.93% to 2.3%.
- The empirical value stands almost constant having maximum value as 44.318 W/m^2K.

5.1.2 At tilt angle 5° and air velocity 1.35 m/sec

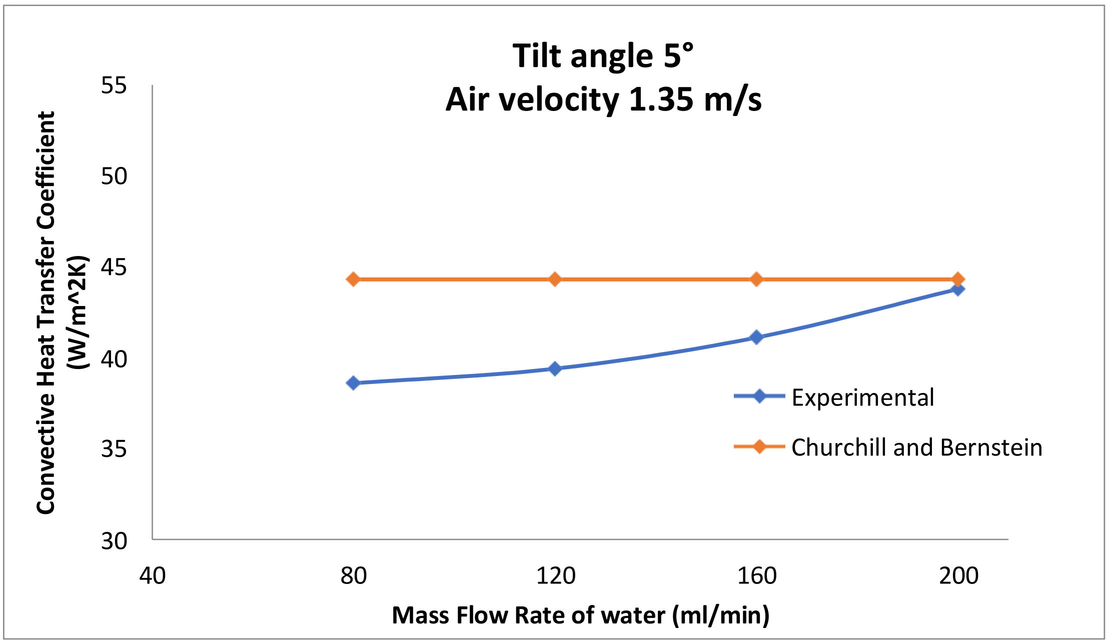

Figure 5.2. h at 5° tilt angle and 1.35 m/s air velocity

Table 5.2. Obtained values of h at 5° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^2K	Empirical value of heat transfer coefficient in W/m^2K
80	38.597	44.312
120	39.401	44.306
160	41.112	44.311
200	43.778	44.311

Observations

- Heat transfer coefficient increases with increased mass flow rate of heating fluid.
- At 5° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 38.597

W/m^{2}K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 43.778 W/m^{2}K.

- At 5°, tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein correlations under forced convection were observed to be within the range 12.9% to 1.2%.
- The empirical value stands almost constant having maximum value as 44.312 W/m^{2}K

5.1.3 At tilt angle 10° and air velocity 1.35 m/sec

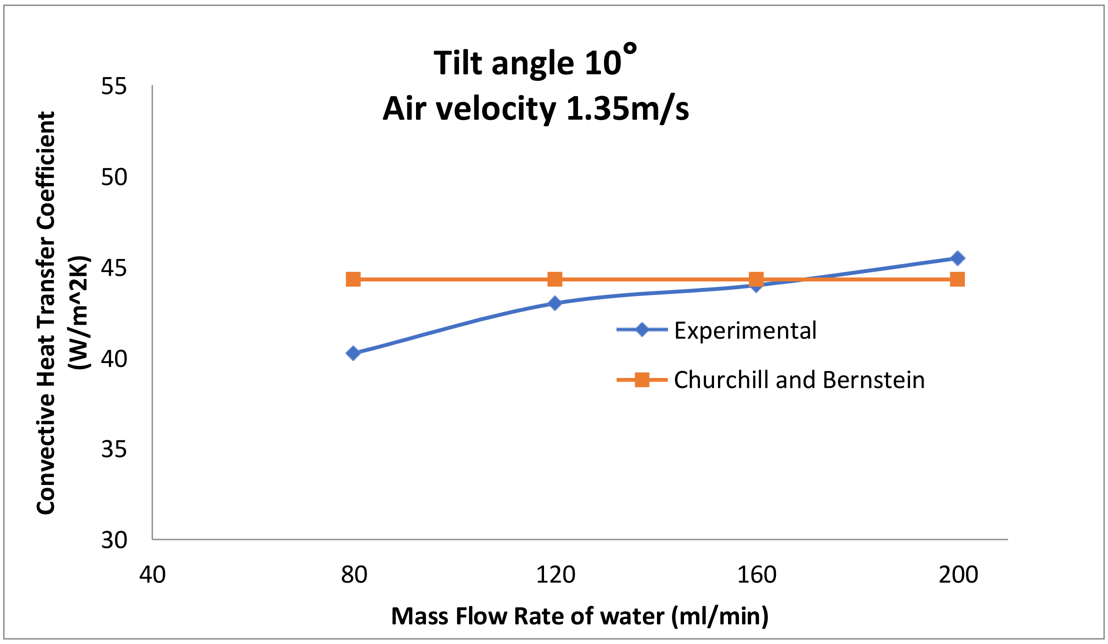

Figure 5.3. h at 10° tilt angle and 1.35 m/s air velocity

Table 5.3. Obtained values of h at 10° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	40.237	44.307
120	42.981	44.305
160	43.98	44.31
200	45.462	44.309

. From the above-depicted graph, the following results were concluded: -

- Similar trends were observed with mass flow rate of water variation with heat transfer coefficient.
- At 10° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 45.462

W/m^{2}K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 40.237 W/m^{2}K.

- At 10° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 9.19% to 0.74%.
- The empirical value stands almost constant having maximum value as 44.31 W/m^{2}K.

5.1.4 At tilt angle 15° and air velocity 1.35 m/sec

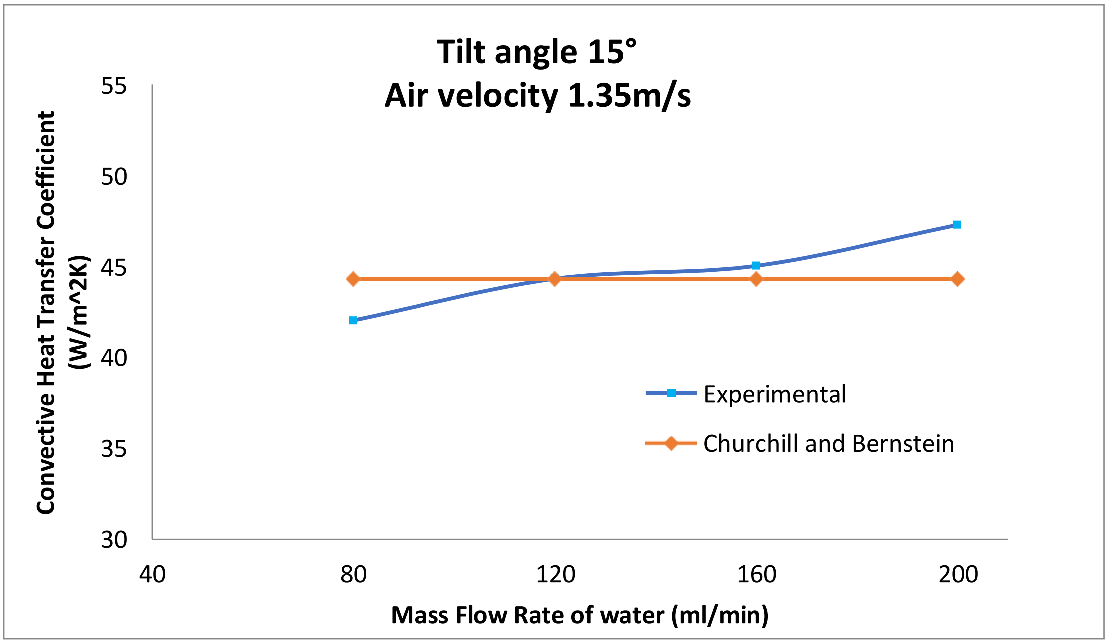

Figure 5.4. h at 15° tilt angle and 1.35 m/s air velocity

Table 5.4. Obtained values of h at 15° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	42.027	44.305
120	44.324	44.306
160	45.028	44.314
200	47.281	44.314

. From the above depicted graph, the following results were concluded: -

- Heat transfer coefficient increases with increased mass flow rate of heating fluid..
- At 15° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was recorded at 200 ml/min mass flow rate of water having value 47.281 W/m^{2}K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 42.027 W/m^{2}K.

- At 15° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein correlations were observed to be within the range 6.7% to 0.04%.
- The empirical value stands almost constant having maximum value as 44.314 W/m^{2}K.
- The experimental value was increasing in 15° tilt angle prior to 10° tilt angle because here the gravitational forces were seen to be in greater magnitude as compared to 10° tilt angle. Moreover, capillary forces inside the wick in addition to gravitational forces due to tilt of heat pipe leads to lower thermal resistance thus increasing the heat transfer coefficient.
- The performance of heat pipe was observed to better at 15° tilt angle as compared to 10° tilt angle. Thus, higher heat transfer coefficient across condenser section were observed in accordance with the above said conclusion.

5.1.5 At tilt angle 20° and air velocity 1.35 m/sec

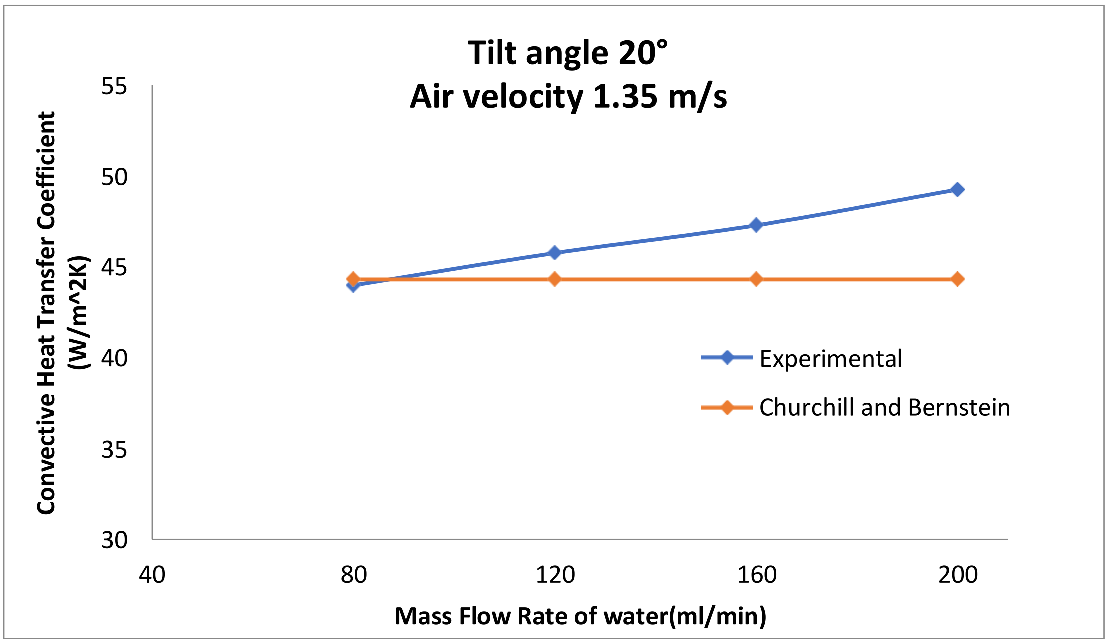

Figure 5.5. h at 20° tilt angle and 1.35 m/s air velocity

Table 5.5. Obtained values of h at 20° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^{2}K	Empirical value of heat transfer coefficient in W/m^{2}K
80	43.983	44.31
120	45.755	44.307
160	47.279	44.309
200	49.25	44.31

. From the above depicted graph, the following results were concluded: -

- Similar trends were observed with mass flow rate of water variation.
- At 20° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was recorded at 200 ml/min mass flow rate of water having value 49.25 W/m^{2}K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 43.983 W/m^{2}K.
- At 20° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein correlations were observed to be within the range 11.15% to 0.74%.
- The empirical value stands almost constant having maximum value as 44.31 W/m^{2}K.

5.1.6 At tilt angle 25° and air velocity 1.35 m/sec

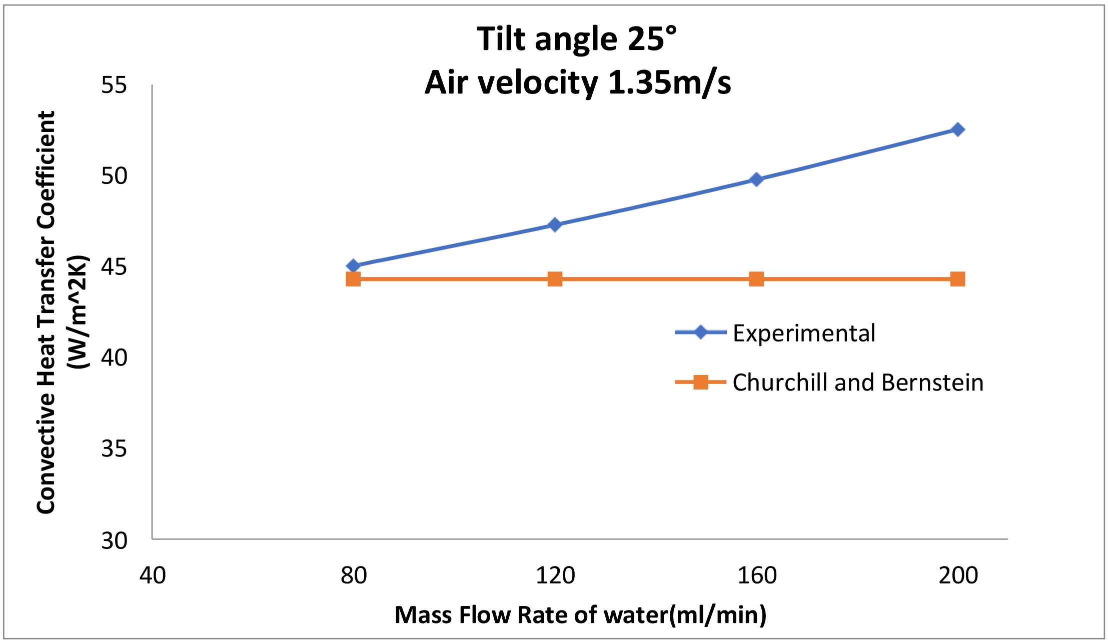

Figure 5.6. h at 25° tilt angle and 1.35 m/s air velocity

Table 5.6. Obtained values of h at 25° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	45.029	44.304
120	47.278	44.308
160	49.768	44.3
200	52.53	44.312

From the above depicted graph, the following results were obtained: -

- Same variation was observed with mass flow rate

- At 25° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 52.53 W/m^2k whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 45.029 W/m^2k.
- At 25° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein correlations under forced convection were observed to be within the range 18.55% to 1.64%.
- The empirical value stands almost constant having maximum value as 44.312 W/m^2k.
- In the above-depicted graph, the experimental value is crossing the empirical value because at 25° tilt angle gravitational forces along with capillary forces inside the wick will help the condensate to return to the evaporator section more quickly as compared to 0° tilt angle. Because of this thermal resistance decreases and heat pipe performance increases leading to high value of heat transfer coefficient.

5.1.7 At tilt angle 30° and air velocity 1.35 m/sec

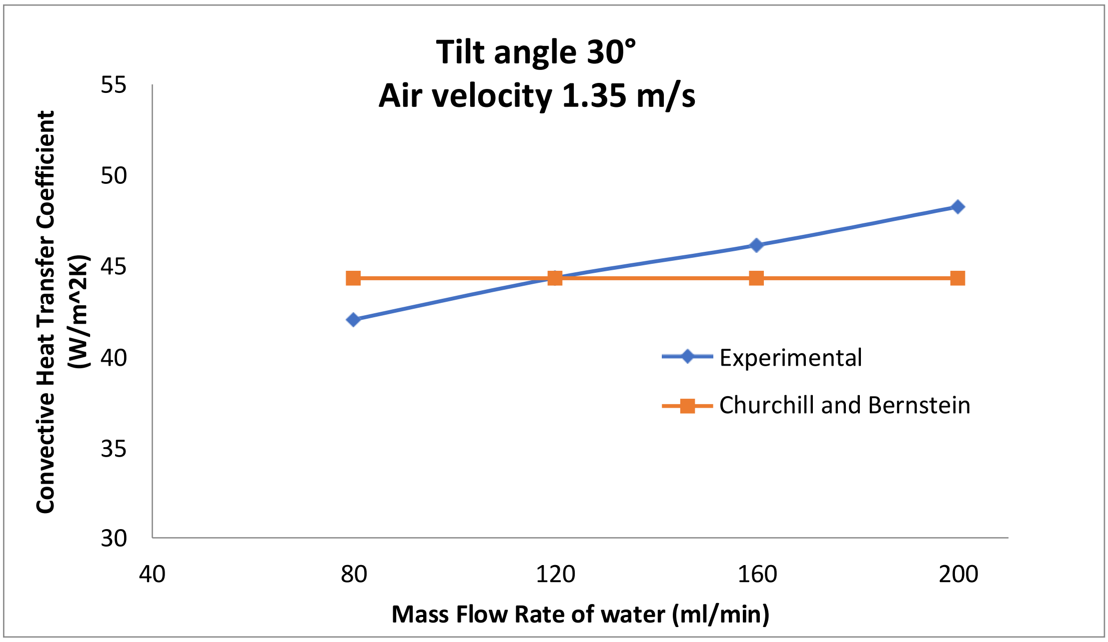

Figure 5.7. h at 30° tilt angle and 1.35 m/s air velocity

Table 5.7. Obtained values of h at 30° tilt angle and 1.35 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^2K**	**Empirical value of heat transfer coefficient in W/m^2K**
80	42.027	44.312
120	44.326	44.306
160	46.126	44.314
200	48.246	44.312

From the above-depicted graph, the following results were concluded: -

- At 30° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 48.246 W/m^2k whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 42.027 W/m^2K.
- At 30°, tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein correlations were observed to be within the range 8.88% to 0.04%.
- The empirical value stands almost constant having maximum value as 44.314 W/m^2K.
- In the above depicted graph, the experimental values start decreasing after 25° tilt angle because beyond 25° tilt angle the condensate returns quickly back to the evaporator section causing flooding of evaporator section which ultimately decreases its performance leading to lower value of heat transfer coefficient. The values of heat transfer coefficient are lower than that at 25°.
- The best thermal performance of the heat pipe was observed in the range of 15-25° tilt angle.

The other set of experiment were conducted at 2.34 m/s Fan speed in which tilt angles varies from 0° to 30° and having same mass flow rate variation ranging from 80 ml/min to 200 ml/min.

5.1.8 At tilt angle 0° and air velocity 2.34 m/sec

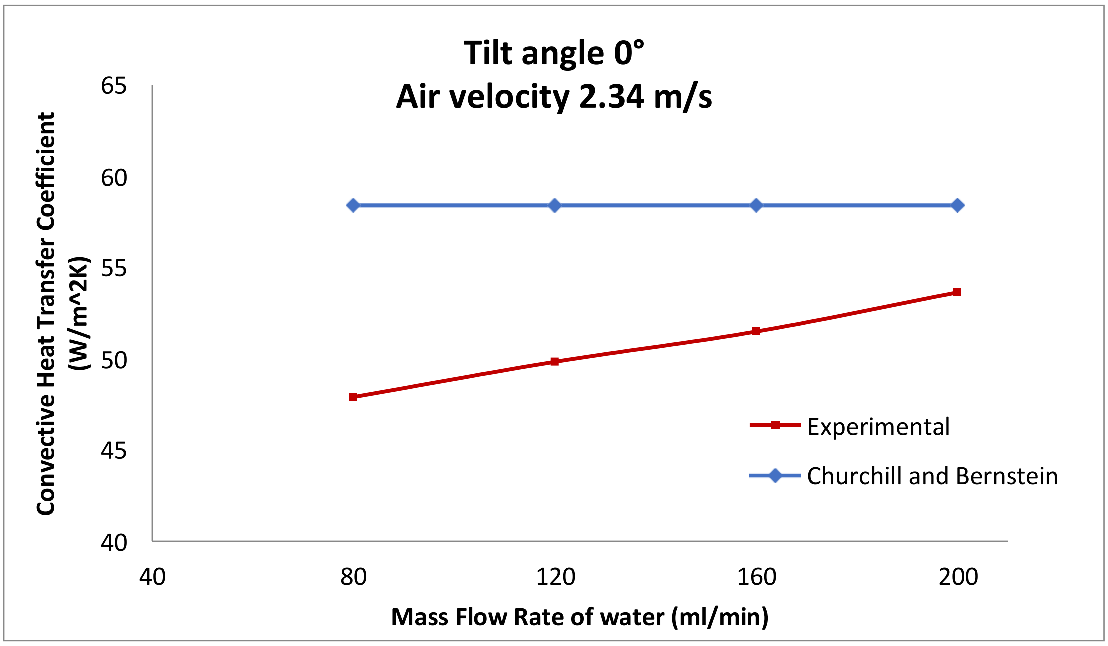

Figure 5.8. h at 0° tilt angle and 2.34 m/s air velocity

Table 5.8. Obtained values of h at 0° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	47.905	58.41
120	49.836	58.404
160	51.495	58.415
200	53.644	58.409

. From the above-depicted graph, the following results were concluded: -

- Similar trends were observed with mass flow rate of water versus condenser side heat transfer coefficient.
- At 0° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 53.644 W/m^{2}K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 47.905 W/m^{2}K.
- At 0° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 17.98% to 8.16%.
- The empirical value stands almost constant having maximum value as 58.415 W/m^{2}K.

5.1.9 At tilt angle 5° and air velocity 2.34 m/sec

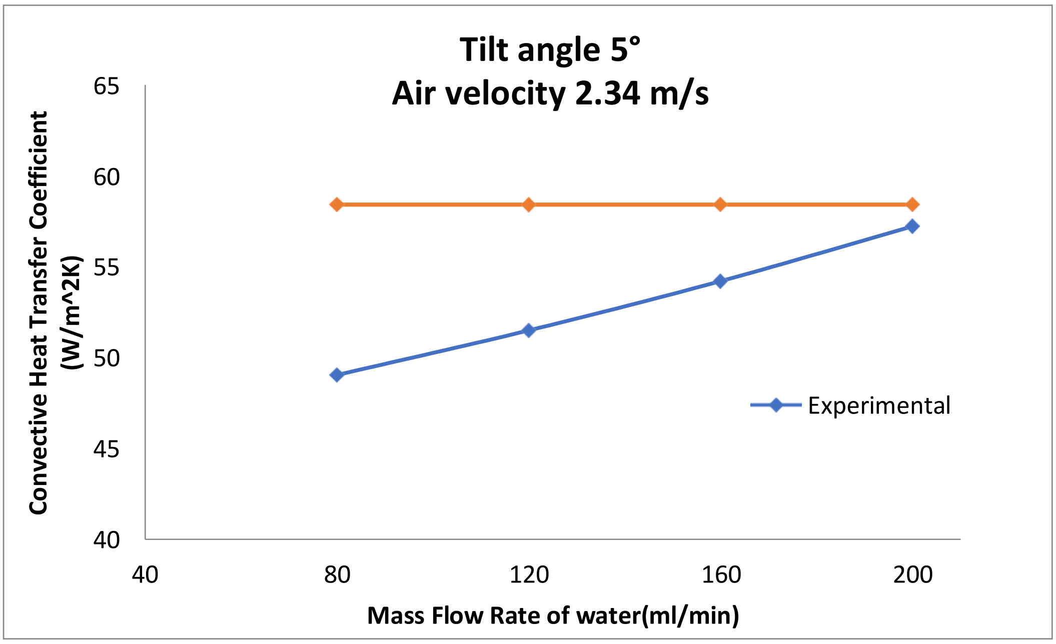

Figure 5.9. h at 5° tilt angle and 2.34 m/s air velocity

Table 5.9. Obtained values of h at 5° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^2k**	**Empirical value of heat transfer coefficient in W/m^2k**
80	49.044	58.417
120	51.497	58.405
160	54.205	58.417
200	57.219	58.413

. From the above depicted graph 5.9, the following results were concluded: -

- At 5° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 57.219W/m^2k whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 49.044 W/m^2K.
- At 5° tilt angle, the percentage deviation between experimental value and empirical value were observed to be within the range 16.05% to 2.05%.
- The empirical value stands almost constant having maximum value as 58.417 W/m^2K.

5.1.10 At tilt angle 10° and air velocity 2.34 m/sec

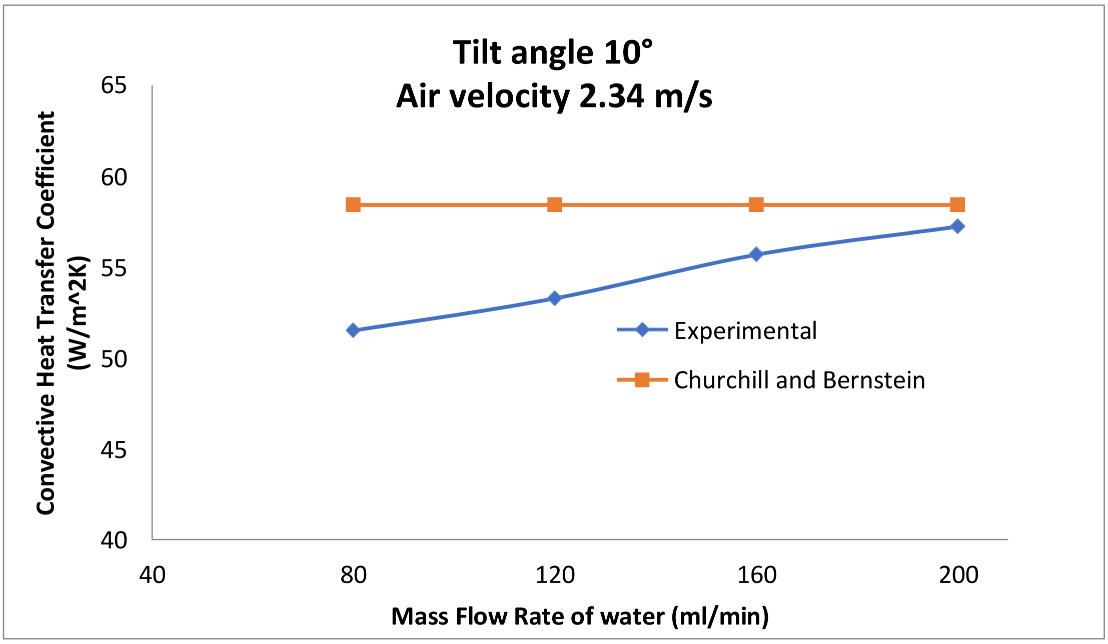

Figure 5.10. h at 10° tilt angle and 2.34 m/s air velocity

Table 5.10. Obtained values of h at 10° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	51.498	58.402
120	53.27	58.407
160	55.671	58.407
200	57.217	58.407

From the above depicted graph 5.10, the following results were concluded: -

- Mass flow rate was seen to have similar trends as discussed above.
- At 10° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 57.217 W/m^{2}K whereas the minimum value obtained experimentally was seen at 80ml/min having value as 51.498 W/m^{2}K.
- At 10° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 11.82% to 2.04%.
- The empirical value stands almost constant having maximum value as 58.407 W/m^{2}K.

5.1.11 At tilt angle 15° and air velocity 2.34 m/sec

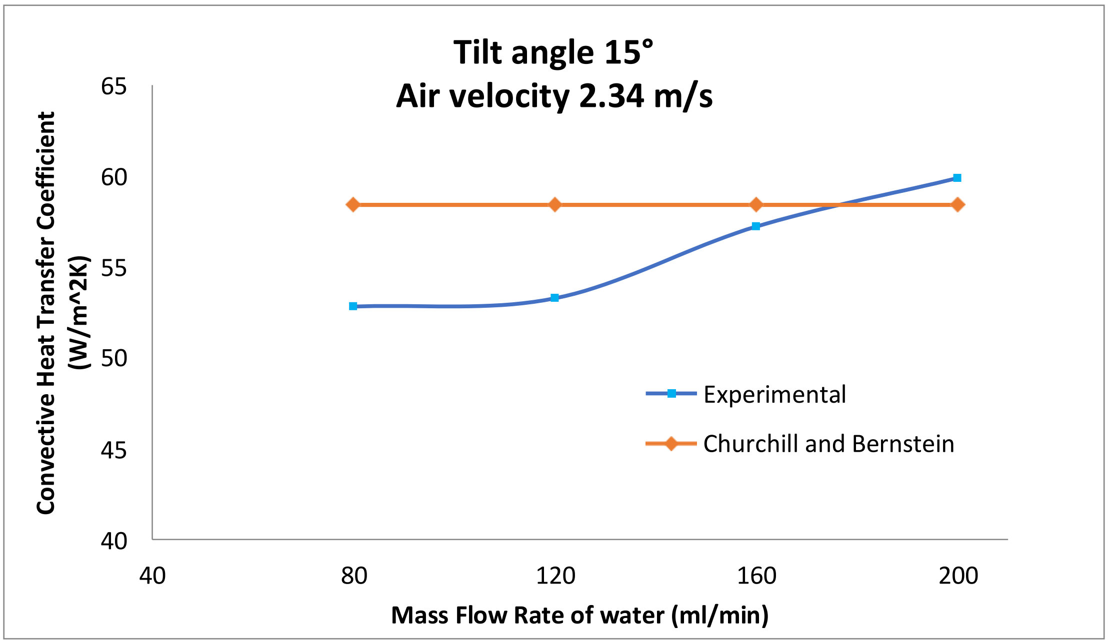

Figure 5.11. h at 15° tilt angle and 2.34 m/s air velocity

Table 5.11. Obtained values of h at 15° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^2K	Empirical value of heat transfer coefficient in W/m^2K
80	52.816	58.404
120	53.27	58.407
160	57.217	58.409
200	59.879	58.416

. From the above depicted graph 5.11, the following results were concluded: -

- Similar trends were seen with mass flow rate variation.
- At 15° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was recorded at 200 ml/min mass flow rate of water having value 59.879 W/m^2k whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 52.816 W/m^2k.
- At 15° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 9.57% to 2.04%.
- The empirical value stands almost constant having maximum value as 58.416 W/m^2k.

5.1.12 At tilt angle 20° and air velocity 2.34 m/sec

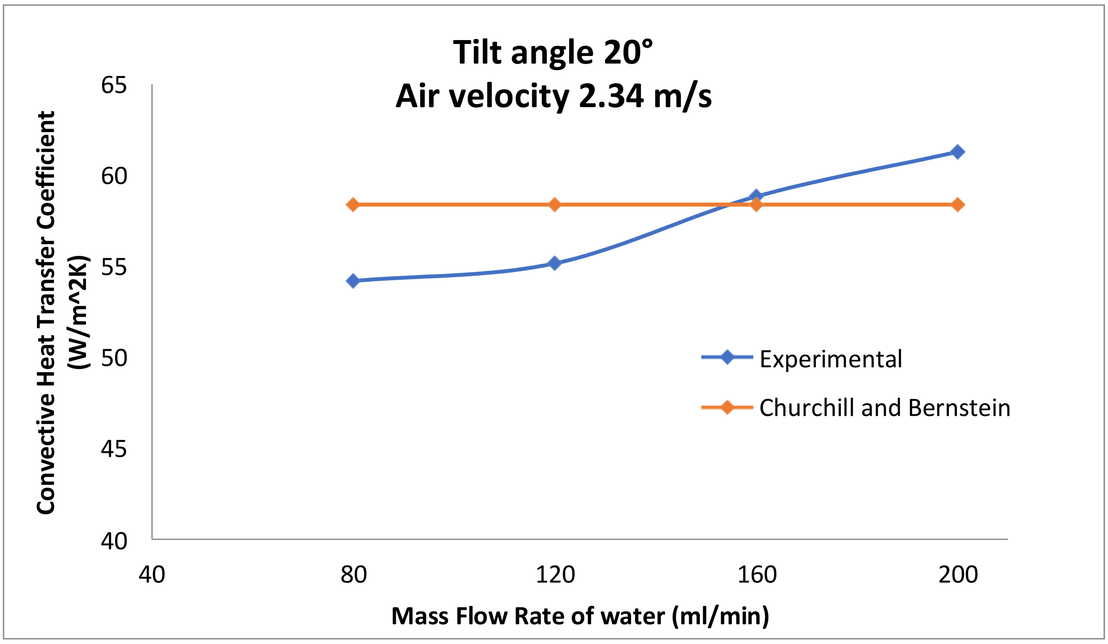

Figure 5.12. h at 20° tilt angle and 2.34 m/s air velocity

Table 5.12. Obtained values of h at 20° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^2K**	**Empirical value of heat transfer coefficient in W/m^2K**
80	54.209	58.405
120	55.173	58.402
160	58.85	58.4
200	61.303	58.405

. From the above depicted graph 5.12, the following results were concluded: -

- Similar trends were seen with mass flow rate of water variation.
- At 20° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was recorded at 200 ml/min mass flow rate of water having value 61.303 W/m^2k whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 54.209 W/m^2k.
- At 20° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 7.18% to 0.76%.
- The empirical value stands almost constant having maximum value as 58.405W/m^2k.

5.1.13 At tilt angle 25° and air velocity 2.34 m/sec

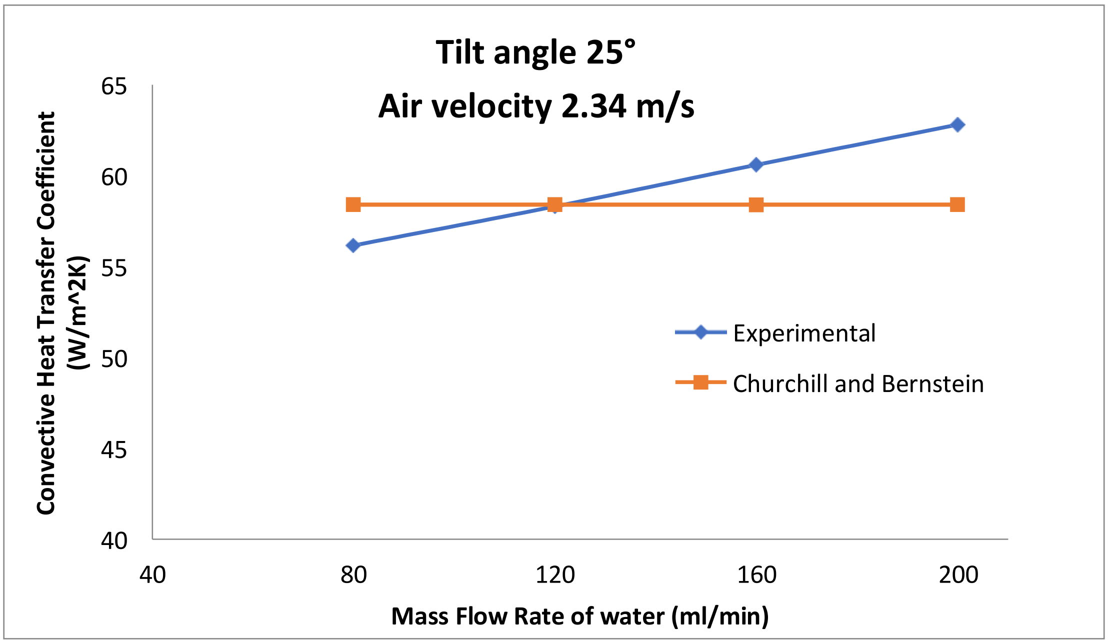

Figure 5.13. h at 25° tilt angle and 2.34 m/s air velocity

Table 5.13. Obtained values of h at 25° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}K**
80	56.155	58.41
120	58.298	58.407
160	60.583	58.391
200	62.793	58.4

. From the above depicted graph, the following results were obtained: -

- Similar trends for mass flow rate variation were seen.
- At 25° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 62.793 W/m^{2}K whereas the minimum value obtained experimentally was seen at 80ml/min having value as 56.155 W/m^{2}K.
- At 25° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 7.52% to 0.19%.
- The empirical value stands almost constant having maximum value as 58.41 W/m^{2}K.
- In the above depicted graph, the experimental value is crossing the empirical value because at 25° tilt angle gravitational forces along with capillary forces inside the wick will help the condensate to return to the evaporator section more quickly as compared to 0° tilt angle. As a result of this thermal resistance decreases and heat pipe performance increases leading to high value of heat transfer coefficient.
- With fan speed of 2.34 m/s we obtained the maximum value of condenser side heat transfer coefficient as 62.793 W/m^{2}K which was highest at this fan speed.

5.1.14 At tilt angle 30° and air velocity 2.34 m/sec

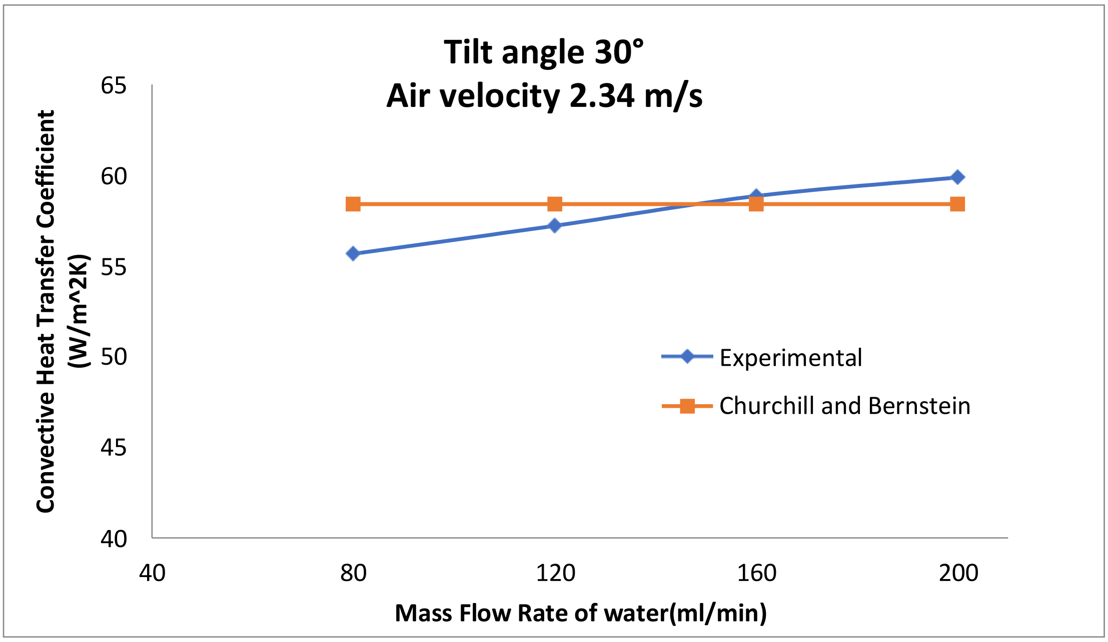

Figure 5.14. h at 30° tilt angle and 2.34 m/s air velocity

Table 5.14. Obtained values of h at 30° tilt angle and 2.34 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^2K**	**Empirical value of heat transfer coefficient in W/m^2K**
80	55.668	58.4
120	57.215	58.404
160	58.851	58.398
200	59.875	58.404

. From the above-depicted graph 5.14, the following results were concluded: -

- Mass flow rate of water was seen to have similar trends with condenser side heat transfer coefficient.
- At 30° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 59.875 W/m^2K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 55.668 W/m^2K.
- At 30°, tilt angle, the percentage deviation between experimental value and empirical value obtained through Churchill and Bernstein correlations were observed to be within the range 4.68% to 0.78%.
- The empirical value stands almost constant having maximum value as 58.404 W/m^2K.

- In the above depicted graph, the experimental values start decreasing after 25° tilt angle because beyond 25° tilt angle the condensate returns quickly back to the evaporator section causing flooding of evaporator section which ultimately decreases its performance leading to lower value of heat transfer coefficient. The values of heat transfer coefficient are lower than that at 25°.
- The best thermal performance of the heat pipe was observed in the range of 15-25° tilt angle.

The other set of experiment were conducted at 1.35 m/s Fan speed in which tilt angles varies from 0° to 30° and having same mass flow rate variation ranging from 80 ml/min to 200 ml/min.

5.1.15 At tilt angle 0° and air velocity 3.36 m/sec

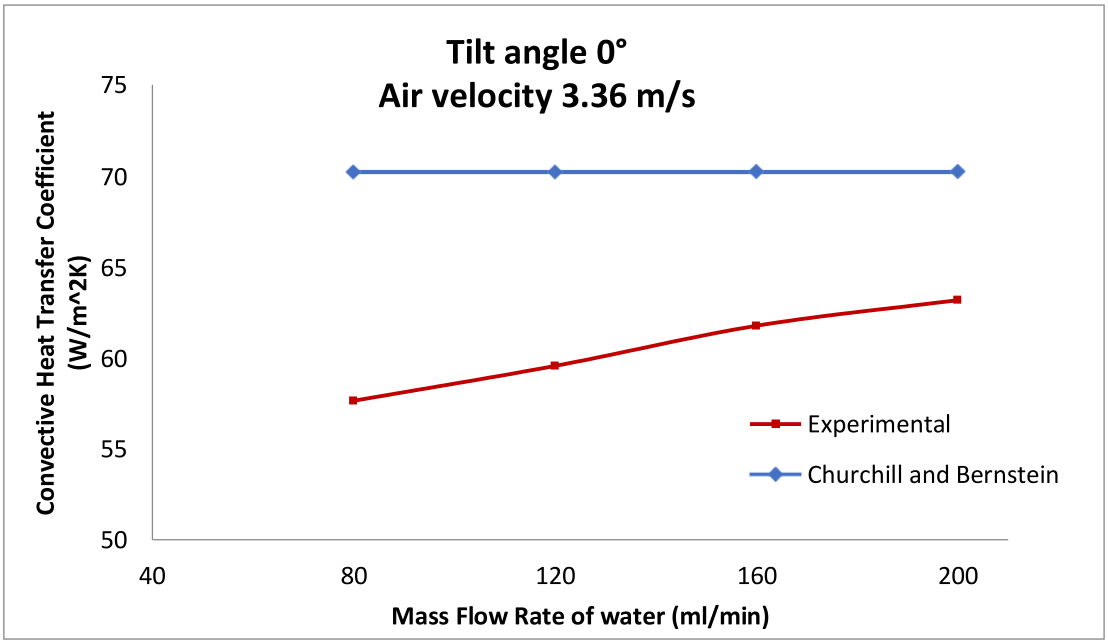

Figure 5.15. h at 0° tilt angle and 3.36 m/s air velocity

Table 5.15. Obtained Values of h at 0° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}k**	**Empirical value of heat transfer coefficient in W/m^{2}k**
80	57.631	70.206
120	59.547	70.201
160	61.751	70.219
200	63.162	70.219

- Performing the experiment at 0° with 3.36 m/s air velocity we got the maximum value of heat transfer coefficient at condenser end obtained experimentally as 63.162 W/m^{2}K at 200ml/min mass flow rate of water whereas the minimum experimental value obtained is

57.631 W/m^2K. The empirical value stands almost constant having maximum value as 70.219 W/m^2K.

- The value of heat transfer coefficient is increasing because as we move from 80 ml/min to 200 ml/min heat at the evaporator zone will be more. So in order to dissipate the requisite heat condenser section has to do more work which ultimately leads to high value of heat transfer coefficient.
- The maximum deviation obtained in the above set of experiments were found to be 17.91% which was observed at 80 ml/min mass flow rate where minimum deviation observed was 10.05% which was found at 200 ml/min respectively.

5.1.16 At tilt angle 5° and air velocity 3.36 m/sec

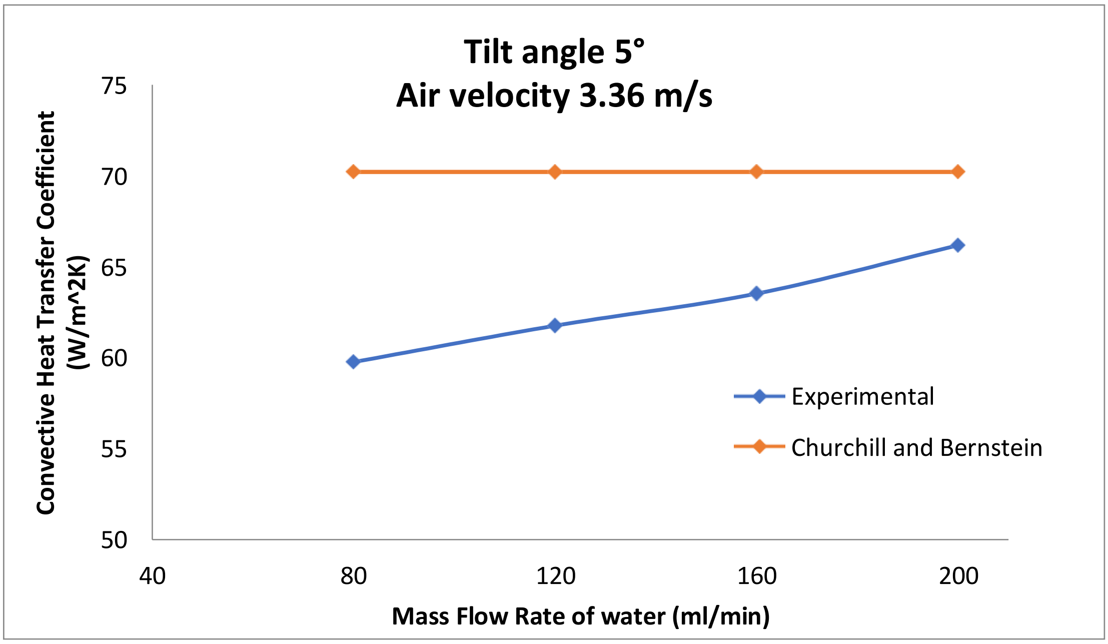

Figure 5.16. h at 5° tilt angle and 3.36 m/s air velocity

Table 5.16. Obtained values of h at 5° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^2k	Empirical value of heat transfer coefficient in W/m^2k
80	59.758	70.215
120	61.756	70.203
160	63.516	70.221
200	66.17	70.215

- The minimum value of heat transfer coefficient obtained in 5° tilt angle was observed at 80ml/min having experimental value of heat transfer coefficient 59.758 W/m^2K. The maximum value of heat transfer coefficient obtained experimentally was observed at

200ml/min having value of 66.17 W/m^{2}K. The empirical value stands almost constant having maximum value at 70.221 W/m^{2}K.

- The maximum deviation was observed to be 14.89% at 80 ml/min mass flow rate of water whereas the minimum deviation stands out at 5.76% at 200 ml/min mass flow rate of water.
- Similar trends in mass flow rate were seen as discussed above.

5.1.17 At tilt angle 10° and air velocity 3.36 m/sec

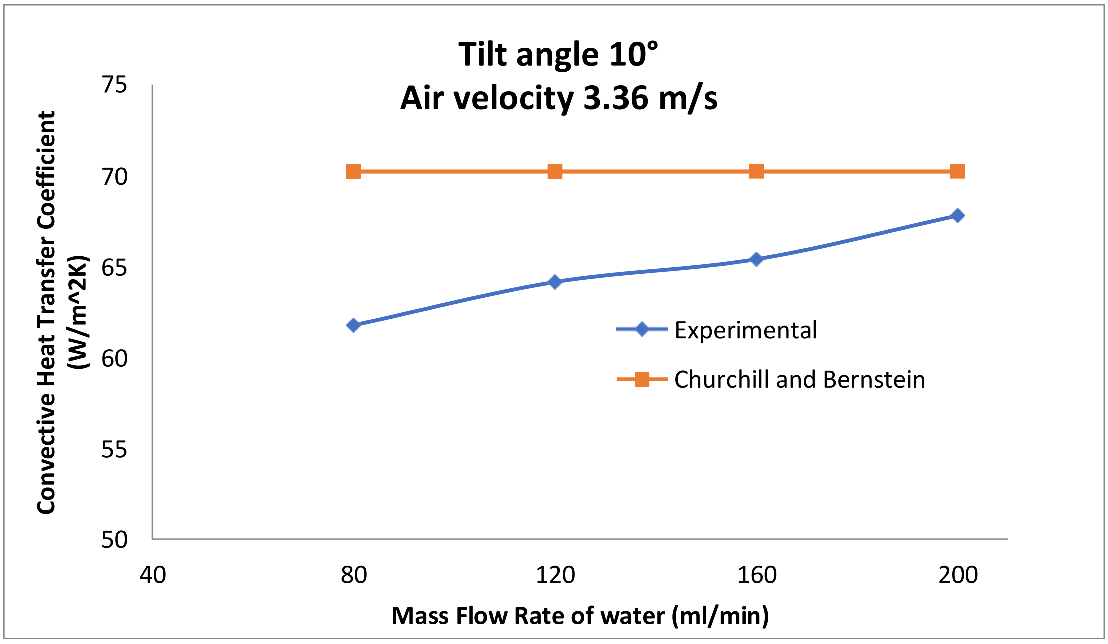

Figure 5.17. h at 10° tilt angle and 3.36 m/s air velocity

Table 5.17. Obtained values of h at 10° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^{2}K	Empirical value of heat transfer coefficient in W/m^{2}K
80	61.756	70.201
120	64.129	70.206
160	65.385	70.215
200	67.781	70.208

- The minimum experimental value at 10° tilt angle was 61.756 W/m^{2}K which was observed at 80 ml/min mass flow rate of water whereas the maximum experimental value of heat transfer coefficient of condenser section was observed as 67.781 W/m^{2}K at 200 ml/min mass flow rate of water.
- The empirical value stands almost constant having maximum value as 70.215 W/ m^{2}K.

- The maximum deviation was seen at 80 ml/min having value of 12.03% whereas the minimum deviation was observed at 200 ml/min having value as 3.46%.

5.1.18 At tilt angle 15° and air velocity 3.36 m/sec

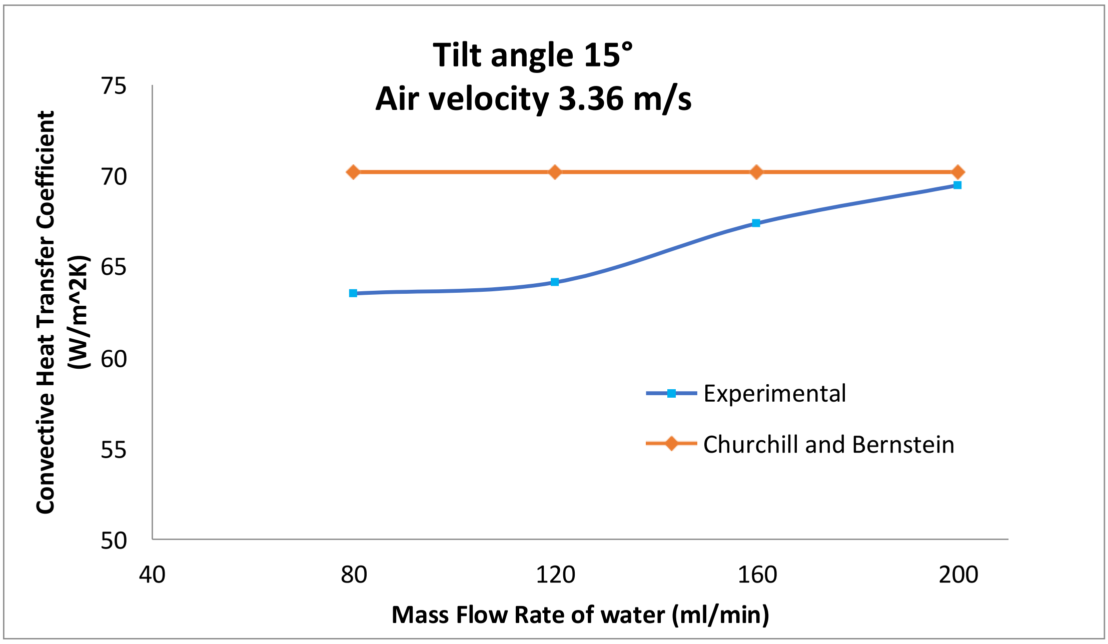

Figure 5.18. h at 15° tilt angle and 3.36 m/s air velocity

Table 5.18. Obtained values of h at 15° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^2K	Empirical value of heat transfer coefficient in W/m^2K
80	63.518	70.213
120	64.131	70.209
160	67.368	70.208
200	69.473	70.201

From the above-depicted graph 5.18, the following results were concluded: -

- Similar trends were seen with mass flow rate variation.
- At this inclination the maximum value of condenser side heat transfer coefficient obtained experimentally was noticed at 200 ml/min mass flow rate of water having value 69.473 W/m^2K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 63.518 W/m^2K.
- At this inclination the deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 9.53% to 1.04%.
- The empirical value stands almost constant having maximum value as 70.213 W/m^2K.

5.1.19 At tilt angle 20° and air velocity 3.36 m/sec

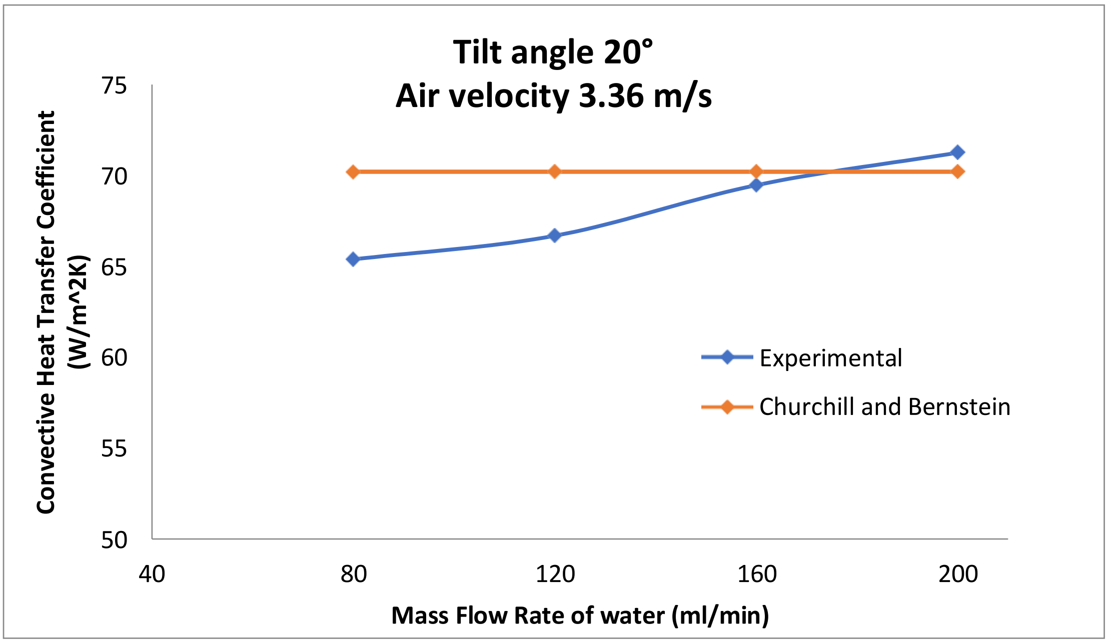

Figure 5.19. h at 20° tilt angle and 3.36 m/s air velocity

Table 5.19. Obtained values of h at 20° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	Experimental value of heat transfer coefficient in W/m^2K	Empirical value of heat transfer coefficient in W/m^2K
80	65.389	70.196
120	66.689	70.208
160	69.473	70.219
200	71.259	70.203

From the above-depicted graph 5.19, the following results were concluded: -

- Mass flow rate of water was observed to be linearly proportional to heat transfer coefficient across condenser end.
- At 20° tilt angle the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 71.26 W/m^2K whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 65.389 W/m^2K.
- At the indicated inclination the percentage deviation between empirical value and experimental value deduced by Churchill and Bernstein were observed to be within the range 6.85% to 1.5%.
- The empirical value stands almost constant having maximum value as 70.219W/m^2K.
- In the above-depicted graph, the experimental value is crossing the empirical value because at 20° tilt angle gravitational forces along with capillary forces inside the wick

will help the condensate to return to the evaporator section more quickly as compared to 0° tilt angle. Because of this thermal resistance decreases and heat pipe performance increases leading to high value of heat transfer coefficient.

5.1.20 At tilt angle 25° and air velocity 3.36 m/sec

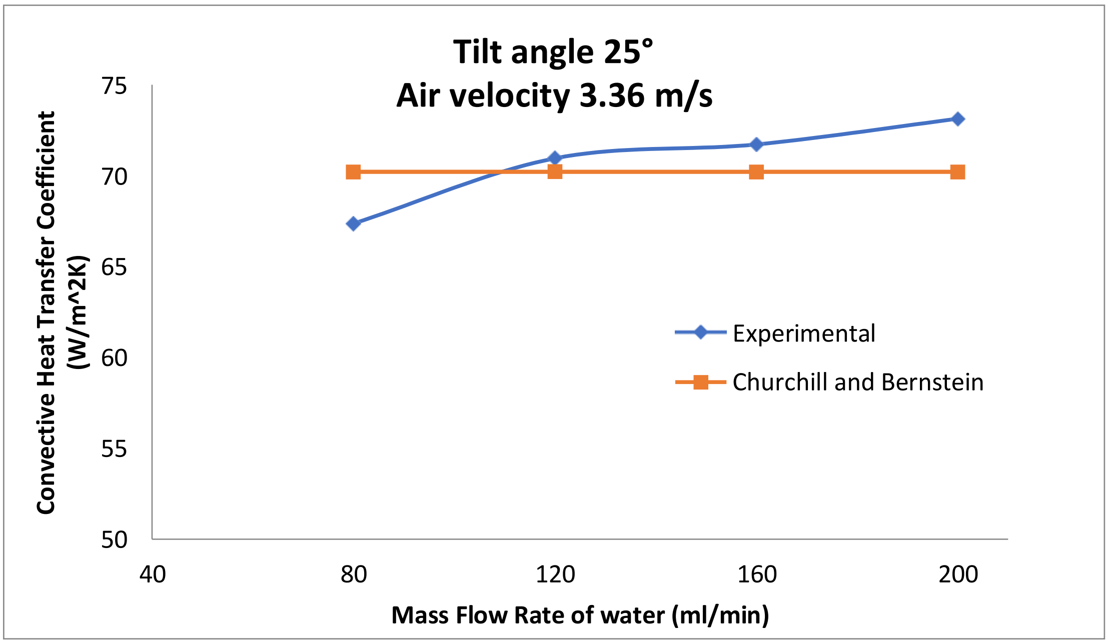

Figure 5.20. h at 25° tilt angle and 3.36 m/s air velocity

Table 5.20. Obtained values of h at 25° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}k**	**Empirical value of heat transfer coefficient in W/m^{2}k**
80	67.366	70.199
120	70.953	70.213
160	71.716	70.203
200	73.127	70.196

. From the above-depicted graph, the following results were obtained: -

- Similar trends were seen with mass flow rate variation.
- At 25° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 73.127 W/m^{2}K whereas the minimum value obtained experimentally was seen at 80ml/min having value as 67.366 W/m^{2}K.
- At 25° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 4.18% to 1.05%.

- The empirical value stands almost constant having maximum value as 70.213 W/m^{2}K.
- In the above-depicted graph, the experimental value is crossing the empirical value because at 25° tilt angle gravitational forces along with capillary forces inside the wick will help the condensate to return to the evaporator section more quickly as compared to 0° tilt angle. As a result of this thermal resistance decreases and heat pipe performance increases leading to high value of heat transfer coefficient.
- With fan speed of 3.36 m/s we obtained the maximum value of condenser side heat transfer coefficient as 73.127 W/m^{2}K which was highest at this fan speed.

5.1.21 At tilt angle 30° and air velocity 3.36 m/sec

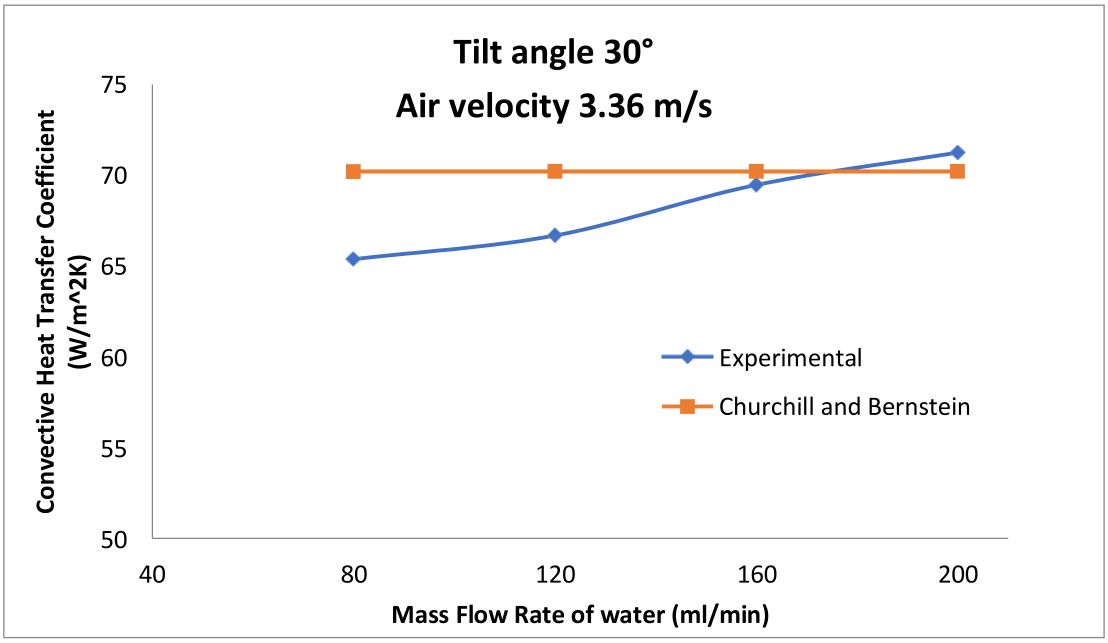

Figure 5.21. h at 30° tilt angle and 3.36 m/s air velocity

Table 5.21. Obtained values of h at 30° tilt angle and 3.36 m/s air velocity

Mass flow rate of water ml/min	**Experimental value of heat transfer coefficient in W/m^{2}K**	**Empirical value of heat transfer coefficient in W/m^{2}k**
80	65.381	70.206
120	66.686	70.208
160	69.473	70.21
200	71.255	70.221

. From the above-depicted graph in 5.21, the following results were concluded: -

- Similar trends were seen with mass flow rate variation.

- At 30° tilt angle, the maximum value of condenser side heat transfer coefficient obtained experimentally was observed at 200 ml/min mass flow rate of water having value 71.255 W/m^2k whereas the minimum value obtained experimentally was seen at 80 ml/min having value as 65.381 W/m^2k.
- At 30° tilt angle, the percentage deviation between experimental value and empirical value deduced by Churchill and Bernstein were observed to be within the range 6.87% to 1.05%.
- The empirical value stands almost constant having maximum value as 70.221 W/m^2k.
- In the above depicted graph, the experimental value starts decreasing prior to 25° tilt angle because beyond 25° tilt angle the condensate returns quickly back to the evaporator section causing flooding of evaporator section which ultimately decreases its performance leading to lower value of heat transfer coefficient that at 25°.
- The best thermal performance of heat pipe was observed in the range of 15-25° tilt angle.

Below table 5.22 depicts the experimental value of heat transfer coefficient at different tilt angles and different air velocities

Table 5.22 Condenser heat transfer coefficient (W/m^2 -K) of copper water heat pipe at different mass flow rate and tilt angles

	1.35m/s				2.34 m/s				3.34 m/s			
Θ^0	80 ml/mi n	120 ml/mi n	160 ml/mi n	200 ml/mi n	80 ml/mi n	120 ml/mi n	160 ml/mi n	200 ml/mi n	80 ml/mi n	120 ml/mi n	160 ml/mi n	200 ml/mi n
0	36.367 W/m^2 K	38.336 W/m^{2}K	40.239 W/m^{2}K	42.983 W/m^{2}K	47.905 W/m^{2}K	49.836 W/m^{2}K	51.495 W/m^{2}K	53.644 W/m^{2}K	57.631 W/m^{2}K	59.547 W/m^{2}K	61.751 W/m^{2}K	63.162 W/m^{2}K
5	38.597 W/m^2 K	39.401 W/m^{2}K	41.112 W/m^{2}K	43.778 W/m^2 K	49.044 W/m^2 K	51.497 W/m^2 K	54.205 W/m^2 K	57.219 W/m^2 K	59.758 W/m^2 K	61.756 W/m^2 K	63.516 W/m^2 K	66.17 W/m^2 K
10	40.237 W/m^2 K	42.981 W/m^{2}K	43.98 W/m^{2}K	45.462 W/m^2 K	51.498 W/m^2 K	53.27 W/m^2 K	55.671 W/m^2 K	57.217 W/m^2 K	61.756 W/m^2 K	64.129 W/m^2 K	65.385 W/m^2 K	67.781 W/m^2 K
15	42.027 W/m^{2}K	44.324 W/m^{2}K	45.028 W/m^{2}K	47.281 W/m^{2}K	52.816 W/m^{2}K	53.27 W/m^2 K	57.217 W/m^{2}K	59.879 W/m^{2}K	63.518 W/m^{2}K	64.131 W/m^{2}K	67.368 W/m^{2}K	69.473 W/m^{2}K
20	43.983 W/m^{2}K	45.755 W/m^{2}K	47.279 W/m^{2}K	49.25 W/m^2 K	54.209 W/m^{2}K	55.173 W/m^{2}K	58.85 W/m^2 K	61.303 W/m^{2}K	65.389 W/m^{2}K	66.689 W/m^{2}K	69.473 W/m^{2}K	71.259 W/m^{2}K
25	45.029 W/m^2 K	47.278 W/m^{2}K	49.768 W/m^{2}K	52.53 W/m^2 K	56.155 W/m^2 K	58.298 W/m^2 K	60.583 W/m^2 K	62.793 W/m^2 K	67.366 W/m^2 K	70.953 W/m^2 K	71.716 W/m^2 K	73.127 W/m^2 K
30	42.027 W/m^2 K	44.326 W/m^{2}K	46.126 W/m^{2}K	48.246 W/m^2 K	55.668 W/m^2 K	57.215 W/m^2 K	58.851 W/m^2 K	59.875 W/m^2 K	65.381 W/m^2 K	66.686 W/m^2 K	69.473 W/m^2 K	71.255 W/m^2 K

The graphs below depict the variation of experimental heat transfer coefficient with tilt angle Θ. It was observed that after a tilt angle of 25° there is a decrease in condenser heat transfer coefficient which was clearly seen at 30° tilt angle due to flooding of evaporator section by quick return of condensate back to evaporator section.

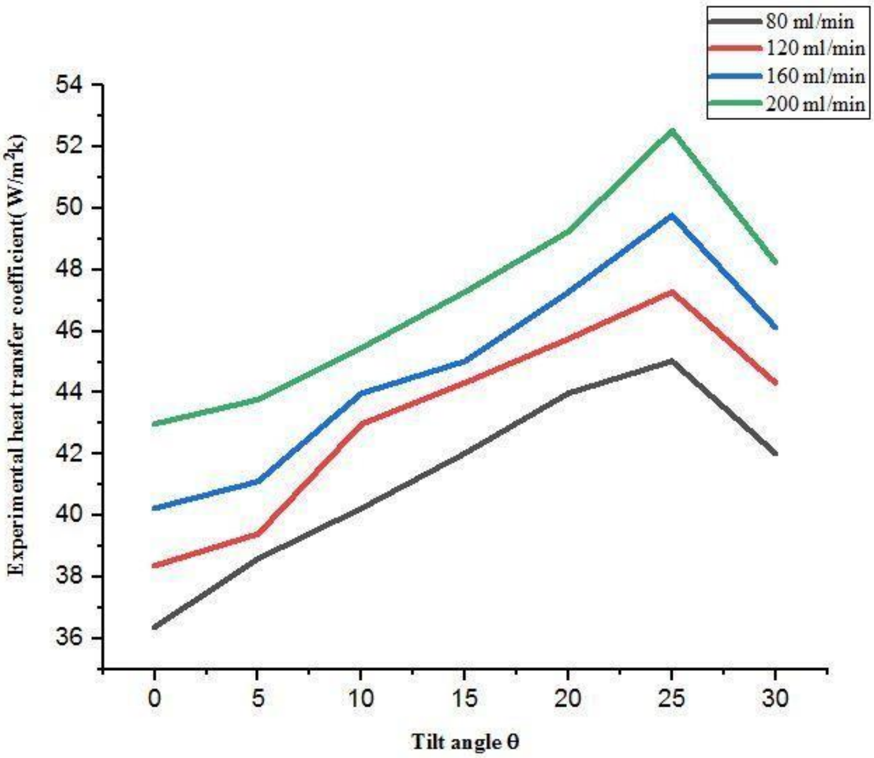

Figure 5.22. Variation of heat transfer coefficient with tilt angle and air velocity 1.35 m/s

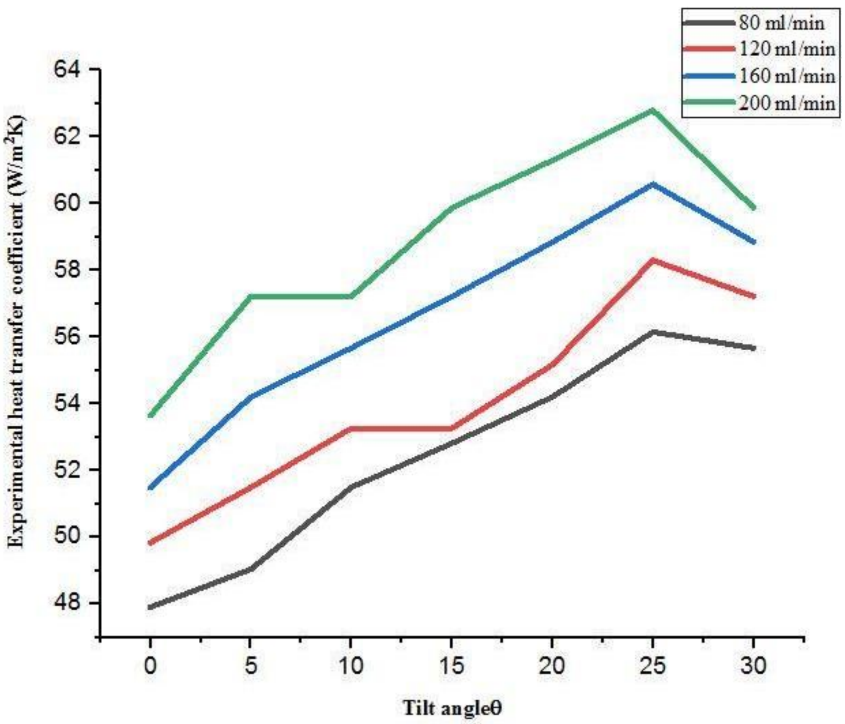

Figure 5.23. Variation of heat transfer coefficient with tilt angle and air velocity 2.34 m/s

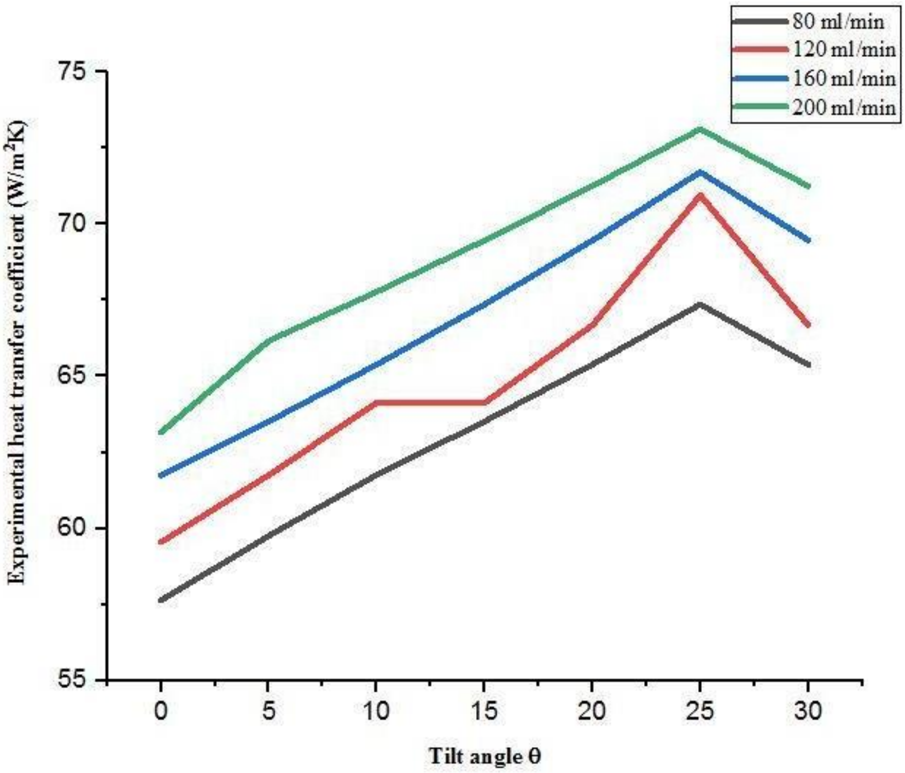

Figure 5.24. Variation of heat transfer coefficient with tilt angle and air velocity 3.36 m/s

Closure

Heat pipe are known to be best heat dissipating devices but at a certain tilt angle the thermal performance increases leading to higher heat transfer. It is suggested to operate heat pipe at a tilt angle where maximum thermal performance is attained. In this findings heat pipe were seen to exhibit maximum heat transfer coefficient across condenser end under forced convection at 25° tilt angle.

Chapter 6
Conclusions and Future Scope

This chapter depicts the conclusion drawn by conducting experimentation on Cu-H_2O heat pipe having annular fins under forced convection at various tilt angles. It was then compared with empirical correlations deduced by Churchill and Bernstein in forced convection. The noticeable observation can be obtained from present investigation are as follows

6.1 Conclusions

- With the increase in the mass flow rate, the rate of heat dissipation by heat pipe also increases.
- The heat transfer coefficient across condenser region having annular fins under forced convection increases with the increase in heat input on the evaporator region.
- The heat transfer coefficient obtained experimentally and empirically were in proper agreement with deviation no more than 18.55%.
- It was also investigated that tilt angle have significant effect on heat transfer coefficient. As tilt angle is increased from 0° to 25°, the heat transfer coefficient across condenser end increases gradually due to the assistance of gravitational forces along with capillary forces.
- However, beyond 25 the heat transfer coefficient starts decreasing because the flooding of evaporator section takes place, which weakens the performance of heat pipe leading to lower values of heat transfer coefficient.
- The maximum heat transfer coefficient was observed at 25° tilt angle having air velocity of 3.36m/s as 73.127W/m^2K.
- It was observed that heat transfer coefficient was proportional to velocity of air. Higher the fan speed more will be the heat transfer coefficient and more will be heat dissipation.
- It was observed that pressure difference across condenser end of heat pipe at air velocity of 1.35 m/s stands constant at a value of 4.98 N/m^2. It was also established that maximum pressure difference was observed at air velocity of 3.36 m/s possessing value of 8.31 N/m^2 and the said value nearly remains constant for particular air velocity. At an intermediate air velocity of 2.34 m/s, the pressure difference across condenser end holds a constant value of 6.64 N/m^2.

6.2 Future scope

Heat pipe has wide applications ranging from electronic industries to aviation industries. Lots of research work has been carried out in this field. The following are the suggested work for future scope.

- Heat pipes working on nanofluids by varying concentrations and different wick materials and shapes can be studied.
- To see effect of coagulation of nanofluids in heat pipes and their clogging in wick needs further study.
- Study of binary fluid heat pipes and effect of binary fluids on performance needs further investigation.

REFERENCES

1. Dunn, P.D. and Reay, D., 2012. *Heat pipes*. Elsevier.
2. R.S., Motors Liquidation Co, 1944. *Heat transfer device*. U.S. Patent 2,350,348.
3. Vasiliev, L.L., 2005. Heat pipes in modern heat exchangers. *Applied thermal engineering*, *25*(1), pp.1-19.
4. Tanaka, O., Yamakage, H., Ogushi, T., Murakami, M. and Tanaka, Y., 1982. Snow melting using heat pipes. In *Advances in Heat Pipe Technology* (pp. 11-23). Pergamon.
5. Hsu, R.T., Acer Inc, 2001. *Heat dissipation system for a laptop computer using a heat pipe*. U.S. Patent 6,288,896.
6. Faghri, A., 2014. Heat pipes: review, opportunities and challenges. *Frontiers in Heat Pipes (FHP)*, *5*(1).
7. Cengel, A., 2007. HEHT TRANSFER.
8. Reay, D., McGlen, R. and Kew, P., 2013. *Heat pipes: theory, design and applications*. Butterworth-Heinemann.
9. Cotter, T.P., 1984. *Principles and prospects for micro heat pipes* (No. LA-UR-84-120; CONF-840578-1). Los Alamos National Lab., NM (USA).
10. Maydanik, Y.F., 2005. Loop heat pipes. *Applied thermal engineering*, *25*(5-6), pp.635-657.
11. Song, F., Ewing, D. and Ching, C.Y., 2004. Experimental investigation on the heat transfer characteristics of axial rotating heat pipes. *International Journal of Heat and Mass Transfer*, *47*(22), pp.4721-4731.
12. Charoensawan, P. and Terdtoon, P., 2008. Thermal performance of horizontal closed-loop oscillating heat pipes. *Applied Thermal Engineering*, *28*(5-6), pp.460-466.
13. Charoensawan, P., Khandekar, S., Groll, M. and Terdtoon, P., 2003. Closed loop pulsating heat pipes: Part A: parametric experimental investigations. *Applied thermal engineering*, *23*(16), pp.2009-2020.
14. Incropera, F.P., Lavine, A.S., Bergman, T.L. and DeWitt, D.P., 2007. *Fundamentals of heat and mass transfer*. Wiley.
15. Senapati, J.R., Dash, S.K. and Roy, S., 2016. Numerical investigation of natural convection heat transfer over annular finned horizontal cylinder. *International Journal of Heat and Mass Transfer*, *96*, pp.330-345.
16. Samana, T., Kiatsiriroat, T. and Nuntaphan, A., 2014. Enhancement of fin efficiency of a solid wire fin by oscillating heat pipe under forced convection. *Case Studies in Thermal Engineering*, *2*, pp.36-41.

17. Noie, S.H., Sarmasti Emami, M.R. and Khoshnoodi, M., 2007. Effect of inclination angle and filling ratio on thermal performance of a two-phase closed thermosyphon under normal operating conditions. *Heat transfer engineering*, *28*(4), pp.365-371.
18. Teng, T.P., Hsu, H.G., Mo, H.E. and Chen, C.C., 2010. Thermal efficiency of heat pipe with alumina nanofluid. *Journal of alloys and compounds*, *504*, pp.S380-S384.
19. Idrus, F., Mohamad, N., Zailani, R., Wisnoe, W. and Abdullah, M.Z., 2014. Thermal Performance of a Cylindrical Heat Pipe For Different Heat Inputs and Inclination Angles. In *Applied Mechanics and Materials* (Vol. 661, pp. 148-153). Trans Tech Publications.
20. Kang, S.W., Wei, W.C., Tsai, S.H. and Yang, S.Y., 2006. Experimental investigation of silver nano-fluid on heat pipe thermal performance. *Applied thermal engineering*, *26*(17-18), pp.2377-2382.
21. Plawsky, J., 2017. Wickless heat pipes in microgravity. *Physics today*, *70*(9).
22. Jose, J. and Baby, R., 2018, August. Recent advances in loop heat pipes: A review. In *IOP Conference Series: Materials Science and Engineering* (Vol. 396, No. 1, p. 012060). IOP Publishing.
23. Huminic, G., Huminic, A., Morjan, I. and Dumitrache, F., 2011. Experimental study of the thermal performance of thermosyphon heat pipe using iron oxide nanoparticles. *International Journal of Heat and Mass Transfer*, *54*(1-3), pp.656-661.
24. Han, X., Wang, X., Zheng, H., Xu, X. and Chen, G., 2016. Review of the development of pulsating heat pipe for heat dissipation. *Renewable and Sustainable Energy Reviews*, *59*, pp.692-709.
25. Chan, C.W., Siqueiros, E., Ling-Chin, J., Royapoor, M. and Roskilly, A.P., 2015. Heat utilisation technologies: A critical review of heat pipes. *Renewable and Sustainable Energy Reviews*, *50*, pp.615-627.
26. Hudakorn, T., Terdtoon, P. and Sakulchangsatjatai, P., 2008. Effect of inclination angle on performance limit of a closed-end oscillating heat pipe. *American Journal of Engineering and Applied Sciences*, *1*(3), pp.174-180.
27. Grooten, M.H.M. and Van Der Geld, C.W.M., 2010. The effect of the angle of inclination on the operation limiting heat flux of long R-134a filled thermosyphons. *Journal of heat transfer*, *132*(5), p.051501.
28. Riehl, R.R. and dos Santos, N., 2012. Water-copper nanofluid application in an open loop pulsating heat pipe. *Applied Thermal Engineering*, *42*, pp.6-10.

29. Senthilkumar, R., Vaidyanathan, S. and Sivaraman, B., 2011. Performance investigation of heat pipe using aqueous solution of n-Pentanol with different inclinations. *Journal of mechanical science and technology*, *25*(4), p.923.
30. Moraveji, M.K. and Razvarz, S., 2012. Experimental investigation of aluminum oxide nanofluid on heat pipe thermal performance. *International Communications in Heat and Mass Transfer*, *39*(9), pp.1444-1448.
31. Ghanbarpour, M., Nikkam, N., Khodabandeh, R., Toprak, M.S. and Muhammed, M., 2015. Thermal performance of screen mesh heat pipe with Al2O3 nanofluid. *Experimental Thermal and Fluid Science*, *66*, pp.213-220.
32. Singh, S., 2018. *Experimental Investigation of Heat Pipe with Annular Fins Under Natural Convection At Diffrent Inclinations* (Doctoral dissertation).
33. Holman, J.P., 2001. *Experimental methods for engineers*.
34. Kumar, V., Gangacharyulu, D. and Tathgir, R.G., 2007. Heat transfer studies of a heat pipe. *Heat Transfer Engineering*, *28*(11), pp.954-965.

Annexure

Table A1: Temperature of heat pipe data at 0° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	60	58	51	50	49	45
120	61	60	49	47	46	44
160	60	59	50	48	47	43
200	61	60	51	49	49	45

Table A2: Temperature of heat pipe data at 5° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	60	58	51	49	48	44
120	60	59	49	47	46	43
160	61	60	48	47	46	43
200	61	60	51	49	48	46

Table A3: Temperature of heat pipe data at 10° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	61	59	51	48	48	46
120	60	59	49	47	45	43
160	60	59	49	48	46	44
200	61	60	51	48	49	46

Table A4: Temperature of heat pipe data at 15° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	61	59	49	48	47	44
120	61	60	49	48	46	42
160	60	59	50	47	48	43
200	61	60	52	50	49	47

Table A5: Temperature of heat pipe data at 20° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	61	59	53	51	49	45
120	61	60	51	49	45	42
160	60	59	51	48	45	43
200	60	59	52	50	49	46

Table A6: Temperature of heat pipe data at 25° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	60	58	53	51	48	45
120	61	59	54	52	50	47
160	61	60	53	51	48	43
200	60	59	52	50	49	47

Table A7: Temperature of heat pipe data at 30° tilt angle and 3.36 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	60	58	51	49	48	44
120	61	60	50	48	45	42
160	60	59	51	50	47	43
200	61	60	52	50	47	44

Table A8: Temperature of heat pipe data at 0° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	60	58	53	51	50	47
120	61	60	51	48	49	44
160	61	60	52	51	49	45
200	61	60	53	50	51	49

Table A9: Temperature of heat pipe data at 5° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	61	59	52	50	48	46
120	61	60	51	49	48	44
160	60	59	52	50	48	44
200	61	60	53	51	50	47

Table A10: Temperature of heat pipe data at 10° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	60	58	53	51	50	48
120	61	60	50	48	47	44
160	62	61	52	51	49	46
200	61	60	53	52	50	49

Table A11: Temperature of heat pipe data at 15° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	62	60	53	51	50	47
120	61	60	51	49	47	44
160	60	59	53	51	48	46
200	60	59	51	50	48	47

Table A12: Temperature of heat pipe data at 20° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	61	59	53	50	50	46
120	61	60	51	49	47	45
160	61	60	52	50	49	46
200	62	61	53	52	50	48

Table A13: Temperature of heat pipe data at 25° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	62	60	53	52	49	46
120	61	59	55	53	53	50
160	62	61	53	51	50	48
200	63	62	54	52	51	48

Table A14: Temperature of heat pipe data at 30° tilt angle and 2.34 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	62	60	53	51	50	47
120	63	62	51	50	47	44
160	63	62	54	51	50	47
200	64	63	54	52	51	48

Table A15: Temperature of heat pipe data at 0° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	63	62	51	48	46	44
120	63	62	53	51	49	46
160	63	62	54	52	51	46
200	61	60	55	54	53	52

Table A16: Temperature of heat pipe data at 5° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	61	59	54	51	52	49
120	62	61	54	53	51	47
160	62	61	53	51	51	49
200	63	62	55	52	53	51

Table A17: Temperature of heat pipe data at 10° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	63	61	55	53	53	50
120	64	63	53	51	50	47
160	63	62	54	52	51	48
200	64	63	55	54	53	51

Table A18: Temperature of heat pipe data at 15° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	64	62	55	54	53	50
120	64	63	52	50	49	47
160	64	63	53	51	49	47
200	64	63	53	50	51	49

Table A19: Temperature of heat pipe data at 20° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	64	62	54	52	51	48
120	64	63	52	51	49	46
160	64	63	53	50	50	48
200	63	62	54	53	52	50

Table A20: Temperature of heat pipe data at 25° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	62	60	53	51	52	50
120	64	63	52	49	48	46
160	63	62	54	53	52	50
200	64	63	53	52	51	48

Table A21: Temperature of heat pipe data at 30° tilt angle and 1.35 m/s air velocity

Mass flow rate ml/min	TS-1 °C	TS-2 °C	TS-3 °C	TS-4 °C	TS-5 °C	TS-6 °C
80	63	61	53	51	51	48
120	62	61	52	49	49	47
160	64	63	53	50	49	46
200	64	63	54	51	51	50

Publication based on Research work

- Mr. Chetan Sharma, Mr. Sumeet Sharma, Dr. Dasaroju Gangacharyulu, 2019, A Review of Heat Pipes: its Types and Applications, INTERNATIONAL JOURNAL OF ENGINEERING RESEARCH & TECHNOLOGY (IJERT) Volume 08, Issue 03 (March – 2019),

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