PERFORMANCE STUDY OF A PULSATED TWO-PHASE LOOP THERMOSYPHON

A Thesis

Submitted by

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Ashiq

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Dedicated to my parents

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ABSTRACT

Thermoloop, a kind of pulsated two-phase thermosyphon (PTPT), is a device with a promise of dealing with very high heat flux requirements for electronics cooling. It comprises of three main components: evaporator, condenser and a liquid reservoir. In the present study, the thermal performance of the thermoloop has been studied to address some issues on design and heat transfer performance both experimentally and analytically. These issues include variations in thermal load, evaporator fill ratio and condenser convection condition of thermoloop.

Outside dimension of the two evaporators employed in the test were 72.5 mm x 60 mm x 20 mm for Prototype-1 and 77 mm x 65 mm x 8 mm for Prototype-2 with inside volumes of 75 cm^3 and 30 cm^3, respectively. Same reservoir was used for the two prototypes with a liquid storage capacity of 80 cm^3. Most of the studies were conducted with prototype-1 using water as the working fluid.

For thermal load variation from 100 W to 250 W, the maximum temperature of the evaporator wall increased from 106^0C to 112^0C and minimum temperature of evaporator wall increased from 73^0C to 95^0C at an evaporator fill ratio of 30%. The values of other functional parameter such as cycle time, height of liquid column in the reservoir and condenser temperature also increased with an increase in thermal load. At a constant heat load and for the same working fluid, the effect of liquid fill ratio on the maximum temperature of the evaporator wall and on cycle time appeared to be insignificant. The addition of fan to cool the condenser increased the operational limit of the device from 125 W to 275 W. Further, the various functional parameters of the thermoloop attained a steady value after 5 to 6 heat transport cycles.

In the analytical part, a simple mathematical model on the thermoloop heat transfer concept is developed based on the hydrodynamics and heat transfer to have a better understanding of the physics involved, which needs to be tested or verified by the experimental data.

NOMENCLATURE

Symbols

A	Area of the forward and return tube
A_C	Condenser surface area
A_E	Evaporator surface area
b	Width of the reservoir
C_P	Specific heat of the fluid at constant pressure
D	Diameter of forward and return tube
F_B	Blasius friction factor
H	Height of liquid in the reservoir
h_{fg}	Enthalpy of vaporization
h_V	Enthalpy of vapor
$\Delta H_{1\text{-}2}$	Elevation Difference between evaporator and condenser
$\Delta H_{2\text{-}3}$	Elevation Difference between condenser and reservoir
$\Delta H_{3\text{-}1}$	Elevation Difference between evaporator and reservoir
L_{FL}	Length of forward tube
L_{RL}	Length of return tube
P_E	Evaporator pressure
P_C	Pressure in the condenser
P_R	Reservoir Pressure
ΔP_C	Pressure drop in the condenser
ΔP_G	Pressure drop due to non-condensable gases
ΔP_{FL}	Pressure drop in the forward line
ΔP_{RL}	Pressure drop in the return line
ΔP_M	Minor Loss
$\Delta P_{1\text{-}3}$	Frictional pressure drop in the forward and part of return line
$\Delta P_{3\text{-}1}$	Frictional pressure drop between reservoir and evaporator
ρ_L	Density of liquid
ρ_V	Density of vapor
μ_L	Viscosity of liquid at condenser outlet

μ_V	Viscosity of liquid at condenser inlet
q	Heat flux
Q	Volume of liquid in the reservoir
Q_C	Heat Dissipated by the condenser
Q_E	Heat taken by the evaporator
Q_{IN}	Heat input by the heater
Q_L	Heat loss to the ambient
Q_{PC}	Heat dissipated by the phase change of vapor
Q_R	Heat loss from the reservoir
Q_{SC}	Subcooling of the saturated liquid in condenser
$\dot{Q}$	Volume flow rate
R_{Th}	Thermal resistance
$\dot{m}$	Mass flow rate
t	Time
T_E	Temperature of the evaporator wall
T_{IN}	Temperature at the condenser inlet
T_{OUT}	Temperature at the condenser outlet
T_R	Temperature of fluid at the reservoir
T_S	Liquid saturation temperature
T_∞	Ambient Temperature
ΔT_{SAT}	Wall superheat
U	Overall heat transfer coefficient
V	Velocity of fluid flow
x	Quality of vapor
$\overline{x}$	Linear average quality of vapor
x_{IN}	Inlet quality of vapor
x_{OUT}	Outlet quality of vapor

Abbreviation

CLPHP	Closed Loop Pulsating Heat Pipe
CMOS	Complementary Metal Oxide Semiconductor

LHP	Loop Heat Pipe
LMTD	Log Mean temperature Difference
PHP	Pulsating Heat Pipe
PTPT	Pulsated Two- Phase Thermosyphon
THTC	Thermoloop Heat Transfer Technology
TIM	Thermal Interface Material

Subscripts

C	Condenser
E	Evaporator
FL	Forward line
IN	Condition at condenser inlet
L	Liquid
M	Minor loss
OUT	Condition at condenser outlet
PC	Phase change
R	Reservoir
RL	Return Line
S	Saturation value
SC	Sub-cooling
V	Vapor

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CHAPTER 1

INTRODUCTION

The major trend in the electronics industry in the recent years has been the development of faster and complex circuit technologies and higher packaging density. This drive towards more compact and faster technologies with shrinking system size has resulted in the tremendous increase of heat flux both at the chip and overall package levels. The significant evolution of the microprocessors has also resulted in the increase in the number of components to be integrated within the system and number of input/ output ports and other connectors. According to the International Technology Roadmap for Semiconductors, heat dissipation of high performance microprocessors is projected to approach the 160 W threshold within the next five years [1]. It also predicts that the resulting heat flux in certain regions of a microprocessor chip is projected to be over 100 W/cm^2. This rapid development in the electronics industry requires, for thermal management, high and efficient heat transfer in a small volume and space due to the closely packed microchips with higher power ratings.

Thermal management of electronic systems is one of the major focal points of the design system and is primarily concerned in keeping the temperature of the various components within a maximum allowable limit. It is argued that in the thermal management of electronic systems, the primary and secondary critical factors that the cooling solution must meet are the device's junction and solder temperatures. The choice of the cooling technique is thus determined not only by the power dissipation, but also by the junction temperature. A failure to maintain this temperature below the allowable limit results in the failure of the whole system. Therefore it is extremely important for effective thermal management of electronics system to precisely control the operating temperature of the critical components.

The most popular cooling devices for low heat dissipation systems today are the fans with heat sinks attached to the microprocessors, because of their low cost, ease of implementation, reliability and efficiency. Although the heat sinks are becoming more and more sophisticated in their design and coolant air velocities have increased significantly to meet heat dissipation requirements over the past decade, at certain point a limit will be encountered where nothing more can be done with a metal plate and fan. The possibilities and limitations of air cooling are discussed extensively in chapter 2.

But the close packaging of microchips and higher power ratings have increased the heat dissipation values to a critical level for high end electronics and computer processor cooling. The heat flux levels for the new designs of high density chips for the new generation of desktop computers has reached a value as high as 80 W/cm^2 [2]. From Fig.1.1 it can be seen that the projected chip heat flux can reach as high a value as 250 W/cm^2 in the year 2010 [3]. Again the board-to-board spacing has been decreased and consequently the available space for the thermal management of these devices has been reduced. Therefore it is readily understood that a challenge of keeping the processor cool with a small size heat sink is becoming more and more difficult.

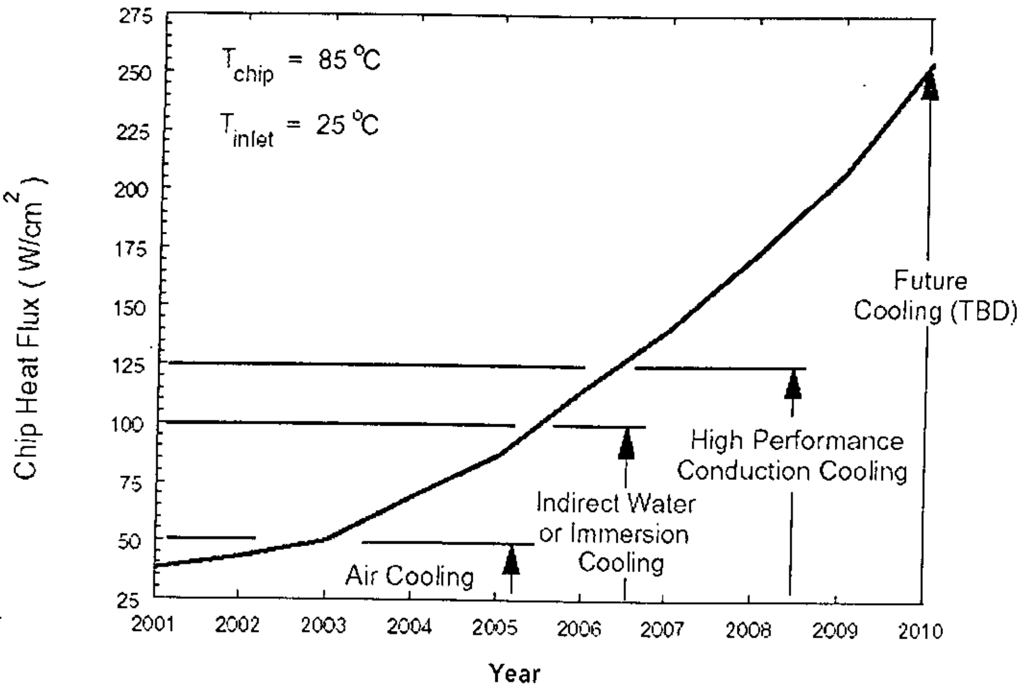


Figure 1.1: Projected Chip Heat Flux and Cooling Technology Limits (2002 thermal management roadmap- National Electronic Manufacturing Initiative – NEMI).

Therefore, new method of computer CPU cooling must be introduced with the capacity to dissipate as much as 200 W/cm^2 while keeping the temperature below 100^{0}C (preferably below 85^{0}C) and also ensuring very high uniformity of temperature over the surface [4]. Although there is no straight and simple answer to the question that whether the air cooling limit is reached, but it is fair to say that the currently available heat sinks are unable to meet the requirements and hampering the development of new generation computers. So searches are being made for highly efficient thermal management alternatives to handle this hugely multiplied heat loads of next generation electronic systems.

Thermoloop Heat Transfer Technology (THTT), also known as Pulsated Two-Phase Thermosyphon (PTPT) is one of the most recent technologies with the promise to be capable of dealing with very high heat flux requirements for electronics cooling and other high heat flux application [5]. It is a very simple, passive device and highly efficient operating against gravity with natural circulation and is capable to overcome the performance limitations of the indirect liquid cooling techniques like heat pipes and thermosyphons

1.1 MOTIVATION AND BACKGROUND

Today considerable interest and significant investment is being made on the search for a new, efficient cooling technology due to the increasing heat dissipation issue from the high end electronic systems. Future processors for high performance computers and servers have been projected to dissipate higher power in the range of 100 to 200 W/cm^2. The waste heat can accumulate and generate unacceptably high temperatures and thermal stress on the chip or in the package, resulting in reliability performance degradation and system malfunction. It is already an acknowledged fact that conventional air cooling techniques are about to reach their limit for cooling of high-end electronics. Experiments have shown that a maximum heat transfer coefficient of 150 W/m^2K can be reached with an acceptable level of noise, by using the standard fans [6]. This is equivalent to about $1W/cm^2$ for a 60^0C temperature difference. According to the various researches carried out by the leading electronics companies of the world like IBM Corporation and Philips, these limitations of the available standard techniques are mainly due to the limited thermal conductivity of air for convection and copper for conduction [7].

Again the limit of heat dissipation of air cooling by parallel plate fin heat sinks for a desktop computer application was reported by Saini and Webb [8] for a 16×16 mm heat source at a temperature difference of 35^0C between the inlet air and heat source. It was reported that the maximum allowable heat load to be 95 watt for a maximum chip heat flux of 37.2 W/cm^2. This can be increased by using a heat sink of greater size or increasing the air flow rate. But this would essentially increase the volume of package which is completely undesirable in this recent trend of miniaturization.

The comparison of heat transfer coefficient attainable for various electronic cooling techniques reveals that much higher heat transfer coefficients can be achieved by phase change liquid cooling than the conventional air cooling techniques. Therefore we can come into a simple conclusion that the future thermal management of electronic application will be based on liquid cooling techniques.

Because of their higher thermal conductivity and higher thermal capacity, liquids are significantly better heat transfer media than air. While air cooling technique is still used as the major means for cooling CMOS circuit technology, it is really being pushed to the limit. From Fig. 1.2 it can be seen that due to the demand of increased packaging density and speed, module heat flux level of recent CMOS circuit technologies has started to increase dramatically from the last decade [7]. Figure 1.2 also shows that the module level heat flux for CMOS technologies is now at the same level as the bipolar technology in the early 90's. But while all the bipolar systems employed some form of liquid cooling, most of the CMOS technologies continue to use air cooling schemes.

But as there is scope of marginal improvement with air cooling, many computer manufacturers are now taking course of liquid cooling. The Institute of Microelectronics (IME) predicts that existing air cooling techniques are reaching their limits due to the unfavorable thermophysical properties of air [7]. Liquid cooling technologies, on the other hand, greatly enhance the cooling capacity, as coolant liquids possess larger heat capacities, high thermal conductivities and higher energy efficiency [9]. Even though liquid cooling can be employed with or without boiling, boiling can greatly reduce the electronics chip temperature compared with single-phase liquid cooling.

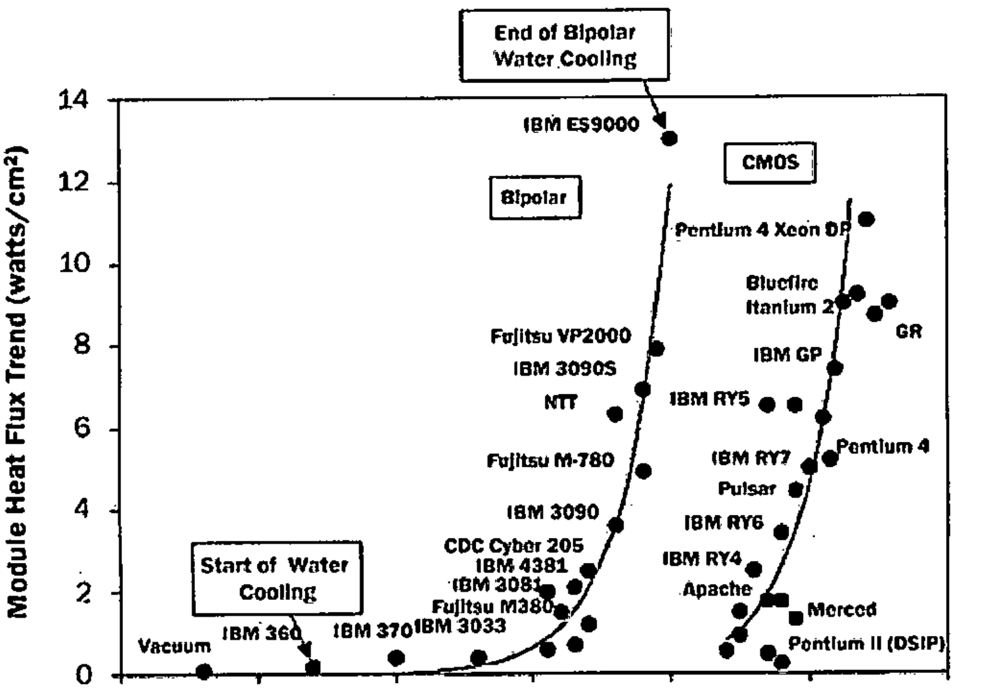


Figure 1.2: Evolution of module level heat flux in high-end computers [Source: IBM]

Liquid cooling techniques employed for cooling of electronic components can be classified into two broad categories- Direct Liquid Cooling and Indirect Liquid Cooling.

1.1.1 Direct liquid cooling

Direct liquid cooling can also be termed as direct liquid immersion cooling, because the liquid coolant comes into the direct contact of the components to be cooled. There is no wall separating the electronic chip and the surface of the substrate of the liquid coolant. High heat removal rate can be achieved by this method as the heat can be removed directly from the chip. Direct liquid immersion cooling has the advantage of greater uniformity of chip temperature than possible by air cooling.

The liquid immersion cooling technique has been used in microwave tubes in 1940s and also used later in cooling of high performance supercomputers [11]. But the implementation of direct immersion cooling in mainframe computers in the 1980s caused significant design complexities and associated increase in costs, which limits their prospect as a heat transfer technique for today's high density miniature electronics. One of the disadvantages associated with direct immersion cooling is that the coolant has to be compatible with the device. Water is the most effective coolant but in most cases water can not be used as the direct immersion cooling liquid due to its electrical and chemical properties. Impinging jets, droplets and sprays are attractive

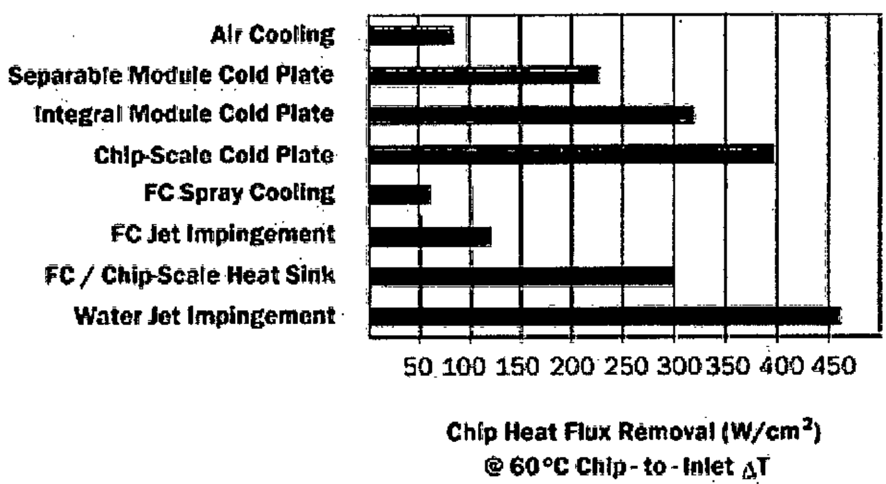


Figure 1.3: The heat removal capacity of different cooling techniques [10]

direct liquid cooling options for removing high heat fluxes because of their associated heat transfer coefficients .These conventional direct liquid cooling techniques are not equally comparable to the performance of indirect liquid cooling schemes as shown in Fig. 1.3. Only direct water jet impingement appears to be a better performer than the best indirect cooling available.

1.1.2 Indirect liquid cooling

Indirect liquid cooling technique is one in which the liquid does not directly contact the component to be cooled. In this method, a good thermal conduction is created between the microelectronic heat source and liquid cooled cold-plate attached to the module surface. Thermal interface materials (TIMs) can be used to increase this thermal conduction between the chip and the plate. In this method, water can be used as the working fluid to take the advantage of its superior thermophysical properties, as there is no direct contact with the components.

The indirect liquid cooling techniques of electronics are drawing more attention of the researcher's to avoid the design complexities associated with the direct liquid cooling. Indirect liquid cooling is capable of removing heat at a rate two to four times than that of air cooling [9]. The indirect liquid cooling techniques can be either Single Phase Forced Convection such as liquid cooled Cold Plates or Two- phase Forced convection such as Heat pipes and Thermosyphons.

Two-phase indirect heat transfer is a very appealing liquid cooling process as high heat flux removal can be realized through vaporization of the fluid in an evaporator attached to the heat source such as the microprocessor in computers. In the recent past, a number of electronic cooling schemes have been developed based on the two-phase heat transfer. Two-phase heat transfer, involving evaporation of a liquid in a hot region and condensation of vapor in a cold region, can provide the removal of much higher heat fluxes than can be achieved through conventional forced air-cooling. This is the reason why considerable researches are being conducted towards these approaches for thermal management of electronics. The most common forms of indirect phase change liquid cooling are the different types of- Heat pipe and Loop thermosyphon.

1.1.2.1 Heat Pipes

Heat pipes provide an indirect and passive means of heat transfer by liquid cooling. It is a simple device that can transfer heat from one location to another with very high thermal conductivity and often termed as 'superconductors' because of their excellent heat transfer capacity. Although the idea was first suggested in 1942 by R.S. Gaugler, it was in 1963 when G. Grover demonstrated the first heat pipe [2].

Heat pipes are sealed aluminum or copper containers which are partially filled with a liquid. The internal walls of the pipes and are lined with a porous medium (the wick) that acts as a passive capillary pump. When heat is applied to one side of the pipe the liquid starts evaporating. A pressure gradient exists causing the vapor to flow to the cooler regions. The vapor condenses at the cooler regions and is transported back by the wick structure, thereby closing the loop. The wick provides the capillary driving force by which the condensate is returned to the evaporator. A simple water-copper heat pipe will, on average, have a heat transfer capacity of 100 W/cm^2 [6]. Heat pipes are mainly used for cooling of electronic equipments such as laptop computers and in cryogenics and space technology.

Loop Heat Pipes: Loop Heat Pipes (LHPs) are particular kind of heat pipe where the condenser and the evaporator are separated and the working fluid is transported between them via tubings. In LHPs, unlike typical heat pipes, the wick structure exists only in the evaporator section [12]. LHPs possess all the advantages of the conventional heat pipes and provide reliable operation over long distance at any orientation. These devices can be considered as one of the most promising thermal control technologies for ground based applications as well as space applications. At present, different designs of LHPs ranging from powerful large size LHPs to miniature LHPs (micro loop heat pipe) have been developed and successfully employed in a wide sphere of applications. The most common working fluids used in LHPs are anhydrous ammonia and propylene. But other options, such as acetone and methanol which represents fewer hazards during manipulation and reduced distillation costs have also been used.

Pulsating Heat Pipes: Closed Loop Pulsating Heat Pipes, which is also known as Meandering Capillary Tube Heat Pipe or Closed Loop Oscillating Heat Pipe, has

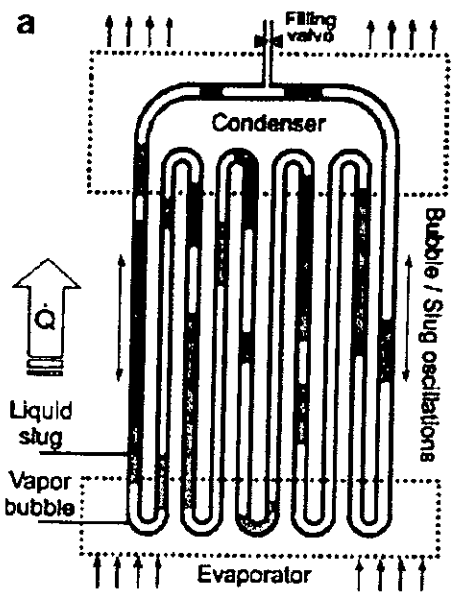


Figure: 1.4 Schematic of closed loop pulsating heat pipe

emerged in the recent years as a new electronics cooling technology. The Pulsating Heat Pipe is an innovating technology that has gained attention in the last 5 years [2]. Through the movement and phase changes in vapor bubbles and liquid slugs, heat is transferred from the evaporator to the condenser, as shown in Fig. 1.4. Higher operating temperatures are achieved when the PHP operates at the vertical orientation, while at horizontal orientation, the operating temperatures are lower. Pulsating heat pipes are capable of higher heat dissipation at relatively lower temperature difference between the evaporator and condenser [13]. The working principle and characteristics of LHPs and PHPs are discussed in detail in chapter 2.

Heat pipes are excellent heat transfer devices but their application is mainly confined to transfer relatively small heat loads over short distances. This limitation of heat pipes is mainly due to the major pressure losses associated with the liquid flow through the porous structure, called the entrainment losses. For the applications involving transfer of large heat loads over long distances, the thermal performance of the heat pipes is badly affected by increase in these losses. For the same reason, conventional heat pipes are very sensitive to the change in orientation in gravitational field. For the unfavourable slopes in evaporator-above-condenser configuration, the pressure losses due to the mass forces in gravity field adds to the total pressure losses and further affect the efficiency of the heat transfer process.

Thus the performance of the heat pipe may be insufficient to meet the continuous need for higher heat removal capacities over longer distance between the processor and the ultimate heat removal location. In addition, the high cost associated with heat pipes in comparison with the conventional forced air cooling places a limit on their use commercially.

1.1.2.2 Loop Thermosyphons

Loop thermosyphon is regarded as a very promising solution for the high end electronics cooling because of its ability to meet the requirement of very high heat flux dissipation and also because of their promise to represent a low cost solution. A simple thermosyphon is a hermetically sealed chamber which is partially filled with a volatile working fluid. A simple thermosyphon differs from a heat pipe because condensate is returned to the evaporator of a thermosyphon by the pressure difference due to the difference in height rather than by capillary force as for a heat pipe. The driving force (pressure) must be greater than the sum of the pressure losses due to wall friction experienced by both vapor and liquid in the loop and other minor losses in fitting, curves etc.

A thermosyphon successfully implements two-phase liquid cooling by indirect contact with electronics. A two-phase thermosyphon basically consists of an evaporator and a condenser, which are connected through a passage, or a loop as shown in Fig. 1.5. The tube carrying the vapor from the evaporator is called rising tube or forward tube and the other tube connecting the condenser and evaporator is called the return tube or the falling tube. The fluid vaporizes in the evaporator as heat is transferred from the source to the evaporator. The vapor then moves to the condenser through the tubing where it condenses. The released heat is dissipated into the ambient from the condenser and the condensed liquid is returned to the evaporator, thus completing a loop. The density difference between the liquid and vapor creates a pressure head, which drives the flow through the loop and no other driving force is needed. Thus the major limitation of miniature electronic applications cooled by loop thermosyphons is that they are not capable to heat transfer and mass downwards, i.e. their performance is dependent on orientation.

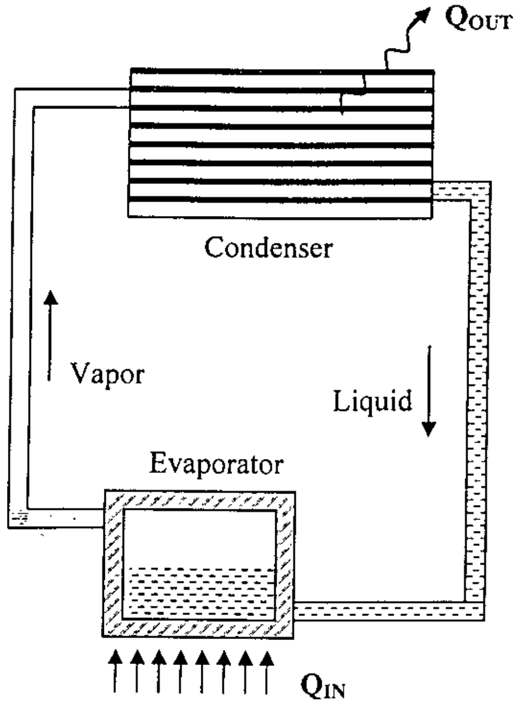


Figure 1.5: Schematic of an Indirect Thermosyphon Loop

The advantages of thermosyphons are as follows-

- Thermosyphon is a passive heat transfer device and it has no mechanical parts, which significantly improves its reliability issues. No energy is added to circulate the fluid and transfer the heat to the condenser by virtue of a pump.
- There is no geometric constraint on the shape of the evaporator. In some applications, this reduces the conduction resistances in transferring heat to the working fluid.
- There is no capillary limit in thermosyphons impeding the return of condensate as in the case of heat pipe. Thermosyphons can, therefore, move heat large distances and dissipate it efficiently unlike the heat pipes.
- Its quiet operation, in addition to the reduction in noise by replacing the heat sink fan, is a positive step for consumer and office desktop systems.
- It has flexibility in design and integration. The connection from evaporator to the condenser can be quite flexible, allowing a thermosyphon of a given design to function in many geometric configurations.

- When compared with liquid phase pumped cooling technologies, a thermosyphon has the advantages of simplicity, small size and low cost.

1.2 PULSATED TWO PHASE THERMOSYPHON (PTPT)

A Pulsated Two-Phase Thermosyphon (PTPT) named as 'Thermoloop' by Alam Thermal Solution [5] is a very recent technology with the promise of dealing efficiently with very high heat flux requirements for electronics cooling and other high heat flux application. It is a very simple and highly efficient technology and its operation is independent of gravity. It has the capability to overcome the performance limitations of the indirect liquid cooling techniques like heat pipes and simple loop thermosyphons.

The Thermoloop or PTPT consists of an evaporator connected to the heat source, a condenser which can be placed anywhere respect to gravity and a flexible or adjustable reservoir, separated from the evaporator as shown in Fig. 1.6. The evaporator is connected to the condenser through a vapor line and to the reservoir through a liquid return line. There are two check valves that control the flow in a fixed direction. The line that transports the vapor from the evaporator to the condenser is called 'the vapor line' or 'the forward line' and the line which brings back the liquid from the condenser to the evaporator through the reservoir is known as 'the liquid line' or 'the return line'. One check valve is placed in the return line between the condenser and the reservoir.

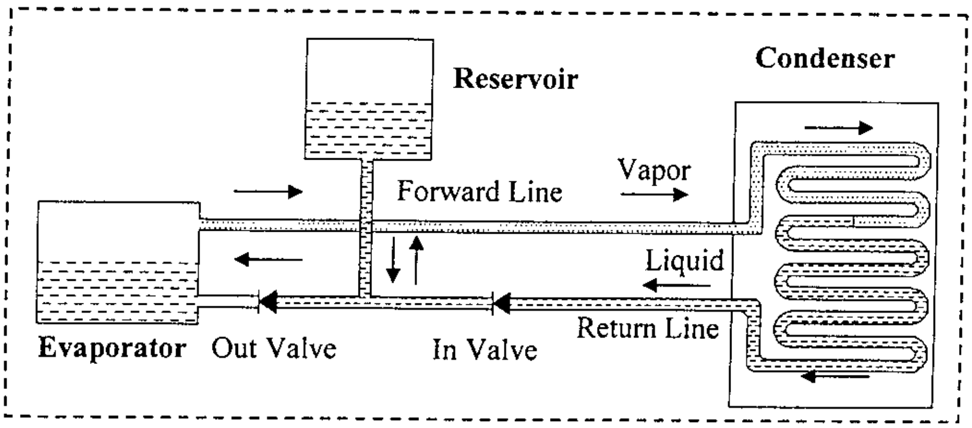


Figure 1.6: Simple schematic diagram of Thermoloop

The other check valve is inserted also in the return line between the reservoir and the evaporator. Unlike the heat pipes like LHPs, there is no wick structure in the thermoloop.

The important features of thermoloop are as follows

- Thermoloop can function against gravity where the circulation of the working fluid is forced without ant external work, but by the oscillation of pressure.
- Thermoloops are very similar in working principle with loop heat pipes and pulsating heat pipes.
- It has the capability to overcome high pressure drops with simple technologies, thus is capable to transport heat over a longer distance than heat pipes.
- As the Thermoloop device is very compact, can dissipate heat over longer distances and has a low cost of realization, it can be a very good choice for computer cooling and future electronic cooling demand.
- The device has no complex parts or manufacturing processes involved and can be made of cheap materials.
- It can be applied for wide range of cooling application ranging from cooling semiconductors, air conditioning to thermal management of outer space components.

When heat is supplied to the evaporator partially or fully filled with the working fluid, the temperature of the liquid starts to rise and when it exceeds the boiling point, the liquid starts evaporating and the pressure inside the evaporator increases. The device then starts operating in a periodic fashion by repeating the following two states-

a. Vapor transfer state
b. Condensate return state

(a) Vapor Transfer State: The liquid in the forward line and the condenser does not move forward until the pressure inside the evaporator is higher than the condenser. This state begins when the liquid inside the evaporator starts evaporating and the evaporator pressure starts to increase. The liquid in the forward line is pushed by the elevated pressure towards the condenser and then from the condenser to the reservoir where it is stored. The vapor can not move through the return line because it is closed by the check valve. Thus as the evaporator pressure increases, the condensed liquid moves into the adjustable reservoir, which accommodate the incoming liquid without increasing the system pressure. This process of pushing the liquid out of the condenser and forward line to the reservoir and replacing them by vapor from the evaporator continues as long as the rate of evaporation is higher than the rate of condensation. In this way the evaporator is gradually being empty while the reservoir is being filled and this continues until all the liquid inside the evaporator is vaporized. The liquid stored in the reservoir can not move back into the evaporator during this state because the one way out valve is closed as the pressure inside the evaporator is still.

(b) Condensate Return State: As only a fixed volume of the liquid is transported into the reservoir, the vapor transfer state ends and the liquid stored in the reservoir has to return to the evaporator to complete the cycle. This state is termed as the condensate return state. The pressure gradient between the reservoir and the evaporator must now change its sign so that the return of the liquid from the reservoir is possible. This means that either the pressure in the accumulator must increase or the pressure in the evaporator should decrease. In the thermoloop device, the pressure inside the evaporator decreases to the saturation pressure corresponding to condenser temperature as there is no liquid inside the evaporator.

Thus the pressure in the evaporator decreases to a lower value than the pressure in the reservoir. So the cold liquid from the reservoir enters into the evaporator as the out valve is now opened. In this way the cycle is completed. Thus, this device uses a two-phase fluid and operates in a periodic fashion with two sequential states of vapor transfer and condensate return and thus with periodic pulsation of pressure. This is why the device is termed as Pulsated Two Phase Thermosyphon or PTPT.

1.3 SCOPE OF THE PRESENT WORK

Thermal Management Roadmap- 2002, published by the National Electronic Manufacturing Initiative (NEMI) has identified the two phase loop thermosyphons as one of the most promising cooling technologies and selected it as an advanced cooling technology development activity [3]. This motivation of developing an efficient future electronic cooling solution has led to the development of thermoloop technology. In this current work, the thermal performance of the thermoloop has been studied to address some of the key design and performance issue for the practical implementation of the device.

Though quite a few experimental investigations have been performed till date with loop thermosyphon as the means of electronic component cooling, but most of them have been done with big size apparatus. Very few experimental tests have been performed on the realization and feasibility of thermoloop as a cooling device for high end electronics cooling. This work proposes the experimental investigation of thermoloop device to determine the influence of various geometric and functional parameters on the performance of the thermoloop.

There is a considerable lack of knowledge on how the various geometric and functional parameters influence the performance of thermoloop device and how to address those problems. In this work, an analytical model of the working principle of thermoloop has been proposed for better understanding of the mechanism. Though there are very few previous works on the effect of fill ratio and condenser condition on the thermal performance of heat pipes, no such comprehensive work was found on the heat transfer performance of thermoloop. One of the major objectives of this work was to determine how the evaporator fill ratio and condenser convection condition influences the heat transfer performance of the thermoloop. Though tests were conducted for two working fluids (water and ethanol) and two prototypes of the thermoloop device, the present study represents the detailed experimental study on the influence of these variations for prototype-1 with water as the working fluid.

Indirect liquid cooling techniques are now considered as the potential solution to break the thermal barrier of high heat dissipation of next generation electronics components. Therefore, the present work also addressed the issue of the heat transfer performance of the thermoloop at various thermal loads. With the aim to understand the physics of this system, variation of other functional and system parameters were also analyzed. The outcomes of this study have also enabled us to understand the effect of different performance indicators on the heat transfer characteristic of the thermoloop in order to find the optimal design configurations.

CHAPTER 2

LITERATURE REVIEW

The following are the conclusions of the Thermal Management Roadmap- published by the National Electronic Manufacturing Initiative (NEMI) in 2002 [3]

- Chip size may decrease with continued increase in circuit density resulting in higher heat flux.
- All new electronic products will most likely be air-cooled, including most computers, for the next few years.

- Portable (laptop) computers will need enhanced cooling technology in the near future despite the emphasis on low power dissipation.
- Cost will be a significant challenge for all future thermal designs and the speed to accomplish new designs will be vital to their success.
- High heat flux cooling capability is required for all high performance electronics.
- High thermal conductivity interface material is needed for heat sink applications.
- New cooling technology/system will be needed to handle increased heat load at product level.

According to this thermal management roadmap, significant improvement of the cooling technology will be needed in the area of indirect and direct liquid cooling along with several other techniques. From the point of view cooling of micro electronics components, two phase heat transfer devices seem to be the most suitable and promising scheme because of their high cooling capacity, greater energy efficiency, possible miniaturization of the device, lower maintenance requirement, safety, low cost and reliability. Different passive two-phase devices such as wicked heat pipes (flat heat pipe and loop heat pipe), wickless heat pipes such as pulsating heat pipes and wickless two-phase loop thermosyphons represent the most promising alternative for thermal control of electronic equipments. For this reason, a number of studies are being conducted to realize the applicability of these devices over the past few years.

2.1 STUDY OF LHPs AND PHPs

During the last decade, extensive researches have been carried out on a number of small scale prototypes of heat pipes on the applicability of those for electronics cooling. Loop heat pipes (LHPs) and pulsating heat pipes (PHPs) have been the most studied capillary driven devices with the ability to operate against gravity.

There has been an increasing interest in LHPs for cooling of high end electronics. The first actual application of LHPs in electronic cooling dates back to the end of seventies, when they were used for cooling of a unit of powerful transistors. Cooling of laptop computers had been the new sphere of LHPs application, owing to the development of miniature, compact, fairly efficient and cost effective devices [2]. The first experience in this direction came in 2001, when a number of compact coolers created on the basis of LHP dissipated a heat of the level 25-30 W from a laptop CPU [3].

LHPs for cooling of notebook computers are usually miniature size, with outer diameter from 2 to 4 mm for circular shape and thickness from 2 to 4 mm for flat rectangular shape. Copper tube and water filling are the best possible combination so far on the ground of compatibility and thermophysical properties [14]. Now copper-water LHPs with a flat oval evaporator 3.8 mm in thickness which can dissipate up to 100 W are being used largely in the laptop computers.

In the recent years, pulsating heat pipe (PHP), also called oscillation heat pipe (OHP) or meandering heat pipe, a particular wickless device capable of operating irrespective of gravity has emerged as a new cooling technology. Since the invention of PHP by H. Akachi in 1990, its operating principle and mechanism has been the subject of extensive research. The high heat flux removal capacity shown by some prototypes of PHPs has increased the interest in these devices. PHP has no complicated wick structure, is easy to produce and is capable of transporting higher heat rates at a relatively lower temperature difference between the evaporator and the condenser [14].

Wang et al. [15] reported an experimental study on the operational limitation of closed loop pulsating heat pipes (CLPHPs) for three operational orientations. They investigated the effects of inner diameter, operational orientation, filing ratio and heat input flux on thermal performance and performance limitation of CLPHPs. The CLPHPs were reported to obtain the best thermal performance and maximum performance limitation when they operated in the vertical bottom heat mode with 50% filling ratio.

2.2 STUDY OF LOOP THERMOSYPHONS

In spite of the excellent thermal performance shown by the different heat pipes, their relatively high cost in comparison with the conventional forced air cooling with heat sinks place a limit on their use commercially. In this regard, concentration is increased on the development of two phase heat transfer devices without any capillary structure such as two phase loop thermosyphons. In the recent years, there has been tremendous improvement in the designs of loop thermosyphon which has drawn considerable interest in them.

Gavotti and Polasek [16] have given a detailed outline of the applicability of loop thermosyphon for the thermal control of electronic equipments. They have shown that loop thermosyphons are able to dissipate heat fluxes from electronic equipments up to a maximum value of 70 W/cm^2, depending primarily on the choice of the working fluid.

Tuma and Mortazavi [17] reported the test results on the use of thermosyphons for electronics cooling for two different test conditions. They tested with an evaporator of $40\times40\times4$ mm^3, made of copper and two different condensers having volume of 440 cm^3 and 220 cm^3 respectively. The peak power attained by the thermosyphons was 170 W for the 1^{st} condenser (volume of 440 cm^3) with two fans of 2800 rpm and 190 W for condenser volume of 220 cm^3 and two fans of 4200 rpm. The results are very competitive with emerging forced convection technologies.

Krustalev [18] also reported the test results of using loop thermosyphons for electronics cooling. The tests were carried out for an evaporator and condenser of same dimension (65×90 mm) where the condenser was located about 60 cm above the evaporator. The tests were performed with methanol as the working fluid. He reported a maximum heat flux removal of 16 W/cm^2, when the temperature gradient between the evaporator and the condenser was 22^0 C. Similar works are also reported on the possibility of loop thermosyphon for electronics cooling by Palm and Tengblad [19].

A comprehensive experimental and numerical study on the performance of a compact thermosyphon for desktop computer cooling was reported by Pal et al. [4], in association with Maryland University and Hewlett Packard. They performed the actual implementation of a compact thermosyphon on a current microprocessor (Fig. 2.1). Their work described the design, construction and performance assessment of a two-phase compact thermosyphon for a Hewlett Packard Vectra VL800 PC with a peak microprocessor power dissipation of 80 W.

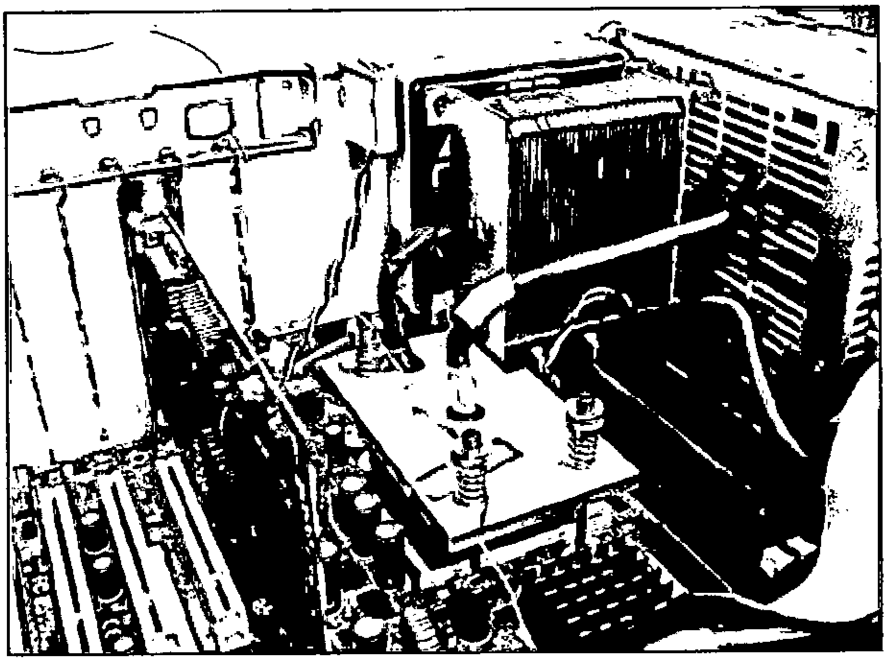


Figure 2.1: Hewlett Packard Vectra PC with thermosyphon assembly [4]

The thermosyphon for the Vectra included an evaporator of 3.2 x 3.2 x 2.9 cm and a condenser 2.6 x 8.2 x 7.5 cm. The condenser was cooled by the system fan of 9.2 x 9.2 cm. Using de-ionized water as the working fluid, the temperature of the evaporator base plate bottom was measured at 57°C under a chip power of 85 W and a local ambient temperature of 23°C. Since the specification called for a maximum case temperature of the CPU chip below 70°C, the thermosyphon was considered to be a great success.

2.3 STUDY ON CHOICE OF WORKING FLUID

One of the advantages of using a thermosyphon loop than immersion boiling is that the fluid may be chosen more freely as the liquid is not in direct contact with the components during normal operation. One may thus chose a fluid which needs small diameters of tubing and which gives low temperature differences in boiling and condensation and allows high heat fluxes in the evaporator. A thermosyphon may also be hermetically and permanently sealed which reduces the risk of leakage and allows the use of fluids with higher vapor pressures.

Palm et al. [20] has reported the results of the influence of the choice of working fluid on the design and performance of loop thermosyphon. They also reported how these goals are fulfilled for the case of closed external two phase thermosyphon loops. They found that high-pressure fluids gives better performance and more compact designs as high-pressure results in higher boiling heat transfer coefficients and smaller necessary tube diameter.

Apart from the properties related to the performance of the loop thermosyphon (pressure drop, heat transfer coefficients of boiling, condensation heat transfer etc.), there are several other requirements which should be met, as identified by Palm et al. are

- The fluid should not be harmful to people during production, normal operation or in case of a breakdown (sudden leak, fire, etc).
- It should not be harmful to the equipment in which it is installed. This means that it should not be explosive or flammable, not corrosive or otherwise incompatible with the materials of the equipment.
- It should not be harmful to the global environment. This means that it should have zero ozone depletion potential (ODP), it should not contribute to the greenhouse effect, not be hazardous to animals or plants or have decomposition products, which have such effects. Preferably, it should be a naturally occurring substance to eliminate the risk of unknown environmental effects.
- From an operational point of view, the fluid should be able to withstand the environment of the equipment for a long period of time without decomposing.
- Finally, it should have a low price and be readily available.

It is necessary to make some kind of compromise as no fluid meets all these requirements. This is done by considering various geometric and functional parameters of the system such as size, material, amount of heat to be transferred, application etc.

2.4 STUDY OF THERMOLOOP OR PTPT

The major limitation associated with the possible application of loop thermosyphon for micro electronics cooling is that they are not gravity independent i.e. the condenser must be placed at a higher location with respect to evaporator. For this reason, some special two phase thermosyphon devices, operating in pulsating manner and capable of transporting heat and mass downwards were developed in the recent past, primarily for solar heating and cooling.

The idea of a Pulsated two phase loop thermosyphon, a particular pump-less heat transfer device operating against gravity without any capillary structure was first realized by Sasin et al. [21] in the Moscow Power and Engineering Institute. They, in fact, started this work with the idea of realizing some unsteady heat transfer device for the application of exploiting geothermal and solar energy [22]. Their invented device, operating on the principle of periodic pulsation of pressure and without any capillary structure, was able to transport heat and mass downwards with the condenser located below the evaporator.

2.4.1 Analytical study of PTPT

Then Filippeschi [23] analyzed all the periodic two phase heat transport devices that operate against gravity and given these particular devices the generic name of Periodic two phase thermosyphon (PTPT). He reported a detailed analysis of the classification criteria of all the PTPT devices considering a large number of applications and their operating modes. The complex mechanism involved in the heat and mass transfer of PTPT devices was also explained using a mathematical code.

Filippeschi classified the PTPT device mainly on the basis of different modes in achieving the return of the liquid back into the evaporator from the condenser. He divided the all PTPT devices in three main categories

▪ **Group A: The liquid returns by a decrease in the pressure in the evaporator:** In this kind of device, the liquid stored in the reservoir or accumulator returns to the evaporator because the pressure inside the evaporator decreases. It can be obtained in two different ways: by periodically interrupting the supply of heat to the evaporator or by emptying the liquid inside the evaporator. The first one is typical of solar application as the solar heat input is discontinuous and periodic by nature.

The second type of device was first developed by Tamburani [24], who noted that a natural decrease in pressure occurs inside the evaporator when all the liquid has evaporated and the evaporator is empty. The pressure falls to the saturation pressure relative to the temperature of the cold sink, which is lower than the pressure inside the reservoir. But this kind of device presents a problem that when the first droplets of the condensed liquid return to the empty evaporator, they fall directly on the overheated surface and get evaporated. So the pressure inside the evaporator again increases and remaining liquid collected in the reservoir can not return to the evaporator to complete the heat transport cycle. The solution proposed for this problem is to use an intermediate vessel (delaying tank) between the reservoir and the evaporator, connecting it with the evaporator through a hydrosyphon, as shown in Fig. 2.2. This solution permits the return of the liquid from the reservoir to be delayed for certain period, until a sufficient volume of liquid, capable of quickly cooling the evaporator wall has been transferred from the reservoir. Although Tamburani presented this device for cooling of electronic equipments, no experimental results were reported.

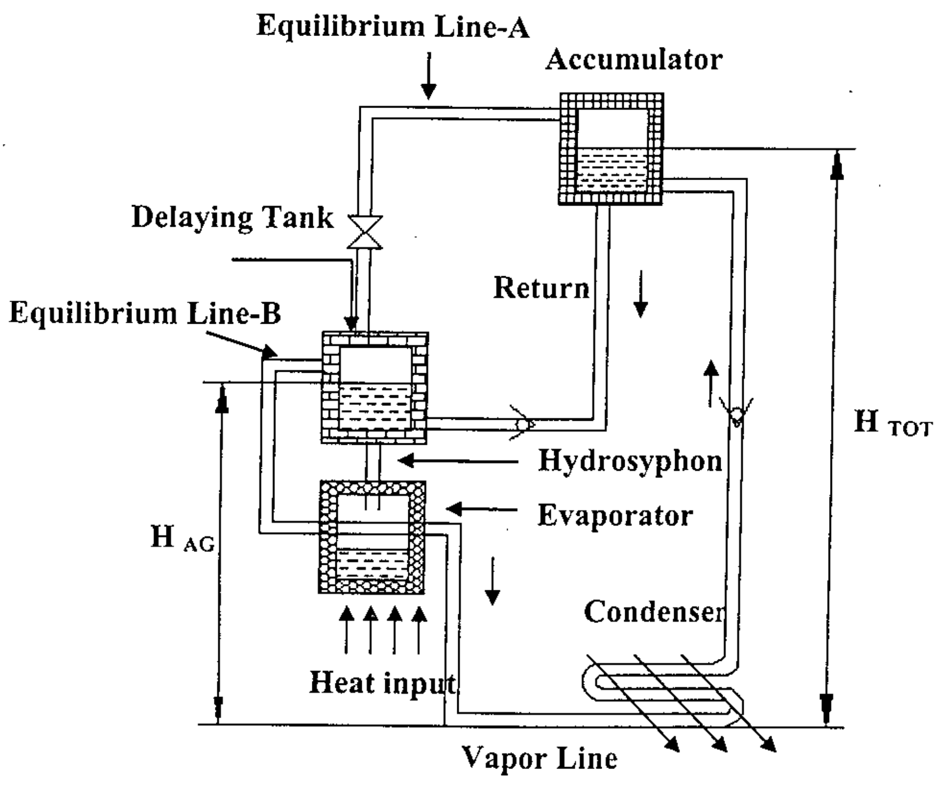


Figure 2.2: Schematic diagram of first kind of PTPT with delaying tank [24]

▪ **Group B: The liquid returns by an increase in the pressure in the reservoir:** The pressure pulsation occurs inside the reservoir instead of the evaporator in this kind of PTPT. These devices were reported to show good thermal performance for electronics cooling because the temperature of the evaporator, to which the chip is connected, remains constant throughout. But in order to create this pressure pulsation inside the reservoir, a periodic supply of heat to the reservoir is needed. Ogushi et al. [25] reported the only device tested of this type, which was proposed for electronics cooling in space application

Group C: There is no difference in pressure: This kind of PTPT device are actually gravity assisted, as the reservoir is placed over the evaporator. Most of these kind of devices were developed in the 1980s for solar heating and water storage applications but they have also been proposed for electronic equipment cooling. The main feature of this device is that the evaporator and the reservoir are connected through an equalization line, which is opened and closed periodically by means of valves, as shown in Fig. 2.3. A number of devices of this kind were reported for different cooling applications ranging from air conditioning to micro-electronics cooling.

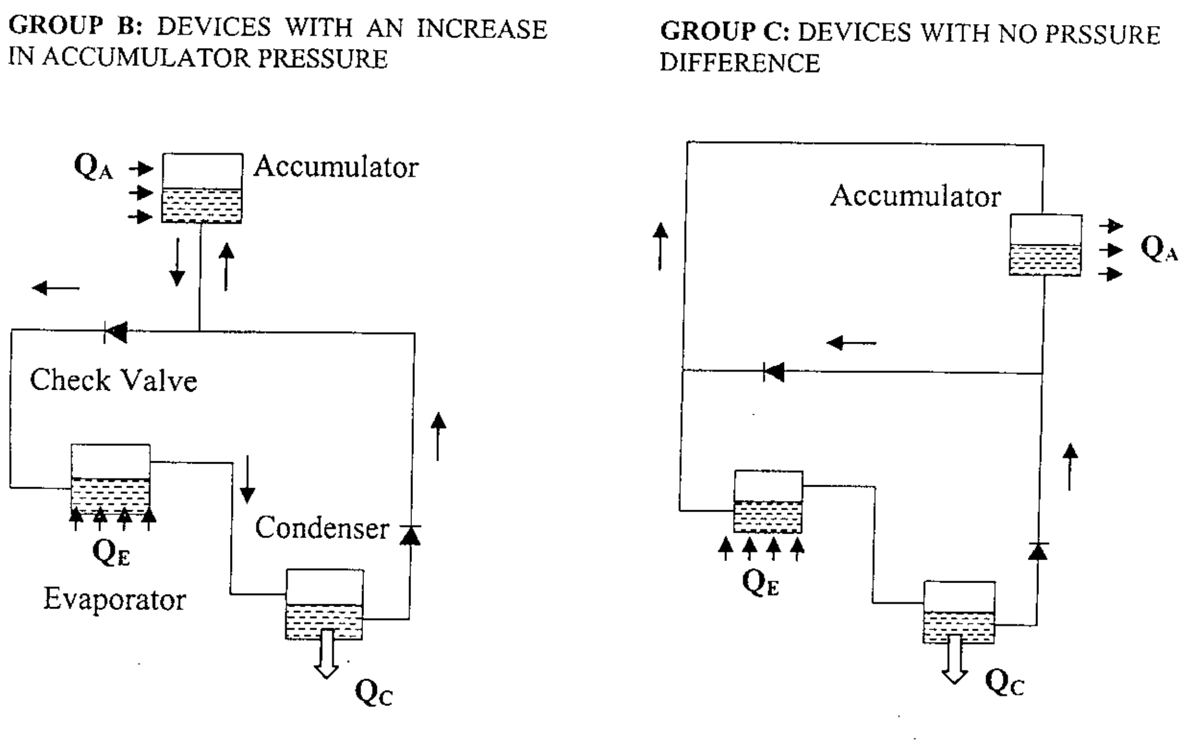


Figure 2.3: Schematic diagram of 2nd and 3rd kind of PTPT

2.4.2 Experimental study on PTPT

Fantozzi et al. [26] reported an experimental work on the upward and downward two phase heat transfer with miniature size PTPT. It presented an experimental analysis on the influence of the distance between the evaporator and condenser on the thermal resistance of the apparatus. The PTPT device was investigated with the condenser positioned both above and below the evaporator (Fig. 2.4).

The evaporator had an internal volume of 80 cm^3 with a maximum evaporator fill ratio of 50% and the condenser dimension was $64\times78\times8$ mm^3, made of aluminum. The condenser was cooled by an external fan and the experiments were performed with a fill ratio of 30%. The heat flux removed was reported to be 11 W/cm^2, with a flat heat exchange surface. The device had shown a stable periodic heat transfer which was weakly influenced by the position of the condenser with respect to the evaporator. In contrast, a classical loop mini- thermosyphon was reported not to achieve a stable functioning for a level difference between the evaporator and condenser lower than 0.37 m.

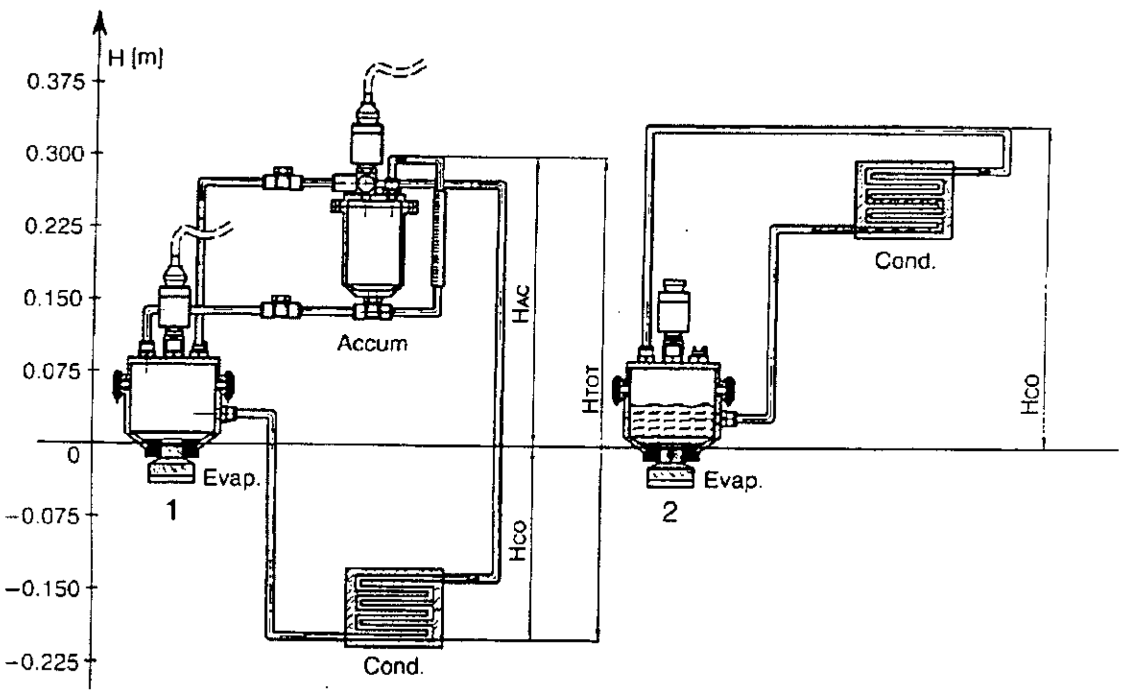


Figure 2.4: Experimental setup showing level differences between the components in the case of PTPT (1) and loop thermosyphon (2) [27]

Some previous experimental works have shown that a PTPT device can well suit to electronic equipment cooling but all the tests have been made with a big size apparatus. The relevant reduction of the scale factor could generate unknown problems and a thermal behavior really different could be observed. Fantozzi et al. [27] analyzed the feasibility study and the realization of a cooling device for desktop computer based on a PTPT heat transfer model. Using a computer code created by the authors in cooperation with Moscow Power Institute and validated by the data collected in previous experimental tests, a mini experimental apparatus based on a PTPT heat transfer regime has been realized (Fig. 2.5). First experimental results were reported and discussed.

The PTPT simulator code has been suited for the working fluid FC-72, a low boiling temperature dielectric fluid. The evaporator of the PTPT device is the simplest one with a flat exchange surface and a bottom heat regime of heat transfer. All the experiments are carried out filling the evaporator with 4×10^{-6} m^3 of working fluid FC-72 at the environmental condition (22 °C). They conducted the simulation tests on two different setup, one of group A type and other of Group B type. The starting data used for the simulation test for these two devices are given in Table 2.1.

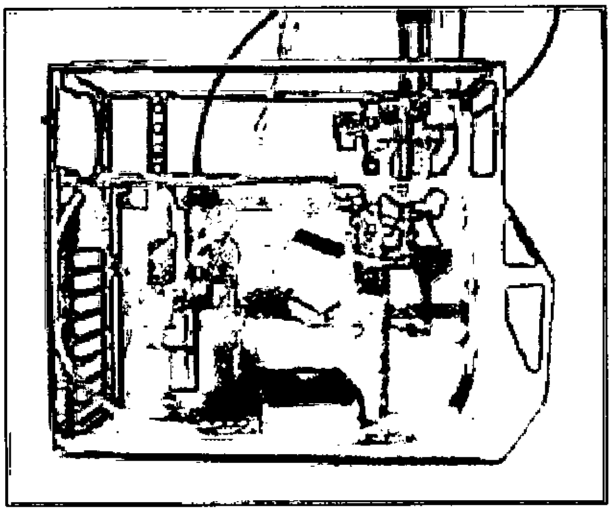


Figure 2.5 – Experimental setup inserted into a desktop computer case. By Fantozzi et al. [28]

Table 2.1: Starting input data for the PTPT simulator code

Starting input data for PTPT Simulator Code			
Evaporator volume	10×10^{-6} m^3	Connecting line's diameter	4 mm
Accumulator volume	5×10^{-6} m^3	H_{tot}	250 mm
Delaying tank volume	4×10^{-6} m^3	ΔH_{ag}	100 mm
Hydrosyphon height	23 mm	T_{env}	30 ^{0}C
Evaporator liquid filling	0.40 %	Cooling air velocity	2.5 m/s
Condenser length	2 m	Heat power rate	40 W

The simulation tests pointed out that both PTPT devices can be used as CPU coolers. The maximum temperature of the evaporator wall was reported by the simulation test to be 65^{0}C for both the devices, with a maximum oscillation of 8^{0}C for the A device and 12.5^{0}C for the B device. The numerical results showed that the heat transport cycle time increases with increasing fill ratio. A good qualitative accordance between the simulation tests and the experimental one has been obtained. The experimental tests have however shown as this PTPT can be a really low cost alternative solution for the desktop computer processor cooling, but some experimental investigations had to be made in order to optimize it.

Alam (2006) came up with a design alternative to name it the thermoloop and presented a detailed analysis of the working principle, performance dependence and its comparison with the heat pipe [5]. He also reported some test results of thermoloop device for the cooling of high end computers. Alam reported that with the help of the two flow controllers and the adjustable reservoir, thermoloop device can produce much stronger vapor transport and condensate return forces, and thereby higher heat flux dissipation capacity. He also provided a conceptual sketch of the thermoloop device used inside a computer to provide thermal solutions for the microprocessor and other microchips (Fig. 2.6).

His test setup consisted of an evaporator of (5×5×1) cm copper block with boiling enhancement characteristics inside and a 0.55 cm diameter copper tube, 170 cm long as the condenser. The forward tube and the return tube had the same diameter (0.55 cm)

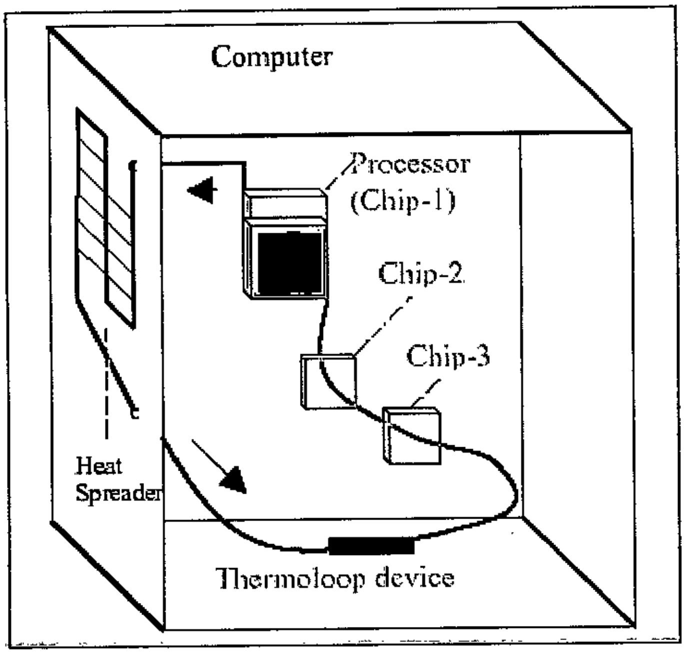


Figure 2.6: Conceptual sketch of a thermoloop device inside a computer

but different length, the return tube (50 cm) being longer than the forward line (30 cm). The check valves had an open pressure of 50 gm/cm^2. The adjustable reservoir was made up of rubber bladder, capable of holding 50 cm^3 of liquid when inflated. The experiments were carried out with two working fluids- water and acetone, at different device orientations.

He observed from the test that the evaporator temperature at different thermal loads for the two working fluid vary depending on the thermal load. The maximum thermal load was observed to be 200 W for water and 150 W for the acetone. He concluded that the maximum thermal load is limited by the condenser heat dissipation capacity and by using stronger air flow and higher number of fins, this limit can be expanded. Alam's test results also reported that the device had shown a similar behavior when placed in a vertical position (condenser at a lower position with respect to the evaporator), except that it had a longer cycle period. The condenser tube length filled with vapor was found to be almost proportional to the thermal load (Fig. 2.7).

Alam also reported the test results of cooling a Pentium 4, 3.2 GHz microprocessor, generating 80-100 W of heat, with acetone as the working fluid. The computer had two

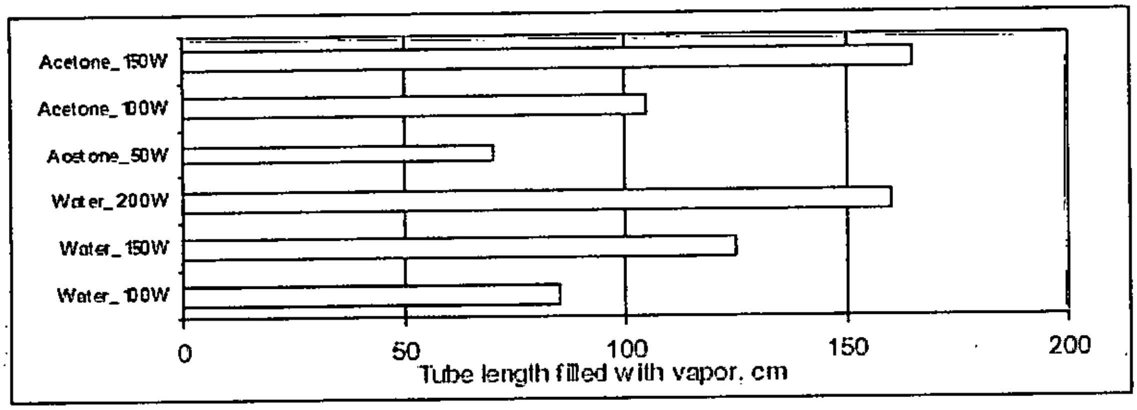


Figure 2.7: Comparison of length of condenser tube filled with vapor at various thermal loads for the two working fluids

more microchips each producing 10-20 W of heat. The thermoloop device was shown to keep the maximum evaporator temperature below 65 ^{0}C (the maximum allowable microprocessor temperature is 70^{0}C), when the machine was run for months using the thermoloop device. He proposed for a different phase change fluid with lower boiling point to meet the problem of narrow temperature margin of maximum allowable microprocessor temperature.

CHAPTER 3

EXPERIMENTAL SETUP AND PROCEDURE

Measurement of various functional parameters was carried out to characterize their effect on the overall heat transfer performance of the thermoloop device under the present study. Two different prototypes of the device of varying dimensions and materials were developed and tested under different condenser conditions. The thermoloop device consists of evaporator, condenser, tubing, reservoir and one way valves. The two prototypes tested had different evaporators, condensers and different forward and return line lengths. Most of the tests were conducted using the prototype-1 and with water as the working fluid. These have been provided by Alam Thermal

Solutions, Inc., USA. These components were assembled to develop two different configurations using the locally available facilities.

3.1 EXPERIMENTAL SETUP

Various components of the thermoloop test setup were connected by different types of connectors and proper adhesive to maintain firm and leak-proof joints. The condenser convection condition was varied by using a forced draft fan. A photographic view of the experimental setup of the thermoloop device is shown in Fig. 3.1.

As heat is applied to the evaporator by the heater connected to it, the temperature of the liquid stored in the evaporator increases. When the temperature exceeds the boiling point of the liquid, it evaporates and vapor bubble starts to form. Pressure increases and vapor pushes the liquid in the forward line towards the condenser, and then from the condenser to the reservoir through the return line. Flow can not occur in the opposite direction due to the two check valves placed on either side of the reservoir. Thus the liquid coming out of the condenser is stored in the reservoir as long as evaporation is continued in the evaporator. When the pressure inside the evaporator falls below the pressure in the reservoir, the check valve between the reservoir and the evaporator

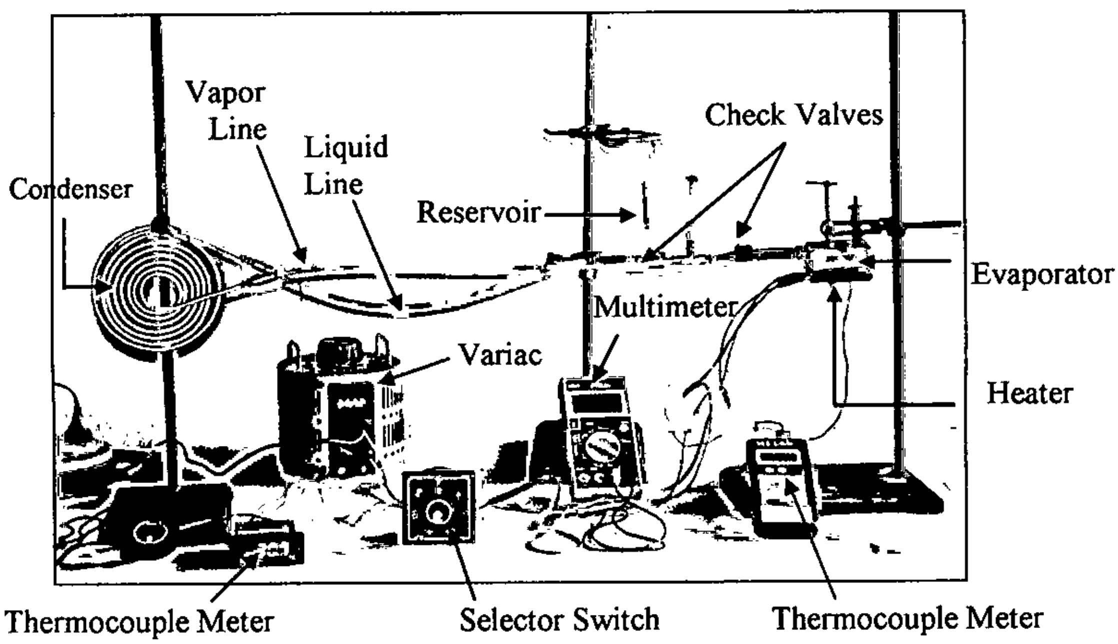


Figure 3.1: Front View of the experimental set-up

opens and the liquid in the reservoir flows back into the evaporator. In this way, the cycle completes and a new repeating cycle starts.

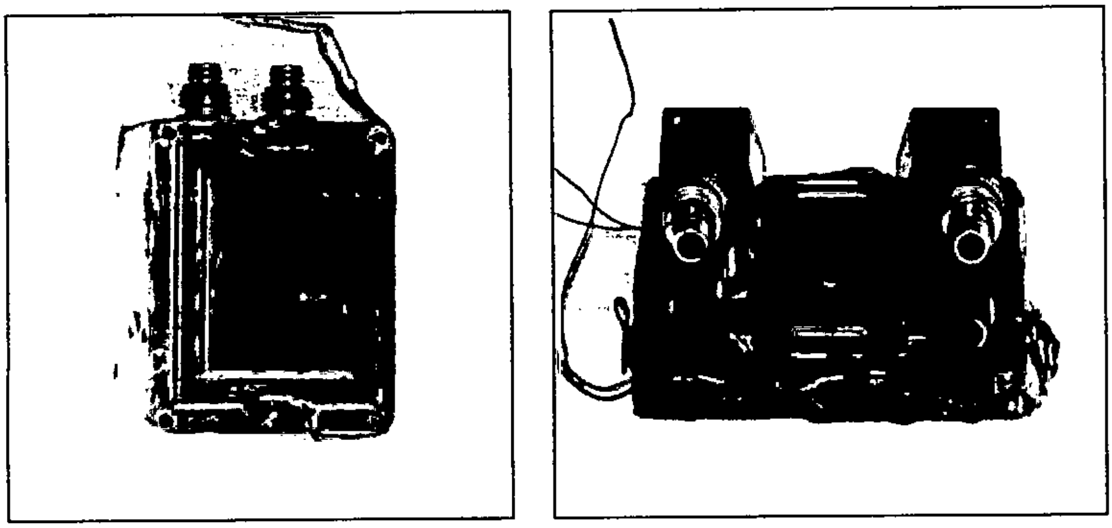
3.1.1 Evaporator

Evaporator is the most important component of the thermoloop as the efficiency of the device is determined by the efficiency and boiling enhancement characteristics of the evaporator. In the present study, two prototypes of the evaporator of different dimensions, shape and material were realized (Fig. 3.2). Both of the evaporators had a flat bottom surface to attach the resistive heaters (serving as the heat source) easily. The internal surfaces of the evaporators were grooved to enhance the boiling heat transfer features (Fig. 3.3). Facility was provided for both the evaporators to connect the forward and return lines easily and without any leakage.

The shape and dimension of both of the evaporators were chosen to ensure that they are compact and can be miniaturized for the practical application in electronics cooling. Considerable thought was given in their design to make measurement and instrumentation easier. Proper care was taken in perfectly sealing the evaporators to ensure that they can withstand the high pressure developed inside during the evaporation of the fluid. Gaskets were used for the evaporator of prototupe-1 to prevent leakage and consequent pressure drop during the boiling process. It will be shown later that the slightest leak in the evaporator can lead to drastic consequences. One thermocouple was connected to the evaporator to give the wall temperature for the prototype-1. There were two thermocouples connected to the evaporator wall for the prototype-2. The dimension and other geometric features of the evaporators are given in Table 3.1.

Table 3.1: Geometric features of the two evaporators employed in the test

Dimension	Prototype 1	Prototype 2
Outside dimension (mm)	72.5×60×20	77×65×8
Inside volume of the Evaporator (cc)	75.0	30.0
Material	Aluminum	Copper
Internal surface	Grooved	Grooved



(a) (b)

Figure 3.2: Evaporators used for **(a)** Prototype-1 and **(b)** Prototype-2

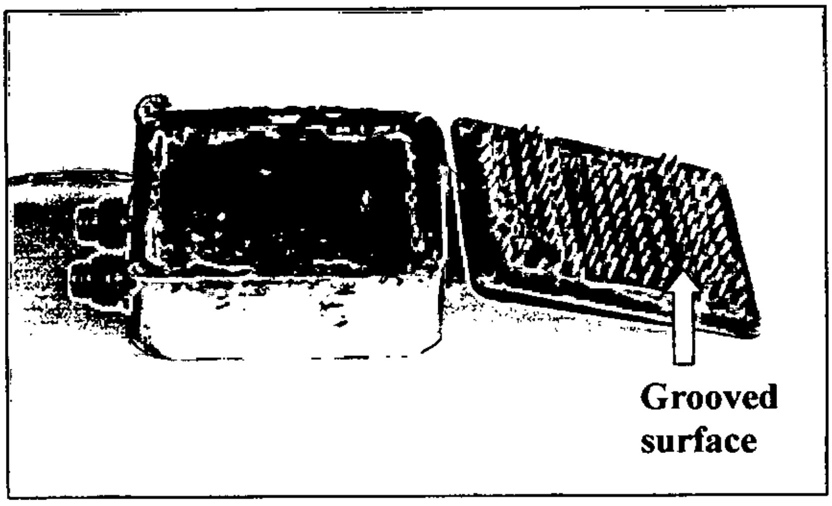


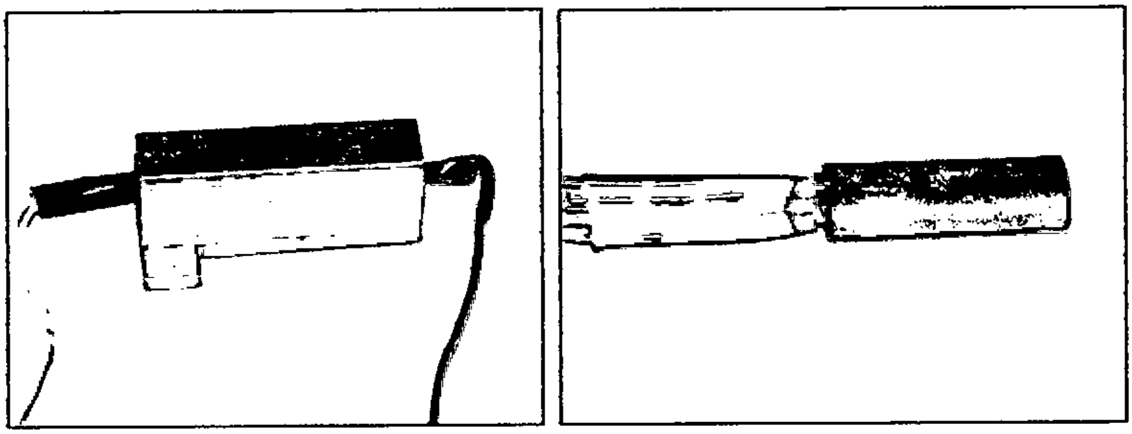
Figure 3.3: Internal surface of the evaporator, with boiling enhancement features.

Table 3.2: Specification of the heaters

Features	Heater type-1	Heater type-2
Heating surface	Flat	Round
Length (mm)	50	55.6
Heating capacity (W)	50	150
Material	Copper	Aluminum
Number of heaters	2	2

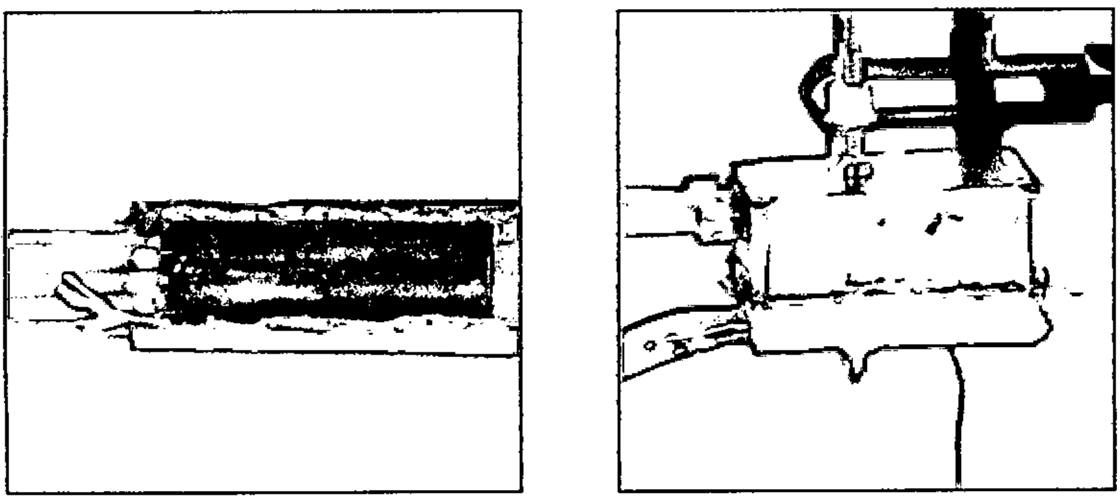
To supply the required heat flux to the evaporator, two different types of AC heaters were employed in the current study. Both these heaters were resistive heating elements whose output can be controlled by connecting them to the electric line through a variac. The characteristics of these two kind of heaters employed are provided in Table 3.2.

The flat heaters were made of copper and had a capacity of 50 W. For higher heat input, they were connected in parallel. The heaters of higher heating capacity (150 watt



(a) (b)

Figure 3.4: **(a)** Flat heater of 50 W capacity, **(b)** Cylindrical heaters of 150 W capacity pressed inside a copper box



(a) (b)

Figure 3.5: **(a)** The cylindrical heater with the copper cover placed inside the rectangular mild steel box with insulation, **(b)** Heater- evaporator assembly

each) were of cylindrical shape and had a diameter of 8 mm each. As the evaporator surface is flat, these two round heaters were pressed into a close rectangular copper cover (5.56×2×1.2 cm) with flat surface to facilitate the connection with the evaporator [Fig. 3.4 (b)]. This copper casing along with the two round heaters was placed inside a rectangular mild steel box (7.13×2.65×1.45 cm). There was thermal insulation between this mild steel box and the copper casing to minimize the heat loss from other surfaces other than the surface connected with the evaporator as shown by Fig. 3.5 (a).

The heaters were connected to the evaporators by clamping [Fig. 3.5(b)]. To minimize the thermal contact resistance between the evaporator and the heaters, a thin layer of thermal interface material (TIM) were used between them. The thermal interface material used was Thermolite® which had a contact resistance less than 0.375 ^{0}C-cm^2/ W.

1.2 Condenser:

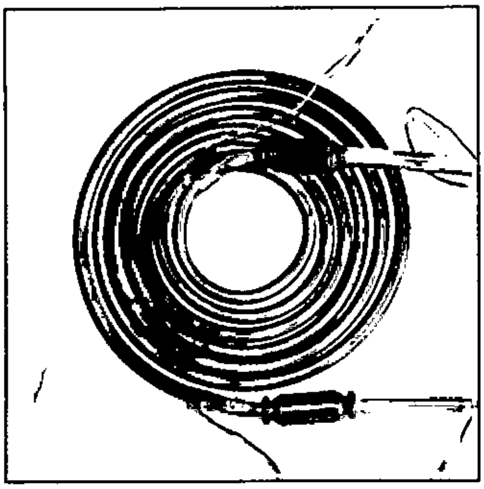
Fantozzi et al. [28] have reported that the design of the condenser has a significant effect on the heat transfer characteristics of pulsating thermosyphons. They concluded from experimental results that, if the condenser is designed for a cooling power lower than the input power of the evaporator, it is possible to obtain a cooling of the reservoir and of environment, otherwise an environmental heating is realized.

In the present study, like the evaporator, two prototypes of the condenser of different dimensions and material were realized (Fig. 3.6). Both of these condensers had the same spiral shape, but with different number of turns and had different wire diameter (Table 3.3). The coil diameter, material and total length of the two condensers were also different. The condensers were connected to the forward and return tubes with copper connectors, making the joint firm and leak-free. Two thermocouples were connected to the condenser to measure the inlet and outlet temperatures of fluid.

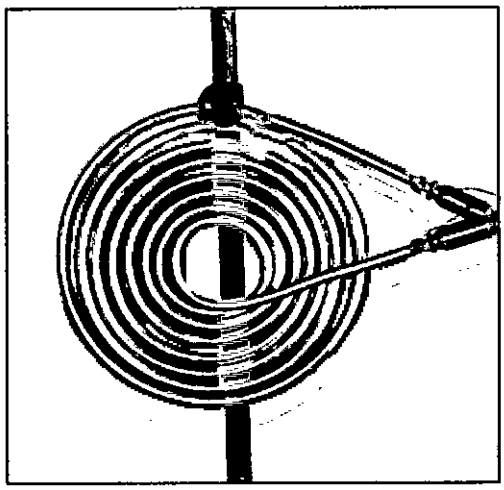
To increase the rate of heat dissipation from the condenser, the convection condition of the condenser was changed to forced convection by employing a forced draft fan (Fig. 3.7). The fan was placed at a distance of 2 cm from the condenser. It was a 12 volt DC

Table 3.3: Geometric features of the two condensers employed

Dimension	Prototype 1	Prototype 2
Tube diameter (mm)	6.52	6.35
Total length (mm)	1900	2410
Volume (cc)	30.05	35.3
Material	Copper	Aluminum
Number of Turns	7	8
Coil diameter (mm)	145	152

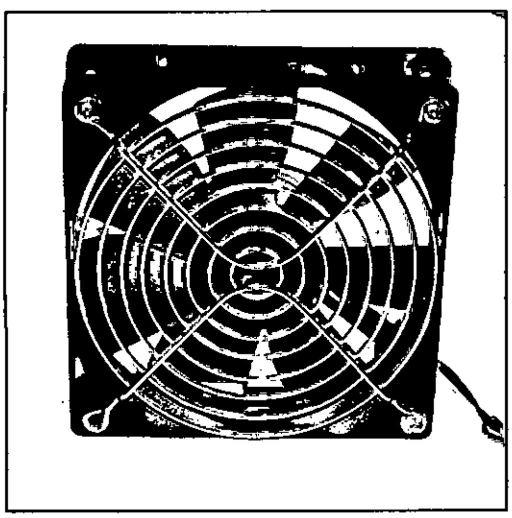


(a)

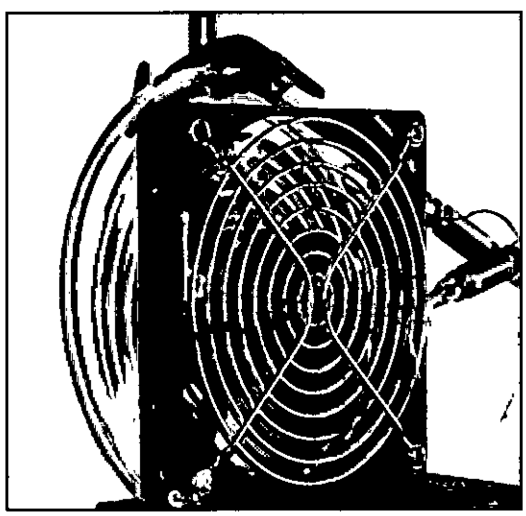


(b)

Figure 3.6: Spiral shaped condenser made of **(a)** Copper and **(b)** Aluminum



(a)



(b)

Figure 3.7: **(a)** Fan used for forced convection, **(b)** Fan placed in front of the condenser during the test

fan which provided an average air velocity of 4.75 m/s. The power consumption of the fan was 3.25 W.

3.1.3 Forward and Return Lines

The forward and return tubes transport the vapor and liquid, respectively, between the evaporator and the condenser and hence form the most important loop of the thermosyphon. The tubing of the first prototype had the same outside diameter, thickness and length for the forward and return lines. The check valves and the reservoirs were inserted at certain distances into the return line [Table 3.4]. The lines were connected to the condenser via copper connectors and to the evaporators with metal nuts. All the joints in the forward and return lines were maintained firmly affixed to prevent any kind of leakage. The reservoir and the check valves (described in sections 3.1.4-3.1.5) were properly glued after inserting them into the return line. The flow of the liquid through the two lines was possible to observe visually because both the tubing was made of transparent poly-ethylene material, capable of withstanding high temperature and pressure.

Table 3.4: Geometric features of the forward and return tubes

Dimension	Prototype 1	Prototype 2
Length of forward tube (mm)	560	600
Length of return tube (mm)	560	645
Outside diameter of forward tube and return tube (mm)	6.35	6.35
Volume (cc)	8.20	8.20
Thickness of forward and return tubes (mm)	1.016	1.016
Length of return tube from condenser to the reservoir (mm)	460	520
Length of return tube from reservoir to the evaporator (mm)	90	125

3.1.4 Reservoir

Fantozzi et al. [28] has pointed out that it is possible to control the temperature of the evaporator by correctly designing the reservoir's surface and its heat loss with the environment and its temperature. In the present study, a rectangular reservoir

(6×2.7×6.2 cm) with curved upper part and made of plastic was used to store the liquid being pushed out of the forward and return lines and the condenser. It had a small opening at the top so that the pressure inside the reservoir is always atmospheric when empty. The reservoir was fitted at a height of 3.5 cm and 5.5 cm above the return line for prototype-1 and prototype-2 respectively. It was placed at a distance of 46.0 and 52.5 cm respectively from the condenser for the two prototypes. The total liquid storage capacity of the reservoir was about 80 cm^3. As the reservoir was transparent like the tubing, the accumulated liquid height in the reservoir was visible from outside. During the experiments, the reservoir was kept straight to measure the accumulated liquid column height which served as a measure of the system pressure. The reservoir was not thermally insulated in the present study.

3.1.5 Flow Controllers

To maintain the flow of the working fluid in the desired direction, two similar kind of one-way check valves (non-return valves) were inserted into the return line on either side of the reservoir. The check valves were placed at a distance of 5 cm from the reservoir in each side. The check valve placed between the reservoir and condenser was called the 'In-Valve'. Its function was to permit the flow only from the condenser towards the reservoir. The other valve, placed between the reservoir and the evaporator was called out-valve and it allowed the flow only from the reservoir towards evaporator. The check valves were made of plastic and had the dimension of 3.5 cm in length and 2 cm in diameter.

3.1.6 Working Fluid

The properties, especially the boiling point and latent heat of vaporization of the working fluid determines the nature and extent of the heat transfer and the heat transfer performance of the device. The maximum temperature of the evaporator wall and the condenser inlet and outlet temperatures depend on the boiling point of the working fluid. The two working fluids used in the current study are water and ethanol. Water was selected because of its excellent thermophysical properties and wide availability. Ethanol was selected for its low boiling point (which is necessary for the device to operate below the maximum permitted temperature of various electronic components) and it's well known suitability for this kind of passive heat transfer devices.

But the most suitable fluid to be used for electronics component cooling by thermoloop is known to be different types of dielectric fluid such as FC-70 or FC-72 etc. FC-72 is an ideal fluid to use in low temperature heat transfer applications. Its high dielectric constant also means that it will not damage the electronic equipments in the case of a leakage or other possible failure. FC-72 liquid is chemically stable, compatible with sensitive materials, nonflammable and practically non-toxic. But as the primary objective of the present study is to determine the influence of various geometric and functional parameters on the performance of thermoloop, the selected fluids (water and ethanol) will suffice to meet the requirements.

3.2 DATA ACQUISITION

The data acquisition for this study consisted of temperature, voltage, current and height measurement. A K-type thermocouple was connected to the evaporator wall and the temperature was measured by a DIGI-SENSE® K-type thermocouple thermometer (Model: 91000-00, Range: -50 to1300^0 C, Accuracy: ±0.5% of reading or ±1^{0}C). The T-type thermocouples connected to the condenser were connected to an Omega DP 115 TC-type thermocouple thermometer (model- DP 115, range -200 to 200 ^{0}C, accuracy: ±1^{0}C) via selector switch. The power supply to the evaporator was changed by varying the voltage with the help of a variac. The input voltage to the heater was measured using a DT 890BT digital multimeter. The amount of current drawn by the heater was measured by HIOKI 3261 digital clamp on meter with a resolution of 0.01 ampere. The height of the liquid collected in the reservoir was measured using measuring tape. All these readings were taken simultaneously at a fixed interval of time.

3.3 EXPERIMENTAL PROCEDURE

For the present study, the experimental set-up consisting of the test specimen and other equipments was mounted on two vertical tubular frames placed at a certain distance. The evaporator and the condenser were attached firmly to these frames by clamping.

The reservoir was also held by a clamping frame to keep it upright, so that it does not tilt or get lowered by the weight of the accumulated liquid.

The experiments were conducted using water and ethanol as the working fluid. Before every test, all the working fluid inside the device was drained out completely. The working fluid was then injected using a syringe and the whole of forward and return lines and the condenser were filled. The fill ratio of the evaporator was varied from 20% (15 cc) to 80% (60 cc) for the first prototype and from 30% to 50% for the second prototype. After filling the evaporator with the desired liquid fill ratio, it was connected to the forward and return lines firmly. Considerable care was taken to eliminate the formation of bubble or gas pockets in the forward and return lines and also in the condenser. All the tests were performed by placing the thermoloop setup horizontally (i.e. keeping the evaporator and condenser at the same level of elevation).

The experimental setup along with the measuring devices was placed on a test table (Fig. 3.1) in the heat transfer laboratory of Mechanical Engineering Department, Bangladesh University of Engineering and Technology. After connecting all the devices, the setup was checked for any possible leakage. Then the power connection to the heater was switched on and the desired heat input was ensured by setting the variac to the required level and taking consequent current reading. After the power input is set, the temperature of the evaporator and condenser and the height of the accumulated water in the reservoir were recorded at a regular interval of one/two minutes. For the forced convection condition of the condenser, a fan was placed at a distance of 2-3 cm from the condenser.

The total time required for completion of every heat transport cycle was recorded for all the tests performed, along with readings of maximum wall temperature of the evaporator, maximum inlet and outlet temperature of the condenser and maximum height of the liquid collected in the reservoir for every cycle. As the forward and return lines were transparent, the motion of the liquid through these lines and its nature of oscillation due to pressure pulsation were also closely observed to understand the physics of the system. Almost all the experiments were carried out for minimum of 12-15 heat transport cycles. Experiments were stopped if the evaporator temperature exceeded 150^0C due to reasons such as leakage or evaporator dry out. During some tests, the experiments were continued for about four-five hours to observe the performance of the device in long period of operation.

3.4 UNCERTAINTY ANALYSIS

In most cases, presentation of experimental results without performing the uncertainty analysis associated with the test result is not acceptable. The benefits and importance of uncertainty analysis have become widely recognized nowadays. This is a very powerful tool of identifying the source of error in the experiments. The results of the uncertainty analysis performed on the primary measured and calculated quantities in the present study are presented in Table 3.5 and Table 3.6 respectively. The calculation of uncertainties in the present study is presented in details in the Appendix B.

Table 3.5: Uncertainty in the measured values

Measurands	Precision limit, P	Bias limit, B	Total limit, W
Time (t)	0.25%	0.02%	0.25%
Temperature (T_E)	1.5%	0.5%	1.58
Temperature (T_C)	1.83%	0.5%	1.9%
Diameter (Di)	1.31%	0.94%	1.61%
Length (L)	0.71%	0.09%	0.71%
Width (b)	2.78%	1.39%	3.10%
Height (H)	1.72%	0.86%	1.92%
Area (A)	2.62%	1.87%	3.22%
Volume (V)	5.16%	2.58%	5.77%

Table 3.6: Uncertainty in the calculated quantities

Quantity	Total Uncertainty (U)
Volume flow rate (Q)	7.35%
Mass flow rate (m)	7.35%
Velocity (v)	6.61%
Pressure drop (ΔP_{CON})	10.0 %
Pressure drop (ΔP_{FL})	7.11%

CHAPTER 4

MATHEMATICAL MODELING

4.1 MATHEMATICAL BACKGROUND

A Pulsated Two-Phase Thermosyphon (PTPT) or thermoloop is very similar in working principle to a heat pipe as both are passive indirect cooling devices and in both the cases pressure is the driving force to move the vapor from the evaporator to the condenser. But thermoloop differs from a heat pipe in the manner the condensate is returned to the evaporator. In thermoloop, condensate is returned to the evaporator by pressure difference between the reservoir and evaporator rather than by capillary force as in a heat pipe. The driving force (pressure) must be greater than the sum of the pressure losses in the components, pressure losses due to wall friction experienced by both vapor and liquid in the loop and other minor losses in fittings, bends etc. For the devices that utilizes the capillary force as the driving force for the return of the liquid, the pressure and temperature variations are considerably low as the thermal resistances are very small. But for thermoloop, the pressure and temperature variations are much higher due to higher thermal resistances in the system.

In a PTPT, both the mass and heat flow depend on time as the device operates in a periodic manner. Sašin et al. [21] developed a time dependent mathematical model for the first time for PTPT device, which was based on the result of the experimental observation. They investigated the thermal behavior of the device by dividing the heat transfer cycle into four different parts. Then Fantozzi et al. [28] proposed another mathematical model based on the original model of Sasin et al. and it was verified by experimental results. They remodeled the original one of Sasin et al. and classified the heat transfer cycle into two distinct portions – a transfer operation from evaporator to reservoir and another return operation from reservoir to evaporator. They divided the whole system into seven control volumes to determine the energy and mass transfer rates. Lumped capacitance method was used to determine the properties within each of the control volumes with several assumptions for the first heat transport cycle. They performed validation of their model by experimental data and reported good qualitative accordance, the error being less than ±2%.

In the present study, an analysis of the physics involved in the working principle of the thermoloop is investigated and a simplified mathematical model has been developed for the prediction of the thermal behavior of the device. The operating processes of thermoloop have been divided into two regimes, the vapor transport regime and liquid return regime. Heat and mass balance has been performed on different components individually, and the pressure pulsation and the variation of operating temperature have been investigated mathematically. Several assumptions have been made to avoid the complexity of two- phase periodic heat transfer process.

4.2 ASSUMPTIONS FOR THE PRESENT STUDY

For the present mathematical model, following assumptions have been made to facilitate the mathematical modeling of a two-phase periodic heat transfer process without unwarranted complexity.

a) The condition in the evaporator is always saturation condition.

b) There are some non-condensable gases present in the system.

c) The properties are considered constant for the vapor transport state, after a stable periodic operation is reached.

d) The upper part of the reservoir is open to atmosphere.

e) The heat loss from different components of thermoloop, other than the condenser is by natural convection.

4.3 ANALYTICAL STUDY OF THERMOLOOP OPERATION

As pointed out by Fantozzi et al [26], a pulsating two-phase thermosyphon device has three main components between which heat and mass are transferred. They are- firstly the evaporator, which receives heat from an external source, then the condenser, which dissipates heat and finally the reservoir, which stores the working fluid and supplies it to the evaporator for completing the heat transfer cycle. The operation of the thermoloop device can be divided into two separate phases- transport of fluid from evaporator to reservoir through the condenser, called the vapor transport state constitutes the first part of the cycle; and then return of the liquid from the reservoir to the evaporator constitutes the second part and is known as the liquid return state.

Figure 4.1 shows the schematic arrangement of the various thermoloop components for the present study. Thermoloop device starts functioning as soon as the heat flux is supplied to the evaporator and the temperature of the evaporator wall exceeds the

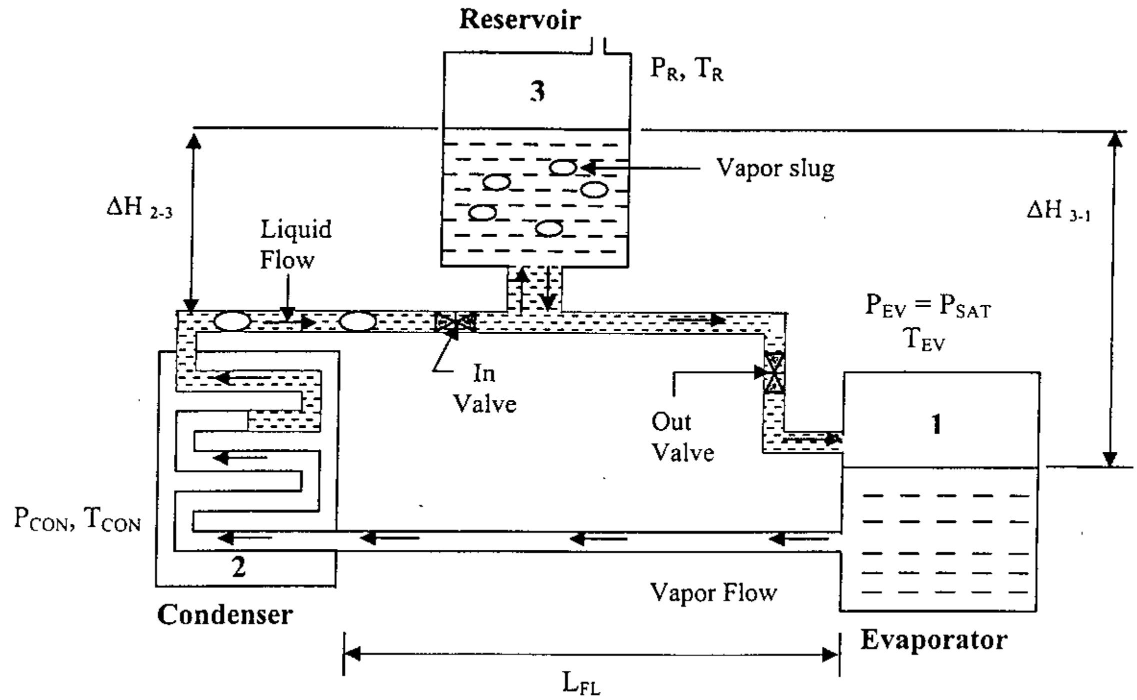


Figure: 4.1 Schematic arrangements of the various components of thermoloop

boiling point of the liquid. Vapor bubble forms and the pressure inside the evaporator increases in an isochoric process. The liquid in the forward (liquid) line is pushed towards the condenser by this pressure and the vapor from the evaporator replaces liquid in the forward line and a part of the condenser. The movement of liquid or vapor in the other direction (return line) is prevented as the check valve between the evaporator and reservoir remains closed. This vapor starts to cool and is subsequently condensed in the condenser.

The liquid in the condenser will not move towards the reservoir unless the pressure in the evaporator is higher than the pressure in the reservoir and summation of pressure drop in the condenser, forward line and return line. Therefore, for the motion of the liquid from condenser to reservoir, the following condition must be satisfied.

$$P_E(t) \geq \rho_{E,V} \cdot g\, \Delta H_{1\text{-}2}(t) + \rho_{C,L1}\, g\, \Delta H_{1\text{-}2}(t) + \rho_{C,L2}\, g\, \Delta H_{2\text{-}3} + \Delta P_C + \Delta P_{1\text{-}3} + \Delta P_M + P_R(t) + \Delta P_G \quad (4.1)$$

where,

P_E = Pressure inside the evaporator

= $P_{SAT}(T_1)$ = Saturation pressure at temperature T_1

$\Delta H_{1\text{-}2}$ = Difference in elevation between evaporator and condenser

= 0, as the condenser and evaporator was placed at same height in the present study

$\Delta H_{2\text{-}3}$ = Difference in elevation between condenser and reservoir

ΔP_C = Pressure drop in the condenser

$\Delta P_{1\text{-}3}$ = Pressure loss due to friction in the forward and part of return line from evaporator to reservoir

ΔP_M = Minor losses in the line due to fittings, curves etc. between evaporator and reservoir

P_R = Pressure in the reservoir

= P_∞ = atmospheric pressure, as there was an opening in the upper portion of the reservoir

ΔP_G = Pressure increase in the return line and reservoir due to non-condensable gases present

$\rho_{E,V}$ = Density of vapor at evaporator condition

$\rho_{C,L1}$ = Density of liquid at condenser inlet condition

$\rho_{C,L2}$ = Density of liquid at condenser outlet condition

Here the pressure, density and the level of liquid in different components are expressed as a function of time because all of them vary during the cycle. The first two terms of Eq. (4.1) represents the pressure drop due to difference in elevation between evaporator and condenser (positive in the case of condenser placed above the evaporator) which is zero in our case. The third term [$\rho_{C,L2}\, g\Delta H_{2\text{-}3}$] represents the pressure drop due to difference in elevation between the reservoir and condenser, as the reservoir is placed above the condenser. The presence of a non-condensable gas in a vapor can significantly decrease the condensation rate below that for a pure vapor. Only a slight amount of air can reduce rate of condensation by up to 50% [29]. In the present study, it was not possible to remove air from the device completely, as a result vapor slug flow coming from the condenser was observed in the return line during the experiments. This caused a reduction in the condensation rate in the condenser and possible increase in pressure in the return line.

So now the condition may be written as –

$$P_E(t) \geq \rho_{C,L2}\, g\, \Delta H_{2\text{-}3} + \Delta P_C + \Delta P_{1\text{-}3} + \Delta P_M + P_R(t) + \Delta P_G \qquad \textbf{(4.2)}$$

The pressure drop ΔP_C in the condenser mainly depends on mass flow rate and is a function of geometry and size of the condenser. It is very low if the mass flow rate from the evaporator to the condenser is small. Fantozzi et al. [28] used the following correlation to estimate the pressure drop in the condenser assuming a quality of the fluid x = 1 at the condenser entry and x = 0 at the outlet.

$$\Delta P_C = \left(\frac{2 f_B \left(\frac{\dot{m}}{A} \right)^2 L}{D \rho_l} \right) \cdot \left(1 + \bar{x} \left(\frac{\rho_l}{\rho_v} \right) \right) \cdot \left(1 + \bar{x} \left(\frac{\mu_l}{\mu_v} \right) \right)^{-1/4} + \left(\frac{\dot{m}}{A} \right)^2 \cdot \left(\frac{\rho_l}{\rho_v} \right) \cdot \left(\frac{x_{out} - x_{in}}{\rho_l} \right) \qquad \textbf{(4.3)}$$

Where,

μ_L and ρ_L = viscosity and density of liquid at condenser outlet respectively

μ_v and ρ_v = viscosity and density of vapor at condenser inlet respectively

$\bar{x}$ = linear average quality of vapor

f_B = Blasius friction factor

x_{OUT} and x_{IN} = outlet and inlet quality of vapor (assumed 1 and 0 respectively)

D = diameter of the condenser tube

Pressure drop in the condenser was calculated for different tests at different heat inputs and different fill ratios by using Eq. (4.3). For a heat input of 100 watt and fill ratio of 30% and with water as the working fluid, the data obtained from the test were – mass flow rate 2.172×10^{-4} kg/s, friction factor 0.1 (assumed), flow area 2.234×10^{-3} m^2 and average quality of vapor 0.5. Using these values in Eq. (4.3), the pressure drop in the condenser was obtained as 2.92 kPa, which is a fair approximation. Although the actual pressure drop is expected to be little higher, but more accurate approximation could be achieved by using Eq. (4.3) if the actual vapor quality is known and better way to evaluate the mass flow rate of fluid can be opted.

Therefore, the motion of the liquid from condenser to the reservoir starts flowing as soon as the condition expressed by Eq. (4.2) is satisfied. This process continues until all the liquid in the evaporator is vaporized and a definite amount of liquid is transported into the reservoir.

All the liquid stored in the reservoir has to return to the evaporator to complete the an operational cycle, this state being known as the liquid return state. So by the return of the liquid to the evaporator, the initial condition of the device is restored and the device starts operating in a new, repeating cycle. This return state requires that the pressure in the reservoir (P_R) must be greater than the summation of pressure in the evaporator (P_E) and frictional and minor losses in the return line between the evaporator and reservoir. This requires either a decrease in pressure in the evaporator or an increase in pressure in the reservoir. Pressure inside the evaporator decreases in the case of thermoloop, which makes possible the return of the liquid from the reservoir into the evaporator.

After all the liquid in the evaporator is vaporized, the pressure in the evaporator starts to decrease and it can finally reach the saturation pressure (P_S) corresponding to the average temperature of the condenser (T_C). Whenever the pressure inside the reservoir decreases below the pressure in the reservoir and the pressure drops; liquid starts moving towards the evaporator from the reservoir, completing the cycle. When the liquid returns to the evaporator, pressure in the evaporator begins to increase again with temperature and the process starts repeating.

Thus the condition required for the liquid return state can be expressed as

$$P_R + \rho_{R,L}\, g\, \Delta H_{3\text{-}1}(t) \geq P_E(t) + \Delta P_{3\text{-}1} + \Delta P_{M2} \qquad \textbf{(4.4)}$$

Where,

$\Delta H_{3\text{-}1}(t)$ = Difference in level between evaporator and reservoir

$\Delta P_{3\text{-}1}$ = Pressure drop due to friction in the return line between reservoir and evaporator

ΔP_{M2} = Pressure drop due to minor losses in the return line between reservoir and evaporator

P_E (t) = Saturation pressure corresponding to the condenser temperature T_{CON}

$\rho_{R,L}$ = Density of the liquid at the condition of the reservoir

The left hand side of Eq. (4.4) represents the pressure due to the height of the hydrostatic liquid column head in the reservoir and the difference in elevation between the reservoir and evaporator; which is positive in this case, as the reservoir is placed at a higher location that the evaporator.

The conditions required for the operation of thermoloop are expressed through Eqs. (4.1) - (4.4) in terms of pressure. The temperature of the device at various points also varies significantly. The variation is much higher than the variation of pressure and temperature in any capillary device like the heat pipes. This variation of pressure and temperature of the system during the heat transport cycle was expressed for PTPT by Fantozzi et al. [27] by a pressure-temperature diagram, which is a very helpful tool for the understanding of the processes involved.

At the beginning of the cycle, the temperature in the evaporator is the saturation temperature and the liquid evaporates in the saturation temperature $T_1 = T_S$. As the vapors move towards the condenser, there is a heat loss due to the surroundings as the forward line is not thermally insulated. Therefore both the temperature and pressure of the vapor decrease as it moves through the forward line towards the condenser. The temperature of the vapor entering the condenser is denoted as T_2 (Fig 4.2), which is marginally lower than the evaporator temperature T_1 and is verified by the experimental results. In the condenser, the vapor gives up its sensible heat and continues to be cooled to a temperature T_3 which is in the subcooled region. This temperature is the outlet temperature of the fluid leaving the condenser. The pressure drop (ΔP_C) in the condenser is given by Eq. (4.3).

If the heat loss in the return line between the condenser and reservoir is assumed negligible; the temperature of the liquid entering the reservoir from the condenser is at the same temperature as T_3. But in our experiment, the return line was not insulated, so the temperature decreased from T_3 to T_4 as it entered the reservoir. In the present study, also there was no thermal insulation of the reservoir; so it exchanged heat with the ambient. As a result, temperature of the fluid in the reservoir decreased to a value T_5 before the start of the return state.

The velocity of the liquid entering the evaporator from the reservoir can be found out by assuming the flow to be one dimensional and applying the continuity and Bernoulli equation between the inlet of evaporator and exit of reservoir. Denoting the speed of the liquid flow in reservoir exit and evaporator inlet as V_R and V_E and area as A_R and A_E, the continuity equation may be written as

$$Q = V_R A_R = V_E A_E$$

$$V_R = \frac{A_R}{A_E} V_E \tag{4.5}$$

Applying the Bernoulli equation

$$\frac{P_E + \Delta P_{3-1} + \Delta P_{M2}}{\rho g} + \frac{V_E^2}{2g} + Z_E = \frac{P_R}{\rho g} + \frac{V_R^2}{2g} + Z_R$$

or, $$\frac{1}{2g}(V_E^2 - V_R^2) = \frac{1}{\rho g}(P_R - P_E - \Delta P_{3-1} - \Delta P_{M2}) + (Z_R - Z_E)$$

or, $$\frac{1}{2g}.V_E^2 (1 - \frac{A_E^2}{A_R^2}) = \frac{1}{\rho g}(P_R - P_E - \Delta P_{3-1} - \Delta P_{M2}) + \Delta H_{3-1}(t)$$

[from equation (7) and as $\Delta H_{3-1}(t) = Z_R - Z_E$]

or, $$V_E^2(t) = 2g\left[(P_R - P_E - \Delta P_{3-1} - \Delta P_{M2})/\rho g + \Delta H_{3-1} / (1 - \frac{A_E^2}{A_R^2})\right]$$

or, $$V_E(t) = \left[2g.\{(P_R - P_E - \Delta P_{3-1} - \Delta P_{M2})/\rho g + \Delta H_{3-1} / (1 - \frac{A_E^2}{A_R^2})\}\right]^{1/2} \tag{4.6}$$

4.4 HEAT BALANCE FOR THERMOLOOP OPERATION

During operation of a thermoloop, the mass and heat energy balance can be done in two ways. Firstly, the whole device can be separated into different control volumes and heat and mass balance can be performed on the individual control volumes. Secondly, heat and mass balance can be done separately on the three main components of thermoloop: evaporator, condenser and the reservoir. The operating temperature at different locations of the thermoloop can be estimated by solving heat balance equations for a given conditions (thermal load, orientation, convection condition, pressure etc.). All the functional parameters of thermoloop such as pressure, mass flow rate, heat transfer coefficient are functions of the operating temperature of thermoloop.

Fantozzi et al. [28] have conducted the heat and mass balance on the PTPT devices by dividing the system into seven control volumes and setting time dependent equations for each of them. In the present study, heat balance has been performed on the three main components of the system individually. All the equations are simplified for the condition when a steady and stable periodic operation has been reached. This idea was obtained from the mathematical model proposed by Fantozzi et al. [28] and that by Kaya and Hoang [30].

The heat and mass transfer scheme of different components of the thermoloop is shown in the Fig. 4.3. Total heat balance equation for the whole system for one complete cycle can be expressed as-

$$Q_{IN} = Q_C + Q_{FL} + Q_R + Q_{RL} \tag{4.7}$$

where,

Q_{IN} = Heat supplied by the heat source (Heater)

Q_{FL} = Heat loss to the surroundings by the forward line

Q_{SC} = Amount of sub-cooling in the condenser

Q_C = Heat dissipated by the condenser to the sorrounding

Q_R = Heat loss from the reservoir

Q_{RL} = Heat loss to the surroundings by the return line

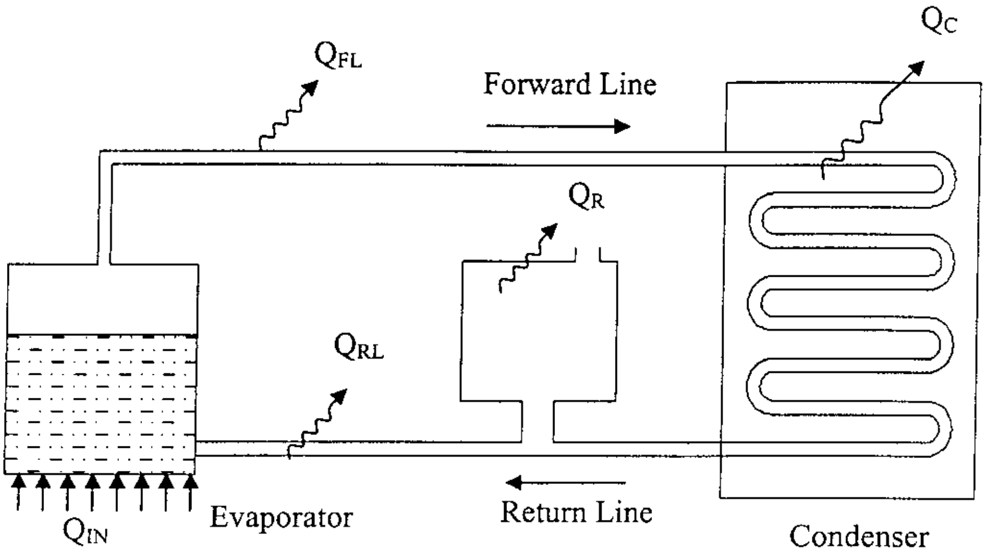


Figure: 4.2 Schematic diagram of heat transfer of thermoloop components

Energy balance equation for the evaporator can be expressed as follows:

$$Q_E = \dot{m}_L(t).h_{fg} + m_V(t).h_v \tag{4.8}$$

where,

Q_E = Heat input to the evaporator

m_L = Mass of the liquid inside the evaporator

m_V = Mass of the vapor inside the evaporator

h_{fg}= specific enthalpy of evaporation

h_V= specific enthalpy of vapor

Heat balance equation for the condenser can be written as –

$$Q_C = Q_E - Q_{FL} - Q_{SC} \tag{4.9}$$

where,

Q_E = Heat input to the evaporator

Q_{FL} = Heat exchange between the forward line and the ambient

Q_{SC} = Amount of sub-cooling in the condenser

Heat transfer from the condenser can be divided into two parts- condensation of the vapor coming from the evaporator into saturated liquid and consequent subcooling of the saturated liquid leaving the condenser. Heat rejection by the condenser can be given as a function of time by the following expression-

$$Q_C = \dot{m}_{V,C}\ h_{fg} + Q_{SC} \tag{4.10}$$

The liquid from the condenser exits at a temperature lower than the saturation temperature, this amount of sub-cooling and hence the temperature of the liquid at the condenser outlet can be estimated from the following equation-

$$Q_{SC} = \dot{m}_C(t).C_P\,[T_{S,C} - T_{C,O}(t)] \tag{4.11}$$

where,

$T_{S,C}$ = Saturation temperature of the fluid in the condenser

$T_{C,O}(t)$ = Temperature of the liquid at the condenser outlet

C_P = Specific heat of the fluid at constant pressure

The liquid from the return line and condenser flows to the reservoir and is stored there to the end of the vapor transfer state. If the return line is not thermally insulated, then the temperature of the liquid drops slightly as it enters the reservoir. Heat exchange also occurs between the ambient and the reservoir by natural convection. Fantozzi et al. [28] have shown that the temperature difference between the reservoir and evaporator is directly proportional to pressure drop. So the temperature of the evaporator can be controlled by correctly designing the reservoir surface and controlling heat loss from the reservoir.

Heat loss from the reservoir can be given as

$$Q_R = U_R.\,A_R.\,[T_5(t) - T_\infty] \tag{4.12}$$

where,

$T_5(t)$ = Temperature of the fluid in the reservoir

U_R = heat transfer coefficient of reservoir

A_R = Area of the reservoir

4.5 THERMAL RESISTANCE OF THERMOLOOP

As previously mentioned, thermoloop can be considered as a thermal machine having three components transferring heat and mass at three different temperature levels. The components are the evaporator or the hot source, condenser or the cold sink and the reservoir working at an intermediate temperature. Several researches have tried to determine the operating temperature and other functional parameters of various thermosyphons by calculating the thermal resistance of the device. This can be a very effective approach in order to improve the performance of the thermoloop by minimizing all the thermal resistance associated with thermoloop operation.

Khodabandeh [31] has presented an analysis of the thermal performance of a closed advanced two-phase thermosyphon loop by investigating the thermal resistance of the device at different operating conditions. He calculated the overall thermal resistance of the device as a sum of the thermal resistances of five major components. He found that the thermal resistance between the heat source (heater) and the evaporator is the highest of all in spite of close contact and using high conductive thermal interface material.

The overall thermal resistance can be defined as the difference between the temperature of the vapor inside the evaporator and the cold sink divided by the heat removed by the condenser.

So, overall thermal resistance of thermoloop-

$$\left(R_{TH}\right)_{Overall} = \frac{T_{SOURCE} - T_{SINK}}{\dot{Q}} \tag{4.13}$$

So the determination of the total thermal resistance of thermoloop, both experimentally and by mathematical model can be very helpful in increasing the thermal efficiency of the device. It will be helpful also in the determination of the effectiveness of various components and in identifying the means to minimize their limitations. For example, the thermal resistance of condenser, evaporator and reservoir can be calculated using Eq. (4.13) and search can be made for minimizing those resistances.

An analytical model of the thermoloop is presented above to predict the behavior of thermoloop operation. To avoid complexity, single phase flow relations have been used in determining some pressure drops, which would have been more accurate if the two phase flow correlations were used. Heat balance of three main components of the thermoloop responsible for the heat transfer has been performed separately. The pressure drop characteristics of the thermoloop required for the heat transport states has been presented mathematically. Though this analytical model has not been verified or tested by the experimental results, it is expected to give fairly accurate and reasonable prediction of thermoloop behavior.

CHAPTER 5

RESULTS AND DISCUSSION

The objective of this study is to evaluate the thermal performance of the thermoloop prototype as a function of the operating and geometric parameters. The performance parameters of the thermoloop considered are the wall temperature of the evaporator, cycle frequency, condenser temperatures, heat transfer limit of the device and pressure developed in the system. The heat transfer performance of two thermoloop prototypes is evaluated for two condenser convection conditions and two working fluids, water and ethanol. But majority of the tests were performed for prototype-1 and with water as the working fluid. The heat input to the device was varied in an incremental manner to investigate the performance of the device at different thermal loads. Performance of the thermoloop device for different evaporator liquid fill ratios at all the heat inputs was also investigated in the present study. The height of the liquid column accumulated in the reservoir is taken as an indicator of the pressure developed in the system.

More than 50 complete sets of data were taken during the course of the present study to comprehend the effect of the different parameters mentioned above on the performance of the thermoloop device. The fill ratio was varied from 30% to 60% and the heat input was varied from 75 W to 275 W with an increment of 25 W for forced convection of condenser. The observed results for the variation in these parameters are described in details in this chapter. Results presented in the form of graphs are mostly for prototype-1 with water as the working fluid.

The boiling curve is drawn in the present study for prototype-1 for a fill ratio of 30% and with water as the working fluid. The boiling curve obtained for the experimental data was compared with the curve obtained by using Rohsenow correlation [32]. The saturation temperature of water inside the evaporator was taken as 100^0C considering the pressure increase in the evaporator to be small. The internal bottom surface of the

Table 5.1: Heat flux data for fill ratio of 30% with water for prototype-1 (With fan)

Heat Input Q_{IN} (W)	**Evaporator Wall Temperature T_E (^{0}C)**	**Liquid Saturation Temperature T_S (^{0}C)**	**Wall Superheat, ΔT_{SAT} (^{0}C)**	**Experimental Heat Flux, q_{exp} (W/ cm^2)**	**Heat Flux from Rohsenow Correlation q_{roh} (W/ cm^2)**
100	106.5	100	6.5	3.0	3.8
125	107.5		7.5	3.8	5.8
150	108.4		8.4	4.7	8.1
175	108.9		8.9	5.5	9.6
200	109.2		9.2	6.4	10.6
225	110.6		10.6	7.3	16.3
250	110.8		10.8	8.1	17.3

evaporator was taken as the heat transfer area. The heat flux values are given in Table 5.1 and the resulting plot is shown in Fig. 5.1. Detailed calculation of the heat flux values is shown in Appendix A-2.

5.1 THE NATURE OF HEAT TRANSPORT CYCLE

As already mentioned, the heat transport cycle of thermoloop device comprises of two repeating states- vapor transfer state and condensate return state. When the condensate returns to the evaporator from the reservoir, the cycle is completed. The variation of the evaporator wall temperature, condenser temperatures and accumulated liquid column height in the reservoir during the heat transport cycle is recorded for each step of heat input. Figure 5.2 shows the variation of evaporator wall temperature with time for a heat input of 100 watt with water as the working fluid at 30% fill ratio.

From Fig 5.2, it is clear that the temperature of the evaporator wall increases gradually up to a temperature of 107^0C and boiling initiates when water is used as the working fluid. It remains almost constant there for a considerable period until all the liquid inside the evaporator evaporates completely. Then the temperature drops to a value of 75^0C when the water from the reservoir flows back into the evaporator- completing the cycle. The same process is repeated in the next cycles. The similar nature in the variation of evaporator wall temperature is observed when ethanol is used as the working fluid. In this case, the temperature of the evaporator wall reaches a maximum value of about 95^0C for the same heat input (Fig. 5.3). In this way, the temperature of the evaporator wall oscillates slightly below and above the boiling point of working fluid. From Fig. 5.2 and 5.3 it can also be observed that the cycle time for ethanol (about 9 minutes) is significantly lower than that for water (about 20 minutes) for the same heat input and evaporator fill ratio.

Height of the liquid column accumulated in the reservoir can serve as an indicator of the variation of pressure within the evaporator. Figure 5.4 shows the variation of water column height in the reservoir with time for a heat input of 100 W with water as the working fluid at 30% fill ratio. As soon as the temperature of the evaporator starts to increase with the heat input, liquid in the forward line starts moving forward towards the condenser and then from condenser towards the reservoir by the associated increase in pressure. The liquid is continuously pushed out from the forward line, condenser and liquid line into the reservoir as long as evaporation of the liquid continues inside the evaporator. The water column height in the reservoir then becomes constant for a while when all the liquid inside the evaporator is completely vaporized and no liquid is pushed out of condenser into the reservoir. As soon as the pressure inside the evaporator decreases below the pressure inside the reservoir, liquid flows back quickly into the evaporator through the out valve, leaving the reservoir almost empty and closing the cycle. The temperature of the liquid after coming back to the evaporator starts to increase and the pressure inside the evaporator begins to increase again as vaporization starts. The pressure inside the evaporator continues to pulsate in this periodic manner.

Nature of variation of condenser inlet and outlet temperature with time for the same test is shown in Fig. 5.5. The condenser inlet temperature varies in similar manner as the evaporator wall temperature. It reaches a maximum point after increasing linearly initially, then remains almost steady at this value for some period and then decreases as the cycle completes. But the condenser inlet temperature reaches its maximum value few minutes (about 2 min in this case) after the evaporator wall temperature reaches its maximum steady value. Also the maximum value at which the condenser inlet temperature remains steady is about 90^0C, which is 16^0C lower than the maximum temperature of the evaporator wall for this heat input with water as working fluid. The condenser outlet temperature remains nearly constant throughout the cycle (the maximum variation is less than 2^0C) for this particular test.

The similar scenario is observed in the case of ethanol as the working fluid. The maximum condenser inlet temperature is about 78^0C, which is about 18^0C lower than maximum temperature of the evaporator wall (Fig. 5.3). Also, the condenser inlet temperature reaches its steady pick few minutes after the evaporator wall temperature reaches its maximum value. The condenser outlet temperature remains almost constant in this case too. But this temperature varied differently for other tests under different convection condition and different thermal load for both working fluids, which will be shown in the later parts.

The maximum heat flux dissipated from the heat source by the thermoloop device in this study was found to be 18.5 W/cm^2 if the heater surface is considered to be the heat transfer area, which is the actual surface area of the heat source. The value of maximum heat flux is 1688 W/cm^2 if the cross sectional area of the tubing is taken as the heat transfer area. The later value is presented here due to the ambiguity in calculating the heat flux found in the literature.

5.2 EFFECT OF THERMAL LOAD ON VARIOUS PARAMETERS

One of the primary objectives of the present study was to determine the effect of thermal load or heat input on the various heat transfer performance parameters of the thermoloop device. The heat input was varied from a minimum 75 W to a maximum of 275 W for the forced convection condition of the condenser with water as working fluid. The power was varied in an incremental manner with an increase of 25 W in each step. Heat input was varied from 75 W to a maximum of 125 W for the natural convection condition with water as the working fluid.

5.2.1 Evaporator Wall Temperature

Evaporator wall temperature is the most important operating temperature of the thermoloop as it resembles the temperature of the microchip to be cooled. Figure 5.6 shows the effect of thermal load on the maximum temperature of the evaporator wall for a fill ratio of 30% with water as working fluid. The maximum temperature of the evaporator is plotted for the first 12 heat transport cycles at 4 different heat inputs. The maximum temperature of the evaporator was found to increase with increase in the heat input. The maximum temperature of the evaporator wall was 112^{0}C for a heat input of 250 W, which was about 6^{0}C higher than the same for a heat input of 100 W (106^{0}C).

The maximum evaporator wall temperature for the same heat input was found to be almost constant, the variation for different cycle being less than 2^{0}C. It can be noted from the Fig. 5.6 that after the initial unsteadiness, the evaporator wall temperature becomes nearly steady after 4-5 heat transport cycles. This capability of thermoloop device to maintain steady maximum temperature of the evaporator wall is a very important attribute in favor of its applicability for electronics cooling, especially when the electronic component to be cooled is in operation for a long time. Same characteristics are observed for all other fill ratios tested. Figure 5.7 shows similar characteristics for an evaporator fill ratio of 60%.

The minimum temperature of the evaporator wall is also a very important parameter because the minimum and the maximum temperature of the evaporator wall indicate the range within which the temperature of the electronic component to be cooled will oscillate. This variation of minimum wall temperature for a number of heat transport cycles at 4 thermal loads is shown in Fig. 5.8 for a fill ratio of 30% with water. The figure shows that the minimum wall temperature increases after the 1st heat transport cycle, and then attains a steady value after 4-5 cycles. From the Fig.5.8 it is also seen that the minimum temperature of the evaporator wall increases with increasing thermal load. The values of the maximum and minimum evaporator wall temperature for 4 different heat transport cycles and 4 heat loads are presented in Table 5.2.

Table 5.2: The maximum and minimum temperature of evaporator wall for 4 different heat loads at a fill ratio of 30% with water as the working fluid (Prototype-1, with fan)

Heat Input Q_{IN} (W)	Cycle Number	Temperature of the Evaporator Wall (^0C)		
		T_E (max) (^0C)	T_E (min) (^0C)	Difference in Temperature (^0C)
100	2	105.4	70.8	34.6
	4	105.5	74.8	30.7
	6	105.4	73.8	31.6
	8	105.8	73.7	32.1
150	2	106.9	72.3	34.6
	4	108.4	76.7	31.7
	6	108.4	79.1	29.3
	8	108.4	79.0	29.4
200	2	109.2	73.5	35.7
	4	109.1	82.5	26.6
	6	109.9	88.4	21.5
	8	109.2	86.9	22.3
250	2	109.8	85.8	24.0
	4	111.3	91.3	20.0
	6	110.1	95.8	14.3
	8	110.8	97.1	13.7

It can also be noted from Fig. 5.7 and Fig. 5.8 and Table 5.2 that the range within which the evaporator wall temperature oscillates decrease with an increase in the thermal load. The steady evaporator wall temperature varies from 91^{0}C to 111^{0}C (range 20^{0}C) for a thermal load of 250 W while it varies from 74^{0}C to 106^{0}C (range 32^{0}C) for a thermal load of 100 W. Similar characteristics are obtained for other fill ratios and for other heat inputs tested.

5.2.2 Condenser Wall Temperature

The variation of condenser inlet and outlet temperature for different thermal loads is also shown in Fig. 5.9 and Fig. 5.10. The maximum temperature of the condenser inlet is nearly constant for all the fill ratios and all heat inputs, which is about 90^{0}C for water. The variation of minimum condenser inlet temperature is plotted in Fig. 5.9 for 4 different heat loads with water as the working fluid and when the condenser was cooled by a fan. From the Fig. 5.9 it is seen that the minimum temperature at the condenser inlet increases with increasing heat input. The minimum condenser inlet temperature attains a steady value after 5-6 cycles. The minimum temperature at the condenser inlet is about 26^{0}C for a heat load of 100 W while it is about 63^{0}C for a heat load of 250 W. So the heat dissipation capacity of the condenser reaches a limit as the heat input to the device increases. The variation in the minimum condenser inlet temperature is very similar in nature to the variation of the minimum evaporator wall temperature for different heat inputs (Fig. 5.8).

The maximum temperature of the condenser outlet is another crucial parameter because it signifies the efficiency of the condenser. The lower the maximum temperature of the condenser outlet, the higher it is able to dissipate heat. Figure 5.10 shows the effect of thermal load on the condenser outlet temperature for 50% fill ratio with water as the working fluid. From the Fig. 5.10 it is seen that the maximum temperature of condenser outlet increases with increasing thermal load. Condenser outlet temperature is almost constant at a value lower than 30^{0}C for heat inputs of 100 W and 150 W. But the steady condenser outlet temperature is as high as 65^{0}C and 87^{0}C for the heat inputs of 200 W and 250 W respectively.

In the present study, the values of condensation heat transfer, Q_C and overall heat transfer coefficient of condensation, U were calculated for 4 different heat inputs and 2 fill ratios with water as the working fluid. The following equations were used to determine these values-

$$Q_C = \dot{m}\,h_{fg} + \dot{m}\,C_P(T_{SAT} - T_{OUT}) \tag{5.1}$$

$$Q_C = U.A_C.LMTD \tag{5.2}$$

The values of these parameters and amount of heat loss to the ambient from the evaporator and forward line for two fill ratios (30% and 50%) and forced convection condition at condenser are shown in Table 5.3 for 4 heat loads. From this table it can be seen that, about 90% of the heat dissipated by the condenser is due to the phase change of vapor by condensation and the remaining 10% heat is dissipated by subcooling of the saturated liquid to the condenser outlet temperature. The total amount of heat dissipated by the condenser at different heat load is nearly same for the two fill ratios. The maximum value of U is 621 $W/m^2.K$ for 250 W heat load at 50% fill ratio. The amount

Table: 5.3 Calculation of condensation heat transfer (Q_C) and overall heat transfer coefficient (U) for forced convection condition with water as working fluid

Fill Ratio	Heat Input Q_{IN} (W)	Heat Loss Q_L (W)	Condenser Heat Dissipation (W)			Log Mean Temperature Difference (LMTD) (^0C)	U ($W/m^2.K$)
			By Phase Change Q_{PC} (W)	By Subcooling Q_{SC} (W)	Total Heat Dissipation Q_C (W)		
30%	100	16.1	64.3	9.2	73.5	21.6	106.9
	150	19.1	98.8	13.3	112.1	21.1	167.1
	200	21.7	157.9	16.3	174.2	18.1	302.6
	250	24.5	203.5	11.5	215.1	13.4	504.8
50%	100	16.2	70.1	10.2	80.3	21.6	116.9
	150	19.0	104.2	14.4	118.6	21.3	175.2
	200	21.7	155.9	13.2	169.06	18.7	284.3
	250	24.4	200.0	9.4	209.4	10.6	621.3

of heat loss to the ambient from the evaporator surface, the forward line and connectors is also presented in the table, which is about (10-16) % of the heat input to the evaporator. The variation of condenser heat dissipation (Q_C) and overall heat transfer coefficient of condensation (U) is plotted against heat input to the evaporator (Q_{IN}), as shown in Fig. 5.11 for a fill ratio of 30% with water as the working fluid. It can be noted from this figure that both condenser heat dissipation and overall heat transfer coefficient increases with increasing heat load. The maximum value of Q_C was 215 W for a heat load of 250 W at 30% fill ratio.

In the determination of the above parameters, temperature of the fluid at the condenser inlet was assumed to be 2^0C lower than the temperature of the fluid at the evaporator which is in saturation temperature. The values of enthalpy of evaporation were taken at inlet temperature of the condenser. The calculation for the determination of the above parameters is presented in detail in Appendix A-3.

5.2.3 Liquid Column in the Reservoir

In this study, the height of the water column accumulated in the reservoir serves as an indicator of the pressure developed within the system. The higher the pressure is developed in the evaporator due to evaporation of the liquid inside it, the greater the amount of liquid accumulated in the reservoir. Figure 5.12 shows the effect of variation in thermal load on the maximum height of water column accumulated in the reservoir for different cycles at 30% fill ratio with water as working fluid. The maximum height of water column increased with an increase in the thermal load. The maximum water column height was also observed to be constant for all the cycles for same heat input. For a thermal load of 250 W, the maximum height is about 4.2 cm while it is about 3.7 cm for 200 W, 3.2 cm for 150 W and about 2.8 cm for a heat input of 100 W. Similar trend of the effect of thermal load on the maximum liquid column height was observed for the other fill ratios tested.

5.2.4 Cycle Time or Frequency

The cycle time for a heat transport cycle is taken as the summation of the time required for vapor transport state and condensate return state. The lower the cycle time, the quicker the device completes these two repeating states. In the present study, it was found that the cycle time is greatly influenced by the variation in thermal load. Cycle time is reduced with an increase in the thermal load as shown in Fig. 5.13 for a fill ratio of 30%. Again like the maximum and minimum evaporator wall temperature and condenser temperatures, cycle time attains a steady value after 4-5 cycles for all the heat input. This trend was observed for all the fill ratios tested, as shown in Fig. 5.14 for a fill ratio of 60%.

This result is different from the simulation tests performed by Fantozzi et al. [28] as shown in Fig. 5.15. From the Fig. 5.15 it can be seen that the simulated results showed almost equal cycle time for all the heat transport cycles for a constant heat load. Alam [5] also reported similar results showing nearly equal cycle time for all the heat transport cycles beginning from the 1st cycle. In the present study, the cycle time reduces after the 1st cycle because when the liquid from the reservoir returns into the evaporator, it is already at a very high temperature. So the liquid is evaporated very quickly and the pressure develops in the system rapidly. Then as the minimum and maximum temperatures of the evaporator wall, condenser temperatures and pressure in the system attains a steady value after few cycles, the cycle time also becomes constant as all the process parameters are then constant. The same trend was observed for all other fill ratios tested, as shown in Fig. 5.13 and Fig. 5.14.

The steady cycle time is about 800 s, 370 s, 200 s and 140 s respectively for heat inputs of 100 W, 150 W, 200 W and 250 W for a fill ratio of 30% with water as the working fluid, as can be seen from Fig 5.16. From this figure, it can be noted that the cycle time decreased rapidly as heat input was increased from 100 W to 150 W, but decreased only moderately as heat input was increased from 150 W to 200 W. After the high heat input of 225 W, cycle time became almost constant.

5.3 EFFECT OF EVAPORATOR FILL RATIO ON VARIOUS PARAMETERS

Polasek and Rossi [14] identified that the optimum filling ratio of the evaporator of a tubular loop thermosyphon influences its maximum thermal performance. They presented the experimental results showing the comparison of thermal resistance in a tubular copper thermosyphon with two initial fill ratios: 25% and 75%. They concluded that for loop thermosyphons in electronic application, an initial evaporator fill ratio of 25% should be adopted.

Yang et al. [33] found that a filling ratio of 50% showed the maximum performance limit and lowest thermal resistance in all orientations for Closed Loop Pulsating Heat Pipe. Murthy [34] reported that the two-phase spreader performance was fairly insensitive to the variation in liquid fill volume. Ramaswamy et al. [35] noticed a similar trend in their study on the performance of a compact two-chamber two-phase thermosyphon.

Fantozzi et al. [27] also reported the results of the simulation test of the effect of fill ratio on the wall temperature of the evaporator. They observed that the maximum evaporator wall temperature increases with an increase in the evaporator fill ratio (Fig. 5.17). They also showed that the cycle time is also increased with increasing evaporator fill ratio.

Any previous work on the study of the influence of evaporator fill ratio on the heat transfer performance of the thermoloop device has not been found. In the present study, the effect of evaporator fill ratio on the heat transfer performance of the thermoloop has been studied. The total volume of the device filled with liquid excluding the evaporator was about 45 cc (condenser, forward line, part of the return line) and the volume of liquid in the evaporator was varied from 22.5 cc to 45 cc. The device was tested mostly for 30%. 50% and 60% fill ratio at different heat inputs. Only for the heat input of 75 W with water as the working fluid, fill ratio was varied from 20% to 80%.

5.3.1 Cycle Time

Figure 5.18 shows the effect of evaporator fill ratio on the cycle time required at three different thermal loads (100 W, 150 W and 250 W) with water as the working fluid for a number of heat transport cycles. It shows that for the heat input of 100W, the cycle time is about 100 sec lower for 50% fill ratio than 30% and 60% fill ratios, where the later two have nearly equal cycle time. For the heat inputs of 150 W and 250 W, the cycle times for all three fill ratios are very close. The results indicate that the variation in cycle time for different fill ratios decreases with an increase in the thermal loads. At high thermal loads, the cycle time becomes almost independent of fill ratios.

5.3.2 Evaporator Wall temperature

Fantozzi et al. [28] have reported the simulation results of the effect of fill ratio on evaporator wall temperature for the miniature PTPT device as shown in Fig. 5.17. The figure shows that the evaporator wall temperature increases with increasing fill ratio for a particular heat transport cycle. The maximum wall temperature increased from 63^0C at 30% fill ratio to 66^0C at 80% fill ratio, a difference of 3^0C.

In the present study, influence of evaporator fill ratio on the maximum temperature of the evaporator wall at two heat inputs (100 W and 250 W) is exhibited in Fig. 5.19. The results of this study were different than that obtained in the simulation test by Fantozzi et al. [28]. For the heat input of 250 W, wall temperature is maximum for 50% fill ratio which is about 2^0C higher than the maximum wall temperature for 30% and 60% fill ratios. But for the heat input of 100 W, evaporator wall temperature is the maximum for 60% fill ratio, whereas the wall temperature is nearly equal for 30% and 50% fill ratios. The variation in the maximum wall temperature is again less than 2^0C for this case. Thus it can be concluded from the present work that at a given load, the maximum temperature of the evaporator wall is fairly insensitive to the variation of evaporator fill ratio for thermoloop.

5.3.3 Liquid Column in the Reservoir

Figure 5.20 depicts the effect of fill ratio on the maximum height of the water column in the reservoir for three different fill ratios (30%, 50% and 60%) at a heat input of 100 watt for water. It can be seen from this figure that the maximum water column height increases with increasing fill ratio. The height of the liquid column is about 3.9 cm, 3.5 cm and 2.75 cm for 60%, 50% and for 30% fill ratio respectively. The similar trend was obtained for other heat inputs as well. The reason is that the higher the fill ratio, the greater the amount of liquid to be evaporated and consequently the higher is the pressure. Therefore, more liquid is pushed out of the liquid line and condenser into the reservoir, thereby increasing the liquid column height in the reservoir.

5.4 EFFECT OF CONDENSER CONVECTION CONDITION ON VARIOUS PARAMETERS

Pal et al. [4] has reported the results of the numerical simulation performed on a compact thermosyphon to reflect on the efficiency of the condenser in natural convection condition and the effect of the system fan on the performance of the condenser. They adopted a system based numerical approach to model the inside of the PC cabinet. For the case with fan, they found that for an average condenser tube wall temperature of 80^0C, the net heat dissipation from the condenser is about 66 W. The same boundary condition was used for the case where the system did not have a fan. It was found that the condenser subjected to natural convection is able to dissipate 60% of the amount of heat dissipated by the condenser subjected to forced convection. So they concluded that the presence of fan creates a big difference in heat dissipation and improves the performance of the condenser a great deal.

In the present study, the effect of convection condition of the condenser on the overall heat transfer performance of thermoloop has been investigated. Also the temperature at the condenser inlet and outlet was measured and its impact on the system was analyzed. A cooling fan was mounted at a distance of 3 cm from the condenser which generated an average axial air velocity of 4.75 m/s.

5.4.1 Evaporator Wall Temperature

The variation of evaporator wall temperature for both natural convection and forced convection condition of the condenser at a heat input of 100 watt with water as working fluid is shown in Fig. 5.21 for a fill ratio of 30%. It can be seen from this figure that the temperature of the evaporator wall did not vary significantly for the variation of cooling condition of the condenser for this heat input. The maximum temperature of the evaporator wall is a fraction higher for the natural convection condition than when a fan is used to cool the condenser. The minimum temperature of the evaporator wall temperature is also higher in the case of natural convection condition. Thus throughout the operation, the temperature of the evaporator wall is slightly higher when the condenser is cooled by natural convection than when a fan is used.

5.4.2 Operating Limit of Thermoloop for Natural Convection Condition

At low heat input such as 75 W, performance of thermoloop was found to be relatively independent of the convection condition of the condenser. But higher heat input has resulted in comparatively poor performance of thermoloop under natural convection of the condenser. Figure 5.21 has shown that the temperature of the evaporator wall was greater in the case of natural cooling of condenser than when a fan was used. So in order to find the heat transfer limit of the device, it was tested at a higher heat input.

Performance test of the device at a higher load of 150 W with water as working fluid has resulted in the complete failure of the device when the condenser was cooled by natural convection, as shown in Fig. 5.22. The process had to be stopped as the dry out of the evaporator occurred and the temperature of the evaporator wall reached as high as 170^{0}C. The temperature at the condenser inlet and outlet was nearly equal (both at 97^{0}C) and the liquid stored in the reservoir was extremely hot. The pressure inside the system was very high as the condenser could not dissipate enough heat from the working fluid. As a result, return of the liquid from the reservoir to the evaporator did not occur and temperature of the evaporator started to rise. Figure 5.22 shows the comparison of the evaporator wall temperatures at two different cooling conditions of the condenser. From this figure it can be seen that the temperature of the evaporator increases up to a very high temperature of 170^{0}C in a very short time and the process had to be stopped.

More tests were performed with a heat input of 135 W with water as working fluid and for fill ratios of 30% and 50% to further investigate the thermal limit of the device without cooling fan. These tests also had to be stopped as the temperature of the evaporator reached a very high value (above 160^{0}C) within a very short time (Fig. 5.23). So the maximum thermal load at which the prototype-1 can function without fan was found to be 125 watt whereas the same device was operated for a thermal load of 275 W without deterioration of performance when a fan was used.

Condenser cooled by natural convection has higher overall thermal resistance than the forced convection condition due to greater thermal resistance between the outer surface of the condenser and the ambient.

It can be expressed mathematically as –

$$(R_{TH})_C = \frac{T_{OUT} - T_\infty}{\cdot Q_C} \tag{5.3}$$

where $(R_{TH})_C$ is the thermal resistance between condenser and ambient, T_{OUT} is the temperature of the outer surface of the condenser, Q_C is the amount of heat dissipated by the condenser and T_∞ is the temperature of the ambient air. From Eq. (5.3) it can be seen that, the higher the temperature of the condenser surface, the higher is the thermal resistance. Temperature of the condenser is very high for natural convection condition, leading to higher thermal resistance and consequently to poor performance.

The value of thermal resistance and heat dissipated by the condenser under forced convection and natural convection conditions for 3 different heat input at 30% fill ratio with water as the working fluid is presented in Table 5.4. The detailed calculation is presented in Appendix A-4.

Table 5.4: Thermal resistance of condenser under forced convection and natural convection conditions for 3 different heat input at 30% fill ratio with water

Heat input (W)	Condenser Outlet Temperature T_{OUT} (^{0}C)		Condenser Heat Dissipation (W)						Thermal Resistance R_{Th} (m^2.K/W)	
			By Phase Change (Q_{PC})		By Subcooling (Q_{SC})		Total Heat Dissipation, (Q_C)			
	With Fan	No Fan	With Fan	No Fan	With Fan	No Fan	With Fan	No Fan	With Fan	No Fan
75	23.2	75	57.2	56.7	8.2	2.6	65.4	59.3	0.018	0.89
100	23.4	95	64.3	62.7	9.2	1.8	73.5	64.5	0.019	0.97
125	24.7	96	87.5	72.7	12.5	1.9	100	74.6	0.027	0.99

From Table 5.4, it can be seen that the amount of heat dissipated by the condenser is lower for natural convection condition than when it is cooled by a fan. For the heat input of 125 W, the heat dissipated by the condenser was 100 W for forced convection, while it was only 74.6 W when there was no fan. Heat dissipation due to phase change and amount of subcooling both decreases for natural convection condition, especially at higher heat input. The amount of subcooling without a fan (1.9 W) was only 15 % of the amount of subcooling for forced convection condition (12.5 W) at the same heat input of 125 W, whereas the power consumed by the fan was 3.25 W for the forced convection condition. As a result, the thermal resistance of the condenser is about 40-50 times higher when there was no fan to cool the condenser. Therefore, the thermoloop device reaches its operation limit as it can not dissipate enough heat when it is operated without a fan.

Thus the addition of fan had a very significant effect on the heat transfer performance of the thermoloop. Forced convection considerably increased the thermal load limit in which the device can operate successfully. Thus the inclusion of a fan to cool the condenser is a very important requirement for higher heat dissipation rate and greater operating range of the thermoloop.

5.4.3 Condenser Wall Temperature

Variation of condenser inlet and outlet temperatures for the variation in convection condition at a heat input of 100 W is shown in Fig. 5.24 and Fig. 5.25. Condenser inlet temperature varies in a similar periodic manner for both cases. But the maximum temperature at the condenser inlet is about 10^0C higher in all the cycles when there is no fan to cool the condenser (Fig. 5.24). Convection condition at the condenser showed more pronounced effect on the outlet temperature of the condenser. Figure 5.24 shows the variation in this temperature for both natural and forced convection at a heat input of 100 W with water as working fluid. The condenser outlet temperature is varied in a periodic manner similar to that of inlet temperature under natural convection condition and the maximum temperature of the condenser outlet is only 0.5^0C lower than the inlet temperature. So heat dissipation capacity of the condenser almost reached the limit as both inlet and outlet temperatures of the condenser have reached nearly the same value. But for the case of forced convection, the outlet temperature is very low and it is almost constant for all the cycles at the room temperature of 25^0C. Thus the forced convection condition offers much higher heat dissipation rate compared to the natural cooling of the condenser.

5.4.4 Cycle Time

Convection condition of the condenser also exhibited significant effect on the cycle time or frequency. Figure 5.26 shows the influence of cooling condition of condenser on the cycle frequency of the thermoloop device at a heat input of 125 W with water as working fluid. The cycle time decreases more for forced convection than when there was no fan. The steady cycle time for forced convection condition is about 4 minutes less than that of the natural convection condition. But for a lower heat input of 75 W and 100 W, effect of condenser convection condition on the cycle time was less significant. Cycle times for both these conditions were nearly equal for a heat input of 75 W and only a fraction less (less than 1 minute) in the case of a heat input of 100 W. Therefore, it can be concluded from the results that the effect of condenser cooling condition on the cycle time is more prominent for higher heat inputs. The reason is that as the condenser reaches the limit of its heat dissipation rate (inlet and outlet temperature being equal), its efficiency decreases and more time is needed to reduce the pressure of the system. As a result, cycle frequency increases when there is no fan. The effect of condenser convection condition on cycle time for the thermal load of 100 W with water as the working fluid is shown in Fig. 5.27.

5.5 EFFECT OF LEAKAGE IN THE DEVICE

The most suitable fluid to be used for electronics component cooling by thermoloop is known to be different types of dielectric fluid such as FC-70 or FC-72 etc., as discussed in the section 3.1.6. The two working fluids used in the current study were water and ethanol. Water was selected because of its excellent thermophysical properties and wide availability. Ethanol was selected for its low boiling point (necessary for many electronic components' maximum allowable temperature) and its well known suitability for this kind of passive heat transfer device. Both these working fluids adequately meet the demand of this present work of determining the thermal performance of thermoloop.

But in the case of these fluids other than the dielectric fluids, perfect sealing of the device has to be maintained as any leakage of the liquid will cause serious damage to the electronic components. In the present work, it was found that apart from the possible damage of the electronic components, the leakage significantly affects on the heat transfer performance of the device.

In the present study, during some tests, leak occurred even after the primary checking to prevent leakage. Leak occurred most frequently in the joints, especially in the connection between evaporator and tubing and also in the insertion point of the two check valves in the return line. The effect of leakage in any part of the device was reflected in the heat transfer nature of the cycle. The major effect of the leakage observed was the notable increase in the cycle time. The liquid coming out of the condenser towards reservoir was observed to move back and forward from condenser in an oscillating manner. The reason is that due to the leakage in the device, the high pressure developed by the evaporation of the liquid could not be maintained within the system. The pressure dropped due to leakage and so the liquid coming out of condenser started to oscillate in and out of the condenser. This pressure pulsation and flow oscillation continued for a long time and the cycle completed with a much higher cycle time. During one or two tests, the oscillation of the liquid back and forward in the return line continued for very long time and the temperature of the evaporator started to shoot off to a very high value, so the process had to be stopped.

Thus the prevention of leakage is extremely important even if the device is operated with dielectric fluid. Leakage not only increases the chance of possible damage of the electronic components and environment, but also can cause the failure of the proper functioning of the device.

5.6 EFFECT OF FLOW CONTROLLERS

The two flow controllers, placed on either side of the reservoir are very important for the proper functioning of the thermoloop device because they ensure the liquid and vapor flow in the desired directions, which is discussed earlier in the section 3.1.5. The first check valve (in valve) maintains that the liquid flow occurs only from condenser to the reservoir and the second check valve (out valve) allows flow from reservoir to the evaporator only. By the proper functioning of these check valves, the return of liquid to the evaporator is made possible.

In the present study, the effect of failure of any of these check valves on the performance of thermoloop device was observed during some tests. After continuous use of the same check valves for a number of tests, it was observed that the check valves fail to maintain the flow in the desired direction. This was true especially for the case of the one way in valve which had to be changed after a few tests as it failed after continuous operation for a number of tests. Back flow from the reservoir through the check valve to the condenser occurred, which is in complete disagreement with the working principle of the thermoloop. The one way out valve also had to be replaced after a certain number of tests, but at a lower frequency than the in valve.

In the case of failure of the in valve, liquid from the reservoir started flowing back into the condenser. But due to the high pressure in the evaporator, this liquid was again pushed back from the condenser towards the reservoir. This pulsation of pressure and oscillation of the liquid continued for a significant period of time. When all the liquid inside the evaporator was vaporized, then pressure started to decrease in the evaporator and the liquid oscillating in the return line between condenser and reservoir flowed back into the condenser. It then started oscillating in and out of the condenser in the forward line, just in front the condenser inlet. This continued for a certain period of time and then the temperature of the evaporator started to rise rapidly and within a few moments, it reached a value as high as 160^0C and the process had to be stopped. Failure in the out valve also caused the failure of the device.

For the proper functioning of the thermoloop device, the flow controllers play an important role by maintaining desired flow directions of liquid and vapor. The plastic flow controllers that were used in the present study failed to work properly after continuous operation and had to be replaced. Therefore, depending on the working fluid and pressure in the system, selection of stronger and proper metal check valves with the optimum opening pressure is a very important design requirement.

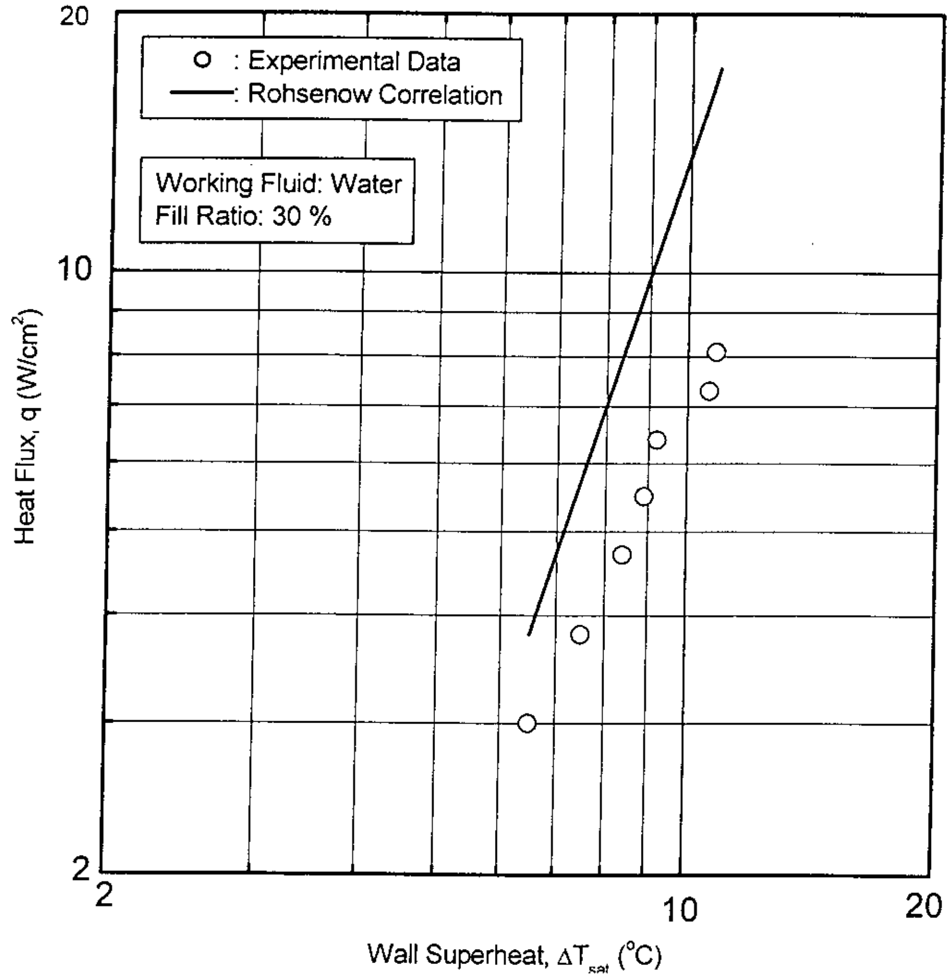


Figure 5.1: Boiling curve for fill ratio of 30% with water as the working fluid (Prototype-1, with fan)

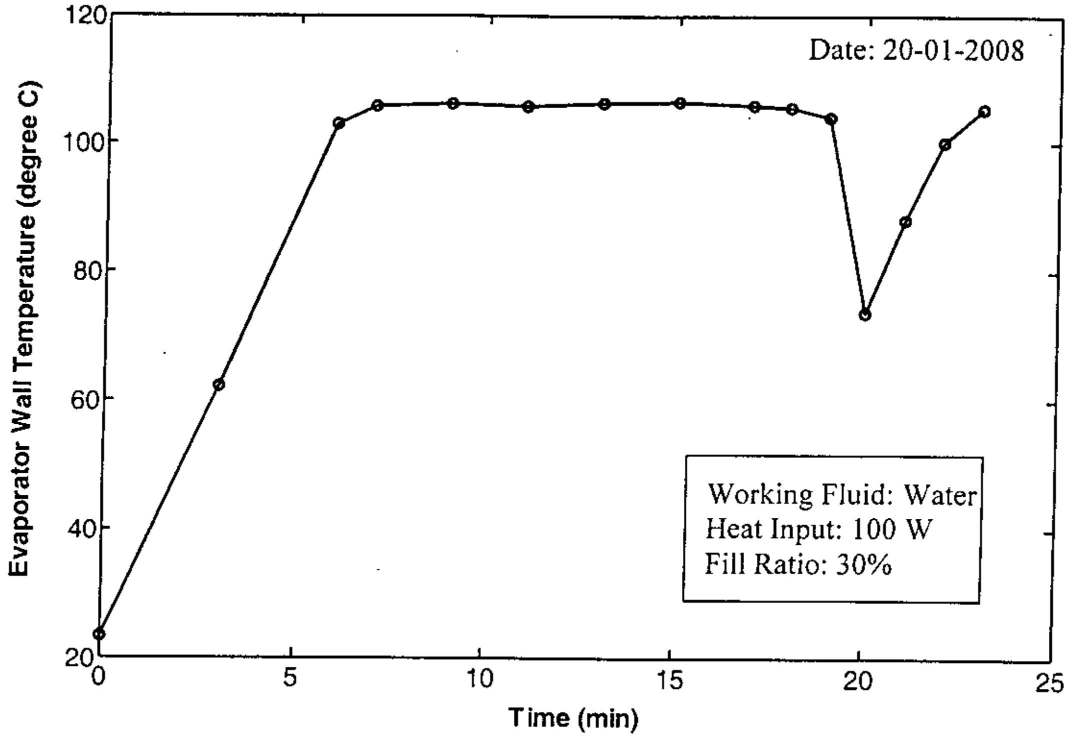


Figure 5.2: Variation of evaporator wall temperature for a heat input of 100 W with water as the working fluid at 30% fill ratio (Prototype -1, with fan)

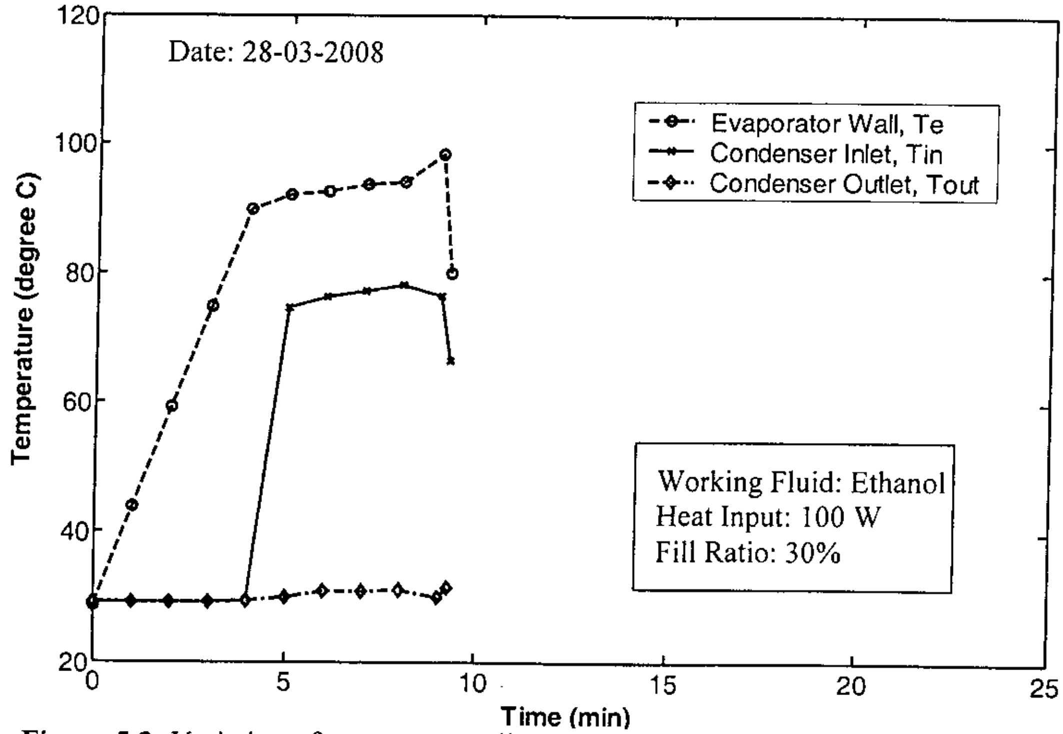


Figure 5.3: Variation of evaporator wall temperature and condenser inlet and outlet temperature for a heat input of 100 W with Ethanol (Prototype -2, with fan)

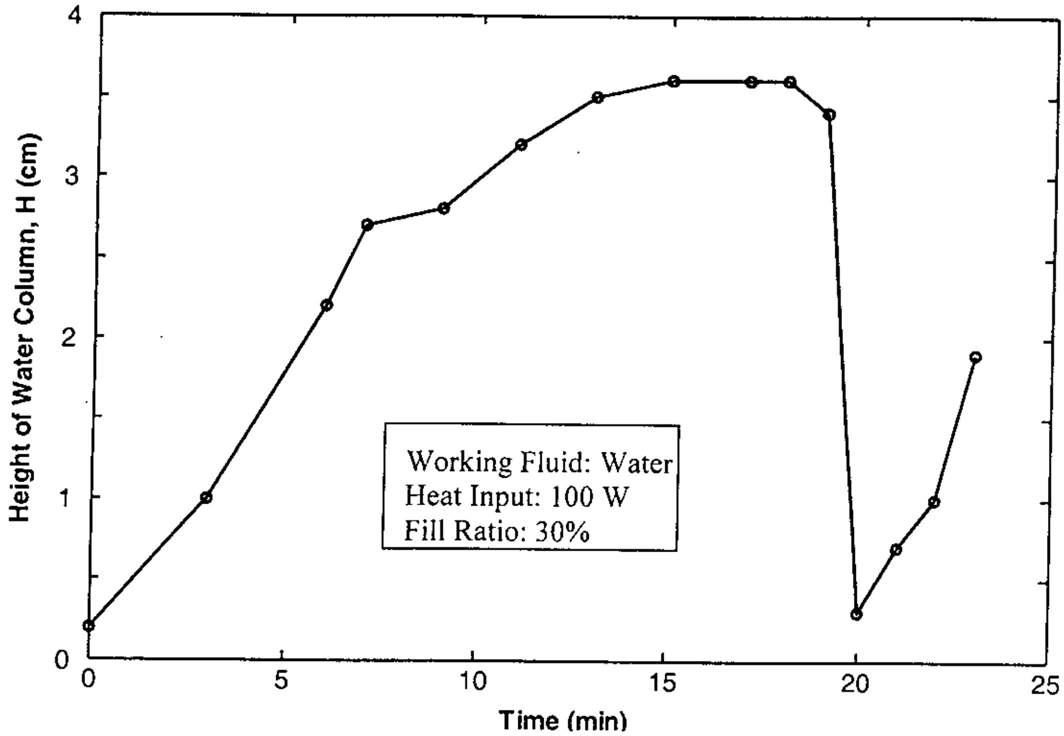


Figure 5.4: Variation of height of water column in the reservoir for a heat input of 100 W with water as the working fluid at 30% fill ratio (Prototype-1, with fan)

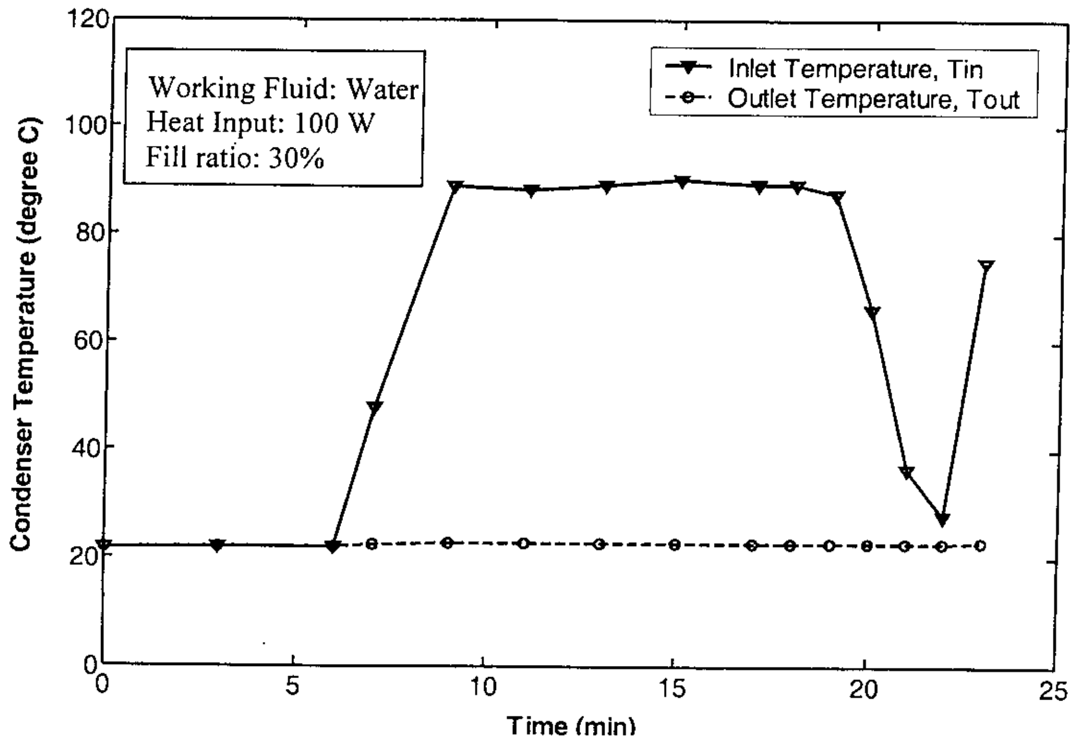


Figure 5.5: Variation of condenser inlet and outlet temperatures with time for a heat input of 100 W with water as the working fluid at 30% fill ratio (Prototype -1, with fan)

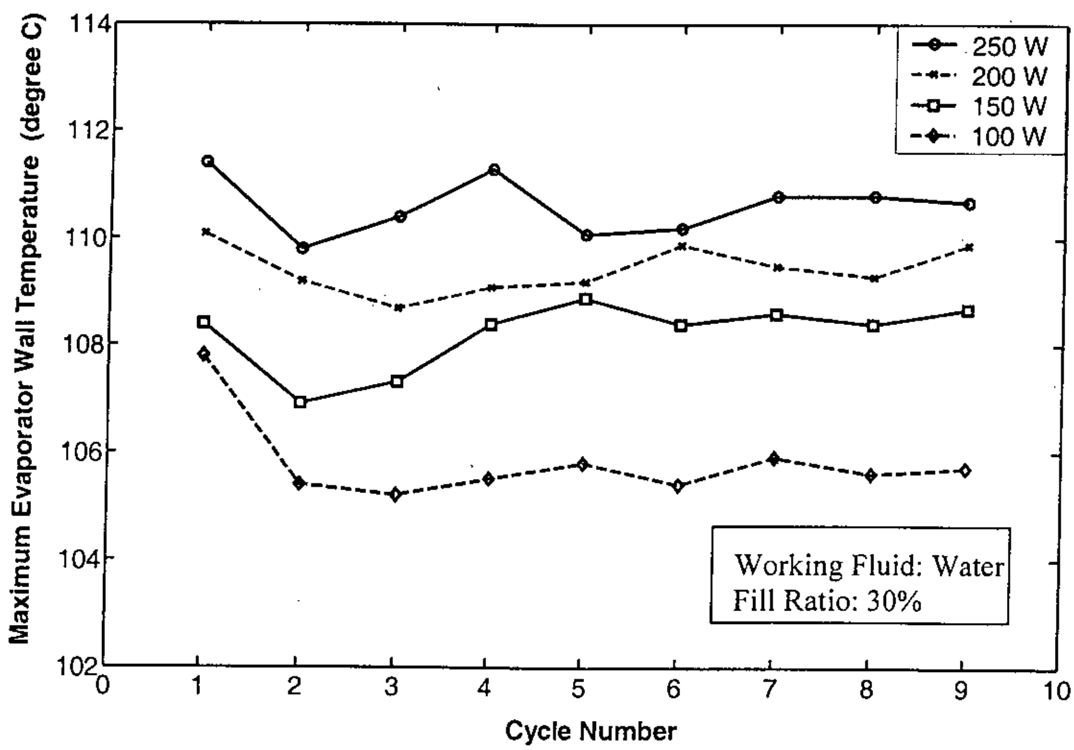


Figure 5.6: Variation of maximum temperature of the evaporator wall for water at 4 different heat loads at 30% fill ratio (Prototype-1, with fan)

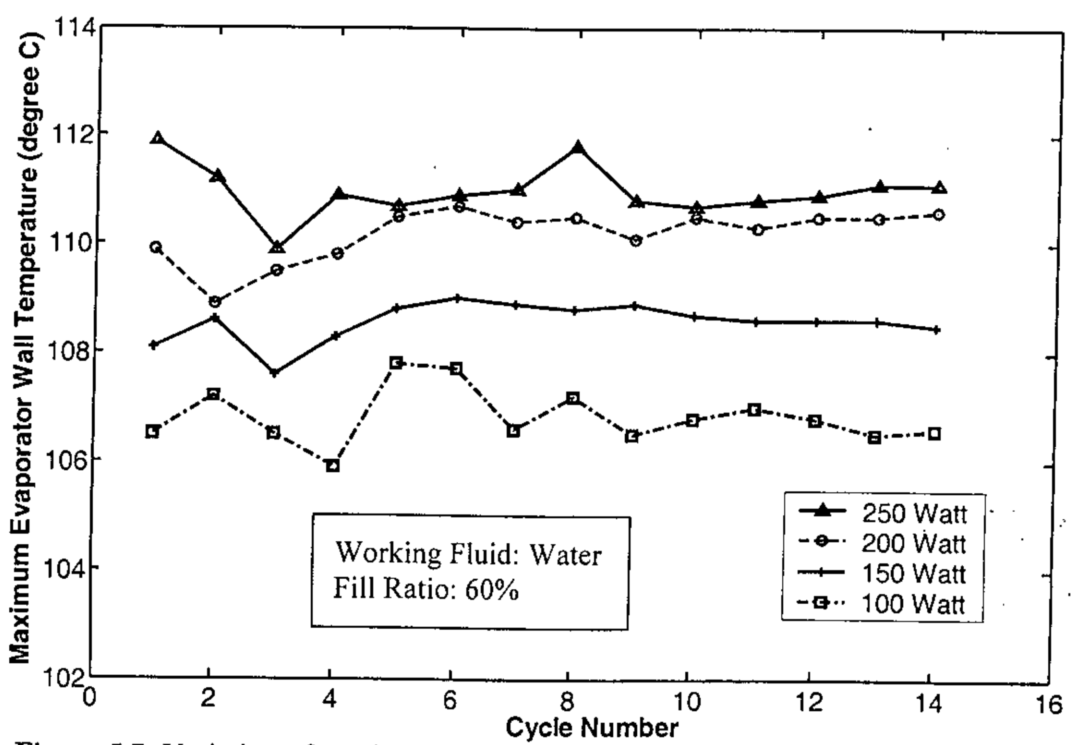


Figure 5.7: Variation of maximum temperature of the evaporator wall for water at 4 different heat loads at 60% fill ratio (Prototype-1, with fan)

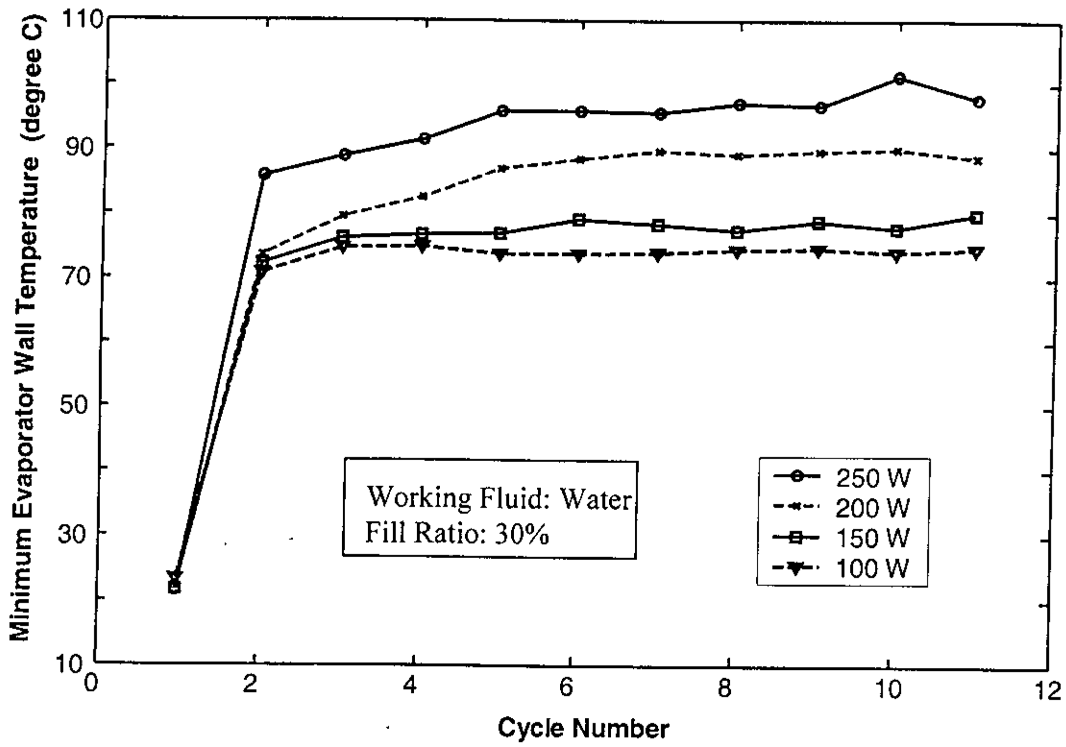


Figure 5.8: Variation of minimum temperature of the evaporator wall for water at 4 different heat loads at 30% fill ratio (Prototype-1, with fan)

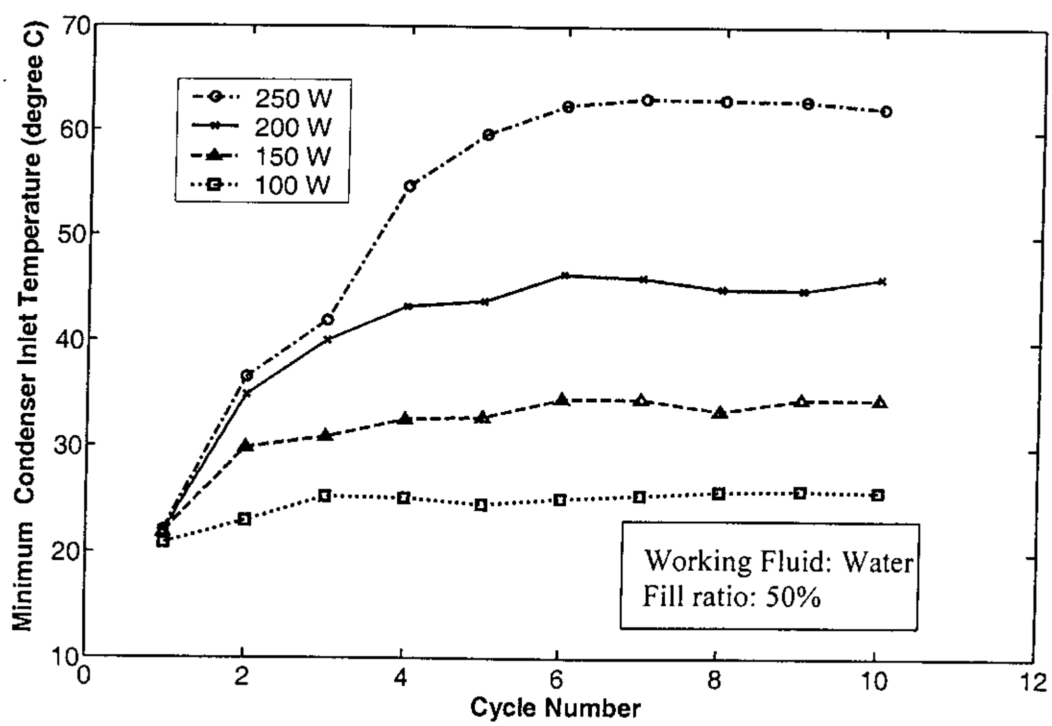


Figure 5.9: Variation of minimum temperature of the condenser inlet at 4 different heat loads with water at 50% fill ratio (Prototype-1, with fan)

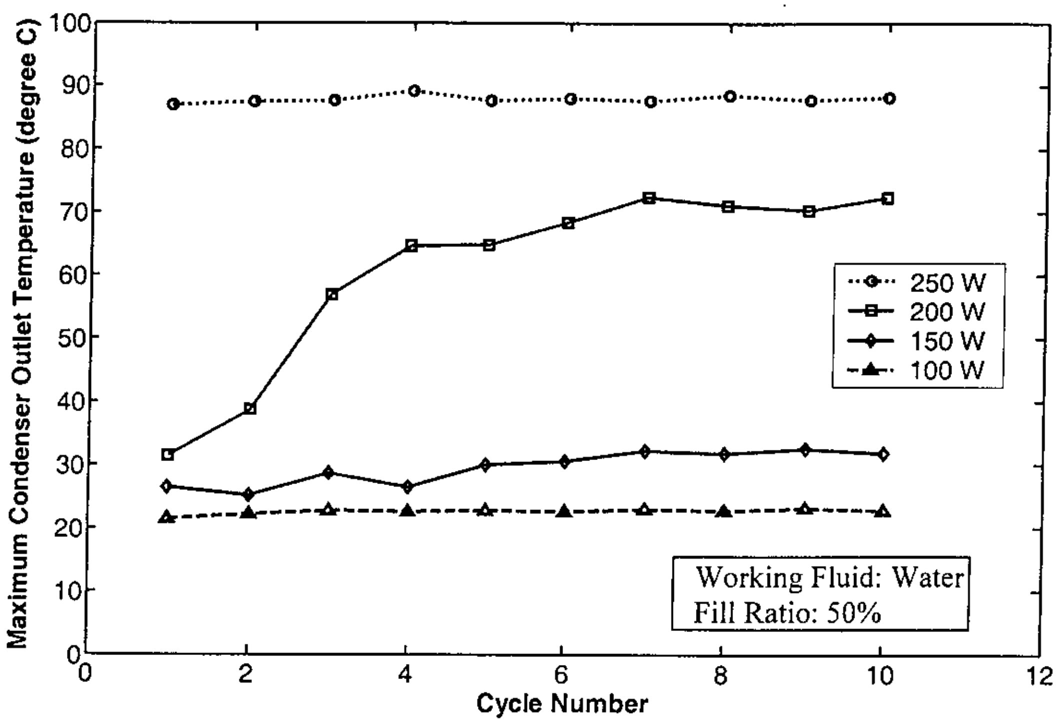


Figure 5.10: Variation of maximum temperature of the condenser outlet at 4 different heat loads with water at 50% fill ratio (Prototype-1, with fan)

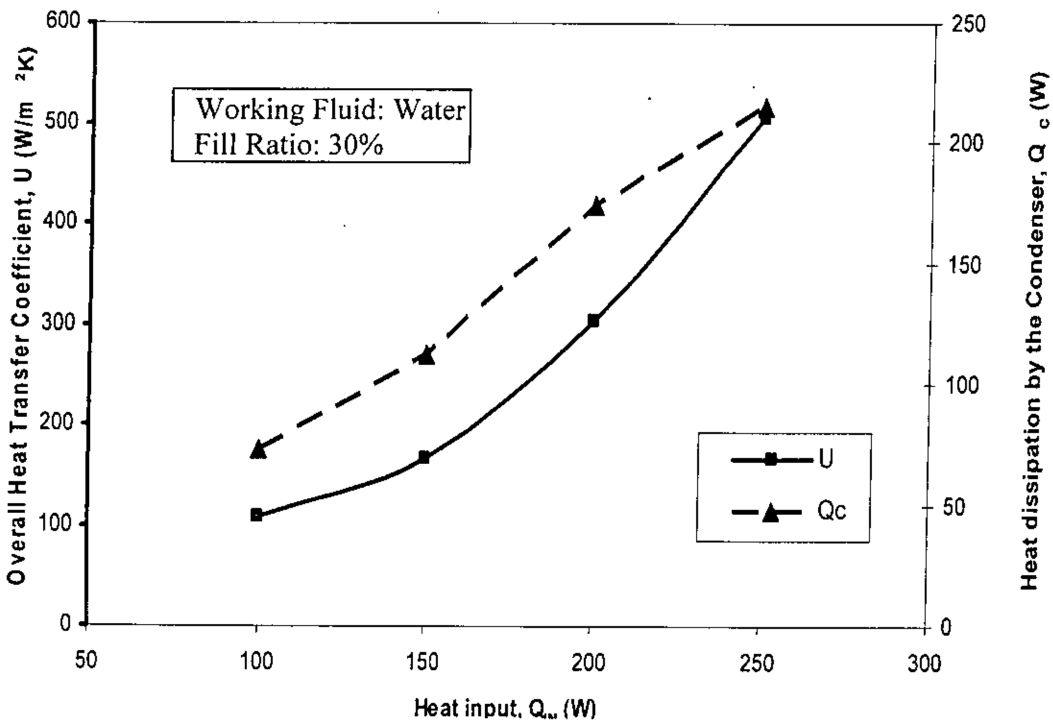


Figure 5.11: Variation of overall heat transfer coefficient (U) and heat dissipation by the condenser (Q_C) with heat loads for water at 30% fill ratio (Prototype-1, with fan)

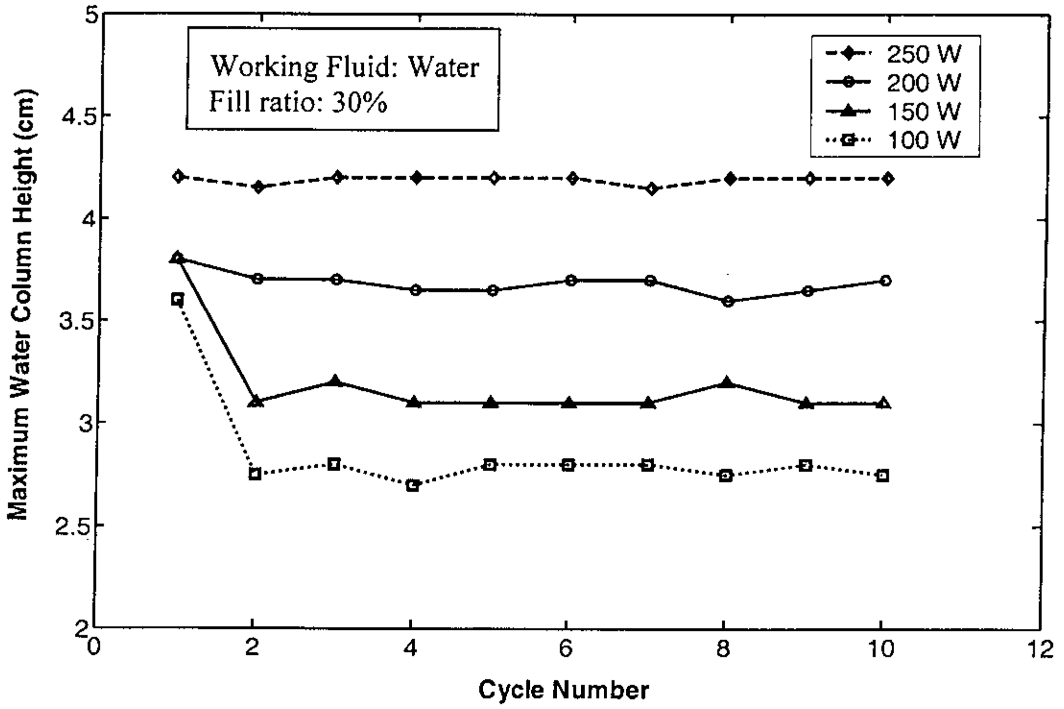


Figure 5.12: Effect of thermal load on the maximum height of the liquid column in the reservoir for water at 30% fill ratio (Prototype-1, with fan)

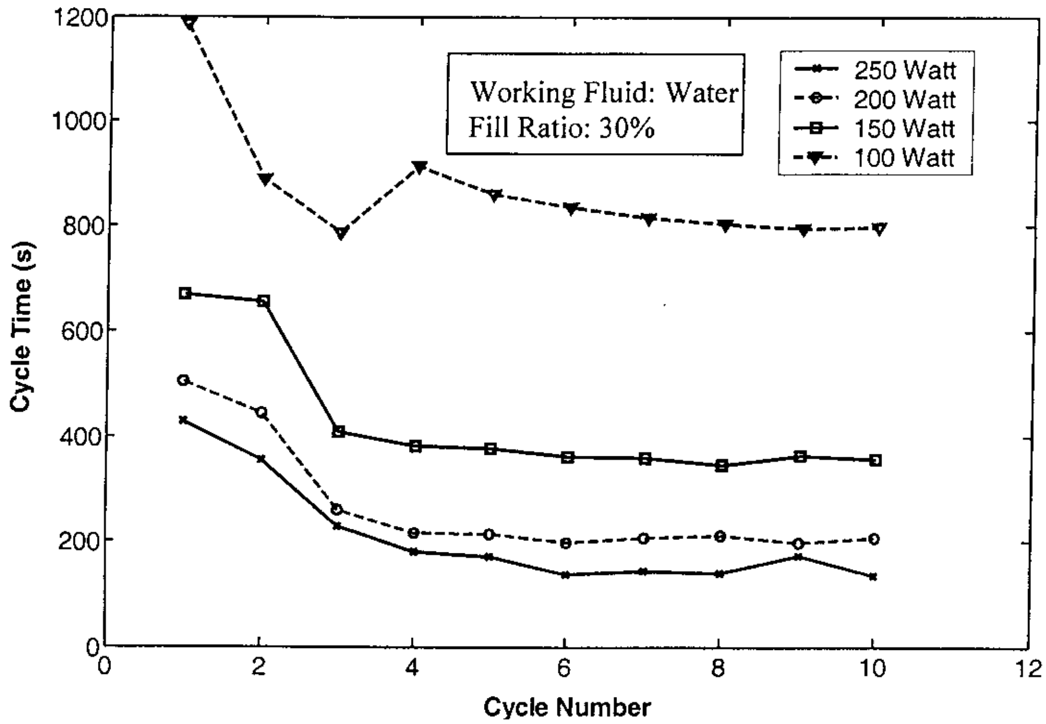


Figure 5.13: Effect on thermal load on the cycle time for 4 different heat loads with water at 30% fill ratio (Prototype-1, with fan)

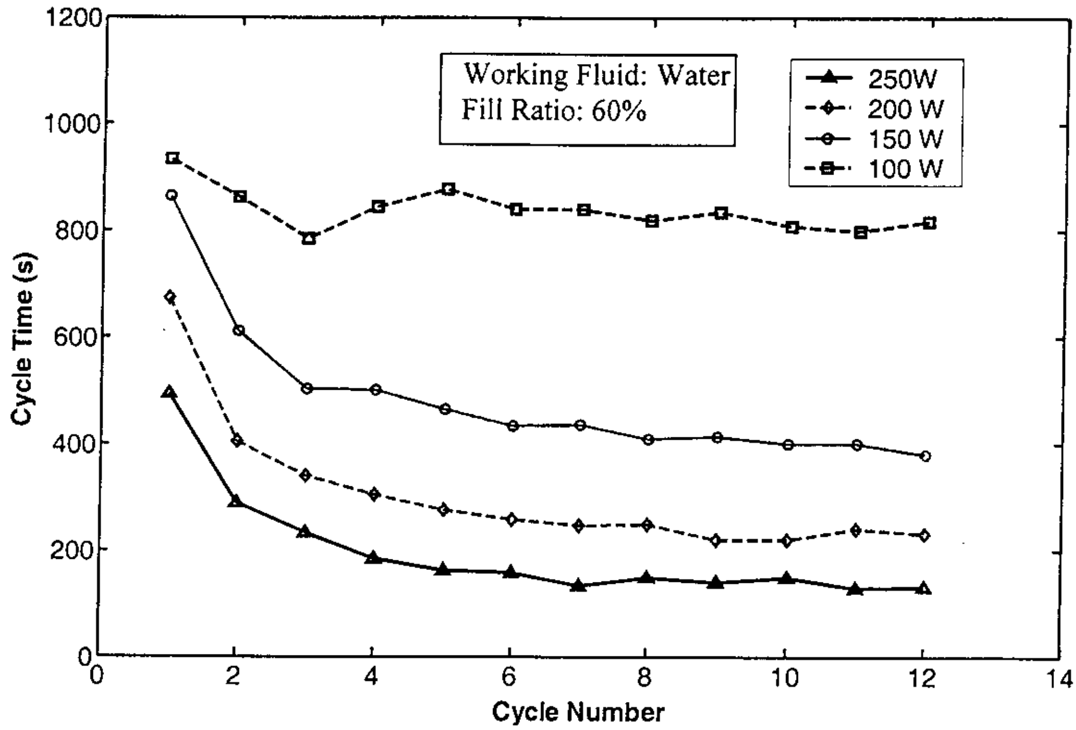


Figure 5.14: Effect of thermal load on the cycle time for 4 different heat input with water at 60% fill ratio (Prototype-1, with fan)

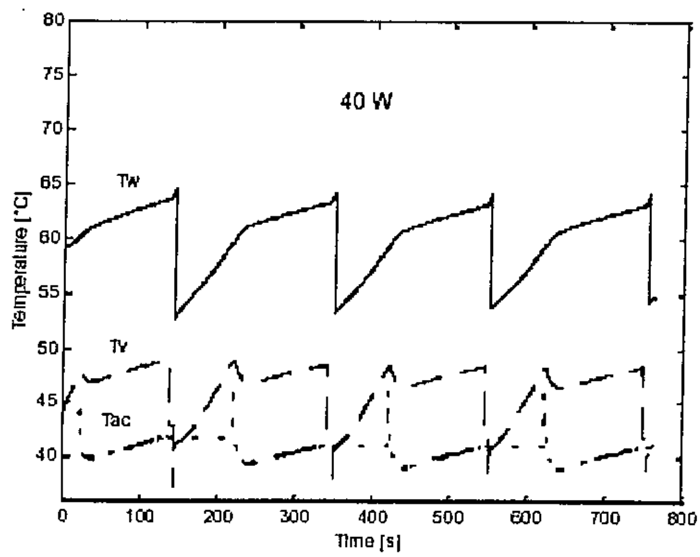


Figure 5.15: Simulation result obtained for PTPT device by Fantozzi et al. [27] showing equal cycle time for all cycles at heat load of 40 W

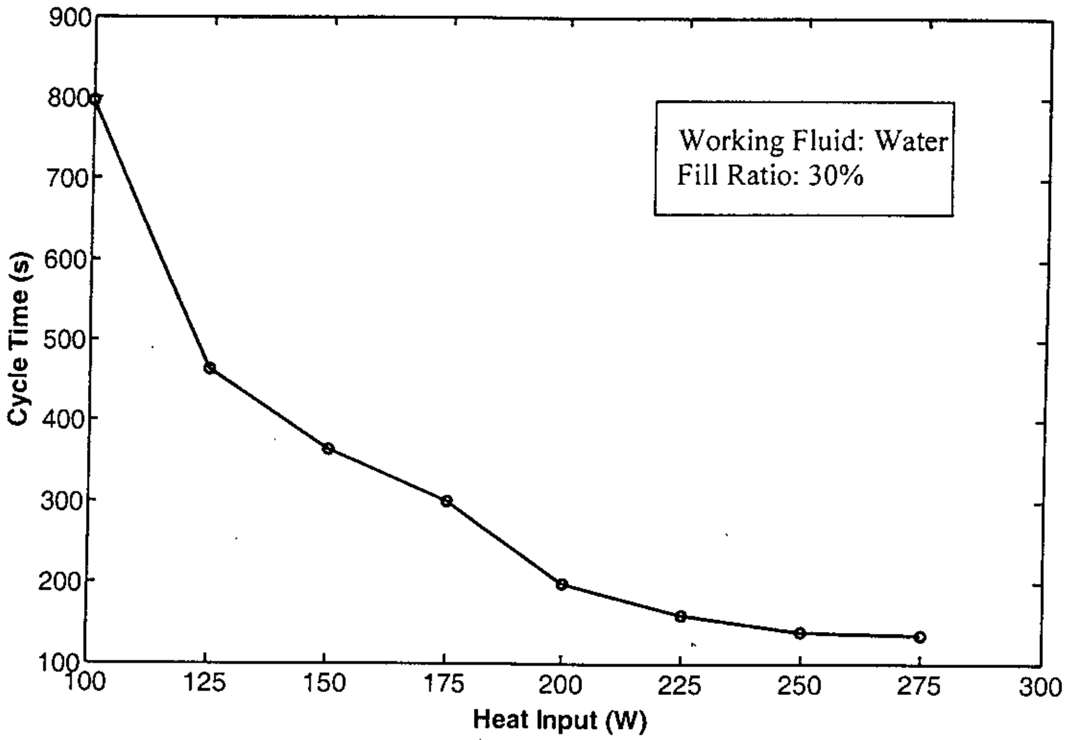


Figure 5.16: Effect of heat input on cycle time with water as the working fluid at 30% fill ratio (Prototype-1, with fan)

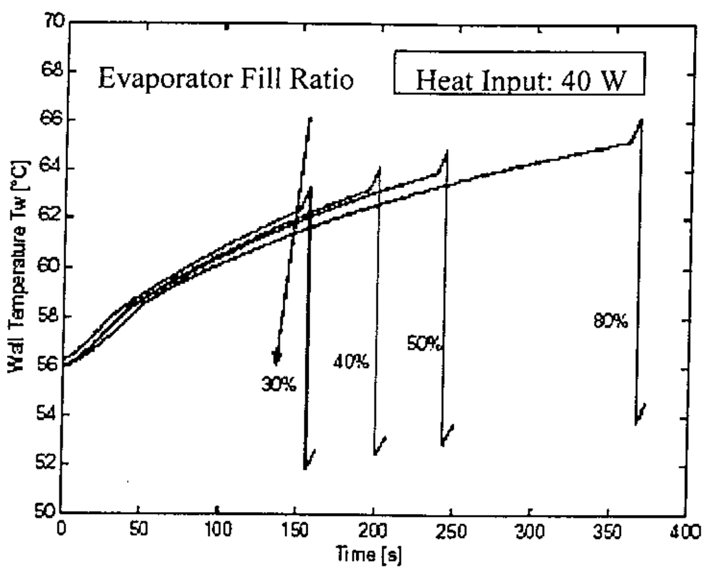


Figure 5.17: Simulation results of the effect of evaporator fill ratio on the temperature of the evaporator wall of PTPT device by Fantozzi et al. [27] at a heat load of 40W

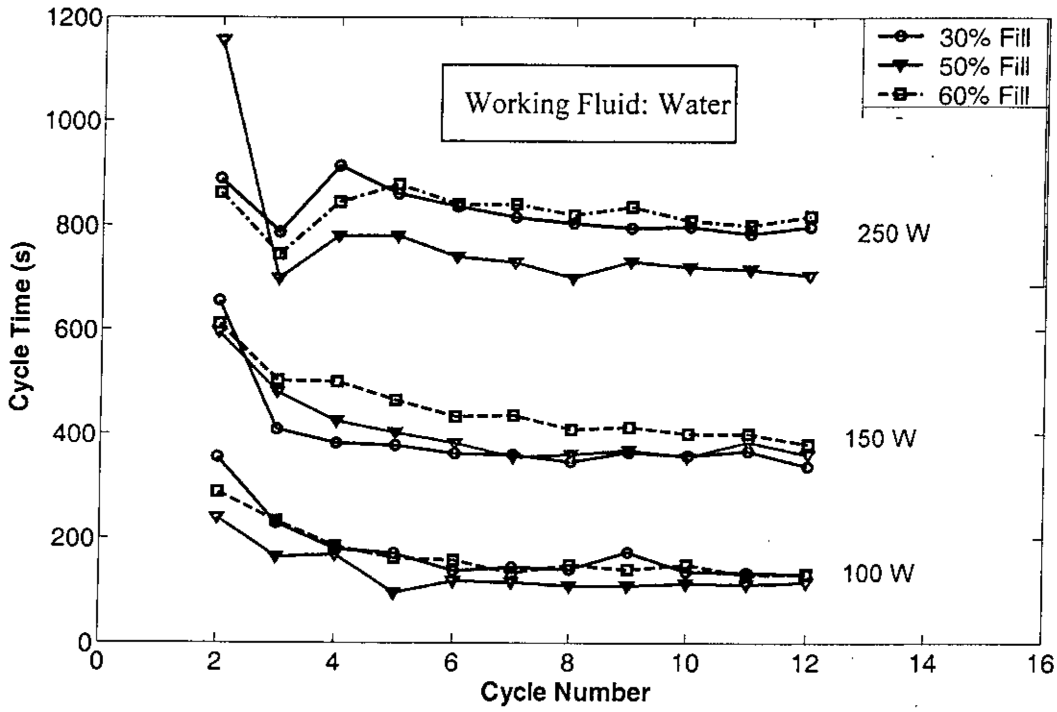


Figure 5.18: Influence of evaporator fill ratio on the cycle time at 3 different heat loads with water as the working fluid (Prototype-1, with fan)

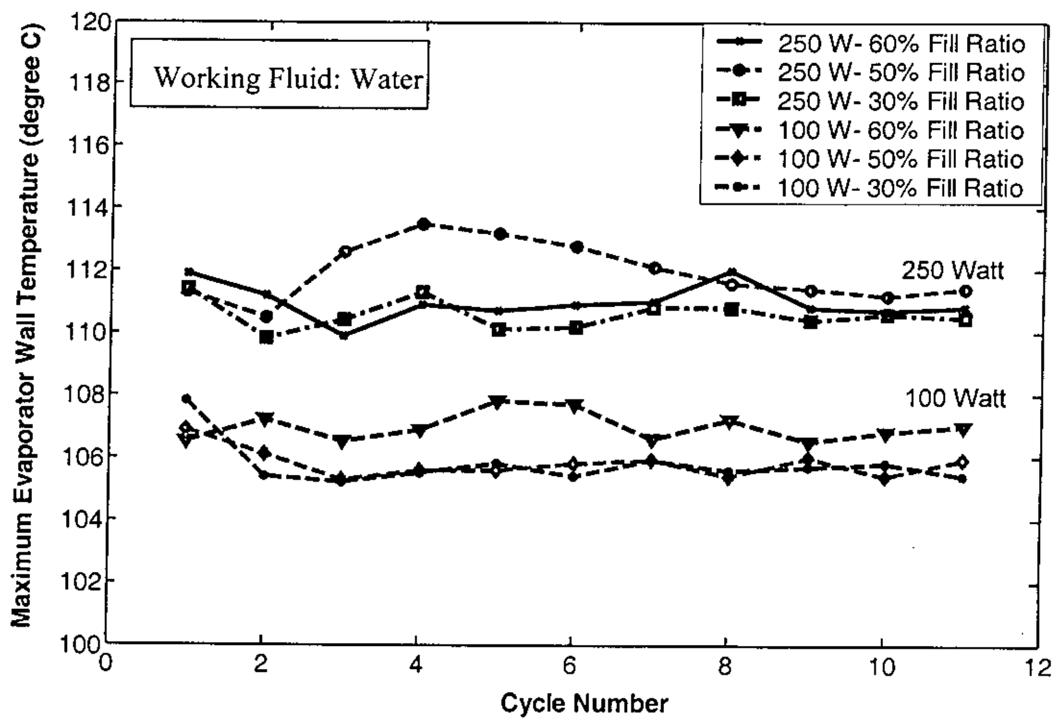


Figure 5.19: Effect of evaporator fill ratio on the maximum temperature of the evaporator wall for heat loads of 100 W and 250 W (Prototype-1, with fan)

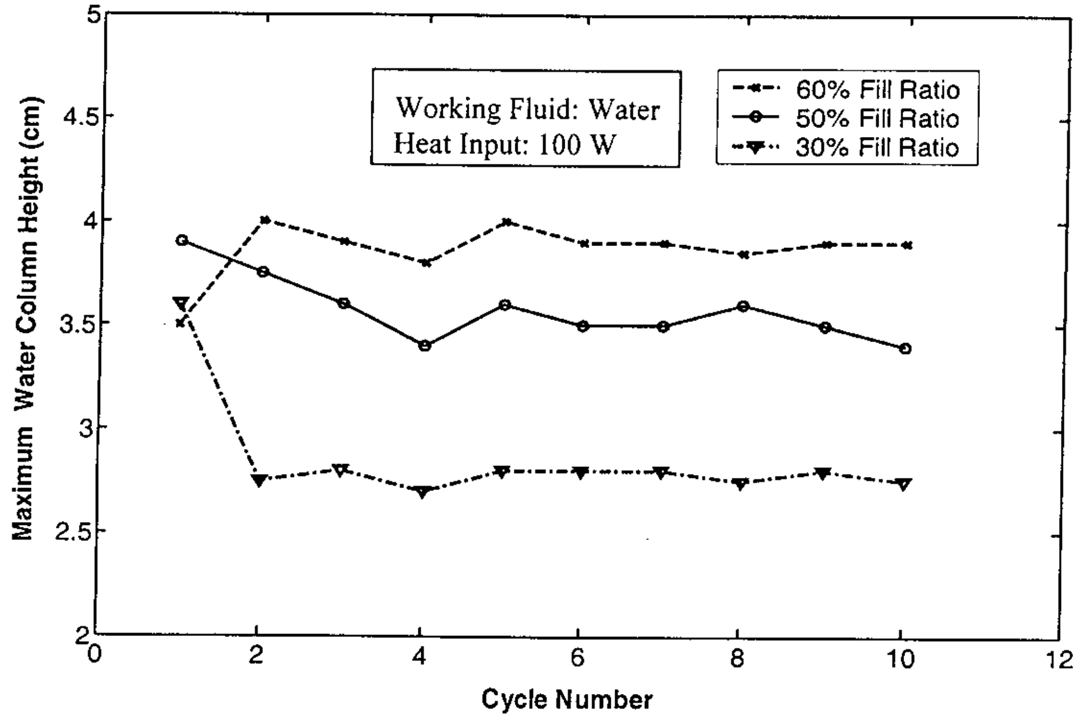


Figure 5.20: Effect of evaporator fill ratio on the height of the accumulated water column in the reservoir at a heat load of 100 W ((Prototype-1, with fan)

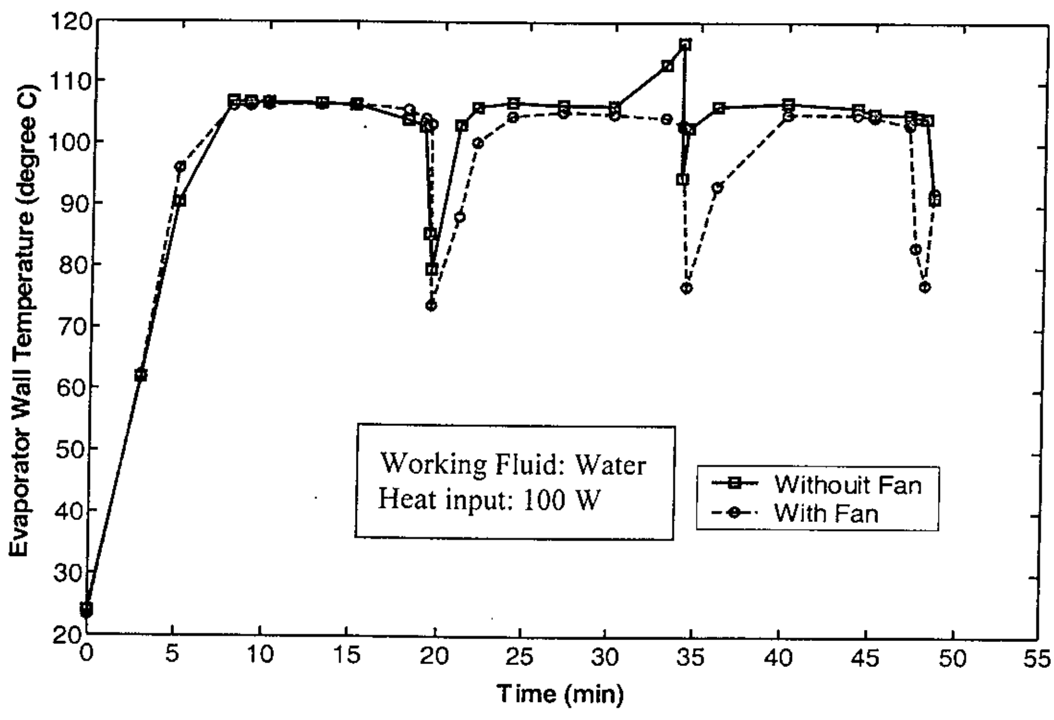


Figure 5.21: Effect of Condenser convection condition on the evaporator wall temperature at a heat load of 100 W with water at 30% fill ratio (Prototype-1)

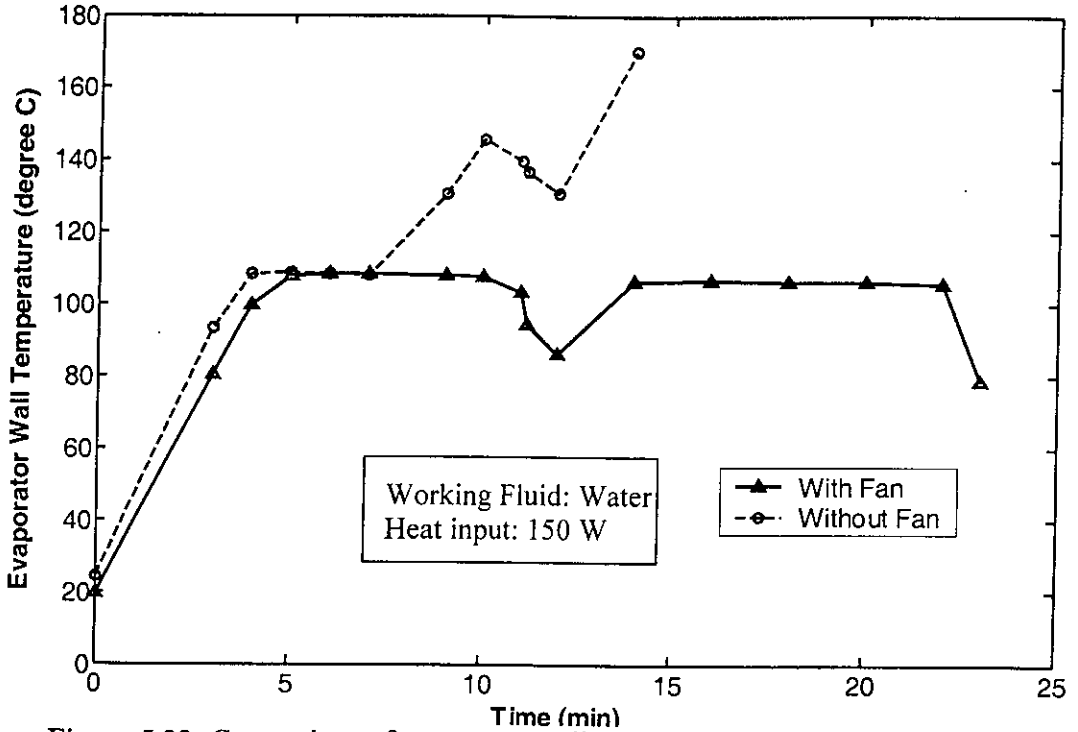


Figure 5.22: Comparison of evaporator wall temperature at different condenser convection condition at heat load of 150 W with water at 30% fill ratio (Prototype-1)

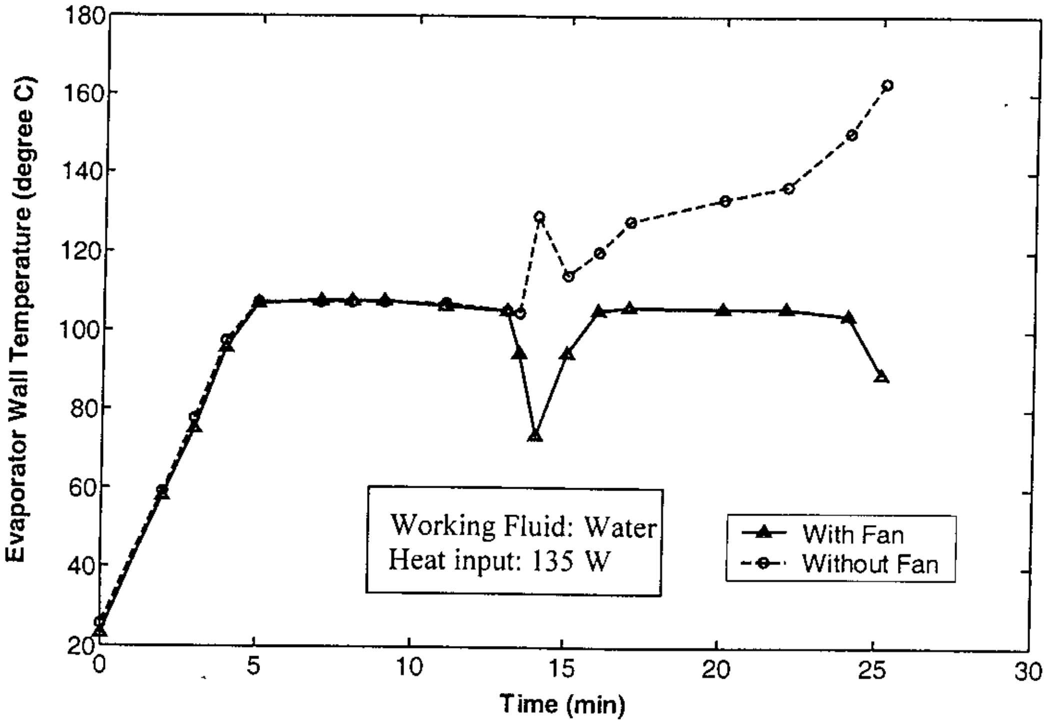


Figure 5.23: Comparison of evaporator wall temperature at different condenser convection condition at heat load of 135 W with water at 30% fill ratio (Prototype-1)

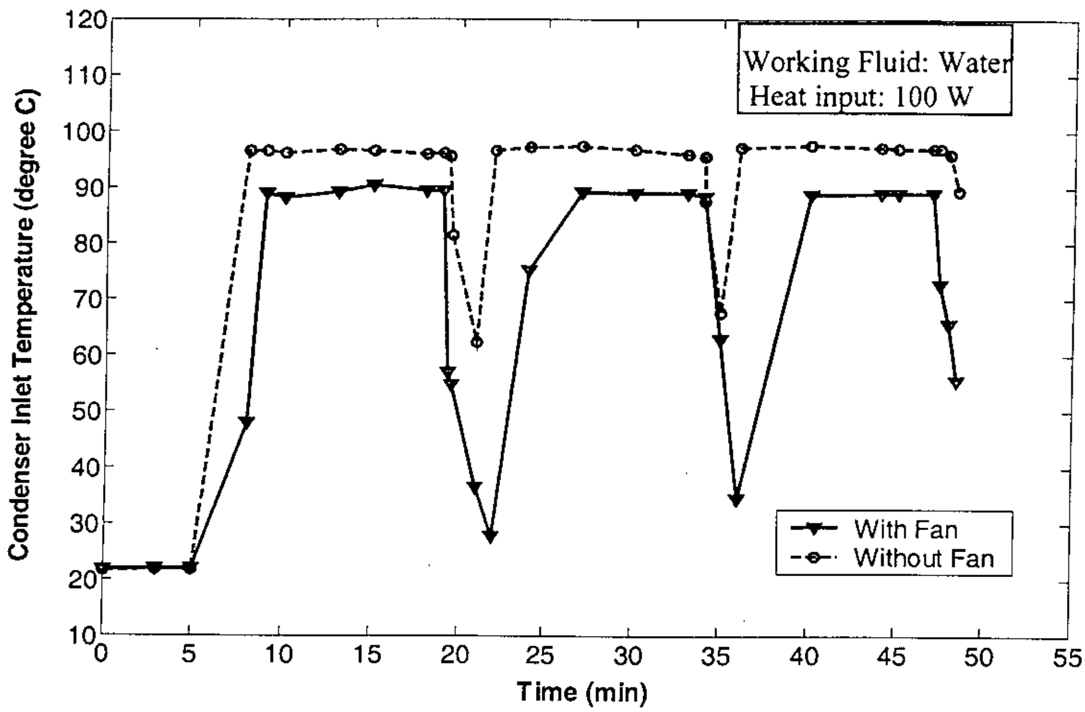


Figure 5.24: Effect of convection condition on condenser inlet temperature for a heat load of 100 W with water at 30% fill ratio (Prototype-1)

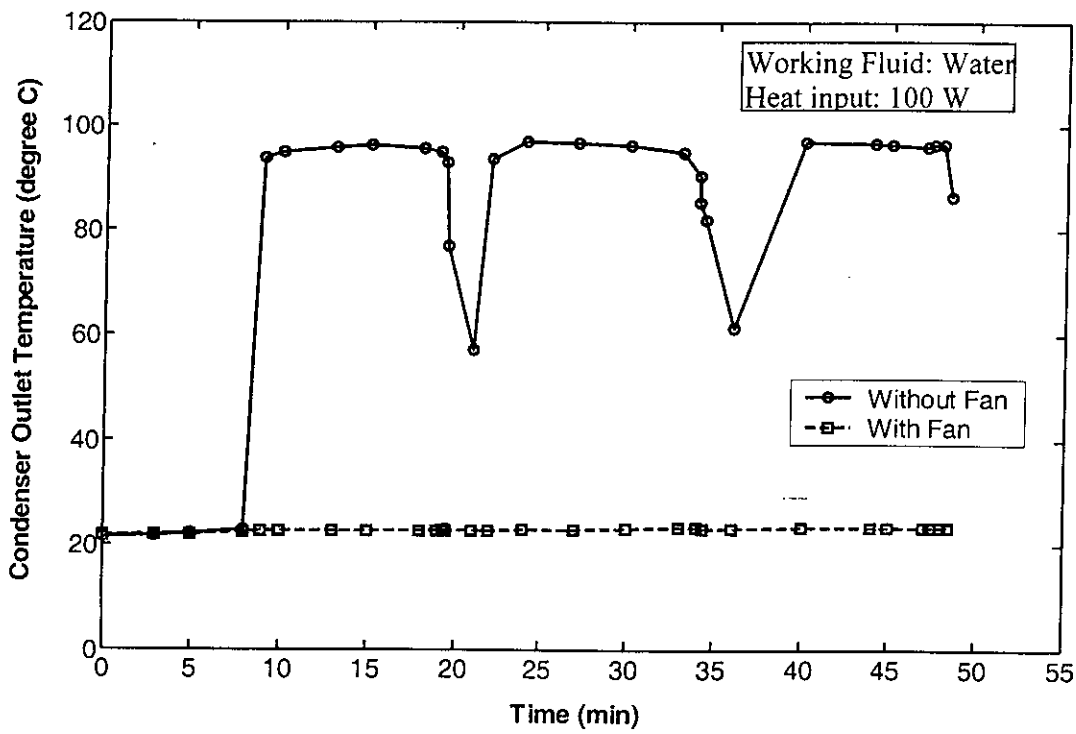


Figure 5.25: Effect of convection condition on condenser outlet temperature for a heat load of 100 W with water at 30% fill ratio (Prototype-1)

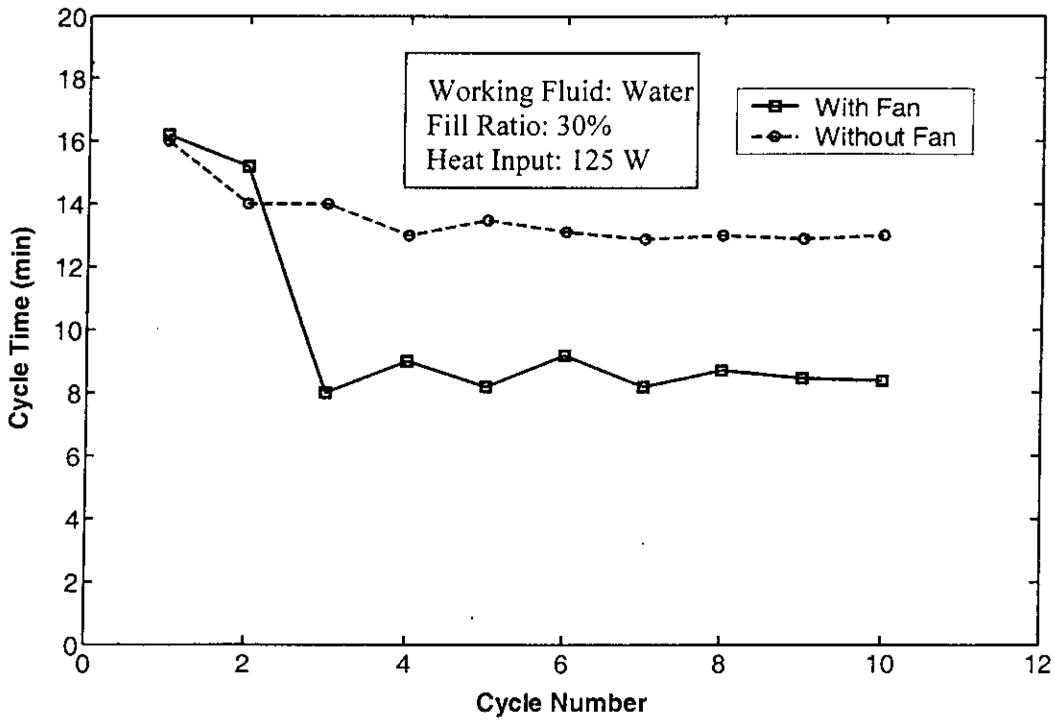


Figure 5.26: Effect of convection condition of condenser on cycle time for a heat load of 125 W with water at 30% fill ratio (Prototype-1)

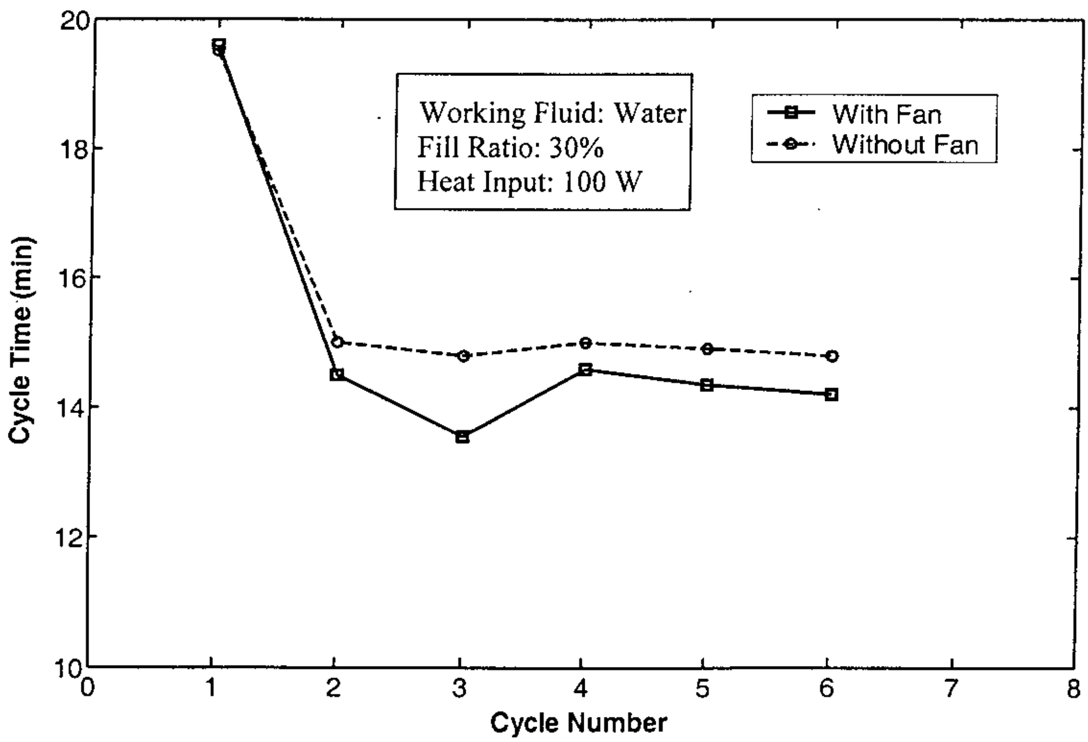


Figure 5.27: Effect of convection condition of condenser on cycle time for a heat load of 100 W with water at 30% fill ratio (Prototype-1)

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

Thermoloop is an indirect, two-phase liquid cooling technology with all the promise to be the future solution of micro-electronics cooling. In the present study, the thermoloop device was investigated experimentally to determine the effect of various functional and geometric parameters on its heat transfer performance. Two prototypes of the thermoloop device have been tested for two different working fluids, water and ethanol.

But majority of the tests was conducted for prototype-1 and with water as the working fluid. A simple model was also developed on the basis of hydrodynamics and heat balance to understand the physics involved in the operation of thermoloop, but it was not verified by experimental data. From the data presented and the subsequent analysis, following conclusions can be drawn-

1. Thermal load significantly influences the heat transfer performance of thermoloop. The thermoloop investigated in this study can transport as high as 215 W of heat for forced cooling of the condenser with water as the working fluid.

2. The maximum and minimum temperatures of the evaporator wall increase with an increase in the thermal load. The maximum wall temperature increased about 6^0C and minimum wall temperature increased more than 22^0C for an increase in the thermal load from 100 W to 250 W under similar conditions.

3. For the same heat input and working fluid, the maximum temperature of the evaporator wall seems to be negligibly influenced by liquid fill ratio; the variation in the temperature was about 2^0C for all the fill ratios.

4. Temperature difference between condenser inlet and outlet decreases with increasing thermal load and hence increases the overall heat transfer coefficient. Overall heat transfer coefficient of condensation was as high as 621W/ m^2K for a heat input of 250 W at 50% fill ratio with water as working fluid.

5. The cycle time for heat transport initially decreases with number of cycles and then becomes constant after 5-6 cycles. It also decreases with an increase in the thermal load and seems to be insensitive to the evaporator fill ratio.

6. The inclusion of fan has a very significant effect on the heat transfer performance of the thermoloop. It increased the heat transfer limit of the device from 125 W to 275 W at the same fill ratio with water as the working fluid.

7. For ethanol as the working fluid, the maximum temperature of the evaporator wall, condenser temperatures and cycle time varied in a similar manner as that with water. But the value of the above mentioned parameters were significantly lower when ethanol was used.

6.2 RECOMMENDATIONS

The present study was conducted for a set of parameters that influence the heat transfer performance of thermoloop. However, further studies are required to supplement the current results and for further enhancement of understanding of the effect of various parameters. The recommendations for future work are listed below:

a. A comprehensive performance study of the device with different working fluids such as FC (Flouro-carbon) family, acetone, ethylene-glycol etc. is required for the determination of the most compatible and suitable working fluid for thermoloop.

b. The maximum operational limit of the device and means to increase the limit for different evaporator and condenser configurations and for different working fluids are required to be investigated.

c. A more accurate study of the device can be achieved by designing and investigating the performance of different components of thermoloop individually for optimum performance.

d. Performance of the thermoloop device at different orientation and at a much miniature scale can be investigated to examine its suitability to use in computer and other micro-electronics cooling.

e. A numerical simulation scheme of the operation of thermoloop can be developed to provide better understanding of the physics involved.

f. The proposed mathematical model can be further improved by considering the two phase flow conditions in calculating the pressure drops and other operational parameters.

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APPENDIX –A

SAMPLE CALCULATION

A1. Calculation of Pressure Drop

The sample data for the heat transfer cycle at a heat input of 100 watt with water as the working fluid and forced convection at the condenser was obtained as follows-

Time of flow, $t = 240$ s

Length of the return tube, $L = 0.56$ m

Diameter of forward/return tube and condenser coil, $Di = 5.334 \times 10^{-3}$ m

Volume of the water collected in the reservoir, $Q = 5.22 \times 10^{-5}\ m^3$

Temperature at the Condenser Inlet, $T_{IN} = 90\ ^0C$

Temperature at the Condenser outlet, $T_{OUT} = 21.7\ ^0C$

Friction factor, f = 0.1 (assumed)

Flow area, $A = 2.234 \times 10^{-5}\ m^2$

Volumetric flow rate = $\dot{Q}\ m^3/s$

The velocity of flow is given by

$$V = \frac{\dot{Q}}{A} = \frac{Q/t}{\frac{\pi}{4}Di^2}$$

$$= \frac{4Q}{\pi Di^2 t}$$

So putting the values, we get

$$V = \frac{4 \times 5.22 \times 10^{-5}}{\pi \times (5.334 \times 10^{-3})^2 \times 240}$$

$$= 9.73 \times 10^{-3}\ m/s$$

Mass flow rate is calculated by

$$\dot{m} = \rho AV = \rho.\dot{Q}$$

$$= 998.5 \times 2.234 \times 10^{-5} \times 9.73 \times 10^{-3}$$

$$= 2.17 \times 10^{-4}\ kg/s$$

The pressure drops in the forward and return lines due to frictional losses are given by the Darcy-Weisbach equation

$$\Delta P_{FL} = \frac{fLV^2}{2gDi}$$

$$\Delta P_{FL} = \frac{0.1 \times 0.56\left(9.73 \times 10^{-3}\right)^2}{2 \times 9.81 \times 5.334 \times 10^{-3}}$$

$$= 1.013 \times 10^{-4}\ Pa$$

In the similar fashion, pressure drop in the return line due to friction is also calculated. The pressure drop in the condenser is calculated by the following formulae

$$\Delta P_C = \left(\frac{2f_B \left(\frac{\dot{m}}{A}\right)^2 L}{D\rho_l} \right) \cdot \left(1+\bar{x}\left(\frac{\rho_l}{\rho_v}\right)\right) \cdot \left(1+\bar{x}\left(\frac{\mu_l}{\mu_v}\right)\right)^{-1/4} + \left(\frac{\dot{m}}{A}\right)^2 \cdot \left(\frac{\rho_l}{\rho_v}\right) \cdot \left(\frac{x_{out}-x_{in}}{\rho_l}\right) \quad \textbf{(A.1)}$$

We assume that linear average quality of vapor, $\bar{x} = 0.5$ and the quality of vapor at outlet and inlet of the condenser, x_{out} and x_{in} respectively as 1 and 0. The values of density and viscosity of liquid and vapor are taken at the temperature of condenser outlet and inlet respectively. The friction factor is assumed to be 0.1.

Putting the values, we get, condenser pressure drop

$$\Delta P_C = [\{6.73 \times (905.43 \times 0.452) + 170.92]$$

$$= 2.928.54 \text{ Pa}$$

$$= \mathbf{2.93\ kPa.}$$

A2. Calculation of Heat Flux

For heat input of 100 watt, amount of heat loss from the evaporator is 11.7 W

Actual heat input to the evaporator, $Q_E = 100 - 11.7 = 88.3$ W

Heat transfer area of the evaporator, $A_E = 0.0615\text{m} \times 0.048\text{m} = 2.952\text{x}10^{-3}\ \text{m}^2$ (Considering the internal bottom surface of the evaporator)

So heat flux, $q = Q_E / A_E = 88.3 / 2.952 \times 10^{-3} = 29.92$ kW/ m^2 = **3.0 W/cm^2**

Now, Rohsenow correlation is given as

$$q = \mu_L h_{fg} \left[\frac{g(\rho_L - \rho_V)}{\sigma}\right]^{1/2} \left[\frac{C_{P,L} \cdot \Delta T_{SAT}}{C_{s,f} \cdot h_{fg} \cdot P_{r,L}{}^{n}}\right]^3 \quad \textbf{(A.2)}$$

Assuming saturation temperature of water to be 100^0C, we get

Density of liquid, $\rho_L = 957.9$ kg/m^3

Density of liquid $\rho_v = 0.5955$ kg/m^3

Viscosity of liquid, $\mu_L = 279 \times 10^{-6}$ N-s/m^2

Specific heat, $C_{p,L} = 4.217$ kJ/kg-K

Enthalpy of evaporation, $h_{fg} = 2257$ kJ/kg

Prandtl Number, Pr = 1.76

Surface Tension, $\sigma = 58.9 \times 10^{-3}$ N/m

Wall temperature, Tw = 106.5 ^{0}C

So, $\Delta T_{SAT} = (106.5 - 100) = 6.5^0$C

Assuming polished surface, we get for water-copper combination

Value of Coefficient, $C_{S,f} = 0.0130$ and n= 1.0.

Putting the above value in Eq. (A2), we get

$$\text{Heat flux, } q = 279\times10^{-6}\times2257\times10^{3}\left[\frac{9.8(957.9-0.5955)}{58.9\times10^{-3}}\right]^{1/2}\left[\frac{4.217\times10^{3}\times6.5}{0.013\times2257\times10^{3}.1.76^{1}}\right]^{3}$$

$= 37.6$ kW/m^2.

= 3.76 W/ cm^2.

A3. Calculation of Heat dissipation by the Condenser and Heat Loss

The outer wall temperature of the bottom surface of the evaporator is taken as same as the thermocouple reading of the inner wall temperature as the wall is only 2.5 mm thick. Temperature of the other surfaces of the evaporator is also assumed the same value to characterize the maximum loss of heat by natural convection.

Heat loss to the ambient:

So, the total surface area of the evaporator, comprising of six surfaces is

$$A_E = 2\,[(0.0725 \times 0.06) + (0.0725 \times 0.02) + (0.06 \times 0.02)$$
$$= 0.014 \text{ m}^2.$$

For a heat input of 100 W. Evaporator wall temperature, $T_E = 105.6\ ^0C$

Ambient temperature, $T\infty = 22\ ^0C$

Assuming, Heat transfer coefficient for natural convection, $h = 10\ W/m^2K$

We get heat loss from the evaporator by convection

$$Q_{L,E} = A_E.\ h.\ (T_E - T\infty)$$
$$= 0.014 \times 10 \times (105.6-22)$$
$$= 11.7\ W$$

Let us assume, 5 % of the heat taken by the evaporator is lost to the surrounding from the forward line, connectors and clamps.

Heat loss from these sources,

$$Q_{L,F} = (100-11.7) \times 0.05$$
$$= 4.41\ W$$

Therefore, total amount of heat loss before the condenser,

$$Q_L = Q_{L,E} + Q_{L,F}$$
$$= 11.28 + 4.44$$
$$= \mathbf{16.11\ W}$$

Heat dissipation by the condenser:

For a heat input of 100 W at a fill ratio of 30%

Temperature of vapor at condenser inlet, $T_{IN} = 104\ ^0C$

Temperature of water at condenser outlet, $T_{out} = 23.7\ ^0C$

Mass flow rate of water, $\dot{m} = 2.86 \times 10^{-5}$ kg/s

Enthalpy of vaporization of water at condenser inlet temperature, $h_{fg} = 2247$ kJ/kg

So, heat released by the condensation of vapor,

$$Q_{PC} = \dot{m}.h_{fg}$$
$$= 2.86 \times 10^{-5}.\ 2247 \times 10^3$$
$$= 64.3\ W$$

Amount of subcooling in the condenser,

$$Q_{SC} = \dot{m}.C_p(T_{SAT} - T_{OUT})$$

$$= 2.86 \times 10^{-5} . 4.2 \times 10^{3} . (100-23.7)$$

$$= 9.2 \text{ W}$$

Therefore, total heat dissipated by the condenser is

$$Q_C = Q_{PC} + Q_{SC}$$

$$= (64.3 + 9.2) \text{ W}$$

$$= \mathbf{73.5\ W.}$$

A4. Calculation of Thermal Resistance

For a heat input of 75 W and natural convection (without fan) of the condenser:

Temperature at the condenser outlet, $T_{OUT} = 75\ ^0C$

Ambient temperature, $T\infty = 22\ ^0C$

Enthalpy of vaporization of water at condenser inlet temperature, $h_{fg} = 2251$ kJ/kg

Mass flow rate of water, $\dot{m} = 2.52 \times 10^{-5}$ kg/s

So, heat released by the condensation of vapor,

$$Q_{PC} = \dot{m}.h_{fg}$$

$$= 2.52 \times 10^{-5} . 2251 \times 10^{3}$$

$$= 56.7 \text{ W}$$

Amount of subcooling in the condenser,

$$Q_{SC} = \dot{m}.C_p(T_{SAT} - T_{OUT})$$

$$= 2.52 \times 10^{-5} \times 4200 \times (100 - 75)$$

$$= 2.6 \text{ W}$$

Therefore, total heat dissipated by the condenser is

$$Q_C = Q_{PC} + Q_{SC}$$

$$= 56.7 + 2.6 \text{ W}$$

$$= 59.3 \text{ W}$$

Thermal resistance between the condenser and ambient

$$(R_{Th})_C = \frac{T_{OUT} - T_\infty}{Q_C}$$

$$= \frac{75-22}{59.3} = \mathbf{0.089\ m^2.K/\ W.}$$

For a heat input of 75 W and forced convection (with fan) of the condenser:

Temperature at the condenser outlet, $T_{OUT} = 23.2\ ^0C$

Ambient temperature, $T\infty = 22\ ^0C$

Enthalpy of vaporization of water at condenser inlet temperature, $h_{fg} = 2251$ kJ/kg

Mass flow rate of water, $\dot{m} = 2.54 \times 10^{-5}$ kg/s

So, heat released by the condensation of vapor,

$$Q_{PC} = \dot{m}.h_{fg}$$

$$= 2.52 \times 10^{-5}.\ 2251 \times 10^3$$

$$= 57.2\ W$$

Amount of subcooling in the condenser,

$$Q_{SC} = \dot{m}.C_p(T_{SAT} - T_{OUT})$$

$$= 2.52 \times 10^{-5} \times 4200 \times (100 - 23.2)$$

$$= 8.2\ W$$

Therefore, total heat dissipated by the condenser is

$$Q_C = Q_{PC} + Q_{SC}$$

$$= 57.2 + 8.2\ W$$

$$= 65.4\ W$$

Thermal resistance between the condenser and ambient

$$(R_{Th})_C = \frac{T_{OUT} - T_\infty}{Q_C}$$

$$= \frac{23.2-22}{65.4} = \mathbf{0.018\ m^2.K/\ W.}$$

APPENDIX –B

UNCERTAINTY ANALYSIS

Experimental results are always subject to errors where the error in the measurement is usually defined as the 'the difference between the true value and the measured value'. But the problem with this definition of error is that in most cases we can not specify the exact value of error in the measurement, rather can only estimate the limit within which the possible error may be bound. For this reason, the term 'uncertainty' comes, which is defined as "a possible value that an error may have" by Kline and McClintock [35]. Although for a single observation, the error is a certain foxed value, the uncertainty may vary considerably depending on the particular circumstances of observation.

The term uncertainty analysis, as explained by Mofat [36] refers to the "process of estimating how great an effect the uncertainties in the individual measurements have on the calculated results". For experimental measurements, this is a very important analysis on the part of the experimenters to assure the readers with a measure of reliability of the test results. Mofat also stresses that the use of uncertainty analysis is not only an essential requirement in reporting results to the technical community, but also has broader uses. This can be very powerful tool in the planning stage of an experimental work to find out the possible sources of error. In the preliminary stages of an experiment, uncertainty analysis can be very helpful in choosing the most reliable technique for a given measurement or to identify the most critical part of measurement in a system.

B.1 UNCERTAINTY TERMINOLOGY OF VARIOUS PRIMARY MEASURANDS

In engineering measurements, the total error in any measurand is usually expressed in terms of two components: a fixed or bias error and a random or precision error. The definition of the terms involved in uncertainty analysis is presented below.

B.1.1 Bias Limit

The bias error is defined as the systematic error which is considered to remain constant during a given test. Therefore, for repeated measurements of a given set, each measurement will have the same bias. There is no statistical relation to define the 'Bias Limit, B'. Instead, it must be estimated and therefore is not an easy matter since the true value is unknown. If the bias error in a result is defined as β, the quantity B is the 95% confidence estimate of the experimenter such that $|\beta| \leq B$.

B.1.2 Precision Limit

The ±P interval about a result (single or average) is the experimenter's 95% confidence estimate of the band within which the mean of many such results would fall, if the experiment were repeated many times under the same conditions and using the same equipment. The precision limit is thus an estimate of the scatter (or lack of repeatability) caused by random error and unsteadiness. The precision limit of a measurand could be calculated as two times the standard deviation of unsteadiness of the population of a set of observations measured with the apparatus in normal running condition. A sufficiently large number of sample (>30) should be taken over a sufficiently long sampling period in order for unsteadiness values to be representative of the process.

B.1.3 Combining Errors

The various sources from which error can arise can be divided into three categories: calibration error, data acquisition error and data reduction error. There will be components in the bias and precision from each of these sources of error. The precision of a given parameter from the N sources of error is given by

$$P = [P_1^2 + P_2^2 + P_3^2 + \ldots.\ P_N^2]^{1/2} \qquad \textbf{(B.1)}$$

Similarly, the bias of a given parameter is given by-

$$B = [B_1^2 + B_2^2 + B_3^2 + \ldots.\ B_N^2]^{1/2} \qquad \textbf{(B.2)}$$

B.1.4 Uncertainty of a Parameter

If a single number (U) is needed to express a reasonable limit of error for a given parameter, then a model for combining the bias and precision is adopted, where the interval

$$\bar{X} = \pm\, U \qquad \textbf{(B.3)}$$

Table B.1: Uncertainty in the measured values

Measurands	Precision limit, P	Bias limit, B	Total limit, W
Time (t)	0.25%	0.02%	0.25%
Temperature (T_E)	1.5%	0.5%	1.6%
Temperature (T_C)	1.8%	0.5%	2.0%
Diameter (Di)	1.3%	0.9%	1.6%
Length (L)	0.7%	0.1%	0.7%
Width (b)	2.8%	1.5%	3.1%
Height (H)	1.8%	0.8%	1.9%
Area (A)	2.6%	1.9%	3.2%
Volume (Q)	5.2%	2.6%	5.8%

Table B.2: Uncertainty in the calculated quantities

Quantity	Total Uncertainty (U)
Volume flow rate ($\dot{Q}$)	7.3%
Mass flow rate (m)	7.3%
Velocity (V)	6.6%
Pressure drop (ΔP_C)	10 %
Pressure drop (ΔP_{FL})	7.1%

Example of calculating the uncertainty incorporated in the various measurands and calculated quantities are discussed in this section. For values of some parameters taken from the material property chart (for example density, viscosity), the uncertainty involved is disregarded.

B.3.1 Uncertainty in the Measurement Flow Velocity:

The velocity of fluid flow inside the forward and return line is given by

$$V = \frac{\dot{Q}}{A} = \frac{Q/t}{\frac{\pi}{4}Di^2}$$

Table B.1: Uncertainty in the measured values

Measurands	Precision limit, P	Bias limit, B	Total limit, W
Time (t)	0.25%	0.02%	0.25%
Temperature (T_E)	1.5%	0.5%	1.6%
Temperature (T_C)	1.8%	0.5%	2.0%
Diameter (Di)	1.3%	0.9%	1.6%
Length (L)	0.7%	0.1%	0.7%
Width (b)	2.8%	1.5%	3.1%
Height (H)	1.8%	0.8%	1.9%
Area (A)	2.6%	1.9%	3.2%
Volume (Q)	5.2%	2.6%	5.8%

Table B.2: Uncertainty in the calculated quantities

Quantity	Total Uncertainty (U)
Volume flow rate ($\dot{Q}$)	7.3%
Mass flow rate (m)	7.3%
Velocity (V)	6.6%
Pressure drop (ΔP_C)	10 %
Pressure drop (ΔP_{FL})	7.1%

Example of calculating the uncertainty incorporated in the various measurands and calculated quantities are discussed in this section. For values of some parameters taken from the material property chart (for example density, viscosity), the uncertainty involved is disregarded.

B.3.1 Uncertainty in the Measurement Flow Velocity:

The velocity of fluid flow inside the forward and return line is given by

$$V = \frac{\dot{Q}}{A} = \frac{Q/t}{\frac{\pi}{4} Di^2}$$

Q = width of reservoir (b) ×height of water column (h) × Thickness (s)

$= 0.058 \times 0.036 \times 0.025 = 5.22\times10^{-5}\ m^3$

So total uncertainty in the measurement of Q is-

$$W_Q = \left[\left(\frac{\partial Q}{\partial b}\times W_b\right)^2 + \left(\frac{\partial Q}{\partial h}\times W_h\right)^2 + \left(\frac{\partial Q}{\partial th}\times W_{th}\right)^2\right]^{1/2}$$

$$= (5.449\times10^{-12} + 5.449\times10^{-12} + 5.449\times10^{-12})^{1/2}$$

$$= \mathbf{3.014\times10^{-6}\ m^3}$$

So, relative uncertainty in the measurement of the volume of liquid collected in the reservoir is

$$\left|\frac{W_Q}{Q}\right| = \frac{3.014\times10^{-6}}{5.22\times10^{-5}} = \mathbf{5.8\%}$$

Now, change in the velocity V with respect to volume Q is expressed as-

$$\frac{\partial V}{\partial Q} = \frac{4}{\pi Di^2 t} = \frac{4}{\pi(5.334\times10^{-3})^2.240} = 186.46$$

Change in the velocity with respect to diameter, Di

$$\frac{\partial V}{\partial Di} = -\frac{8Q}{\pi Di^3 t} = -\frac{8\times5.22\times10^{-5}}{\pi(5.334\times10^{-3})^3.240} = -3.65$$

Change in the velocity with respect to time, t

$$\frac{\partial V}{\partial t} = -\frac{4Q}{\pi Di^2 t^2} = -\frac{4\times5.22\times10^{-5}}{\pi(5.334\times10^{-3})^2.240^2} = -4.05\times10^{-5}$$

Now putting values in equation (B.5), we get

$$W_V = \left[\left(\frac{\partial V}{\partial Q}\times W_Q\right)^2 + \left(\frac{\partial V}{\partial Di}\times W_{Di}\right)^2 + \left(\frac{\partial V}{\partial t}\times W_t\right)^2\right]^{1/2}$$

$$= \left[\left(186.46\times3.014\times10^{-6}\right)^2 + \left(-3.65\times8.60\times10^{-5}\right)^2 + \left(-4.05\times10^{-5}\times0.602\right)^2\right]^{1/2}$$

$$= 6.437\times10^{-4}\ m/s$$

Therefore, the relative uncertainty in the measurement of the flow velocity is

$$\left|\frac{W_V}{V}\right| = \frac{6.437\times10^{-4}}{9.73\times10^{-3}} = \mathbf{6.6\%.}$$

B.3.2 Uncertainty in the Measurement of Pressure Drop in Forward line:

The pressure of the vapor moving from the evaporator towards the condenser drops in the forward line due to frictional losses. This pressure drop is given by

$$\Delta P_{FL} = \frac{fLV^2}{2gDi} \tag{B.7}$$

Putting the values, we get

$$\Delta P_{FL} = \frac{0.1\times 0.56\left(9.73\times 10^{-3}\right)^2}{2\times 9.81\times 5.334\times 10^{-3}}$$

$$= \mathbf{1.013\times 10^{-4}\ Pa}$$

Now, change in the pressure drop with respect to V is expressed as-

$$\frac{\partial \Delta P}{\partial V} = \frac{fLV}{gDi} = \frac{0.1\times 0.56\times 9.73\times 10^{-3}}{9.81\times 5.334\times 10^{-3}} = 0.0104$$

Change in the pressure drop with respect to length (L) is expressed as-

$$\frac{\partial \Delta P}{\partial L} = \frac{fV^2}{2gDi} = \frac{0.1\times\left(9.73\times 10^{-3}\right)^2}{2\times 9.81\times 5.334\times 10^{-3}} = 9.04\times 10^{-5}$$

Change in the pressure drop with respect to diameter (Di) is expressed as

$$\frac{\partial \Delta P}{\partial Di} = -\frac{fLV^2}{2gDi^2} = -\frac{0.1\times 0.56\times\left(9.73\times 10^{-3}\right)^2}{2\times 9.81\times\left(5.334\times 10^{-3}\right)^2} = -9.5\times 10^{-3}$$

Change in the pressure drop with respect to friction factor (f) is expressed as

$$\frac{\partial \Delta P}{\partial f} = \frac{LV^2}{2gDi} = \frac{0.56\times\left(9.73\times 10^{-3}\right)^2}{2\times 9.81\times 5.334\times 10^{-3}} = 5.06\times 10^{-4}$$

So total uncertainty in the measurement of pressure drop is-

$$W_{\Delta P} = \left[\left(\frac{\partial \Delta P}{\partial V}\times W_V\right)^2 + \left(\frac{\partial \Delta P}{\partial L}\times W_L\right)^2 + \left(\frac{\partial \Delta P}{\partial Di}\times W_{Di}\right)^2 + \left(\frac{\partial \Delta P}{\partial f}\times W_f\right)^2\right]^{1/2}$$

$$= \left[\begin{array}{c}\left(0.014\times 6.43\times 10^{-4}\right)^2 + \left(9.05\times 10^{-5}\times 4.03\times 10^{-3}\right)^2 + \left(-9.5\times 10^{-3}\times 8.6\times 10^{-5}\right)^2 \\ +\left(5.06\times 10^{-4}\times 0.005\right)^2\end{array}\right]^{1/2}$$

$$= \mathbf{7.21\times 10^{-6}\ Pa}$$

Therefore, the relative uncertainty in the measurement of pressure drop in the forward line is

$$\left|\frac{W_{\Delta P}}{\Delta P}\right| = \frac{7.21\times10^{-6}}{1.013\times10^{-4}} = \mathbf{7.1\ \%}$$

B.3.2 Uncertainty in the Measurement of Pressure Drop in Condenser:

The pressure drop in the condenser is calculated by the following formulae

$$\Delta P_C = \left(\frac{2f_B\left(\frac{\dot{m}}{A}\right)^2 L}{D\rho_l}\right)\left(1+\bar{x}\left(\frac{\rho_l}{\rho_v}\right)\right)\left(1+\bar{x}\left(\frac{\mu_l}{\mu_v}\right)\right)^{-1/4} + \left(\frac{\dot{m}}{A}\right)^2 \cdot \left(\frac{\rho_l}{\rho_v}\right) \cdot \left(\frac{x_{out}-x_{in}}{\rho_l}\right) \qquad \textbf{(B.8)}$$

The average value of quality of vapor is taken equal to 0.5 and friction factor f_B is taken as 0.1. The density and viscosity of liquid and vapor are taken corresponding to the inlet and outlet temperature of the condenser. Putting the values, we get

$$\Delta P_C = [\{6.73\times(905.43\times0.452)+170.92]$$
$$= 2.928.54 \text{ Pa}$$
$$= \mathbf{2.93\ kPa}$$

Now, $$\left(\frac{\dot{m}}{A}\right)^2 = \left(\frac{\rho\dot{Q}}{A}\right)^2 = \left(\frac{\rho AV}{A}\right)^2 = (\rho V)^2$$

So equation **B.4** can be written as

$$\Delta P_C = \left(\frac{2f_B(\rho V)^2 L_{CON}}{D\rho_l}\right)\left(1+\bar{x}\left(\frac{\rho_l}{\rho_v}\right)\right)\left(1+\bar{x}\left(\frac{\mu_l}{\mu_v}\right)\right)^{-1/4} + (\rho V)^2 \cdot \left(\frac{\rho_l}{\rho_v}\right) \cdot \left(\frac{x_{out}-x_{in}}{\rho_l}\right) \qquad \textbf{(B.9)}$$

So total uncertainty in the measurement of pressure drop in condenser can be expressed as-

$$W_{(\Delta P)C} = \left[\left(\frac{\partial \Delta P}{\partial V}\times W_V\right)^2 + \left(\frac{\partial \Delta P}{\partial Di}\times W_{Di}\right)^2\right]^{1/2}$$

Putting the values, we get

$$W_{(\Delta P)C} = [(291.33)^2 + (22509.3\times8.60\times10^{-5})^2]^{1/2}$$

$$= [(291.33)^2 + (44.43)^2]^{1/2}$$

$$= \mathbf{294.7\ Pa}$$

Therefore, the relative uncertainty in the measurement of pressure drop in condenser is

$$\left|\frac{W_{(\Delta P)C}}{(\Delta P)_C}\right| = \frac{294.7}{2928.54} = \mathbf{10\ \%.}$$

APPENDIX-C

RAW SAMPLE DATA

DATA FOR PROTOTYPE – 01

WORKING FLUID: Water, FILL RATIO: 30%, HEAT INPUT: 100 W,
With Fan

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)	H (cm)
01	0.0	23.3	21.7	21.7	0.2
	3.0	62.3	21.9	21.9	1.0
	6.0	103.1	22.0	22.0	3.0
	7.0	105.8	47.8	22.3	3.5
	9.0	106.2	88.9	22.5	3.6
	11.0	105.8	88.1	22.6	3.6
	13.0	106.3	89.1	22.6	3.6
	15.0	106.5	90.3	22.6	3.6
	17.0	106	89.3	22.6	3.6
	18.0	105.7	89.3	22.6	3.6
	19.0	104.2	56.9	22.8	3.6
	20	73.8	54.6	22.7	0.6
02	21.0	88.2	36.5	22.7	0.7
	22.0	100.3	27.7	22.7	1.0
	23.0	105.4	74.9	22.8	2.0
	27.0	105.3	89.0	22.9	2.2
	30.0	104.9	88.8	23.1	2.4
	32.0	104.6	88.8	23.2	2.5
	33.0	104.4	88.2	23.2	2.7
	34.0	103.2	88.2	23.2	2.7
	34.5	76.9	62.8	23.1	0.7

DATA FOR PROTOTYPE – 01

WORKING FLUID: *Water*, **FILL RATIO:** *30%*, **HEAT INPUT:** *100 W*, *Without Fan*

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)	H (cm)
01	0	24	20.6	20.6	0.2
	5	90.5	21.5	21.5	2.4
	8	106.9	96.5	22.0	3.6
	10	106.7	96.1	95.0	4.1
	12	106.9	96.6	95.8	4.3
	15	106.3	96.4	96.2	4.6
	18	104	95.8	95.7	5.1
	19	85.4	81.4	77.0	0.4
02	20	93.6	72.3	62.3	0.4
	21	103.1	62.2	57.0	0.7
	22	105.9	96.4	93.6	3.1
	24	106.7	97.1	96.9	4.3
	27	106.4	97.2	96.8	4.5
	30	106.1	96.6	96.3	4.9
	33	113.1	95.8	97.0	5.0
	34	94.5	87.4	87.8	0.5
03	35	102.7	76.5	73.7	0.5
	36	106.1	97.1	75.4	3.0
	40	106.8	97.4	97.0	4.5
	44	105.9	97.0	96.7	4.8
	47	104.9	96.9	96.1	4.8
	48	104.5	96.8	96.5	5.0
	49	91.3	89.2	86.6	0.4

DATA FOR PROTOTYPE – 01

WORKING FLUID: Water, FILL RATIO: *50%*, HEAT INPUT: *100 W*, *Without Fan*

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)	H (cm)
01	0	22.4	20.9	20.9	0.1
	4	70.9	21.0	21.0	0.7
	6	95.9	21.0	21.0	1.7
	8	105.9	82.7	21.2	3.2
	11	106.2	89.0	21.4	3.6
	15	106.7	89.1	21.3	3.7
	17	106.8	89.3	21.4	3.7
	20	106.7	89.0	21.4	3.7
	22	106.5	89.7	21.5	3.7
	25	106.0	89.0	21.6	4.0
	27	73.5	56.2	21.6	0.4
02	28	84.9	34.8	21.5	0.6
	30	105.0	23.5	21.6	1.4
	32	104.8	88.6	21.9	2.4
	36	105.1	89.1	22.0	2.6
	40	104.9	88.7	22.0	2.9
	45	104.1	88.9	22.0	3.7
	46	78.9	52.5	22.0	0.4
03	47	88.1	37.5	22.0	1.2
	49	100.8	89.0	22.0	3.2
	50	105.2	88.9	22.6	3.4
	52	105.0	89.4	22.2	3.4
	57	105.2	89.3	22.2	3.6
	58	78.9	49.6	22.2	0.4

DATA FOR PROTOTYPE – 02

WORKING FLUID: Ethanol, FILL RATIO: 30%, HEAT INPUT: 100 W,

With Fan

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)
01	0	28.7	29.1	29.1
	1	43.9	29.1	29.1
	2	59.2	29.2	29.2
	3	74.9	29.2	29.2
	4	89.8	29.3	29.3
	5	92.1	74.6	30.0
	6	92.6	76.4	30.9
	7	93.7	77.2	30.9
	8	94.2	78.2	31.0
	9	98.4	76.5	30.0
	9.5	80.0	66.5	31.5
02	11	90.8	76.0	31.8
	12	89.7	76.7	31.8
	13	90.8	77.4	31.8
	14	94.0	77.6	31.7
	14.5	81.2	67.7	31.6
03	15	86.6	63.9	31.3
	16	87.2	75.7	31.3
	17	87.9	76.2	31.3
	18	88.8	77.0	31.4
	19	92.3	77.9	31.6
	19.5	82.9	62.0	31.7
04	21	87.5	75.5	31.7
	22	88.4	76.3	31.9
	23	89	76.9	31.9
	24	90.5	77.9	31.8
	25	80.9	62.5	31.6

DATA FOR PROTOTYPE – 01

WORKING FLUID: *Water*, **FILL RATIO:** *30%*, **HEAT INPUT:** *150 W*, *With Fan*

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)	H (cm)
01	0	19.4	19.4	18.4	0.1
	3	80.2	18.5	18.4	1.5
	4	99.6	18.4	18.4	2.9
	5	107.8	85.0	19.2	3.5
	6	108.3	89.8	19.2	3.6
	7	108.3	90.3	19.1	3.7
	9	108.1	89.8	18.8	3.7
	10	107.6	89.6	18.8	3.75
	11	94.5	85.4	18.8	0.3
02	12	86.1	42.1	18.8	0.4
	14	106.5	89.2	19.2	1.8
	16	106.7	89.7	19.4	1.8
	18	106.5	89.6	20.0	2.4
	20	106.2	89.8	20.2	2.9
	22	105.6	88.8	20.1	3.1
	23	78.8	62.1	20.0	0.4
03	24	98.6	33.9	19.9	0.6
	25	106.9	89.4	22.4	2.2
	26	106.8	89.6	25.6	2.7
	28	106.2	89.5	23.8	3
	29	105.5	89.6	22.9	3.2
	30	82.6	55.2	21.8	0.4
04	31	102.3	32	21.1	0.6
	32	108.1	89.2	21.3	2.3
	34	107.2	88.9	25.2	3.0
	35	106.5	89.5	24.6	3.1
	36	80.8	52.5	21.9	0.4

DATA FOR PROTOTYPE – 01

WORKING FLUID: *Water*, **FILL RATIO:** *30%*, **HEAT INPUT:** *200 W*, *With Fan*

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)	H (cm)
01	0	22.2	21.1	21.1	0.1
	2	73.8	21.1	21.1	1.2
	3	103.6	44.0	21.1	2.9
	4	109.6	90.1	21.4	3.4
	5	110.1	88.3	21.4	3.4
	6	109.9	88.5	21.8	3.5
	7	109.5	88.2	23.3	3.6
	8	79.9	65.4	25.6	0.3
02	9	98.2	45.4	24.1	0.6
	10	108.2	88.9	26.1	1.6
	11	108.2	88.4	28.1	2.2
	12	108.4	88.3	29.6	2.6
	13	108.3	88.4	34.3	3.0
	15	107.8	88.2	38.8	3.7
	16	81.8	67.8	31.0	0.2
03	17	108.0	37	30.1	1.6
	18	108.7	88.5	56.5	3.2
	20	87.1	67.9	33.7	0.2
04	22	108.8	88.4	56.1	3.4
	23	107.0	88.3	40.0	3.6
	24	88.14	75.2	35.0	0.4
05	26	108.8	88.4	54.7	3.4
	27	107.0	88.1	45.5	3.6
	28	88.9	68.5	38.9	0.3
06	29	108.4	88.9	58.7	3.2
	30	108.0	88.4	48.3	3.7
	30.5	89.4	69.8	39.8	0.4

DATA FOR PROTOTYPE – 01

WORKING FLUID: ***Water*****, FILL RATIO:** ***30%*****, HEAT INPUT:** ***250 W, With Fan***

Cycle No.	Time (min)	T_{EV} (^0C)	$(T_{CON})_{in}$ (^0C)	$(T_{CON})_{out}$ (^0C)	H (cm)
01	0	21.5	20.9	20.9	0.2
	2	75.1	21.1	21.1	1.2
	3	108.8	21.3	21.3	3.1
	4	111.2	89.1	21.3	3.4
	5	111.4	88.9	24.5	3.5
	6	110.7	88.2	45.4	4
	7	85.8	68.9	40.9	4.2
02	8	104.8	44.6	29.0	0.3
	9	109.7	88.9	35.7	2.6
	10	109.7	88.9	61.6	3.1
	11	109.6	88.5	68.0	3.7
	12	109.3	88.1	61.9	4.1
	13	88.8	63.8	53.2	0.6
03	14	110.3	89.6	36.6	2.1
	15	109.7	88.9	62	4.2
	17	89.9	61.1	42.7	0.3
04	18	111.2	89.4	67.2	3.9
	19	109.8	88.9	64.7	4.2
	20	95.8	64.9	52.8	0.4
05	21	110.4	89.4	84.1	4.0
	22	109.2	89	64.1	4.2
	23	100.0	68.9	58.2	0.3
06	24	110.2	89.4	84.7	4.0
	24.5	109.5	88.9	68.7	4.2
	25	96.0	70.5	58.9	0.3