HEAT PIPE FOR ELECTRONIC EQUIPMENT COOLING

by

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CERTIFICATE OF APPROVAL

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It is hereby declared that this thesis or any part of it has not been submitted elsewhere for the award of any degree or diploma.

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LIST OF ABBREVIATIONS OF TECHNICAL SYMBOLS AND TERMS:

SYMBOL	MEANING	UNITS
R	Thermal resistance	°C/W
T_e	Temperature of the evaporator	°C
T_c	Temperature of the condenser	°C
Q	Thermal load	W
U_t	Over all heat transfer coefficient	W/m^2 °C
A_e	Surface area of the evaporator	m^2
θ	Inclination angle	degree (°)
$Q_{Lc,max}$	Heat transport factor	W
N_1	Liquid transport factor	Btu / $hr\text{-}ft^2$
r_c	Capillary pore radius	mm
$r_{h,l}$	Hydraulic radius	mm
t_w	Wick thickness	mm
r_v	Vapour core radius	mm
λ	Heat of vaporisation	J/Kg
σ	Surface tension	N/m
ρ_l	Liquid density	Kg/m^3
μ_l	Liquid viscosity	kg /m-sec
L_e	Evaporator length	mm
L_c	Condenser length	mm
L_a	Adiabatic length	mm
d_v	Vapour core diameter	mm
Q_{max}	Maximum axial heat flux	W/m^2
ρ_v	Vapour density	Kg/m^3
ε	porosity	dimensionless
T_v	Vapour temperature	°C
f_{max}	Maximum hoop stress in the wall	N/m^2
P	Pressure difference across the wall	Kg/m^2
d_o	Tube outside diameter	mm
d_i	Tube inside diameter	mm
t	Tube wall thickness	mm
v	Velocity of cooling air	m/s
k_l	Liquid thermal conductivity	watt /m-K
ρ/f_u	Weight Parameter	sec^2 / ft^2
K	Permeability	m^2
MHP	Miniature heat pipe	
PHP	Pulsating heat pipe	

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ABSTRACT

An experimental investigation has been performed to investigate the heat transfer characteristics of a miniature heat pipe having diameters of 5 and 6 mm, which could be applied for cooling of miniature electronic equipment such as the notebook PC, CPU etc. The experimental parameters are inclination, the diameter of the heat pipe, and the velocity of the cooling air. The working fluid is water. The heat transfer characteristics based on the principle of two-phase change (evaporation/boiling-condensation) at different incline angles and different cooling air velocities are experimentally determined. The wall surface temperature at the heating section decreases with the increase of inclination angle of the heat pipe. The thermal resistance decreases with increase in inclination angle for both tubes at all heat inputs. This implies that the action of gravity, which serves to speed up the flow of liquid from condenser to evaporator, increases with increase in inclination angle. The overall heat transfer coefficient increases with the increase of diameter of the heat pipe and with the increase in inclination angle for both tubes.

CHAPTER 1

1. INTRODUCTION

The continuous increase of the power densities in electronic equipment and packages reduce in size has presently increased challenges in thermal management. Among other cooling techniques heat pipe emerges as the most appropriate technology and cost effective thermal design solution due to its excellent heat transfer capability, high efficiency and its structural simplicity. The ability to transport very large quantities of heat with small temperature differences is the main feature characterizing the heat pipe. The device has other desirable features as well. When heat is concentrated at one location, it can be removed through a heat pipe and distributed over a large region. This feature has become important in the miniaturisation of electronic equipment. In such equipment heat is generated as a by-product; the heat pipe makes it possible to remove a large amount of heat from the interior and transport it to some place where it can easily be removed by a cooling air stream. Consequently, electronic equipment can be constructed in compact form.

Heat pipe is very promising heat transfer device, which makes use of change-of-phase heat transfer in a novel way. It is a form of heat exchanger useful for transporting heat over relatively large distances with a small temperature difference. The heat pipe consists of a hollow tube closed at both ends and partially filled with a liquid that boils at a desired temperature. One end of the tube is immersed in the warm region, and the other in the cold region. The objective is to transfer heat through the pipe from the warmer to the colder region. The process may be visualized by assuming that the tube is in a vertical position with the lower end immersed in the warm region.

The liquid fills the lower end of the tube and starts boiling when the temperature of the warmer region exceeds boiling temperature of the liquid. The accumulation of vapour increases the pressure at the lower end of the tube. This forces the vapour upward, where it condenses since the temperature of the colder region is below the evaporation temperature. Gravity causes the condensed liquid to run down along the inner surface of the tube is shown in Fig.1.1. In this way a steady circulation of the fluid is maintained.

Heat pipes can be designed to operate in a wide temperature range from below 0°C up to around 2000°C. This is accomplished by proper choice of the liquid that the pipe contains and the material of which the pipe is manufactured. Methanol, acetone, water, fluoridated hydrocarbons, mercury, lithium, lead, bismuth, and a variety of inorganic salts can be used as working fluids. The tube itself has been made of copper, stainless steel, nickel, tungsten, molybdenum, tantalum, and a number of alloys. Wicks have been constructed of sintered, porous materials, woven mesh, fiberglass, longitudinal slots, and combinations of these structures. Today heat pipes are widely used in computer, telecommunication and other various electronic equipments. Estimated more than 75% of high performance notebook computers of today has integrated heat pipe in the thermal solution.

Most of the experiment on heat pipe was carried out with copper tube and water as a working fluid. The wick was used in most of the case. It was found that the copper- water heat pipe generated appreciable amount of non-condensable gas and corrosion and erosion of the container and wick. The non-condensable gas was mainly composed of CO_2 with limited presence of O_2 and N_2. To overcome this deficiency, the present study is an experimental investigation on heat pipe for electronic equipment cooling which will be carried out with stainless steel tube without wick and water as a working fluid. The design parameters used for the experiment included the effect of diameter of the heat pipe, angle of inclination and cooling air velocity.

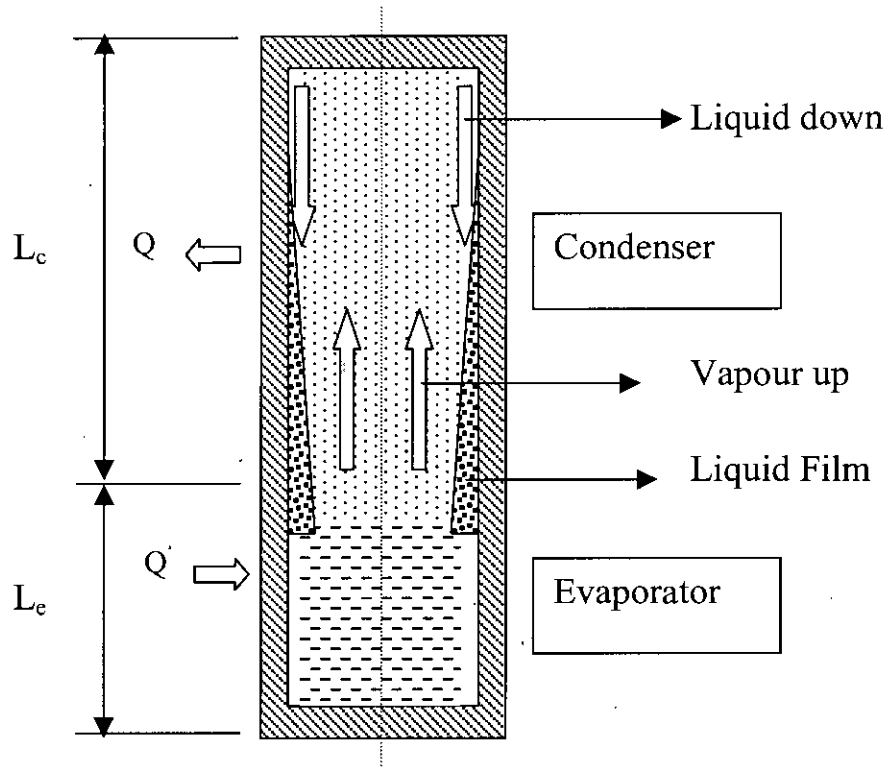


Fig.1.1 Schematic diagram of a Heat Pipe [2]

CHAPTER 2

TYPES OF HEAT PIPE AND ITS APPLICATION

2.1 TYPES OF HEAT PIPES

On the basis of control technique heat pipe can be categorised into four classes:

a. Condenser blocking with non-condensing gases (gas loaded heat pipe)
b. Condenser flooding with excess working fluid (Excess Liquid heat pipe)
c. Vapour flow control. (Vapour flow modulated heat pipe)
d. Liquid flow control (Liquid flow modulated heat pipe)

a. Gas – Loaded Heat Pipe

Figure 2.1 shows several examples of the gas – loaded heat pipe. During the operation of a heat pipe vapour flows from the evaporator to the condenser section. As a consequence any non-condensing gas present in the vapour is swept alone, and since such gas does not condense, it accumulates at the condenser, blocking a portion of the condenser. The volume of the non-condensing gas can be either self-controlled or feedback controlled.

b. The Excess Liquid Heat Pipe

It is closely related to the gas loaded heat pipe. During its operation, excess working fluid in the liquid phase is swept by the vapour to the condenser, thus inactivating portions of the condenser. Fig.2.2 illustrates such a system. A decrease in the vapour temperature in this system will expand the control fluid in the bellow, forcing excess liquid to inactive portions of the condenser.

c. Vapour Flow Modulated Heat Pipe

Here throttling process is applied between evaporator and condenser, which create a pressure difference between the two sections, and hence a corresponding temperature difference. Temperature characteristics of heat pipe can be varied using this principle. Figure 2.3 illustrates a vapour flow modulated heat pipe. An increase in heat load or source temperature raises the temperature of the vapour in the evaporator section, which in turn expands the control fluid so as to partially close the throttling valve and create a temperature and pressure drop. Hence, an increase in the heat load or source temperature is counter balanced by a decrease in the vapour temperature at the condenser section, thus minimising the change of the sink temperature.

d. Liquid flow modulated heat pipe

Figure 2.4 shows a liquid trap diode heat pipe. The liquid trap is a wick-lined reservoir located at the evaporator end of the diode. The trap wick is separated from the wick in the operating portion of the diode and does not communicate with it. In the normal mode of operation the trap is dry and the diode operates as an ordinary heat pipe with the correct amount of working fluid (i.e without excess liquid). Condensation begins to occur on the liquid trap surfaces, as the liquid trap end becomes the cold end of the pipe.

As liquid accumulates in the trap the main heat pipe wick becomes under-filled causing a fairly rapid reduction in pumping capacity. Eventually, the main wick dries out completely and the pipe ceases to function as a conductor and acts as an insulator instead.

Another classification can be made on the basis of structure:

i. Flat plate heat pipe
ii. Flexible heat pipe
iii. Anti-gravity thermosyphon
iv. Rotating heat pipe

i. Flat plate heat pipe

The flat plate heat pipe, one form of which functions in the same manner as the conventional tubular heat pipe, the main difference being the form the wick takes to enable liquid distribution over a wide surface area to be obtained. The main characteristic of this form of heat pipe is its ability to produce a surface with very small temperature gradients across it. This near-isothermal surface can be used to even out and remove hot spots produced by heaters, or to produce a very efficient radiator for cooling of devices mounted on it. Also, by mounting a number of heat-generating components on a plate heat pipe, they can be operated at similar temperatures due to the in-built equalisation process resulting from the fact that the vapour space will be at a fixed uniform temperature.

The wick structure is designed to ensure that liquid can be returned to and distributed over the surface of the plate in cases where operation against gravity is required. If the heat input is via the base, the sintered structure could be omitted, provision for liquid distribution across the bottom plate only being necessary. Location of the evaporator section along one edge of the plate is possible, and where a limited space is available for the evaporator, the vapour could be fed into the flat vapour chamber and the heat distributed over the larger flat surface.

ii. Flexible heat pipe

No work of major significance has been reported on flexible heat pipes during the past three years. However several manufacturers in the United States offer these units, and their desirability where source vibration is encountered, or where difficult assembly problems is not in question. The flexibility may be introduced by incorporating a bellows-types structure between the evaporator and condenser, or a simple flexible tube of some plastic material, with conventional metal sections for heat input and extraction regions.

iii. Anti-gravity thermosyphon or inverse thermosyphon

In the conventional thermosyphon the evaporator must be located below the condenser for satisfactory operation, as the device has to rely on gravity for condensate return. It is therefore ineffective in zero gravity or in cases where liquid has to be returned against a gravity head, however small.

As with conventional heat pipes and thermosyphon, the container is sealed and contains only the working fluid in liquid form. In order to drive the condensate back to the evaporator section, a vapour lift pump is used. This takes the form of a tube with its base in the sump containing the condensate the other end having an aperture leading to the evaporator. Creating two-phase mixture in the tube which, having a lower density than the liquid in the rest of the sump is forced back up to the top of the tube (known as the riser tube).

iv. The rotating heat pipe

The rotating heat pipe is a two-phase thermosyphon in which the condensate is returned to the evaporator by means of centrifugal force. The rotating heat pipe consists of a sealed hollow shaft, having a slight internal tape along its axial length, and containing a fixed amount of working fluid. The rotating heat pipe, like the conventional capillary heat pipe, is divided into three sections, the evaporator region, the adiabatic region and the condenser region. However, the rotation about the axis will cause a centrifugal acceleration $2\omega r$ that cause the condensed working fluid to flow along the wall back to the evaporator region.

2.2 APPLICATIONS OF HEAT PIPE

Heat pipes have found numerous applications. Today heat pipes are widely used in computers, telecommunications and various electronic equipment. Now typical applications of heat pipe [3] are presented below:

a. Notebook application of heat pipes

b. Desktop applications of heat pipes

c. Cooling of power electronic equipment

a. Notebook application of heat pipes

There are three basic designs to dissipate the heat generated by notebook CPUs by means of heat pipes:

i. Passive cooling design by standard heat pipes with wicks:

These designs use heat pipes to enhance the heat transfer from the CPU to the notebook case. At their side opposite to the CPU, heat pipes are usually attached to an aluminium plate that transfers the heat to the chassis or dissipates it under the keyboard as shown in Fig.2.6. Many current notebooks are actually equipped with this kind of heat removing system: however, this design is limited to a thermal performance in the range of 8 W, mainly because the surface temperature of a notebook case cannot exceed 15°C over ambient for ergonomic reasons.

To a certain extent, the thermal performance of such designs can be increased by using the so called Heat Pipe Hinge Systems, by means of which the heat is moved away from CPU, passes through an articulated joint and is then transferred to an aluminum plate located on the back of the display.

Heat pipes for passive cooling of notebooks are usually miniature size, with outer diameter from 2 to 4 mm (circular shape) or thickness from 2 to 4 mm (flat sub-rectangular shape). Heat pipes must also have an effective internal capillary structure, in the form of axial grooves, screen, sintered copper powder or bundle of fine wires. The thermal performance of these miniature heat pipes ranges from 5 to 20 W, depending on their diameter, vapour temperature and mainly wick type.

ii. Passive cooling by pulsating heat pipes (PHPs):

New generations of microprocessors are continuously developed at a faster pace, and CPUs dissipating 25 to 35 W will soon equip every kind of PCs, including notebooks. Hence, none of the above mentioned completely passive cooling designs would be applicable any longer, as they would result in skin temperatures exceeding ergonomic limits.

However, a passive cooling design is still acceptable at 35 W dissipated and 90° C maximum CPU temperatures provided that heat is transferred from the CPU to a finned heat sink via a flat pulsating heat pipe. The heat sink temperature will rise up to around 60°C, but all the other parts of the notebook case will be kept within the temperature ergonomic limits. For the heat transfer from the CPU to a finned heat sink a U shaped, flat PHP was developed by the company Actronics. The PHP body consists of a flat aluminium extrusion whose external rectangular shape incorporates many parallel capillary channels inside is shown in Fig.2.7. A 2mm thick, 50 mm wide and 250 mm long flat PHP with a very compact finned surface soldered to its cooled (condenser) part can transport up to 50 W with very small temperature drop.

iii. Forced air convection with heat pipes

This kind of Remote Heat Exchanger (RHE) as shown in Fig. 2.8 is widely employed in notebooks for its extreme compactness, lightweight and design flexibility. It features a miniature heat exchanger cooled by a micro fan at the cooled end of the heat pipe. This carries the heat away from the CPU through the usual input pad [3]. Sometimes a wider aluminium sheet is used instead of the input pad to gather the heat dissipated by other electronic devices, like video chipsets, memories and the like.

The obvious choice for locating the micro fan (size 20 to 30 mm square) is the back panel of the notebook. Usual dimensions for the miniature heat exchanger are: 20-30 mm width, 10-20 mm height, and 40-60 mm length. The highest thermal performance that can be expected by such cooling systems is around 12 W.

b. Desktop applications of heat pipes

The typical cooling system adopted for desktop computers now-a-days features an electrically driven fan and a finned heat sink directly attached onto the processor. Almost all latest Pentium processors are cooled this way. The finned heat sink can be machined from a standard aluminum extrusion or a die-cast unit. Nevertheless, at present desktop computer manufacturers are actively studying the possibility of switching over a fully passive, fan-less solution, to increase reliability, reduce costs and eliminate the noise. This means that the CPU and other smaller heat sources must be put in condition to dissipate together from 30 to 50 W only by natural convection and radiation. A possible way to solve the problem is to use a heat pipe to remove the heat from the electronic components and carry it very effectively (with a minimum temperature drop) to a location close to the computer case, or, if possible, to the chassis itself or part of it acting as an external heat exchanger.

Solutions of this kind are already in production for top-end PCs, industrial PCs and servers Fig.2.9. Particularly suitable for this job, systems using a flat pulsating heat pipe as the medium to thermally connect the input pad attached to the electronics with the remote dissipating unit are now under investigation as potential highest performers, like it was described for notebook cooling.

c. Cooling of power electronic equipment

Power semiconductor elements (PSEs) press pack (disc type) such as GTO thyristors and diodes or rectangular type such as power modules and IGBT modules with dissipated heat from 100 W up to 2000 W according to the type and rating have been widely used in transport drivers and static devices such as large motors, power control devices such as converters, inverters choppers further in substations, plating equipment, etc. Disc shape PSEs are usually arranged in a stack and various types of phase change coolers have been used for cooling of transport electric drivers in train locomotives, trams, metro train and trolleybuses.

i. Direct immersion thermosyphon coolers

Direct immersion thermosyphon coolers have been widely used in transport electric drivers for many years and company such as Hitachi, Mitsubishi, Toshiba, AEG, Siemens etc. have manufactured and used several designs of direct immersion thermosyphon coolers:

a. Electronic components are cooled either by pool bubble boiling or forced bubble boiling in falling liquid film.

b. Condensation of vapour is executed either in vapor space condenser or in a submerged condenser. Vapour space condensers are either directly adjacent to the evaporator part or they are remote from the evaporator parts and they are connected by tubes for flowing the vapour up and the liquid down as shown in Fig.2.10.

There are a number of benefits of these coolers: good electrical insulation, reduced thermal cycling, compact physical design, no pump required, high thermal efficiency to a unit of weight and built-in space. The disadvantages with these coolers include: the seal must be broken for maintenance; large volume of fluid required, electrical connections must be gas tight.

ii. Indirect coolers from straight tubular thermosyphon

In indirect contact coolers the surfaces of the evaporator parts of the heat pipes are linked to the other surfaces of the device. The heat pipe constitutes an independent compact air cooler, and clamping mechanism are usually used to achieve good contact with the semiconductor device. The proper design of the heat pipe may vary. Because of easy maintenance, lower weight and higher reliability heat pipe assembly [Fig.3.6] was introduced to transport electric drivers approx. 15 years ago and several companies, such as Furukawa, Fujikura, Hitachi Cable, Thermacore, Noren, Isoterix, SVUSS, Elbomec, Xeram, Liang Dao have been manufacturing them in two basic types:

a. all-metallic heat pipe assembly
b. electrically insulated heat pipe assembly

A typical indirect heat pipe assembly of power semiconductor elements is presented in the Fig.2.11. It is composed from several parallel heat pipes or thermosyphon in one or two rows, in which evaporator parts are pressed or brazed into a copper/aluminium input pad (block).

On the other condenser parts, a series of fins is attached. Waste heat from electronic components, which are mounted to the block either one side or both sides, is transferred by heat conduction to the evaporator parts through the heat pipe to the condenser part. The working fluid is usually water, methanol, FC72, mixture of water – ethylene glycol, etc, and its choice depends on the outside air temperature. Waste heat from condenser fins is rejected to a coolant either through free or forced convection. The main advantage of this cooler in comparison to the single-tube heat pipe cooler with face heating [7] is the increased reliability, because if one heat pipe fails in the bundle of heat pipes, the cooler is still operational, only with a smaller efficiency.

Electrically insulated heat pipe assemblies are intended for cooling semiconductor devices, in which evaporator zones are connected with areas of elevated electrical potential and condenser zones are connected to standard assembly areas, mainly in traction drivers. Such a cooler is composed of an aluminium block with a current terminal, a plurality of copper / FC72 heat pipes and a large number of aluminium radiation fins. Between the copper evaporator zone and copper condenser zone an electrically isolated ceramic parts is located shown in Fig.3.7. The voltage standoff is up to 1200 V.

The company ATHERM has developed and manufactured for traction drive applications electrically insulated heat pipe coolers in which an electrical insulation of the input block from the condenser part is achieved by means of an epoxy tube inserted between the input block and the evaporator part of the tube. The voltage standoff is approx. 5000 V.
Heat pipe assemblies of IGBT modules are composed of the aluminium blocks, copper/water heat pipes and aluminium fins. Several power modules or IGBTs are bolted to the aluminium input block. This heat pipe cooler is intended mainly for vertical (block down) operation with forced air-cooling of the finned condenser zone.
The heat pipes can be extended through a panel separating a sealed electronic enclosure from the air enclosure, improving the cleanliness in the electronic compartment. The company HITACHI Cable [9] has tested a heat pipe cooler of IGBT modules used in electronic railcar drive system. Several IGBT modules are attached to the block as shown in Fig.2.13 and the evaporator parts of the heat pipes are inserted into the block from the opposite side. Air-cooled fin arrays are attached to the condenser part at the heat pipes.
Recently IGBT modules have been used in traction drivers and companies such as Fujikura, Furukawa, Thermacore, Noren, Elbomec etc. are now producing heat pipe assemblies with a thermal performance of up to 1500 W in the case of natural air-cooling and up to 7000 W in the case of forced air cooling. Basic differences between heat pipe assemblies of disc type PSE and IGBTs are as follows:

(i). Dissipated heat from the present disc type PSE is from 500 W to 1500W depending on the rating. PSE's and heat pipe assemblies are stacked in a parallel arrangement.

(ii). Dissipated heat from the present IGBT modules is up to 2000 W and several IGBT's modules are clamped by screw into one or both sides of an input pad. It means, that present heat pipe assemblies of IGBT modules should be designed up to 7000 W for forced air convection and they have much larger input pads and more heat pipes (up to twenty) than the case of disc type PSEs.

iii. Indirect loop thermosyphon coolers

GEC ALSHOM with INSA Lyon/32 have performed R and D on various types of thermosyphon coolers of PSEs. In the first one, the PSEs are pressed in a sandwich pattern between two small hollow heat sinks partially filled with liquid. The vapour evaporated in these heat sinks is collected in a vapor tube and carried to a common condenser as shown in Fig.2.14. After condensation of the vapour, the liquid returns by a liquid tube, which guarantees a continual supply to the heat sinks. The advantage of this device is that the condenser only dissipates the mean losses from all the PSEs in a specific operating cycle. French companies XERAM, FERRAZ and DATE have manufactured these separate thermosyphon indirect loops called System Modulaire Fluid (SMF). In the newest innovative version the evaporator is connected with the coolant reservoir through an electrically insulated conduit, so that the reservoir and the condenser are electrically insulated and kept at ground potential.

The Japanese company DEWSO [11] developed an indirect loop thermosyphon cooler of IGBT modules which consists of a multi radiator and a refrigerant bath with multiple power modules, mounted vertically as shown in Fig.3.10.

When compared to the conventional aluminium fin system, this design has only 1/4 of size and 1/5 of weight in comparison to aluminium fin system.

The company FURUKAWA [12] has developed an indirect loop thermosyphon cooler at IGBT modules used in traction drive systems. IGBT modules are attached to one side at almost vertical block Fig.2.15 and the evaporator parts at the thermosyphon loops are inserted into the block from the opposite side. Air-cooled fin arrays are attached to the condenser part of the thermosyphon loops.

iv. Pulsating Heat pipe

ACTRNICS [6] has developed and manufactured pulsating heat pipes (PHPs) for cooling of IGBT modules. The first PHP coolers were created from capillary tube turns, of which one part is embedded into the evaporator aluminium input pad and the other part serves as a condenser-radiating fin as shown in Fig.2.16. The height of input pad is 110 mm and the height of radiating tube turns 200mm. The thermal resistance of this PHP cooler is 0.022 kW the face air velocity 3m/s and the thermal performance 1800W.

Another PHP cooler of IGBT is created from flat extruded aluminium plate with 16 capillary channels inside. This plate is filled with a working fluid and bent as shown in Fig.2.7. One part is soldered into an input and the other part serves as a condenser-radiating fin.

v. Vapour chamber thermosyphon heat spreader

Presently, very efficient air cooler of IGBT modules are created from a massive aluminium block with extruded or folded high axial fins (60 – 100 mm) on the opposite side or are created from an aluminium rectangular channel with plenty of inner sub-channels for air flow.

Improvement in effectiveness of these aluminium air coolers, mainly in spreading locally concentrated thermal load faster, could be by heat pipes:

(i) Heat pipes are inserted into some cylindrical holes through which cooling air flows. Tests at Sumitomo [13] proved, that the surface temperature of the heat sink with heat pipes was approx. 30°C lower than when compared with the heat sink without heat pipes at the thermal performance of 1600 W.

(ii) DORNIER and SVUSS manufactured heat pipes directly into a solid part in contact with IGBT modules. While DORNIER manufactured individual axial heat pipes with a grooved wick, SVUSS drilled cross-holes so as to create a vapour chamber inside the solid part as shown in Fig.3.13. Heat dissipated from IGBT modules, attached on the smooth heat sink surface, is effectively spread to heat sink fins. This means, that the heat pipes serve as a spreader inside an air heat sink.

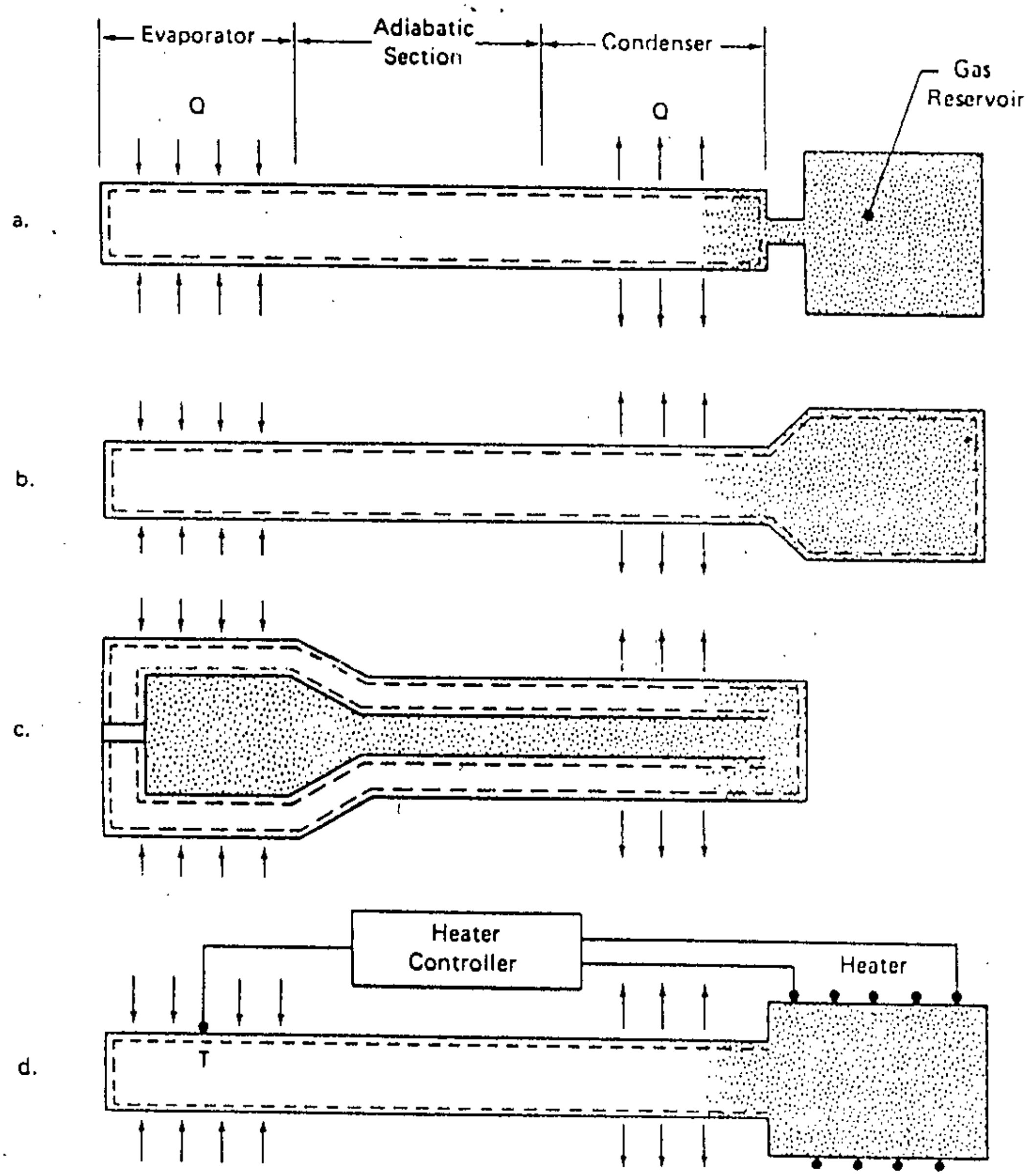
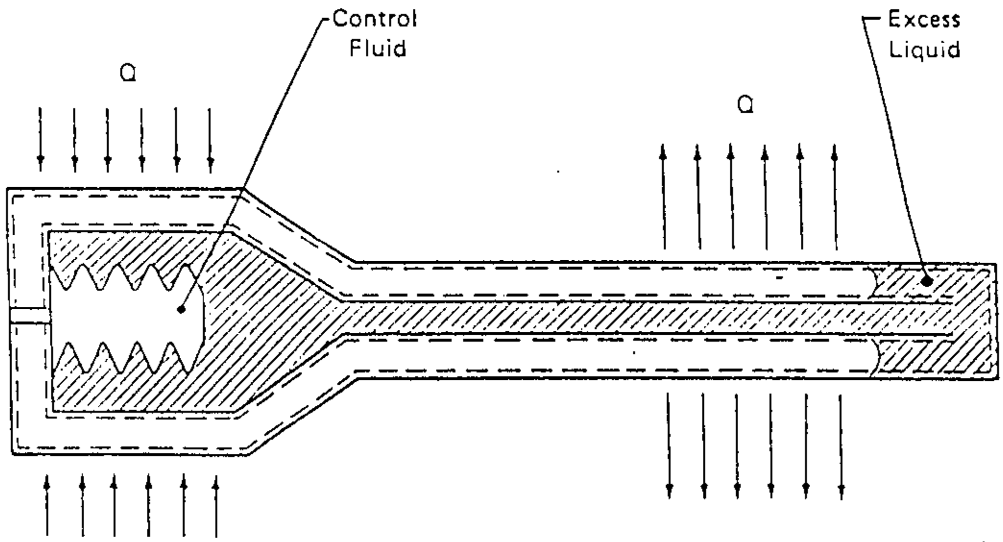
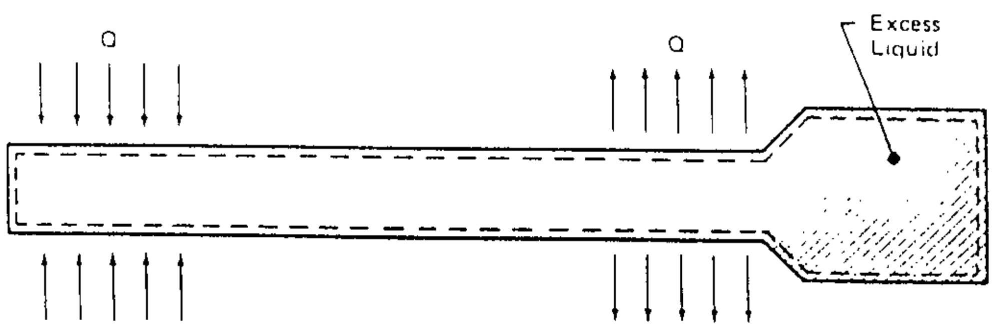


Fig. 2.1 Schematic of gas- loaded Heat Pipes [3]

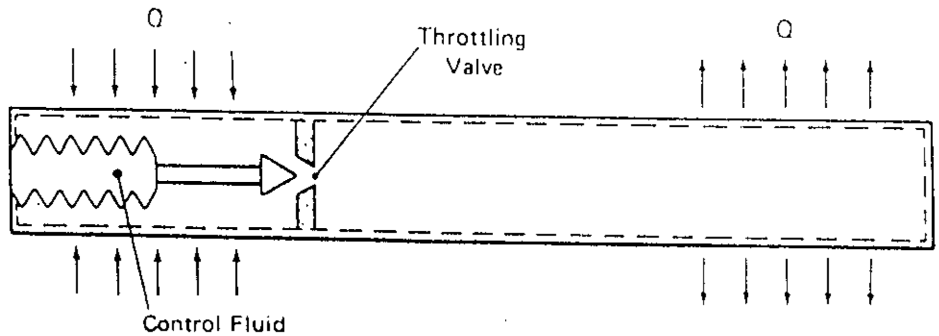


a. Variable Conductance.

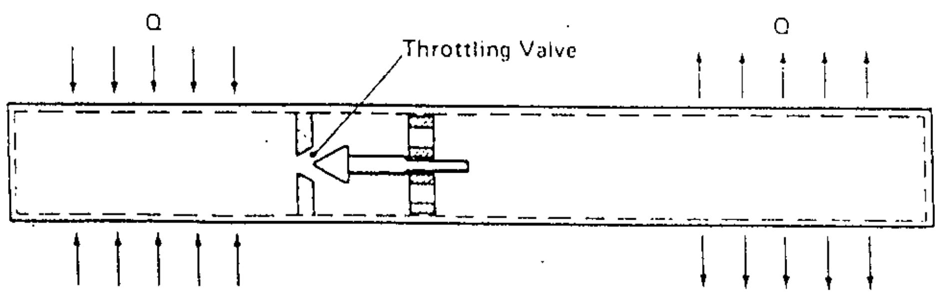


b. Vapour-Modulated Variable Conductance.

Fig. 2.2 Schematics of excess- liquid Heat Pipes [3]

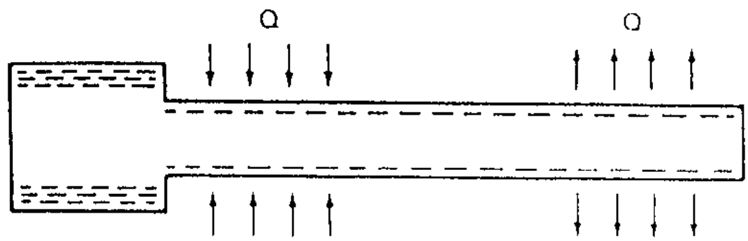


a. Vapour-Modulated Variable Conductance.

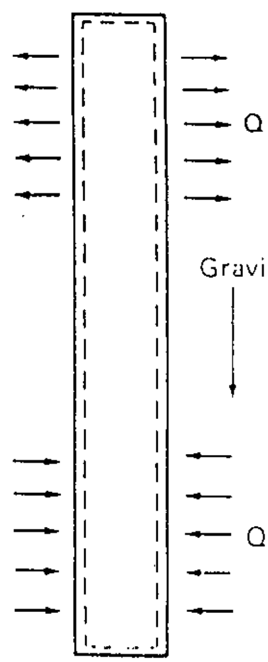


b. Vapour-Modulated Thermal Diode

Fig. 2.3 Schematic of vapour-flow Modulated Heat Pipes [3]

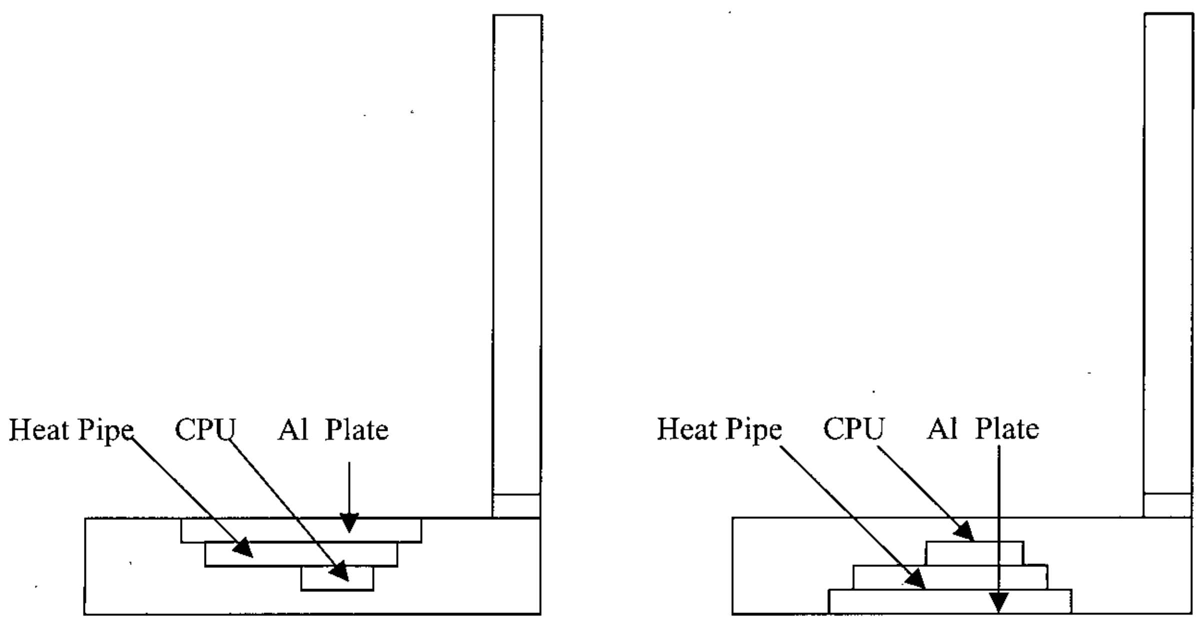


a. Liquid Trap Diode Heat Pipe.



b. Gravity Operated Diode Heat Pipe.

Fig. 2.4 Schematic of liquid-flow Modulated Heat Pipes [3]



(a) Heat Dissipated Under Keyboard, (b) Heat Dissipated On Chassis

Fig. 2.5 Heat Pipe Cooling at Notebooks [2,4]

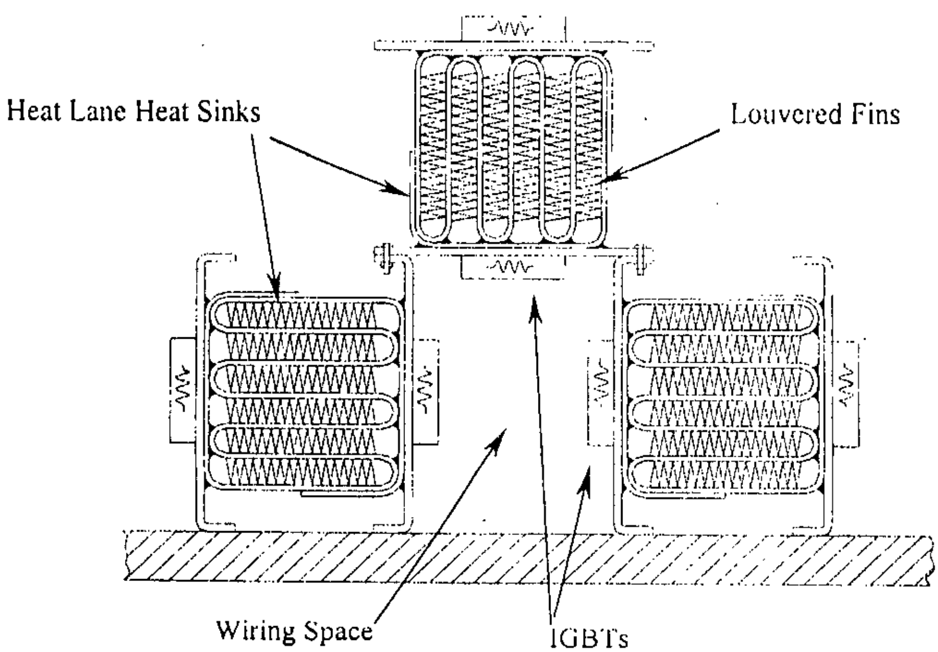


Fig. 2.6 Pulsating Heat Pipe "Heat Lane"[2,5]

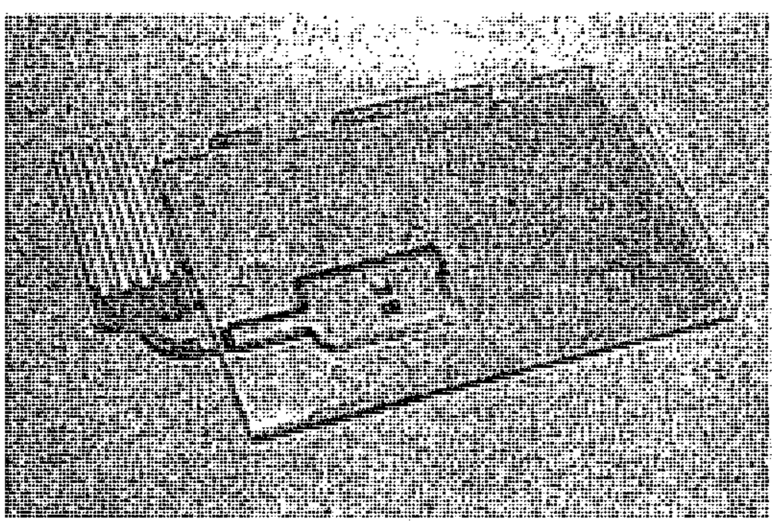


Fig. 2.7 Forced Air Cooling system for Notebooks
(Micro Fan is not shown) [2,6]

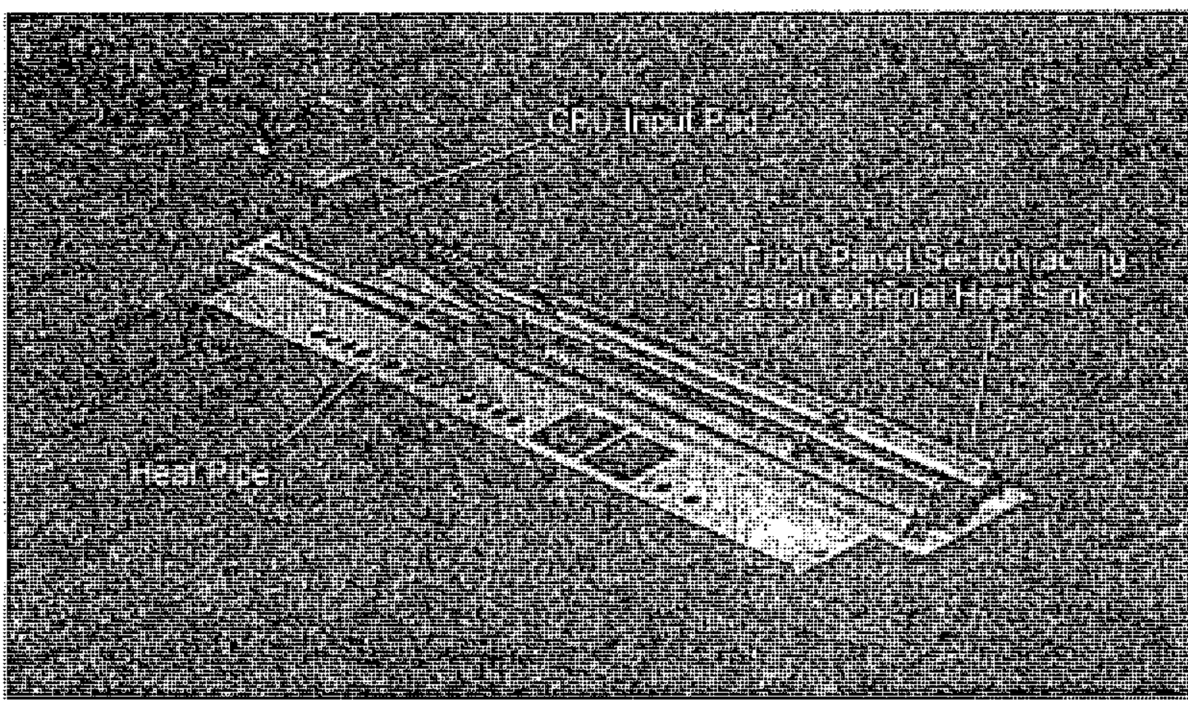


Fig. 2.8 Telecom Computer Heat Pipe Passive Cooling System [2,6]

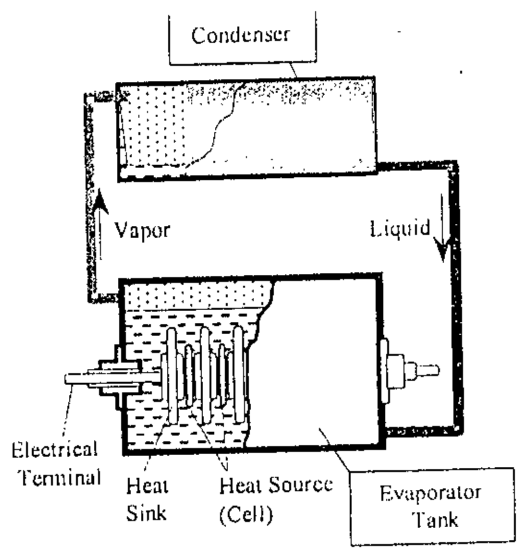


Fig. 2.9 Schematic of an Immersion Thermosyphon Loop [2]

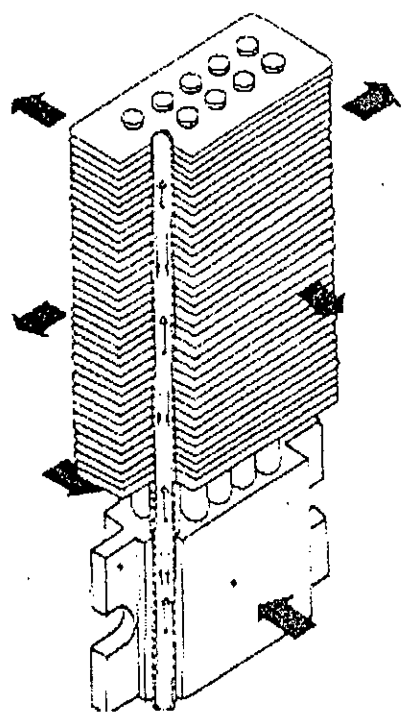


Fig. 2.10 Indirect Tubular Thermosyphon Assembly for Power Semiconductors [2]

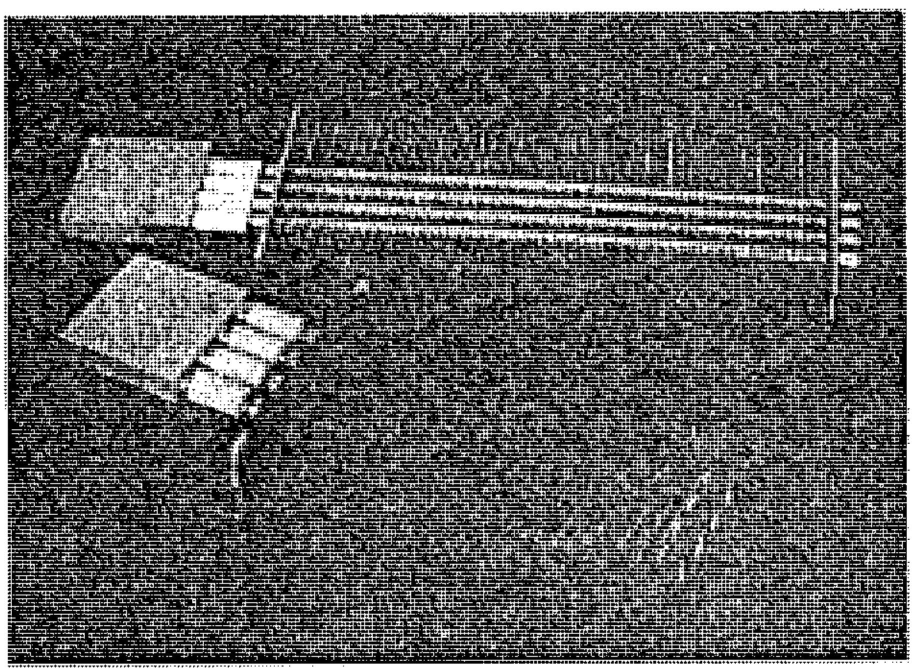


Fig. 2.11 Electrically Insulated Heat Pipe Coolers [2,8]

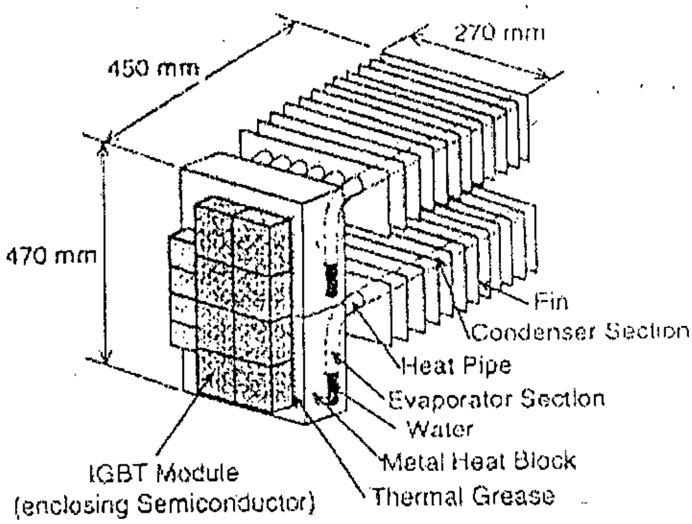


Fig. 2.12 Heat Pipe Cooler of IGBT Modules [2]

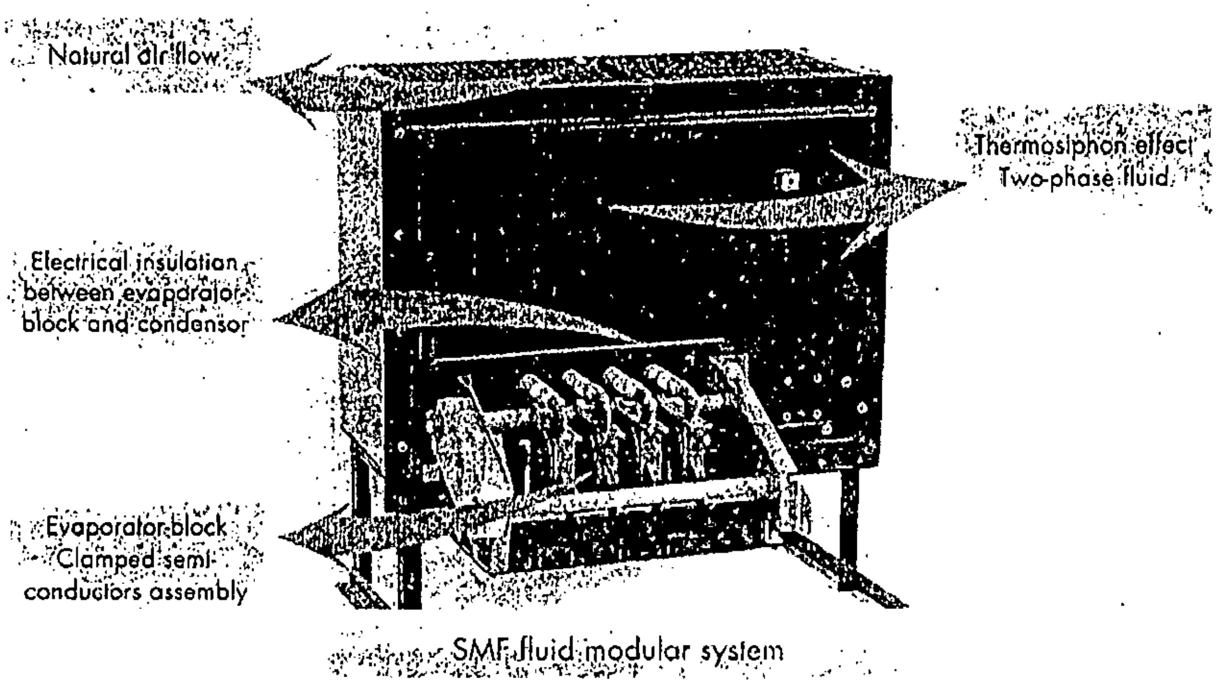


Fig. 2.13 Indirect Thermosyphon Coolers of PSE [2,10]

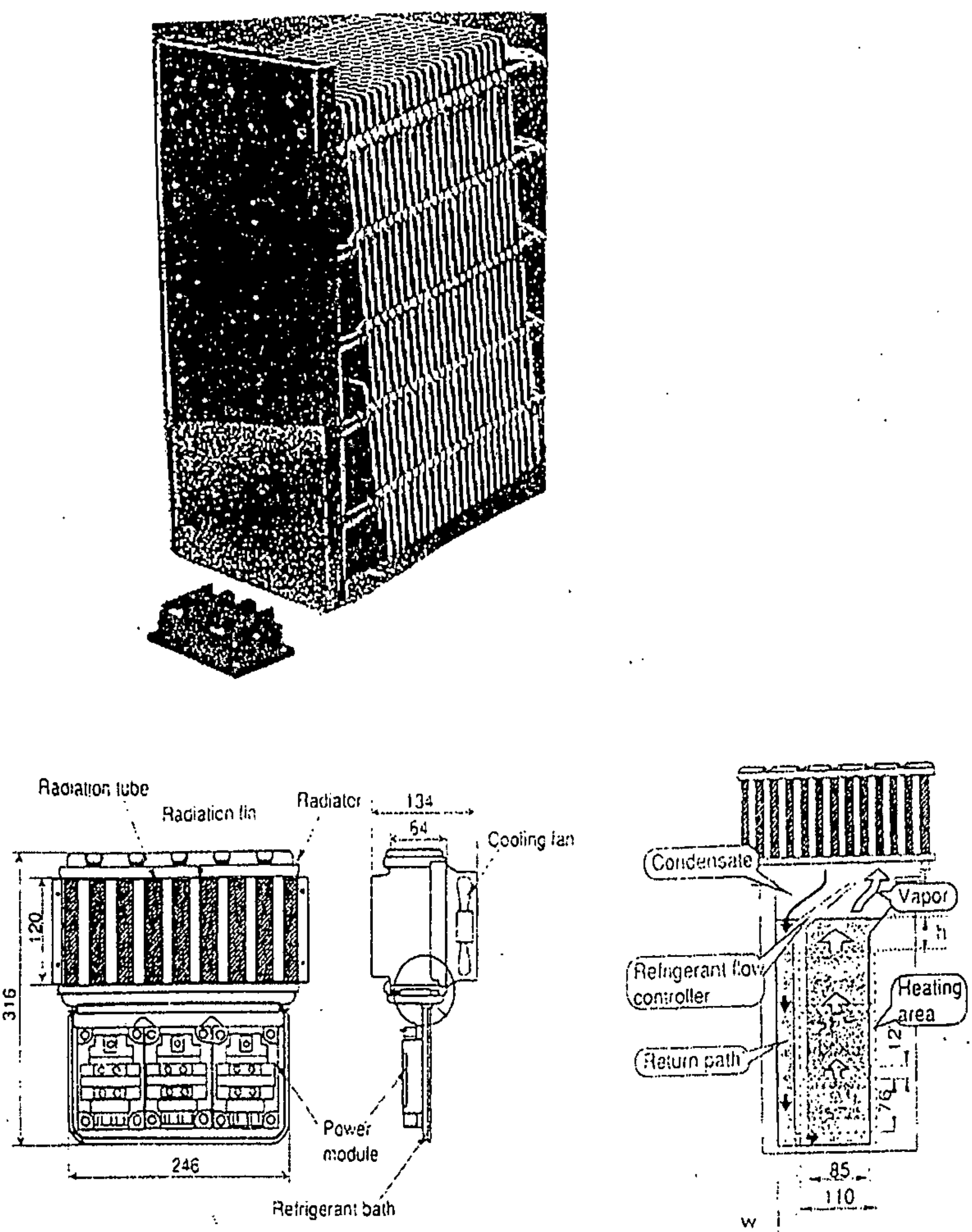


Fig. 2.14 Indirect Loop Thermosyphon Cooler [2,11,12]

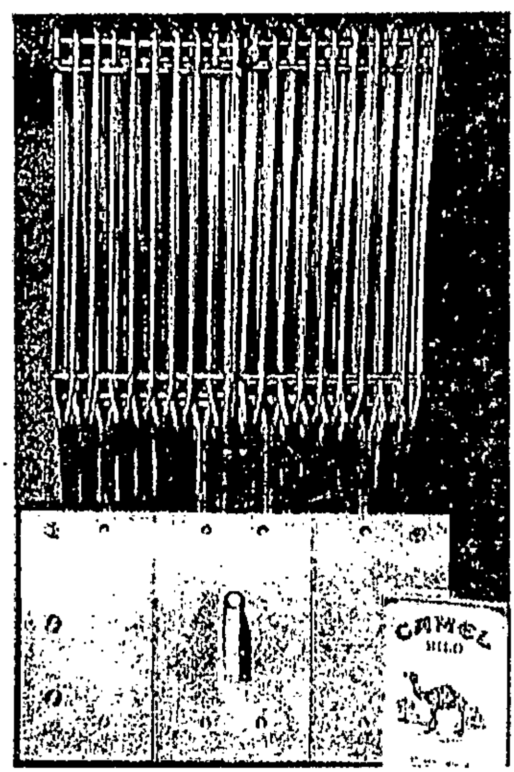


Fig. 2.15 Pulsating Heat Pipe Cooler of IGBT Modules [2,5]

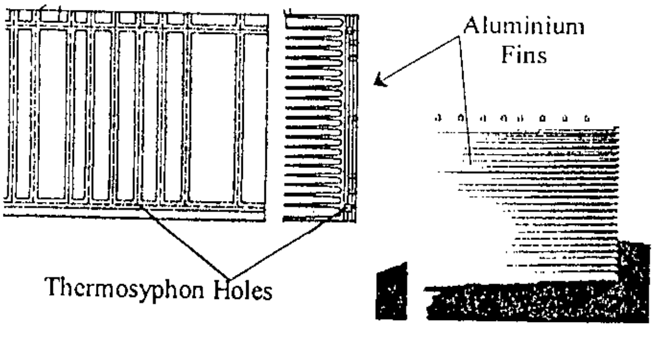


Fig. 2.16 Vapour Chamber Thermosyphon used as a heat spreader [2]

CHAPTER 3

DESIGN OF HEAT PIPE

3.1 DESIGN CONSIDERATION

To design a heat pipe, combination of fluids, wick structure, wick materials and container materials should be selected before detail design calculations are made. For this purpose the operating temperature of the pipe may assumed to be known. The selection process can be made as follows:

a. Working fluids are selected first with the aid of Fig. 3.1 to Fig.3.3. Working fluid with a large liquid transport factor and a large liquid conductance factor are usually preferred. However flammability and toxicity may sometimes also determine the fluid to be selected.

b. Appropriate wick structures are next to determined. Wicks of Fig 3.4 and Fig 3.5 have been used in practice. Simple wicks may be considered for their low cost and complex wicks can be considered for their high performance. Often several possible wick structures may be selected at this stage.

c. Wick and container materials are so selected that they will be compatible with the working fluids. Figure 3.6 and Fig 3.7 shows the material conductance parameter and material weight parameter for several heat pipe material respectively. For a minimum weight, the material with small value of (ρ/f_u) should be chosen and for a small temperature gradient the ultimate strength (kf_u) must be small.

In addition materials and fabrication cost may also play an important role in determining the material to be used.

3.2 FLUID SELECTION

For the successful operation of a heat pipe, its working fluid must be in the liquid state. That is, the selected fluid for the heat pipe must have a melting point temperature below and a critical point temperature above the pipe operating temperature. Within a temperature limit several possible working fluids may exist and a variety of characteristics must be examined in order to determine the most acceptable of these fluids for the application being considered. The prime requirements are

a. Compatibility with wick and wall materials
b. Good thermal stability
c. Wet ability of wick and wall materials
d. Vapour pressures not too high over the operating temperature range
e. High latent heat
f. Low liquid and vapour viscosities.
g. High surface tension

Figure 3.1 shows temperature ranges from melting point to critical point temperature for several fluids. Because of the overlapping of these temperature ranges for different fluids, several fluids can often be used for a given operating temperature. The relative merits of different fluids can best be observed by examining heat pipe theory as follows.

The following simplifying assumptions are made:

i. Vapour pressure losses are negligible.

ii. Wick thickness is much smaller than the vapour core radius.

iii. Heat flux density is uniform at the evaporator or condenser surface.

iv. Thermal conductivity of the liquid-saturated wick is proportional to that of the liquid.

The heat transport factor $(Q\ L)_{c,\ max}$ under these assumptions, can then be written as:

$$(Q\ L)_{c,\,max} = 2\,(\sigma\,\rho_l\lambda\,/\,\mu_l)\;\;(K\,/\,r_c)\,(\,2\pi\,r_v\,t_w\,) \quad \text{................} \quad (3.1)$$

Here the term in parentheses represent, from left to right, the liquid property, the wick property and the wick cross sectional area, respectively, Hence Eqn (2.1) indicates that for a pipe of fixed wick structure and dimensions its heat transport factor $(Q\ L)_{c,\ max}$ is directly proportional to the liquid transport factor N_l, which is defined as $(\sigma\ \rho_l\lambda\ /\ \mu_l\ )$. Figure 3.3 shows the values of liquid transport factor N_l for several fluids.

For a minimum temperature gradient of a heat pipe, the temperature drop across the liquid–saturated wick must be minimum. Based upon the simplifying assumptions (2) and (4) described above, the temperature drop across the wick is proportional to $Q\ t_w\ /\ k_l$, that is

$$\Delta T \propto Q\,t_w\,/\,k_l \quad \text{..........} \quad (3.2)$$

In addition, it can be seen from Eqn. (3.1) that the required wick thickness t_w for the same heat transport is inversely proportional to the liquid transport factor N_l. Equation 3.2 can be written as

$$\Delta\,T \propto\; Q\;/\;(\,k_l\,N_l\,) \quad \text{..........} \quad (3.3)$$

This equation indicates that the temperature drop across a wick is inversely proportional to the liquid property (k_lN_l), which has been called the liquid conductance factor. The value of the liquid conductance factor is shown in Fig 3.2.

From the above examination, it can be seen that for heat pipes with a large heat transport capability but a small temperature gradient, one may select fluids that have a large heat transport factor and liquid conductance factor, as the working fluids. In addition, toxicity and flammability of the liquid may also have to be considered in certain applications.

3.3 WICK SELECTION

The purpose of a heat pipe wick is three fold:

a. To provide the necessary flow passage for the return of liquid from the condenser to the evaporator.

b. To provide surface pores at the liquid-vapour interface for the development of capillary pumping pressure.

c. To provide the heat flow path from the inner wall of the container to the liquid-vapour interface.

It can be seen from the Eqn (3.1) that, for a large heat transport capability the wick structure must have large permeability K and small capillary pore radius r_c. In addition, it can be seen be from the equation, $K=2\varepsilon\, r^2_{h,l}/\,(f_l\, Re_l)$ that the wick permeability K is proportional to the product of the porosity ε and the square of the hydraulic radius $r_{h,l}$. Numerous wick structures both homogeneous and composite as shown in Fig.3.4 and Fig.3.5 respectively have been developed in general, high performance of wick have large value for ε and $r_{h,l}$ but small r_c values. However the other consideration such as self-priming i.e. the capability of filling the wick in liquid without external assistance, boiling susceptibility of the liquid in the wicks, static rising height of the liquid in the wick and cost of wick fabrication must also be considered in the selection of wick structures. In addition, the effect of wick structure on the pipe temperature gradient may also be important.

In view of the above fact, a large number of factors affect the wick selection. It is not possible to give definite rules in the selection of wick structures. However, some remarks on the wick selection are appropriate. If several wick structures satisfy the performance specifications, the simplest wick structure should be used. For this reason, the wrapped screen wick as shown in Fig 3.4a and the composite screen wick as shown in Fig.3.5a are most commonly used in practice. However, when screen wicks are used for cryogenic or moderate temperature pipes, the temperature drop may be excessive and a sintered metal wick as shown in Fig.3.4b may be an alternative. When screen wick and sintered metal wick do not have sufficient heat transfer capability, annular, crescent and artery wicks as shown in Fig 3.4d to Fig.3.4f can be considered. It should, however, be noted that when annular and crescent wicks are used with cryogenic or moderate temperature liquid i.e. those with low conductivity, the temperature characteristics of the pipe will be poor and the boiling limit will likely be reached.

For good temperature characteristic, axial groove or screen covered groove wicks Fig.3.4c and Fig.3.5b may be used. However, unit costs for small quantity production of grooved pipes may be excessive and under this condition, artery wicks may be preferred. Slab and tunnel wicks Fig.3.5c and Fig.3.5d are high performance wicks that have high heat transfer capability and good temperature characteristics, but the cost of fabrication may be relatively high.

3.4 MATERIAL SELECTION

A major factor in the selection of materials for heat pipe wicks and containers is their compatibility with working fluids. This is an important consideration because heat pipes are subjected to continuous performance degradation as a result of a chemical reaction or decomposition of working fluids and corrosion or erosion of the container wicks.

Chemical reaction or decomposition of the working fluid may give rise to a non-condensable gas evolution. A specific example is the hydrolysis of water yielding hydrogen gas in water-aluminium heat pipe. In a conventional heat pipe all non-condensable gas is swept to the condenser end, thus inactivating a portion of the condenser.

Corrosion and erosion of the container and wick may result in a change of the fluid wetting angle and the permeability or capillary pore size of the wick. Solid particles resulting from corrosion and erosion are transported by the flowing fluid to the evaporator region and deposited there. This leads to an increased resistance to fluid flow in the evaporator, which result in a decrease of heat transport capability for the heat pipe. At present there is no definite theory for predicting materials compatibility. Nevertheless extensive tests on heat pipe have been run to empirically determine material compatibility.

3.5 DESIGN OF HEAT PIPE CONTAINERS

The function of the heat pipe container is to isolate the working fluid from the outside environment. It should therefore, be leak-proof, maintain the pressure differential across its walls, and able to transfer heat to and from the working fluid. Selection of the container material depends on several factors. These are as follows:

a. Compatibility (both with working fluid and the external environment).
b. Strength to weight ratio.
c. Thermal conductivity.
d. Ease of fabrication.
e. Porosity.
b. Wetability.

Most of these are self-explanatory. A high strength to weight ratio is more important in spacecraft applications, and the material should be non porous to prevent the diffusion of gas into the heat pipe. A high thermal conductivity ensures minimum temperature drop between the heat sources and the wick. The most widely used design technique for heat pipe containers that must withstand vapour pressure is the ASME code for unfired pressure vessels. The ASME code specifies that the maximum allowable stress at any temperature be one-quarter of the material's ultimate tensile strength f_m that temperature. For round tubes, in which the wall thickness is less than 10% of the diameter, the maximum pressure stress is closely approximated by the simple expression.

$$f_{max} = Pd_o / 2t \qquad (3.5)$$

Where,

f_{max} = maximum hoop stress in the wall

P = Pressure differential across the wall

d_o = tube outside diameter

t = tube wall thickness

The maximum hoop stress in a thick-wall cylinder, subjected to internal pressure, is given by the expression: -

$$f_{max} = P(d_o^2 + d_i^2)/(d_o^2 - d_i^2) \quad (3.6)$$

Where,

f_{max} = maximum hoop stress in the wall

P = pressure differential across the wall

d_o = outside diameter of the tube.

d_i = inside diameter of the tube.

t = thickness of the tube.

Ends of the pipe container can be fitted with hemispherical, conical, or flat end caps. The maximum stress in a thick-walled hemispherical end cap is

$$f_{max} = P(d_o^3 + 2d_i^3)/2(d_o^3 - d_i^3) \quad (3.7)$$

If the wall thickness of the hemispherical end cap is less than 10% of it's diameter, the above equation can be approximated by

$$f_{max} = P d_o / 4t \quad (3.8)$$

The maximum stress in a flat circular end cap can be calculated by the equation:

$$f_{max} = P d_o^2 / 8t^2 \quad (3.9)$$

Where,

f_{max} = maximum stress

P = Pressure differential across the wall

d_o = end cap diameter

t = end cap thickness

For design calculations, the internal pressure of the pipe is equal to the saturation vapour pressure of the pipe working fluid at each operating temperature or its maximum cycle pressure whichever is large. The pressure differential is equal to the vapour pressure minus the ambient pressure. As the vapour pressure is usually much larger than the ambient pressure, the vapour pressure is therefore, approximately equal to the pressure differential. Figure 3.9 contains vapour pressure versus temperature information for several fluids. The maximum allowable stress is equal to one-quarter of the ultimate tensile stress (UTS). Ultimate tensile stress for different material can be found on different books of properties of solids materials. Figure 3-10 contains design curves, which can be used to determine quickly the required tube size when the pipe operating pressure and the ultimate tensile stress are known.

2.6 HEAT PIPE DIAMETER

Although heap pipes can be made of different cross-sectional shapes, the round shapes are the most common configuration. Round tubes and pipes of many materials are readily available, and they are the most efficient configuration from the stress point of view. The size of the pipe diameter necessary for a given application should be determined so that vapour velocity is not excessive. Control of vapour velocity is required since at high Mach numbers the flow compressibility of vapour contributes to a large axial temperature gradient. For convenience the heat pipe can be designed so that its maximum Mach number in the vapour flow passage does not exceed 0.2 under temperature gradient negligibly small.

For a heat pipe whose heat transport mode requirements and consequently maximum axial heat flux Q_{max} are known, the required vapour core diameter d_v, at vapor Mach number M_v equal to 0.2, can be determined from Fig.3.8 and the following equation:

$$d_v = [20\, Q_{max} / \pi\, \rho_v\, \lambda\, \sqrt{\lambda_v R_v T_v}]^{1/2} \qquad (3.10)$$

Where,

d_v = vapour core diameter

Q_{max} = maximum axial heat flux

ρ_v = vapour density

λ_v = vapour specific heat ratio

λ = latent heat of vapourization

R_v = gas constant for the vapour

T_v = vapuor temperature

3.7 DESIGN PROCEDURE

Method of selecting heat pipe working fluids, wick structures, wick material and container materials has been developed in the previous articles. Physical properties of fluids, wick and materials together with the problem specifications form the input for the design theory. The design procedure are as follows:

a. Pipe diameter will first be determined so that the vapour velocity is not excessive.

b. Mechanical design will be used to determine the container details.

c. Wick details will be designed considering the capillary limit.

d. Other heat transport limit, i.e. entrainment and boiling limit, will be checked to ensure that heat pipe will operate within all limits.

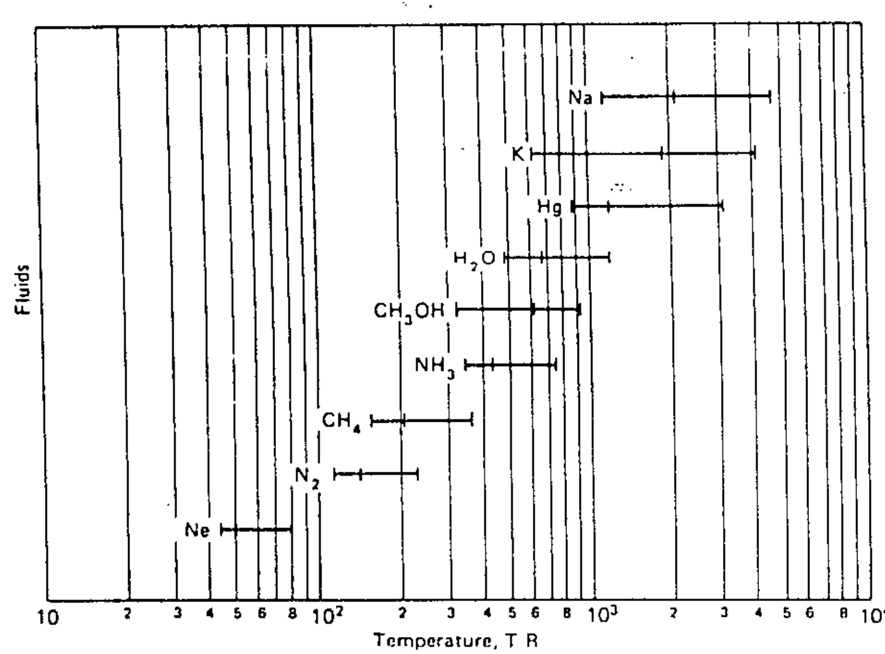


Fig. 3.1 Normal melting point, boiling point, and critical point temperatures for several heat pipe working fluids (1R = 0.5556 K). [3]

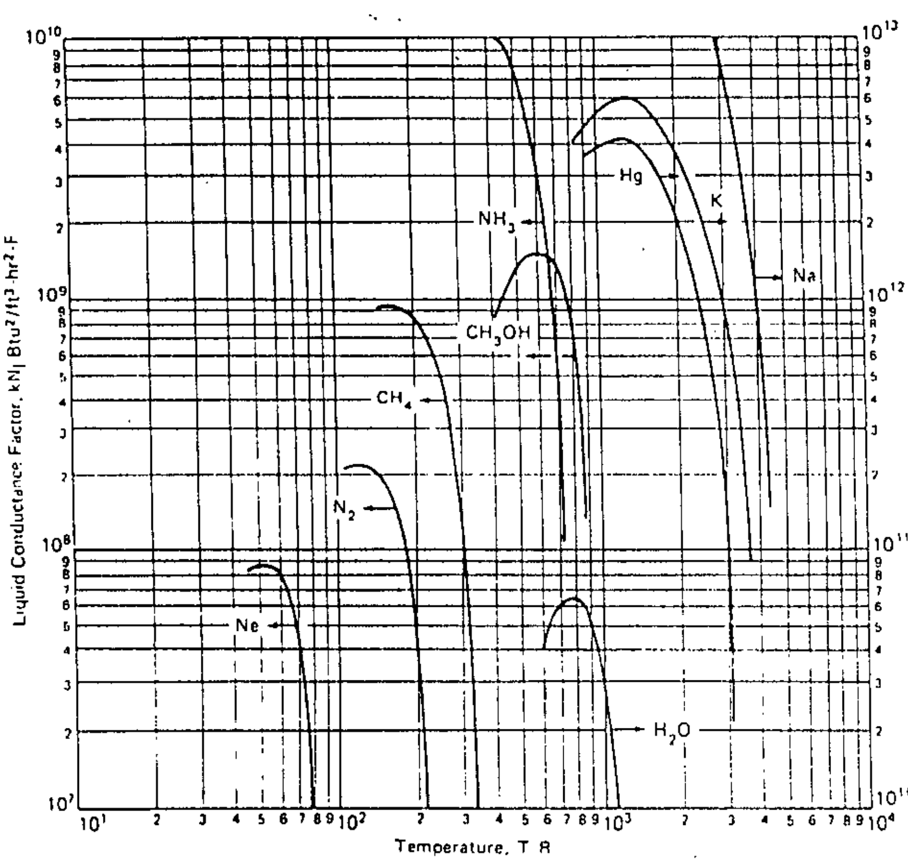


Fig. 3.2 Liquid conductance factor versus temperatures for several heat pipe working fluids (1 Btu ^{2}/ ft ^{3} - hr-F = 5.455 W ^{2}/ m ^{3}-K; 1R= 0.5556K). [3]

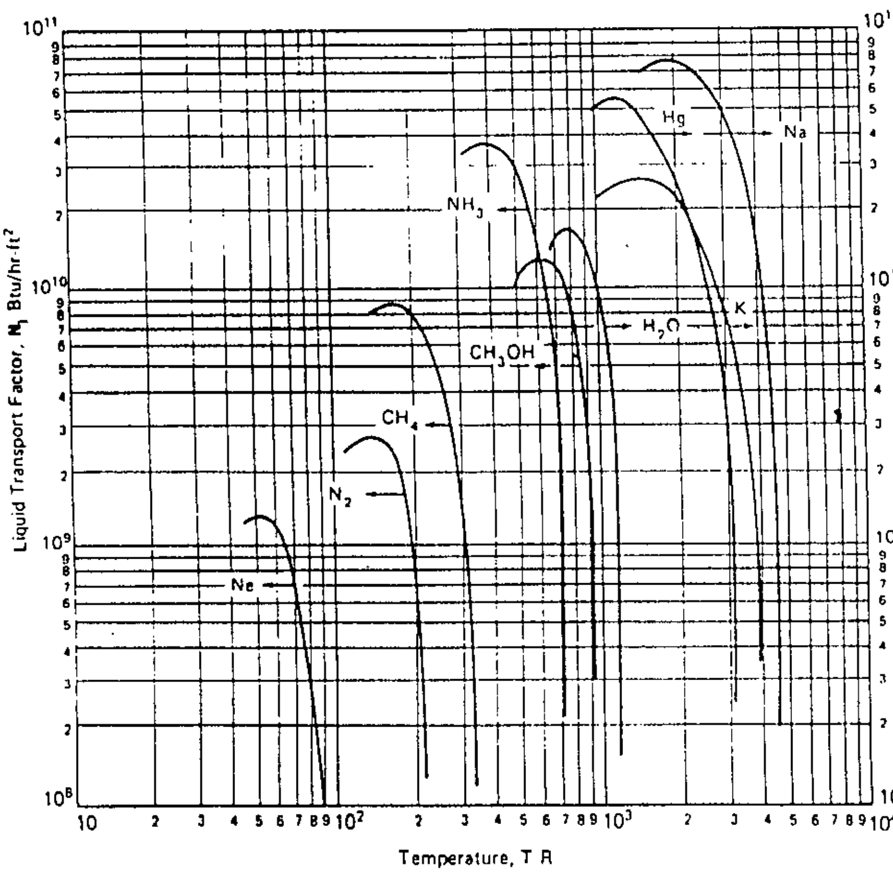


Fig. 3.3 Liquid transport factor versus temperatures for several heat pipe working fluids (1 Btu / hr-ft ^{2} = 3.153 W / m ^{2}; 1R= 0.5556K). [3]

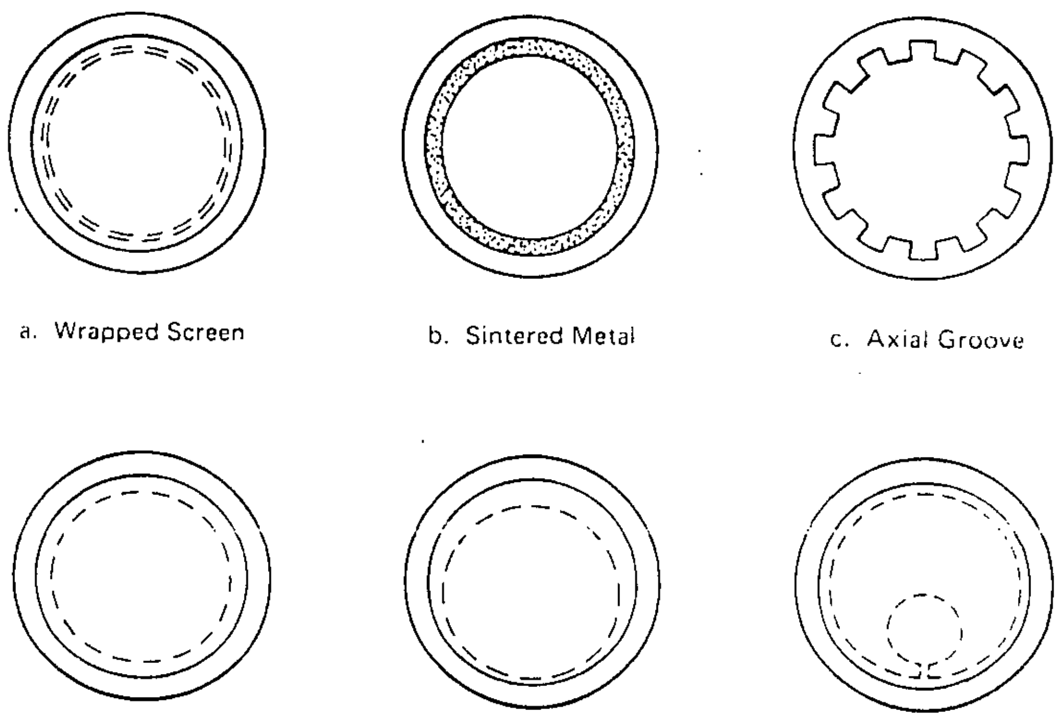


Fig. 3.4 Examples of homogenous wick structures [3]

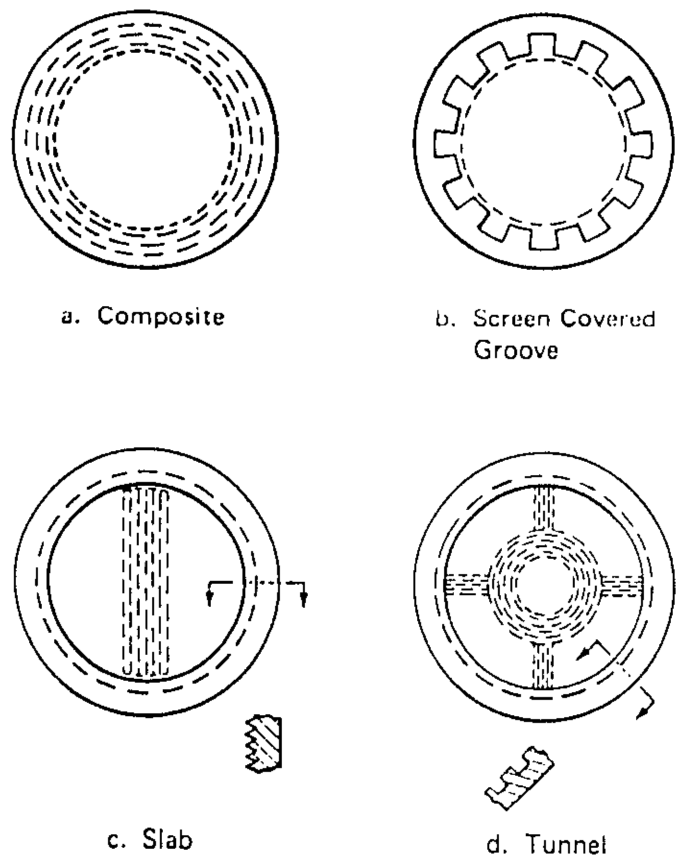


Fig. 3.5 Examples of composite wick structures [3]

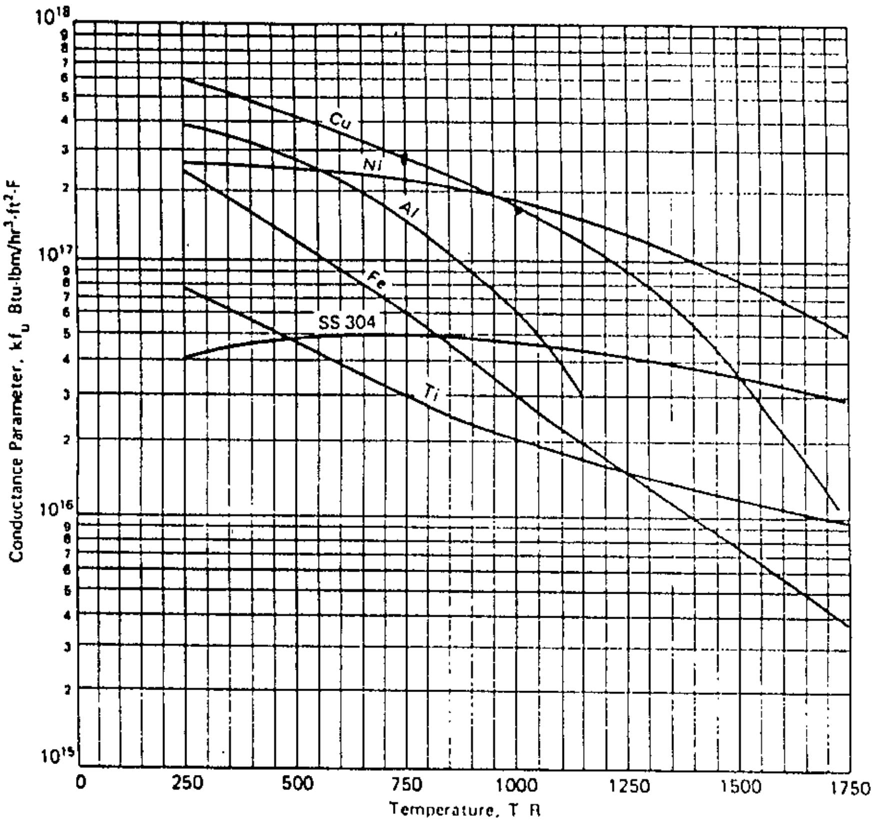


Fig. 3.6 Material conductance parameter versus temperatures for several heat pipe material [3]

(1 Btu-lbm /hr^3.ft^2-F =1.986×10^{-7} W-kg / sec^2-m^2-K;1R= 0.5556K).

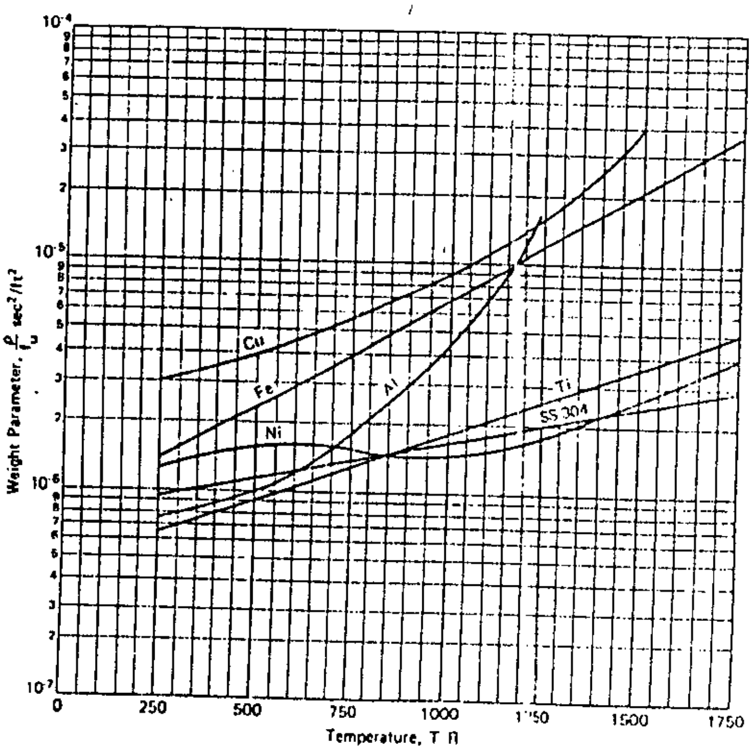


Fig.3.7 Material weight parameter versus temperature for several heat pipe materials (1 sec^2/ft^2 = 10.76 sec^2/m^2; 1R= 0.5556K) [3]

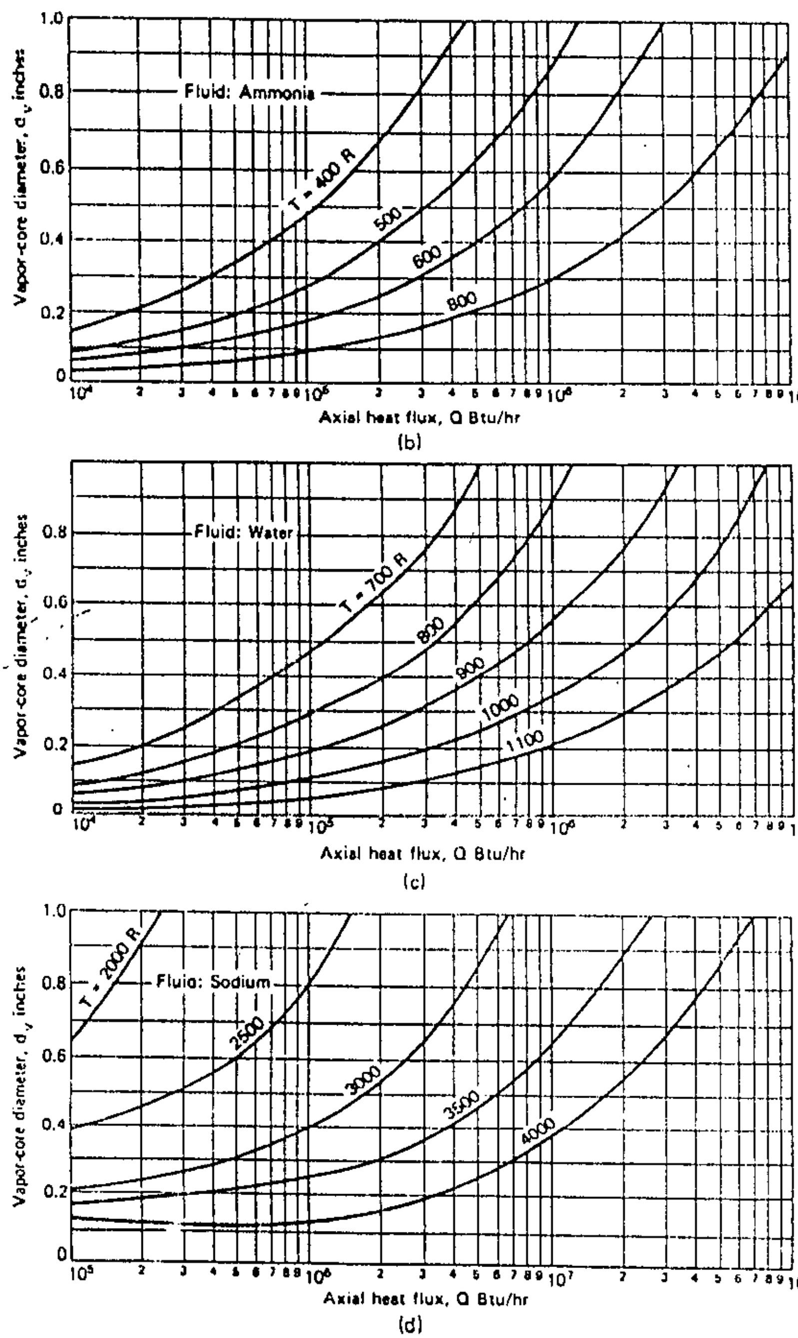


Fig. 3.8 Vapor core diameter vs. heat transfer rate for vapors Mach number of 0.2 (1in = 0.0254 m, 1Btu/hr = 0.2929 W, 1R = 0.5556 K) For Ammonia, water, and Sodium. [3]

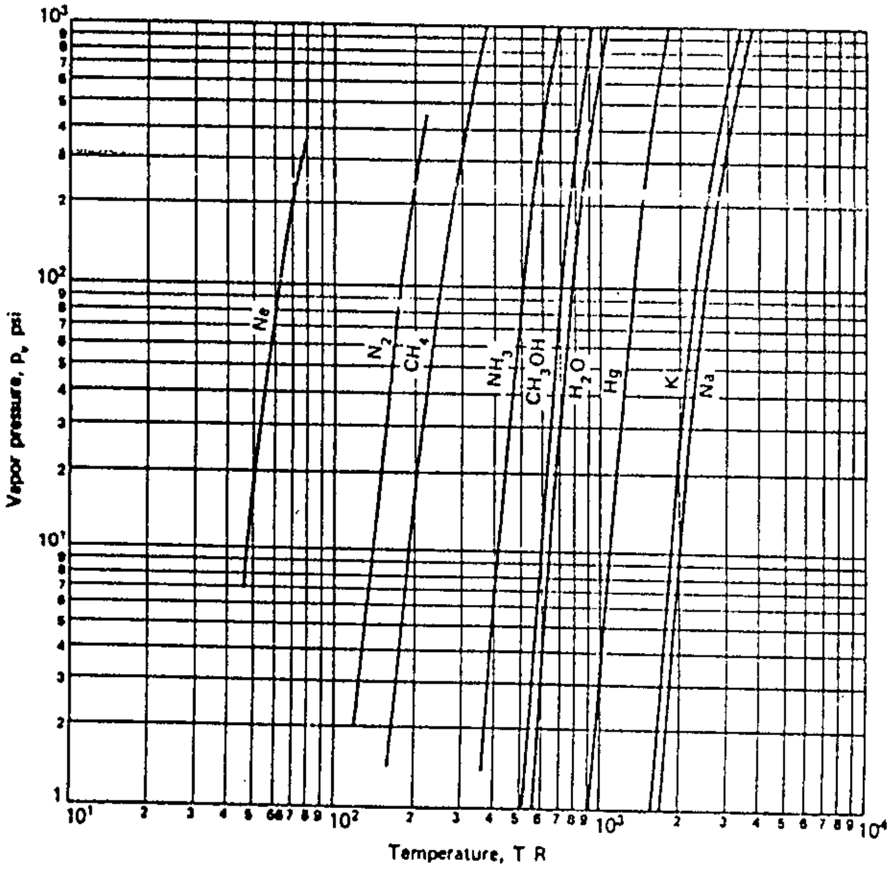


Fig. 3.9 Vapor pressure versus temperatures for several heat pipe working fluids (1 psi = 6.895×10^3 N/m, 1R = 0.5556K) [3]

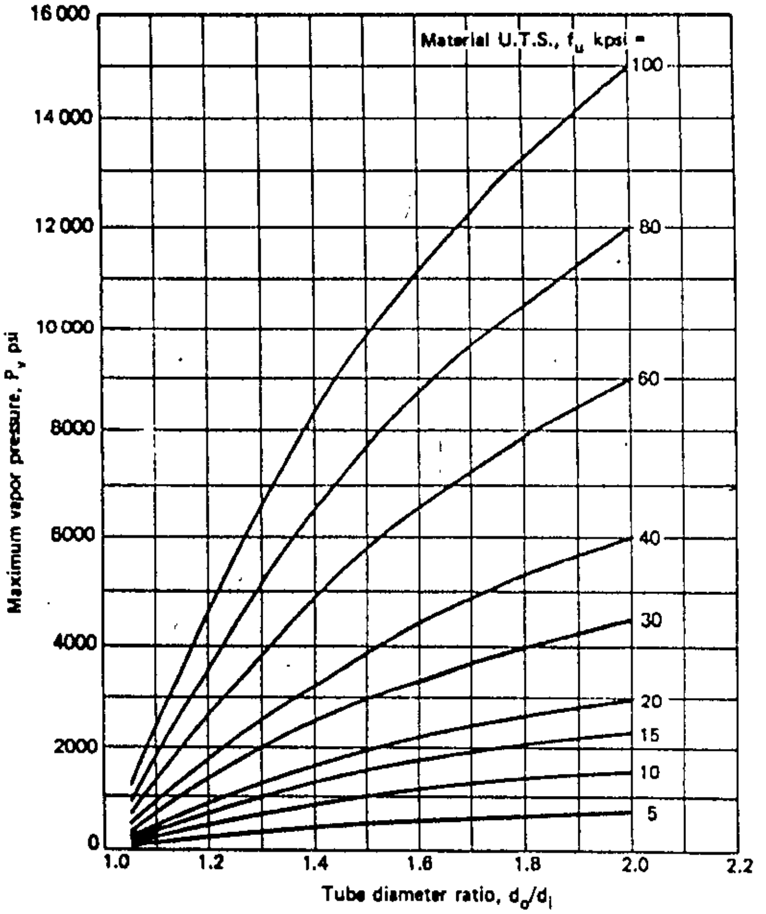


Fig. 3.10 Design chart for heat pipe container tubes [3] (1psi = 6.895×10^3 N/m^2, 1kpsi= 6.895×10^6 N/m^2).

CHAPTER 4

LITERATURE REVIEW

In recent years, the miniature heat pipe used for heat dissipation and homogeneous temperature of computer and numerous electronic instruments have displayed its remarkable effect. The thorough comprehension of the limited heat transfer capability of miniature heat pipe and factors of influence is indispensable for further development and improvement of its performance.

4.1 PREVIOUS WORK ON HEAT PIPE

Experiments on heat pipe first conducted by E. Schmidt a German engineer in 1939. He carried out experiment on heat pipe with a copper tube filled with ammonia or carbon dioxide as working fluid. He reported that a tube filled with ammonia or carbon dioxide near its critical point transfers an amount of heat per unit time that is more than 4000 times larger than the amount of heat transferred by a solid rod of copper with the same dimensions and at the same temperature difference between the hot and cold regions. The limitation of that experiment it can operate only in a vertical position.

In 1942 this limitation was eliminated through an improvement conceived by R. Gaugler at General Motors Corp. He incorporated a wick or porous matrix covering the inside wall of the tube into the design, because the wick contained many capillary passages. Capillary action transported the liquid through the wick from the cold to the hot end of the tube, regardless of the direction in which it was oriented. In the wick the working fluid continuously evaporates at its hot end.

Capillary forces continuously replace the evaporated liquid. The circulation of the fluid in the pipe occurs in such a way that a pressure difference in the vapour causes the vapour to flow from the warm to the cold end of the tube. Capillary forces return the liquid through the wick.
Since the device conceived by Gaugler did not become widely known, M Grover, at Los Alamos Scientific Laboratory, described it again in 1963. He gave it the name " heat pipe" suggesting that it be used to transport heat in power plants for space vehicles. [1]
The first commercial organization to work on heat pipes was RCA. Most of their early support came from US Government contracts. In 1964 to 1966 they made heat pipes using glass, copper, nickel, stainless steel, and molybdenum as wall materials. Working fluid included water, cesium, sodium, lithium and bismuth. Maximum operating temperature of 1650°C had been achieved.

During 1967 and 1968 several articles appeared in the scientific press, most originating in the United States, indicating a broadening of the area of application of the heat pipe to electronic cooling, air conditioning, engine cooling and others. These revealed developments such as flexible and flat plate heat pipes.

In 1968 heat pipe was first used in space for satellite thermal control on GEOS-B, launched from Vandenburg Air Force Base. Two heat pipes were used. The heat pipes were constructed using 6061 T-6 aluminum alloys, with 120- mesh aluminium as the wick material. Freon 11 was used as the working fluid. The purpose of heat pipes was to minimize the temperature differences between the various transponders in the satellite [2].

In 1969 NASA developed a new type of heat pipe (rotating heat pipe) in which the wick was omitted. The rotating heat pipe utilizes centrifugal acceleration to transfer liquid from the condenser to the evaporator, and can be used for cooling motor rotors and turbine blade rotors. GRAY also proposed an air-conditioning unit based on the rotating heat pipe.

Most of the work on heat pipes described so far has been associated with liquid metal working fluids and, for lower temperatures, water, acetone, alcohols etc. With the need for cooled detectors in satellite infrared scanning systems cryogenic heat pipes began to receive particular attention. The most common working fluid in these heat pipes was nitrogen, which was acceptable for temperature ranges between 77 and 100°K. Liquid oxygen was also used for this temperature range. The Rutherford High Energy Laboratory was the first organization in the United Kingdom to operate cryogenic heat pipes, liquid hydrogen units being developed for cooling targets at the RHEL. Later RHEL developed a helium heat pipe operating at 4.2 °K.

By 1970 a wide variety of heat pipes were commercially available from a number of companies in the United States. RCA, Thermo-Electron, and Noren Products were among several firms marketing a range of 'standard' heat pipes, with the ability to construct 'specials' for specific customer applications. During the next few years several manufacturers were established in the United Kingdom and a number of companies specializing in heat pipe heat recovery systems, based primarily on technology from the United States, have entered what is becoming an increasingly competitive market.

The early 1970's saw a considerable growth in the application of heat pipes to solve terrestrial heat transfer problems, in addition to the continuing momentum in their development for spacecraft thermal control. While much development work was concentrated on 'conventional' heat pipes, it was seen the increasing interest in the rotating heat pipe and in research into electro hydro-dynamics for liquid transport. The proposed use of 'inverse' thermal syphons and an emphasis on the advantages (and possible limitations) for gravity–assisted heat pipes have stood out as areas of considerable importance, and as such are given more space in this edition of 'Heat Pipes'. The reason for the growing interest in these topics is not difficult to find. Heat pipes in terrestrial applications have, probably in the majority of cases, proved particularly viable when gravity, in addition to capillary action has aided condensate return to the evaporator. This is seen best of all in heat pipe heat recovery units, where slight changes in the inclination of all in heat pipes used in such heat exchanges can, as soon as reliance on the wick alone effected, can cut off heat transport completely.

Studies on the application of miniature heat pipes having the diameter of 3 or 4 mm for cooling of the notebook PC CPU have been actively conducted by the American and Japanese enterprises specializing in heat pipes recently [14,15]. Kwang Soo Kim, *et al.* [16] also performed an experimental investigation on cooling characteristics of miniature heat pipes having the same dimension [14,15] with woven-wired wick and water as a working fluid. The experimental parameters of their experiment were inclination, structure of the wick and length of the condenser. They claimed that the thermal resistance of a miniature heat pipe of diameter 4 mm is less than that of diameter 3 mm..

Wel and Yuan [17] experimentally investigated the heat transfer characteristic of a closed two-phase thermosyphon working with a binary mixture at different inclination angles. They reported that the temperature of the evaporator section is higher when the thermosyphon is placed vertically. They also reported that the heat transfer rate increases when the thermosyphon is placed at a certain angle and the heat transfer coefficient increases as the inclination angle increases.

Most of the heat pipes used in computer industries is copper-water heat pipe because water is a safe fluid to the environment and also it is the best heat transport fluid in the medium operating temperature range of 50°C to 150°C for computers. MURAKAMI *et al.* [18, 19] experimentally investigated on "statistical prediction of long-term reliability of copper-water heat pipes" from accelerated test data. They provided life test data and life estimation for phosphorous deoxidized copper-water heat pipe and oxygen free copper-water heat pipe. They tested heat pipes, which were of grooved type, diameter of 6.35 mm and length of 200 mm. They found that the phosphorous deoxidized copper-water heat pipe generated appreciable amount of non-condensable gas, while the oxygen-free copper-water heat pipe generated little of non-condensable gas. The non-condensable gas was mainly composed of CO_2, limited presence of O_2 and N_2. They predicted that the oxygen-free copper-water heat pipe would degrade about 5°C in 20 years under the operating temperature of 60° C.

CHAPTER 5

EXPERIMENTAL METHODS

5.1 EXPERIMENTAL APPARATUS

The experimental apparatus are as follows:

a. Miniature heat pipe
b. Heat sink
c. Duct
d. Cooling fan
e. Anemometer
f. Thermocouple and Digital thermometer
g. Ni-Cr thermic wire
h. Power supply unit.
i. Test stand

a. Miniature Heat Pipe

A stainless tube having the outer diameter of 5 mm and 6 mm and length of 150 mm is used as a heat pipe for this experiment. The heat pipe consists of three sections as shown in Fig. 5.1

i. Evaporator section.
ii. Adiabatic section.
iii. Condenser section.

Evaporator section:

Evaporator section is located at the bottom part of the heat pipe except horizontal orientation ($\theta = 0$). The length of the evaporator section is 50 mm.

Adiabatic section:

Adiabatic section is located in between the evaporator section and condenser section. The length of the adiabatic section is 50 mm. Adiabatic section is thermally insulated. Adiabatic section is actually kept with heat pipe to distinguish evaporator section and condenser section.

Condenser section:

Condenser section is the uppermost part of the heat pipe except horizontal orientation. The length of the condenser section is also 50 mm.

b. Heat Sink

A heat sink having length of 50 mm, width of 50 mm and height of 50 mm is made from aluminium casting. Six fins having length of 50 mm, width of 40 mm and thickness of 5 mm is made by machining process. Three small holes are made at the top surface of the heat sink so that thermocouple can get contact with the wall surface of the condenser section and wall surface temperature of the condenser section can be measured. A 5 mm hole is made at the top center above the fins along the length of the heat sink. Details of the heat sink are shown in Fig.5.2a. The heat sink is installed at the condenser section for its cooling purpose.

c. Duct

A duct having length of 100 mm, width of 52 mm and height of 52 mm is made of an acrylic material. Three small holes are made at the top surface of the duct so that the thermocouple can be inserted as shown in Fig.5.2b. Duct is used around the heat sink to enhance airflow.

d. Cooling Fan

A 12-volt DC cooling fan (0.9 Amp, 1.08W, CE Co. Made in Taiwan) is used for this experiment. The dimension of the fan is 50 mm ×50 mm ×10 mm as shown in Fig.5.2c.

e. Anemometer

Anemometer (CE. Made in USA) is used to measure fan speed.

f. Thermocouple and Digital Thermometer

Nine thermocouples of K type and a digital thermometer are used for measuring wall temperature at the various section of the heat pipe.

g. Ni-Cr Thermic Wire

Ni-Cr thermic wires having width of 1.5 mm, thickness of 0.1 mm and resistance of 12.4 Ω/m are wound around the evaporator wall at a constant interval of 1.5 mm.

h. Power Supply Unit

A DC power supply unit (Top ward DC power supply, Model 3303 D) having voltage range 0 ~ 30 V and current range 0~5 Amp is used.

i. Test Stand

Test stand for experimental set up is used to kept and set the miniature heat pipe at various inclination angles. Details of the test stand are shown in Fig. 5.3. Alignment plate is made of mild steel sheet having dimension of 120 mm length, 60 mm width and 6 mm thickness. The alignment plate is connected to the stand with an adjusting knob. The alignment plate can be set any inclination angle. The heat pipe is placed on the alignment plate.

The inclination angle of the heat pipe is measured from the angle disk, which is attached to the stand. A base plate is used to support the stand. The adjusting knob is used to fix the alignment plate at any inclination angle.

5.2 EXPERIMENTAL PROCEDURE

The schematic diagram of the experimental set up is shown in Fig 5.4. Stainless steel tubes having diameters of 5 mm and 6 mm and length of 150 mm were used as heat pipe for this experiment. Both ends of the heat pipe are closed by welding and 75 percent inside volume of the tube filled with water and other end also closed by welding. The condenser section of the heat pipe was inserted into the heat sink. A duct was used around the heat sink to enhance airflow. Nine thermocouples of K type were attached at the wall surface of the heat pipe by heatproof tape. Four units of thermocouple were attached at the evaporator wall, two units were at adiabatic section and three units were at condenser section. Mica sheet is wound around the evaporator section for electrical insulation. Ni-Cr thermic wires are wound around the evaporator wall at a constant interval of 1.5 mm. The evaporator section and adiabatic section was thermally insulated with glass wool to minimize the heat loss from the wall. To perform forced cooling a fan was installed in front of the heat sink inside the duct.The speed of the axial flow fan can be changed and an anemometer was used to measure the speed. After that the heat pipe was placed on the alignment plate of the test stand at a particular orientation. The alignment plate along with heat pipe can be set at any inclination angle. The heat is supplied to the evaporator section electrically through DC power supply starting from 0.5 W with an increase of 0.5 W step wise reaching up to 4.0 W finally. The temperature of the heat pipe wall was recorded in each step by using thermocouple when the temperature of the wall reached at the steady state. It took about one hour to reach the steady state. The performance test for understanding the effects of the inclination as well as the change of airflow rate by a cooling fan at the condenser, experiments were performed with increasing the heat input stepwise by 0.5 W from 0.5 W in the same method with the inclination and cooling air flow rate of each fixed. Experimental parameters and their ranges for the performance test of the miniature heat pipe are shown in table below:

Table 5.1 Experimental parameters and their ranges

Parameters	Condition
Diameter of pipe	5 and 6 mm
Length of pipe	150 mm
Length of evaporator	50 mm
Length of condenser	50 mm
Kinds of working fluid	Water
Inclination angle	0°, 5°, 10 °, 15 °
Charging ratio	75%.

5.3 MATHEMATICAL EQUATION

The mathematical equation that are used to calculate the thermal resistance and the overall heat transfer coefficient are given below:

$$R = (T_e - T_c) / Q \quad (4.1)$$

Where T_e is the wall temperature of the evaporator section of heat pipe, T_c is the wall temperature of the condenser section, and Q is the thermal load (W) imposed on the evaporator.

The overall heat transfer coefficient, U_t is obtained from the following equation:

$$U_t = Q / A_e (T_e - T_c) \quad (4.2)$$

Where A_e is the surface area of the evaporator of the heat pipe

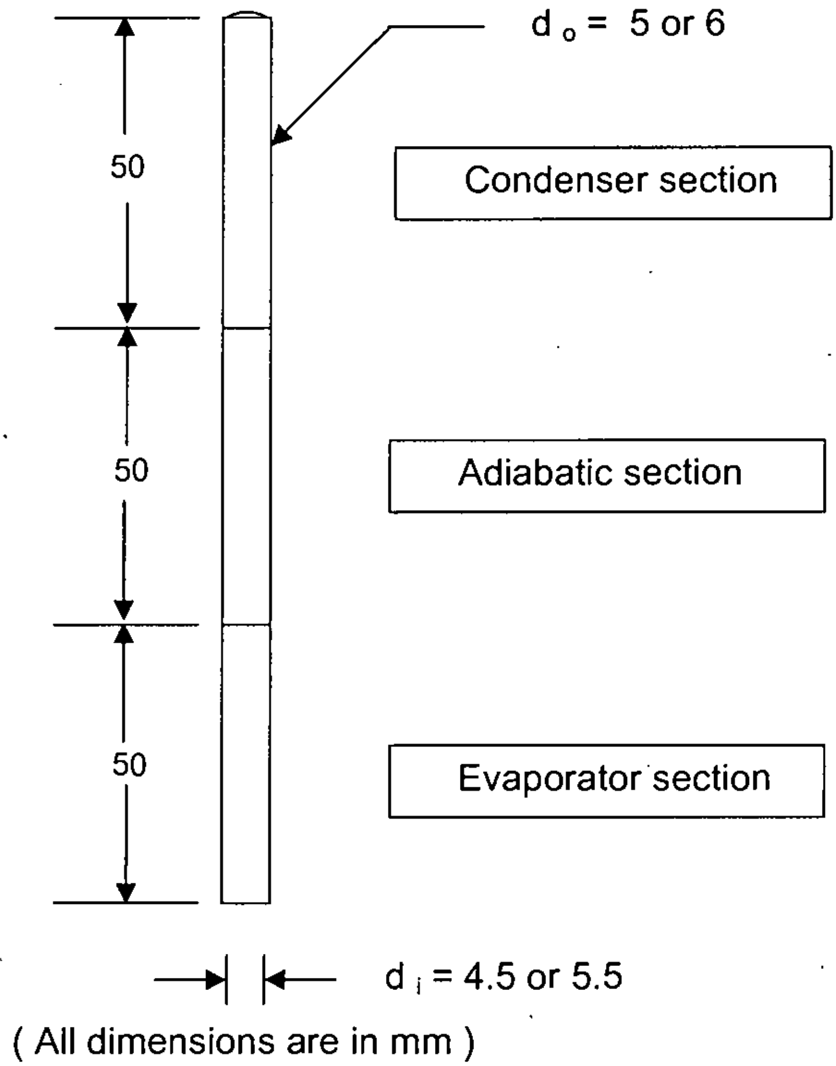
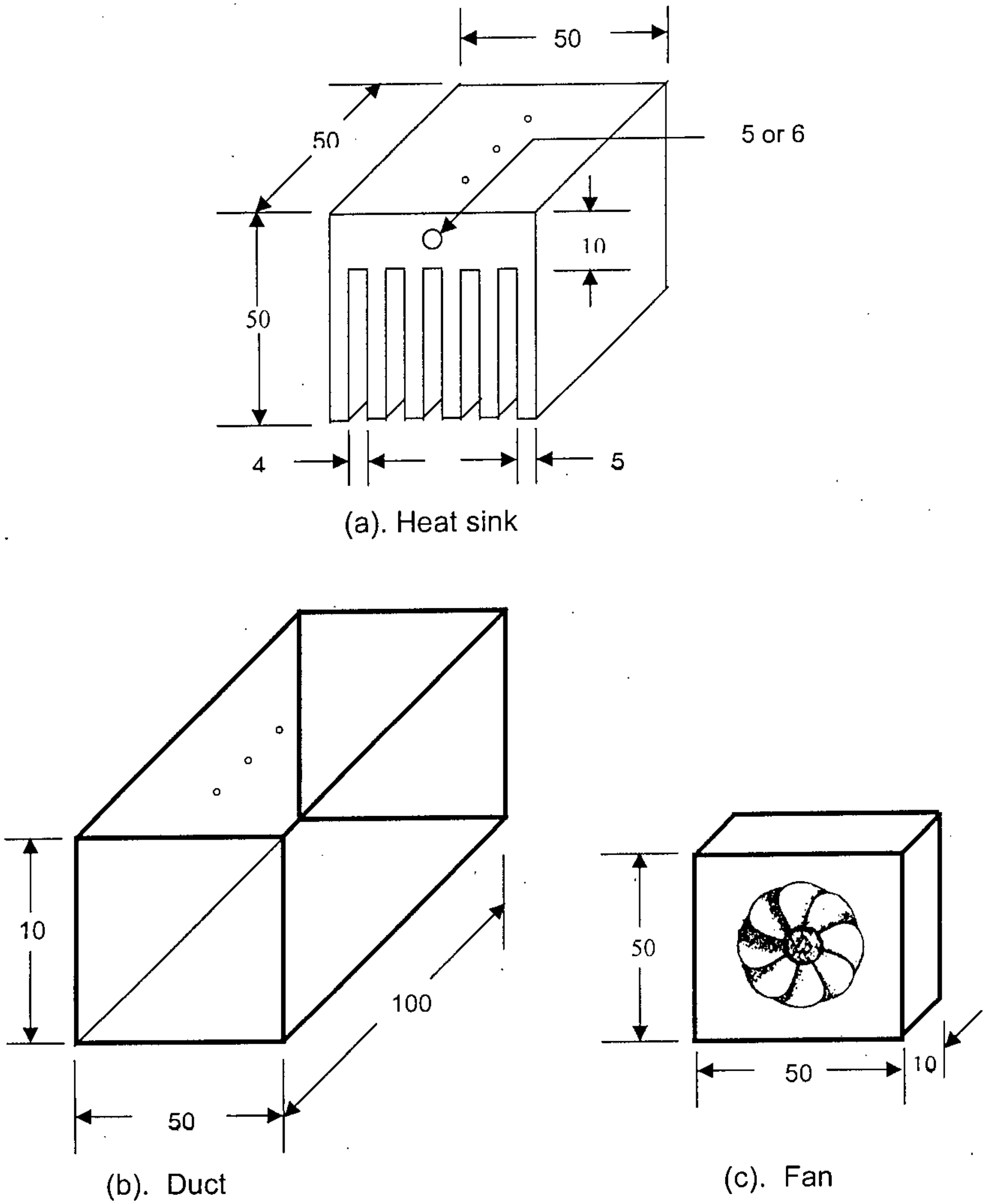


Fig. 5.1 Different sections of the miniature Heat pipe



(All dimensions are in mm)

Fig. 5.2 Details of the condenser section

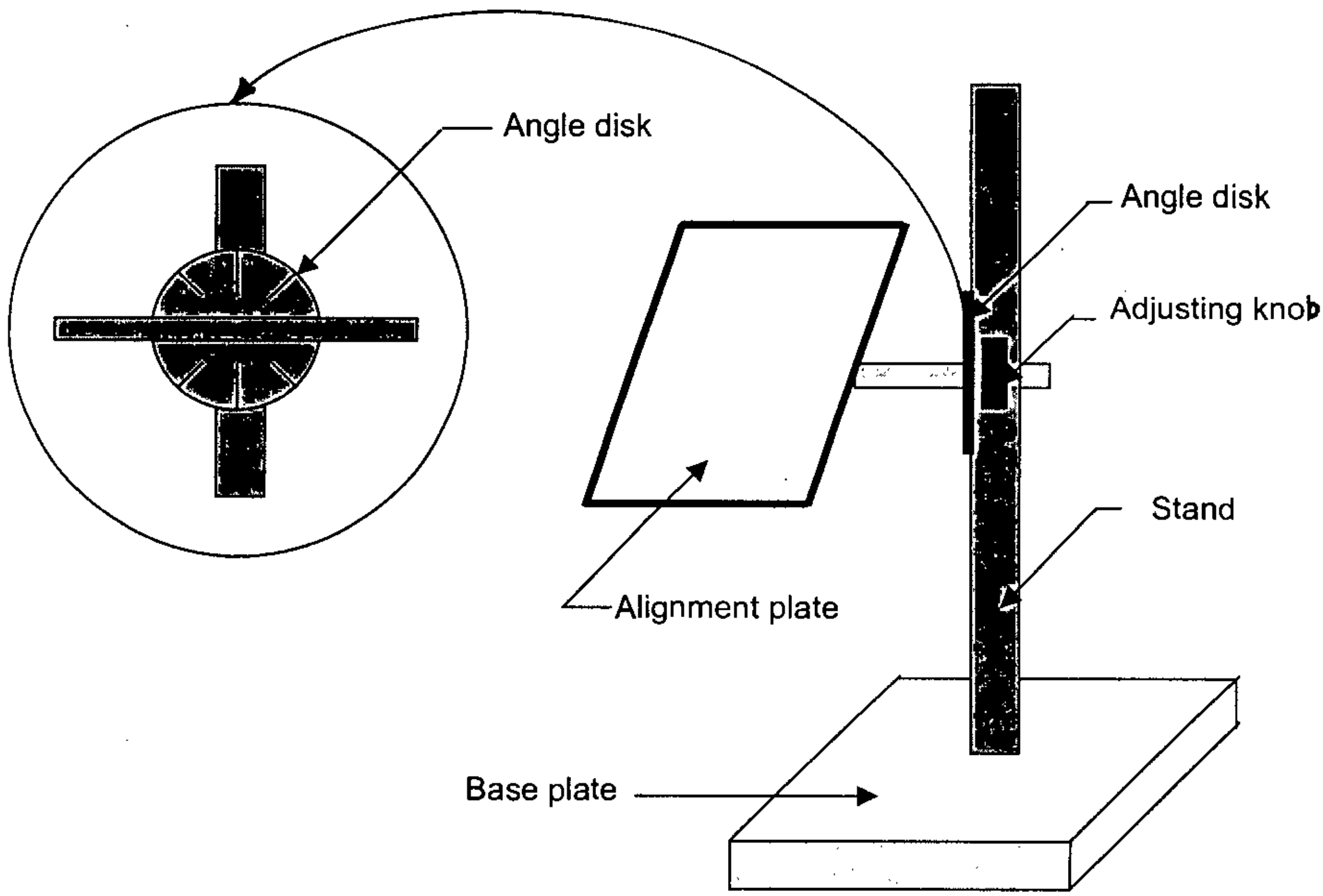


Fig. 5.3 Test stand of the experimental setup

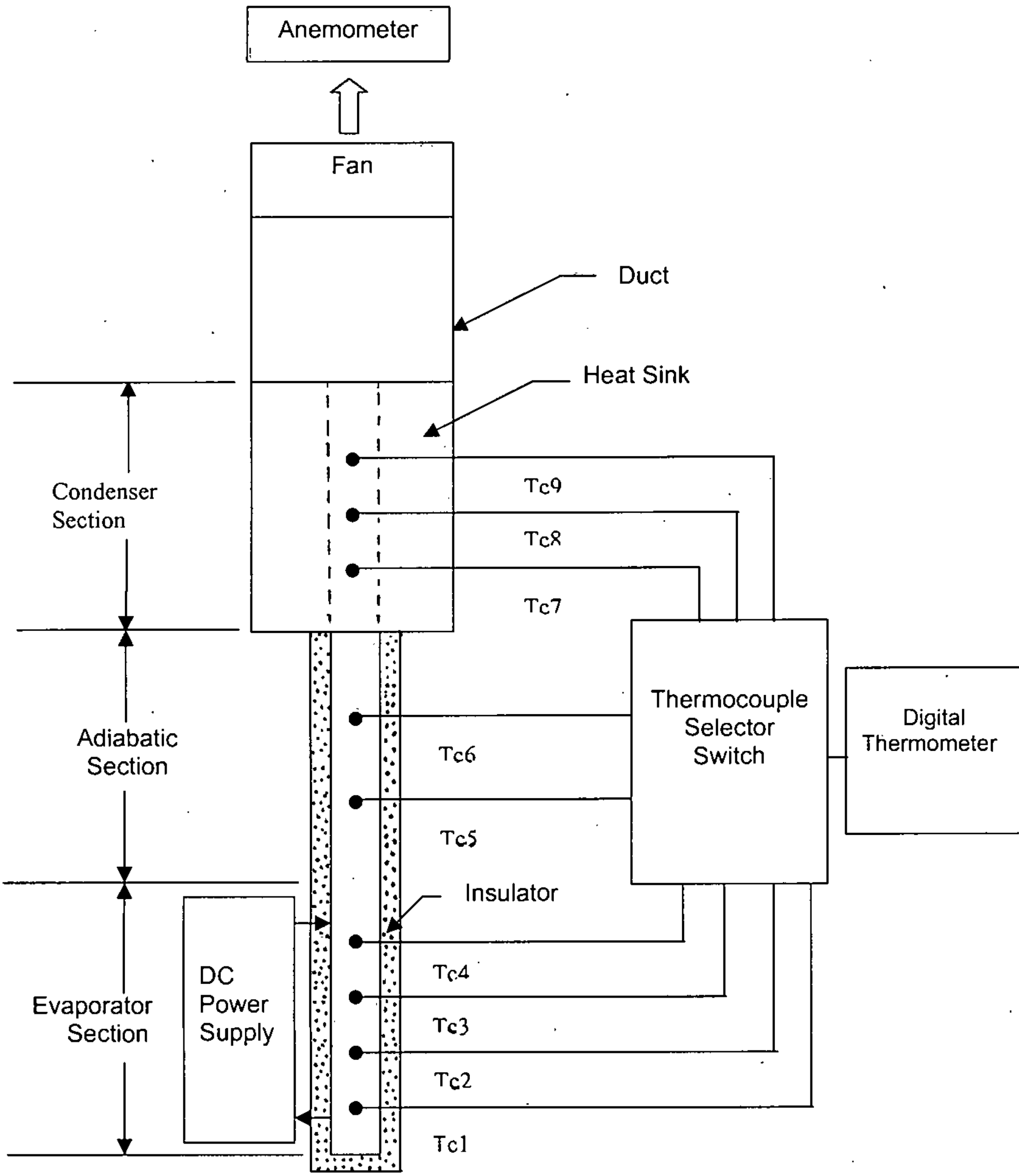


Fig. 5.4 Schematic diagram of experimental setup

CHAPTER 6

6. RESULTS AND DISCUSSION

The heat transfer characteristics of a miniature heat pipe working with water at various power inputs, at different incline angles and at different fan speeds were experimentally investigated. The results and discussion are summarised as follows:

Figures 6.1(a) to 6.1(m) and Figs. 6.2(a) to 6.2(m) show the axial wall temperature distribution of two miniature heat pipes having diameter of 5 mm and 6 mm respectively, at various heat inputs, incline angles and fan speeds. The figures indicate that at a particular heat input the temperature of evaporator section is higher when the heat pipe is placed horizontally and the temperature of the evaporator section decreases as the inclination angle of the heat pipe increases for both tubes. This result is consistent with the finding of Wel and Yuan [17]. At the same heat input, fan speed and inclination angle, the wall temperature of the evaporator section in the 6 mm diameter tube is 5°C to 15°C lower than those for 5 mm diameter tube within the present experimental range.

Figures 6.3(a) to 6.3(d) and Figs. 6.3(e) to 6.3(h) show the effect of inclination angle on thermal resistance of miniature heat pipe having diameter of 5 mm and 6 mm respectively at various power inputs and fan speeds. From the figures it is clear that the thermal resistance decreases with increase in inclination angle for both tubes at all heat input. This implies that the action of gravity, which serves to speed up the flow of liquid from condenser to evaporator, increases with increase in inclination angle.

Figures 6.4(a) to 6.4(h) show the effect of velocity of cooling air in the condenser on thermal resistance within the stable operational zone where no dry out occurs. It is seen in the figures that the thermal resistance is reduced as the condenser heat transfer rate is increased according to the rate of increase in the amount of cooling air flowed in. It has been observed that thermal resistance of a miniature heat pipe having diameter of 6 mm is less compared to that having diameter of 5 mm. It is also shown that the thermal resistance is greatly reduced in the section converting from natural convection cooling to forced convection cooling, but the thermal resistance according to the flow rate of air is not greatly changed in the section of forced convection cooling of 1.5 m/s or greater. This result is consistent with the finding of kwang soo kim *et al* [16]. Therefore, it is necessary to establish the optimum range of the flow rate of cooling air for miniature heat pipes to be applied in view of the problems of acoustic noise and vibration may be caused if the speed of a fan is increased in order to increase the air velocity. In the meantime, it is shown that the thermal resistance is reduced as the thermal load is increased. Because with the increase of thermal load, heat transfer rate is increased due to the increase of vapour density.

Figures 6.5(a) to 6.5(h) and Figs. 6.5(i) to 6.5(p) show the effect of diameter of a miniature heat pipe on overall heat transfer coefficient at various power inputs, fan speeds and inclination angles. From figures it is seen that as the diameter of the heat pipe increases, the total heat transfer coefficient also increases. It is also seen that the heat transfer coefficient increases as the inclination angle increases. This result is also consistent with the finding of Wel and Yuan [17].

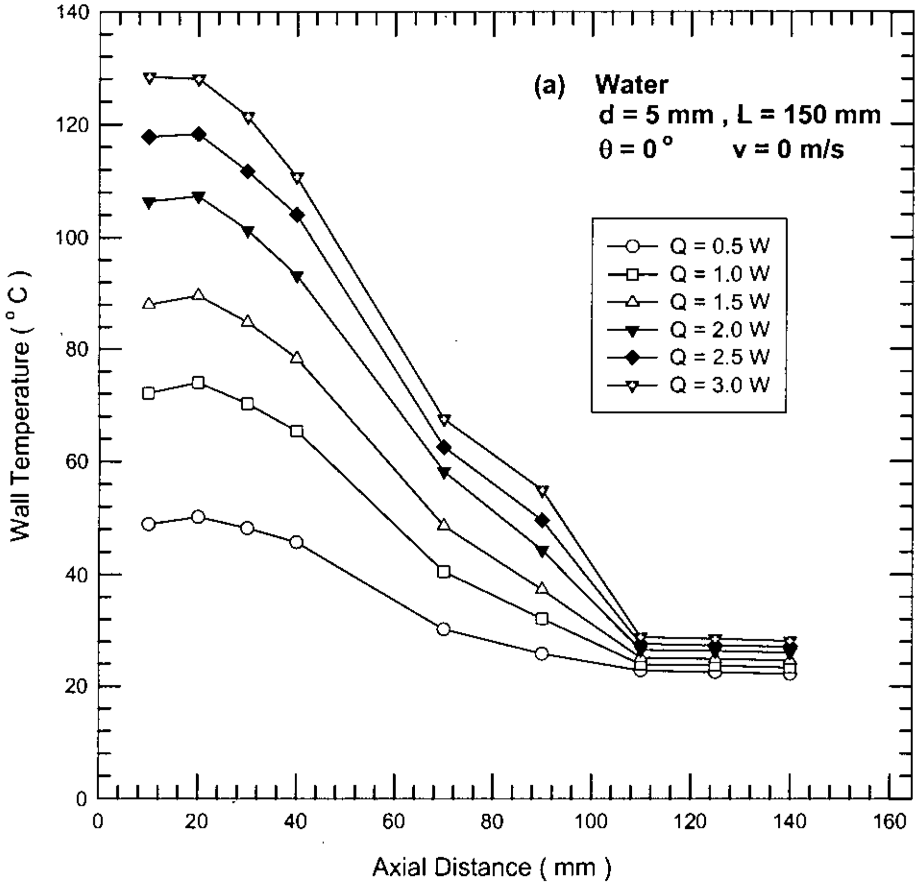


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm]

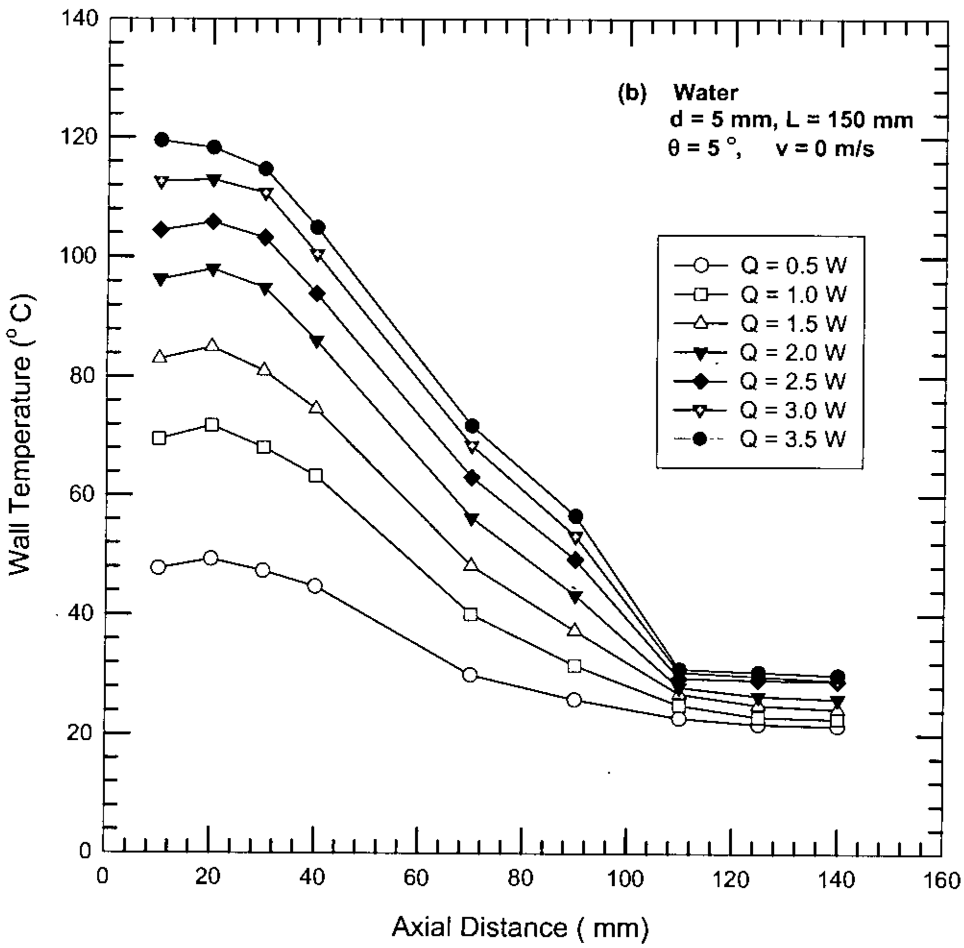


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

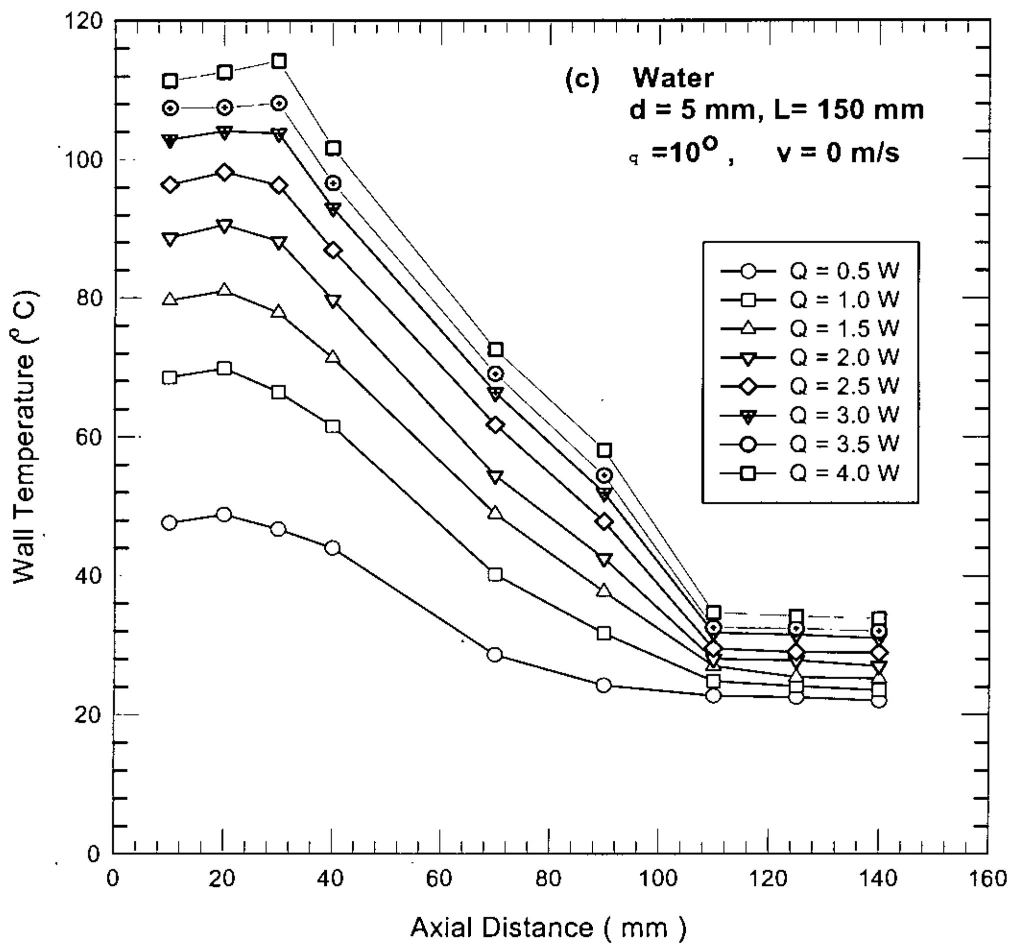


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

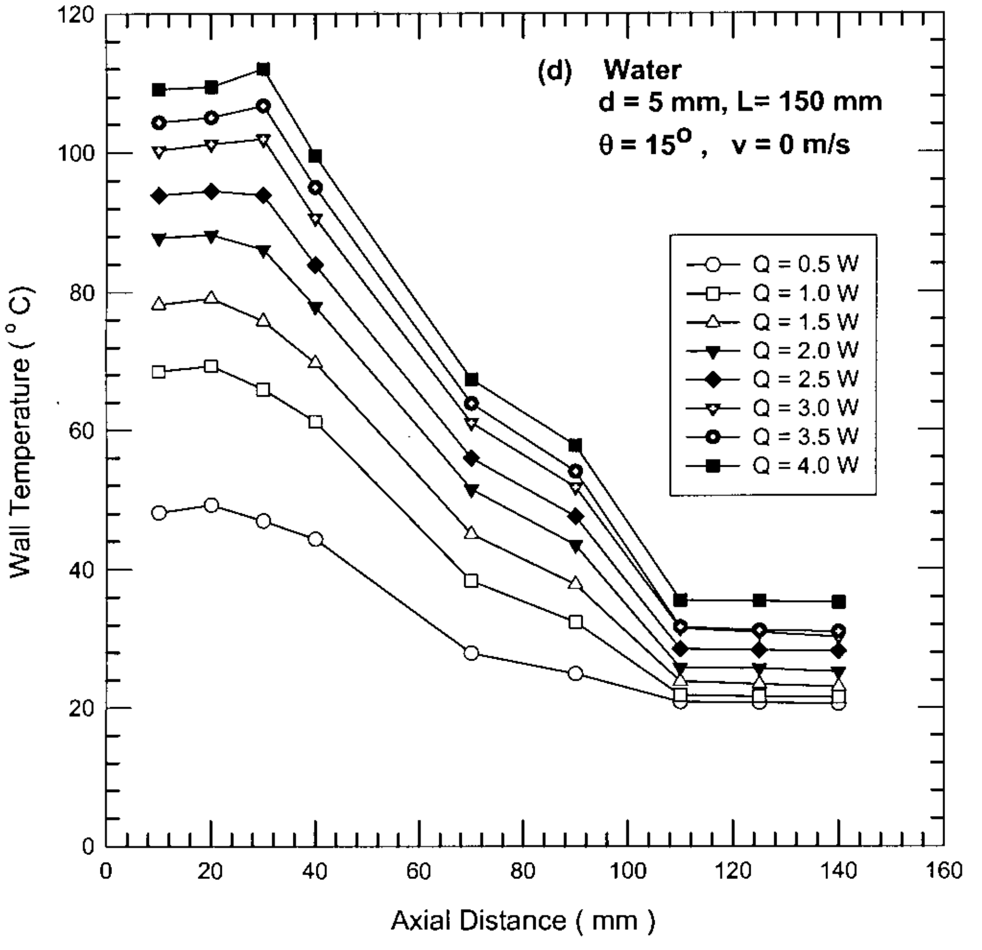


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

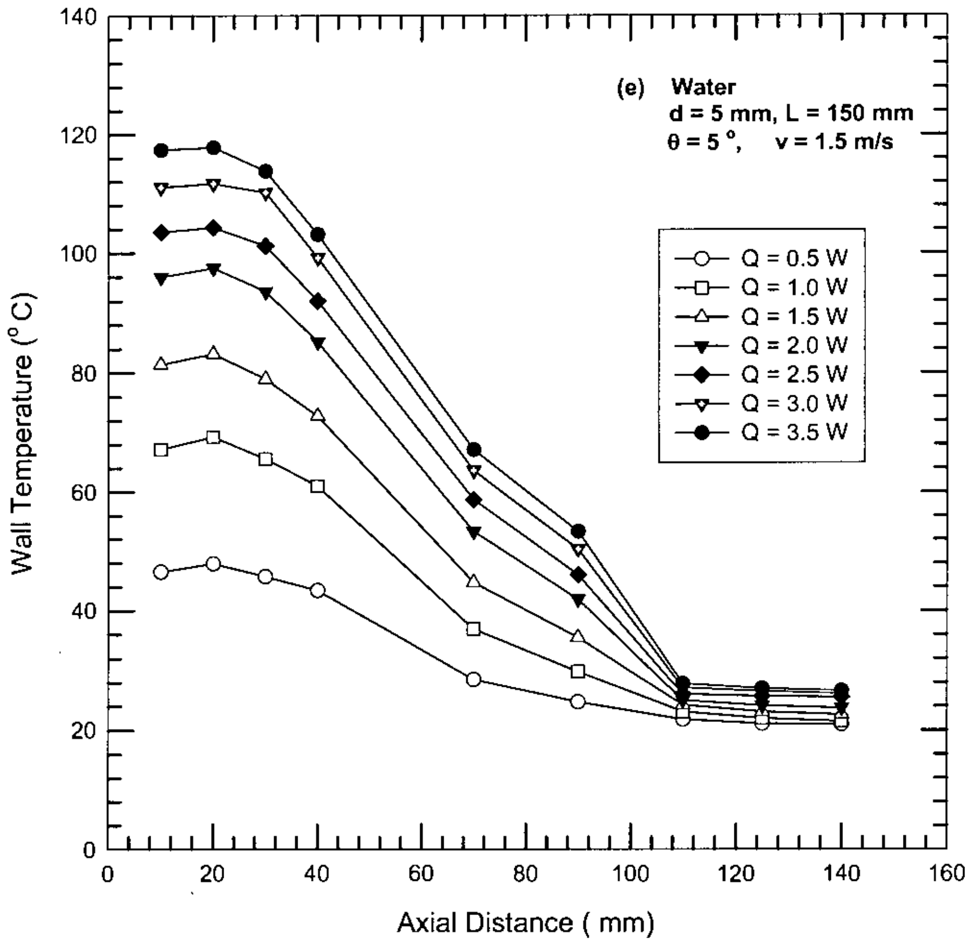


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

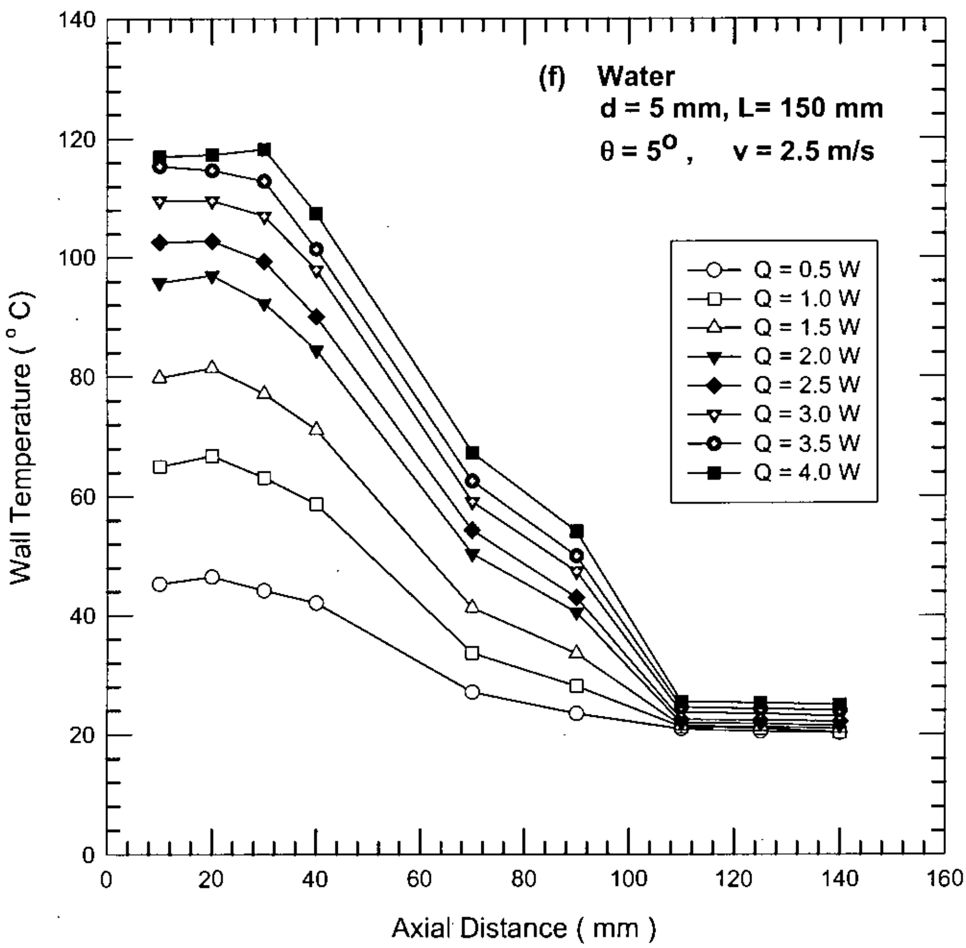


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

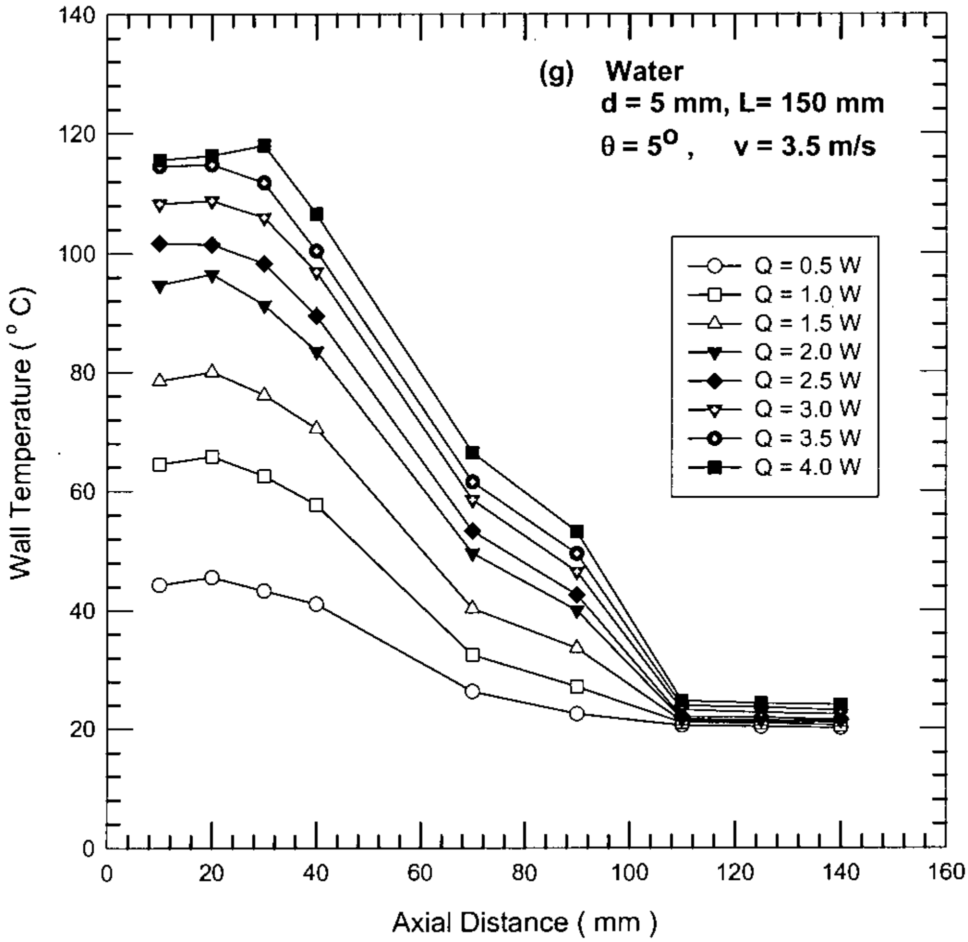


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

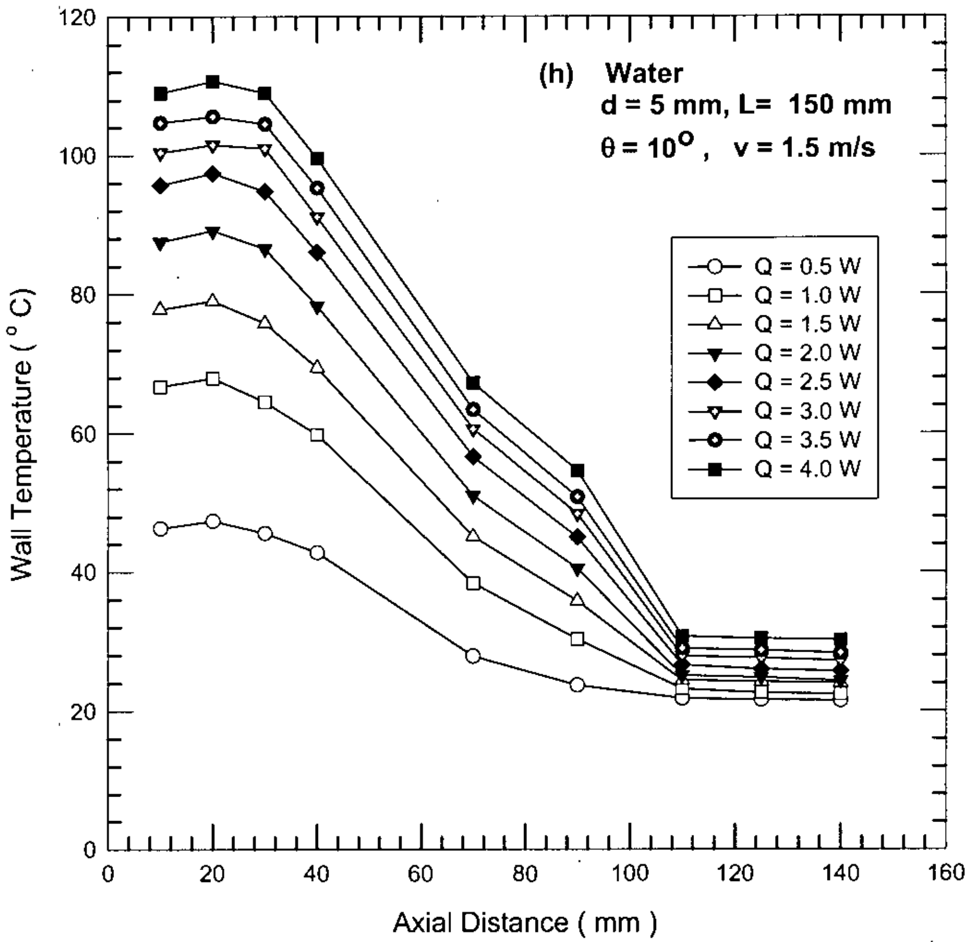


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

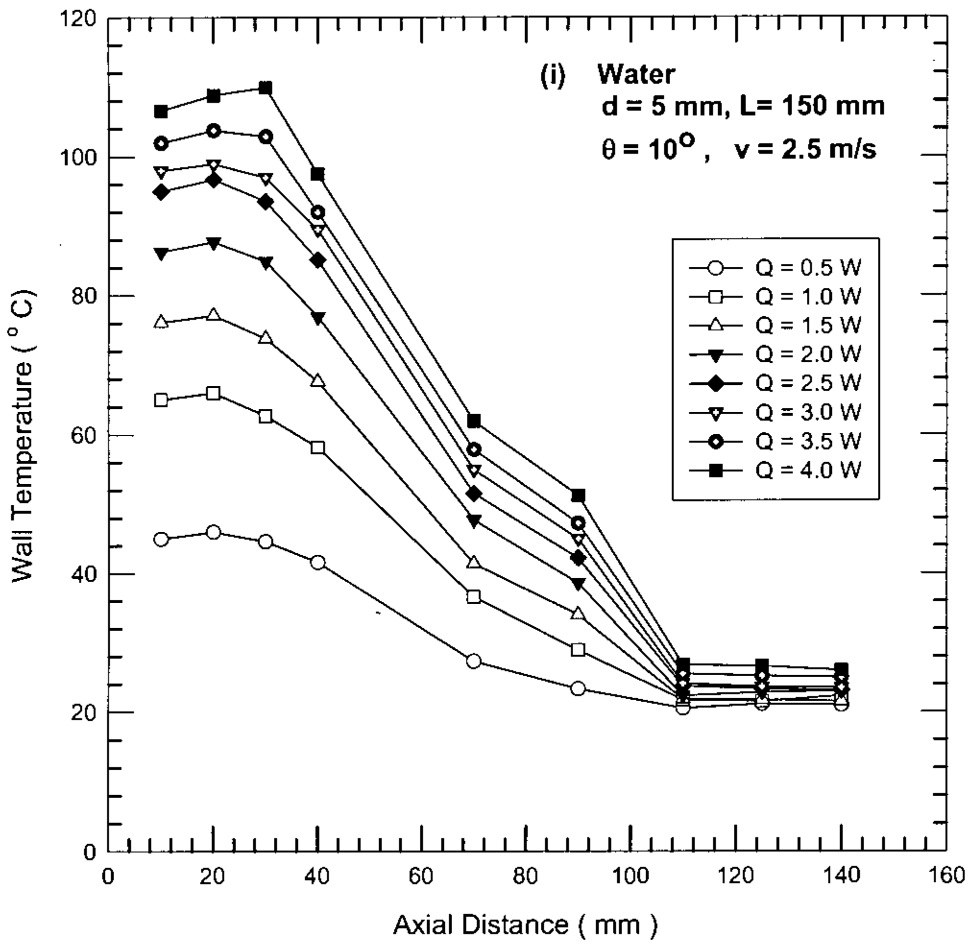


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

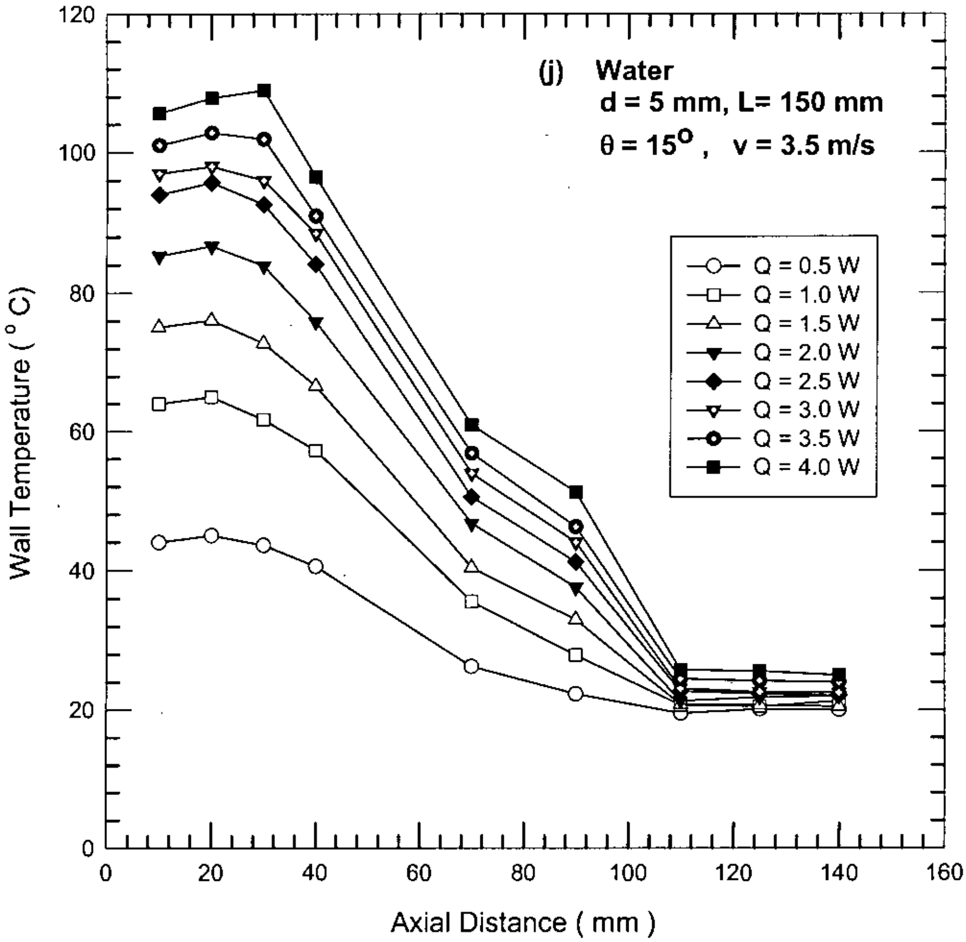


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

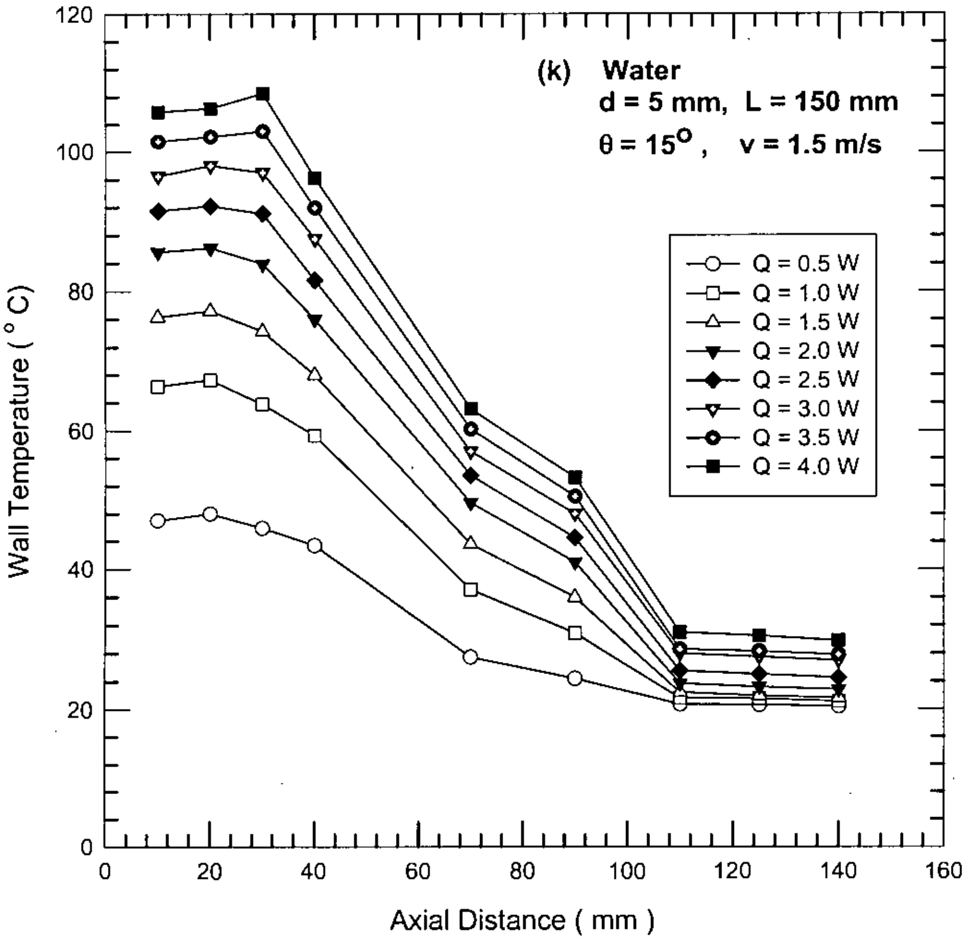


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

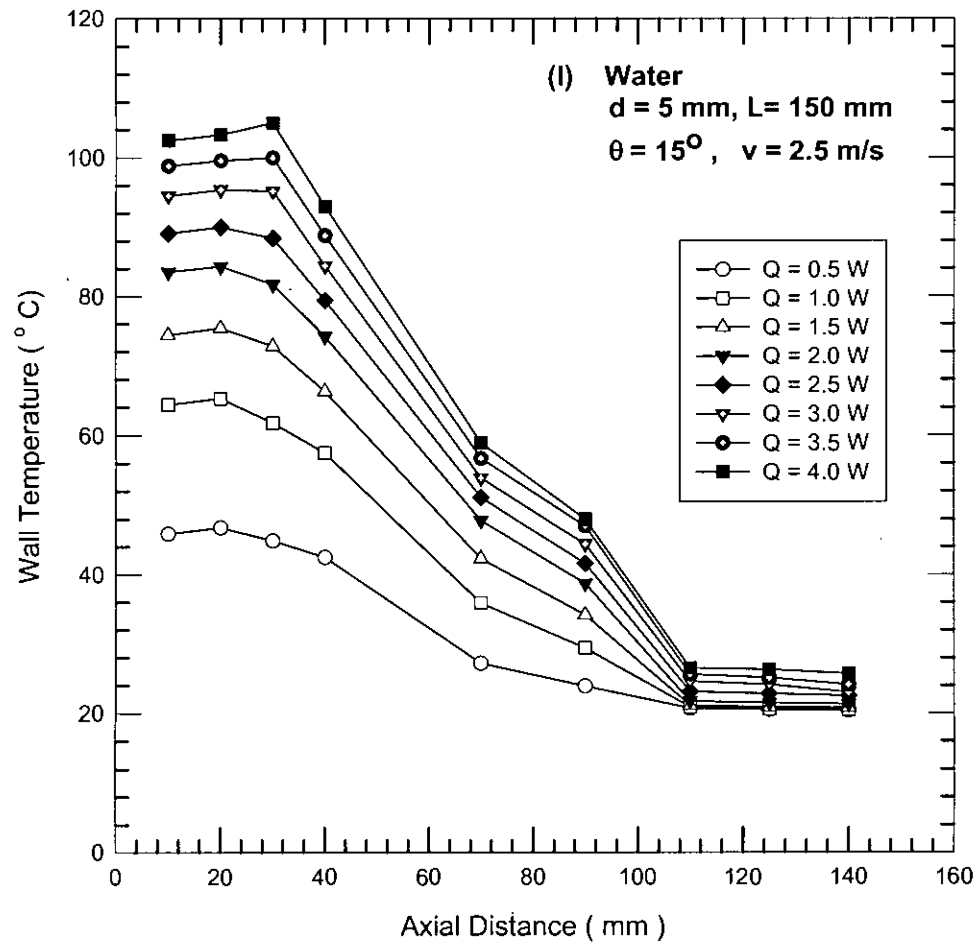


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

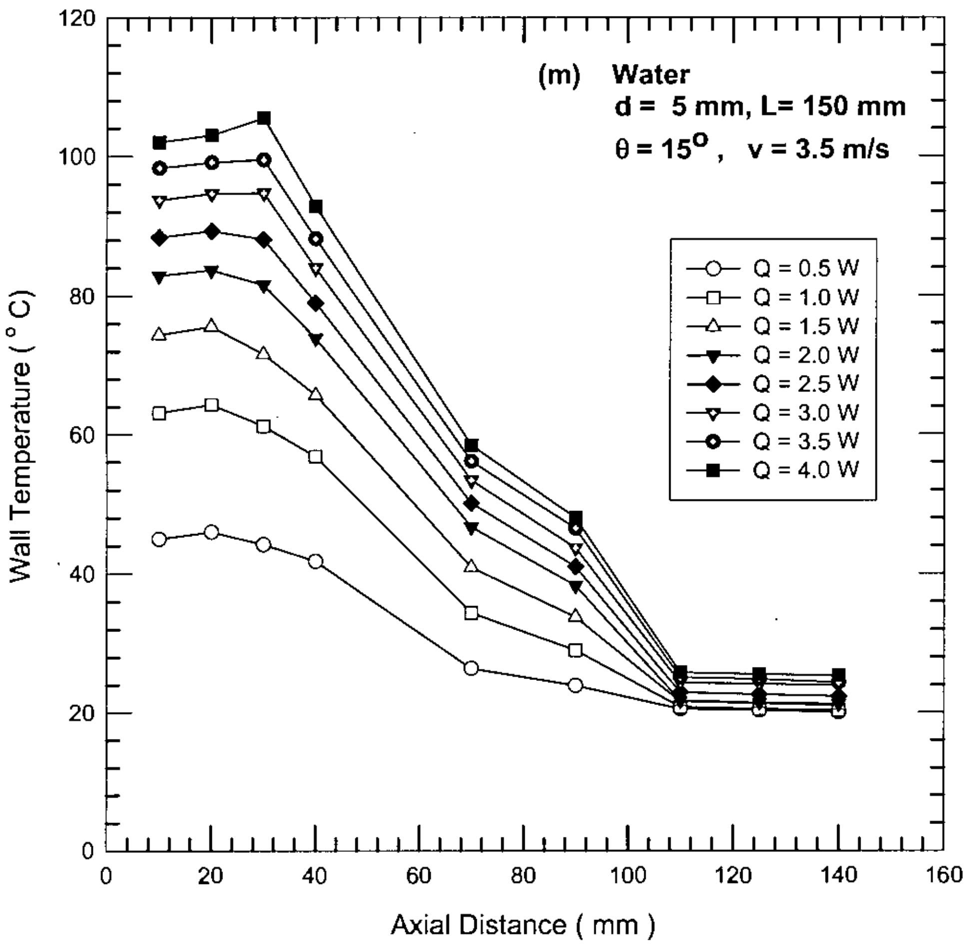
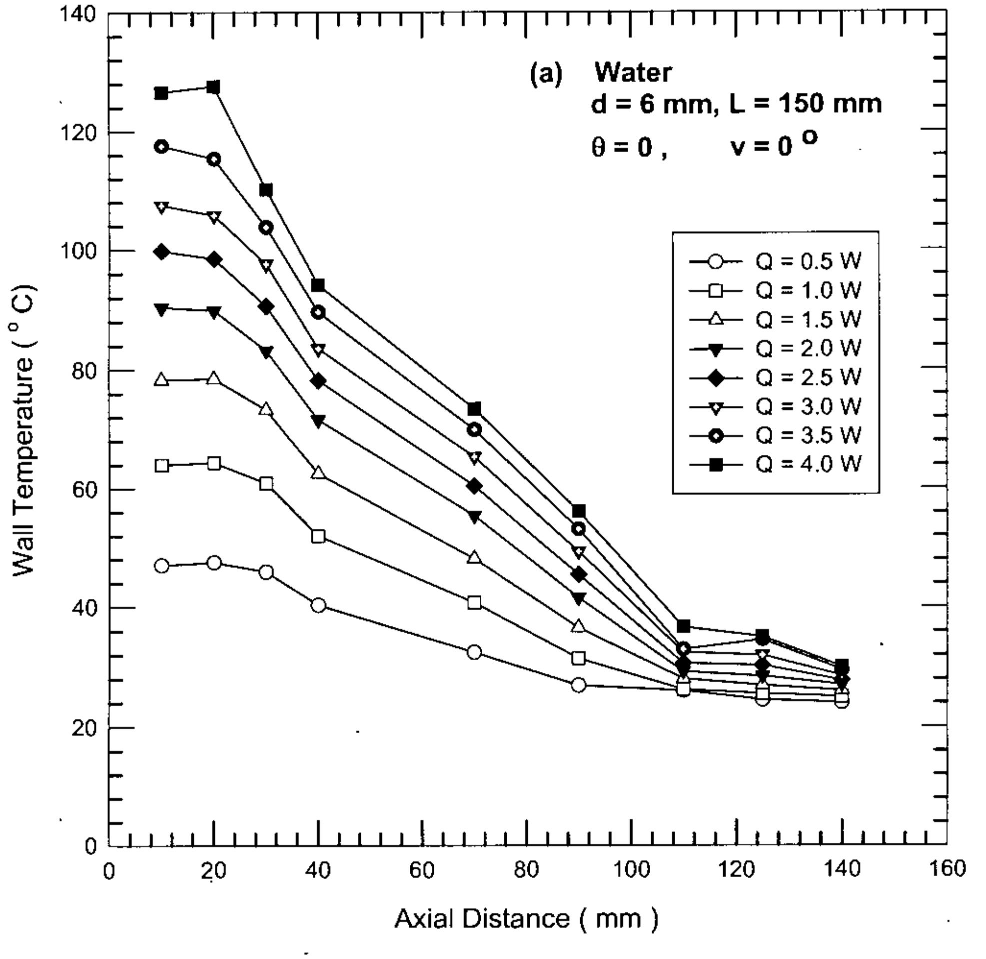
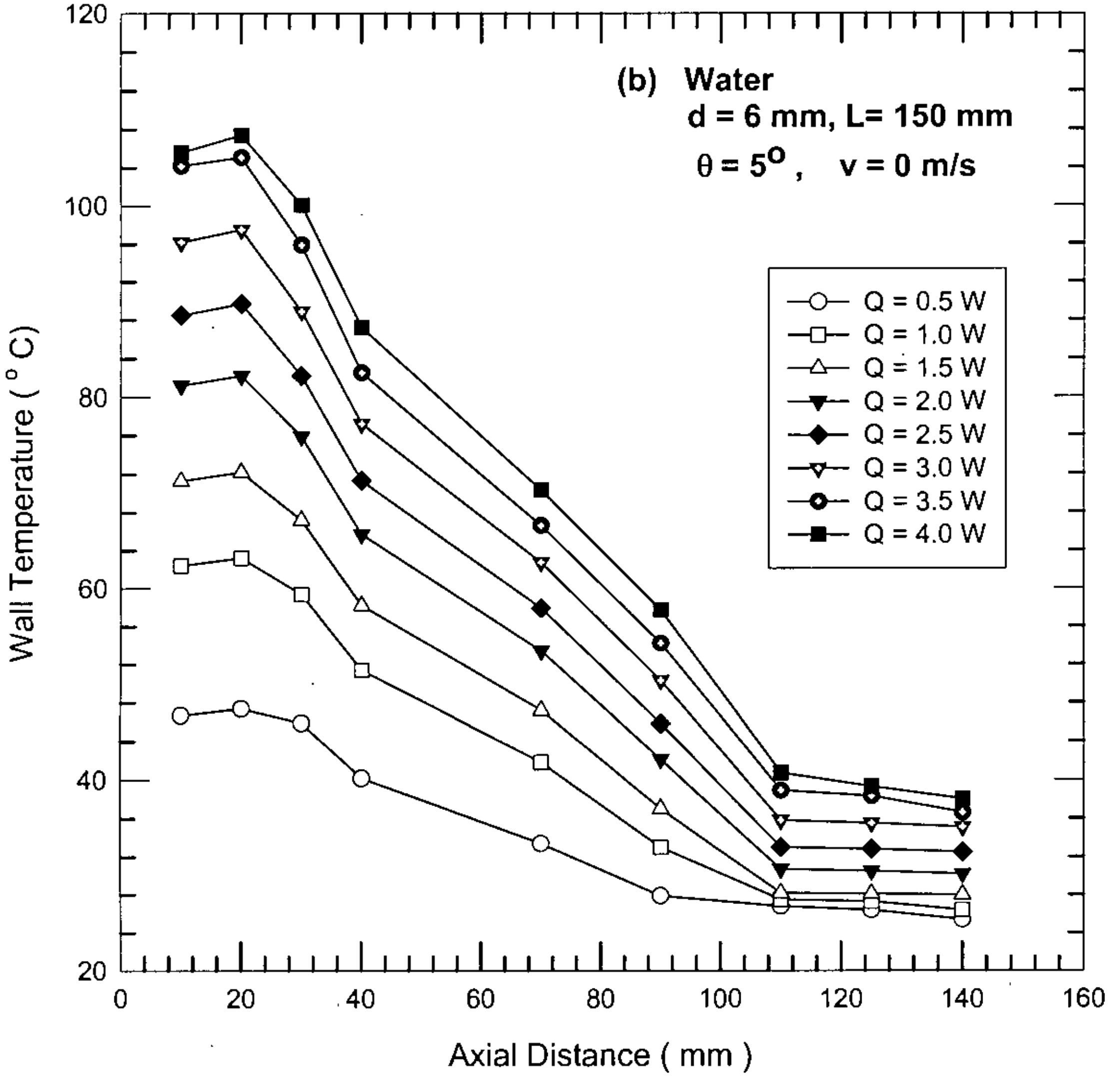
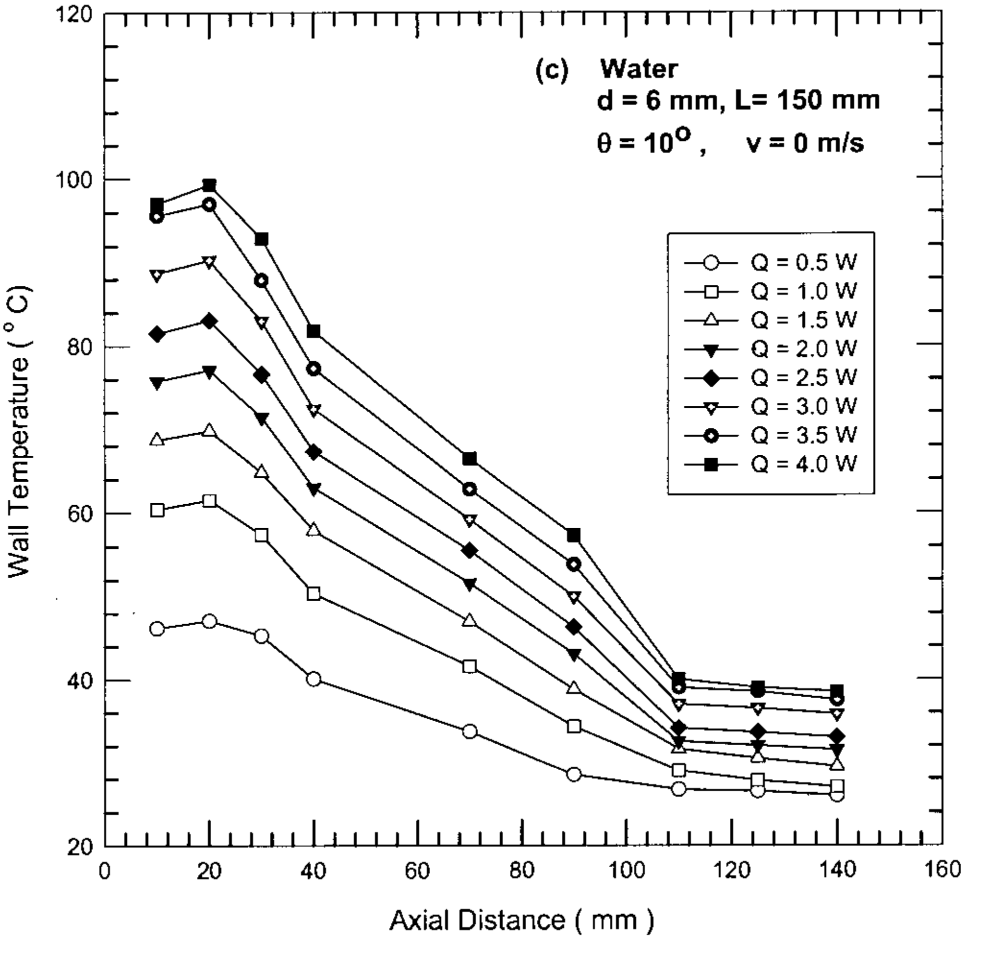


Fig. 6.1 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)


(a) Water
d = 6 mm, L = 150 mm
θ = 0 , v = 0 o
Q = 0.5 W
Q = 1.0 W
Q = 1.5 W
Q = 2.0 W
Q = 2.5 W
Q = 3.0 W
Q = 3.5 W
Q = 4.0 W
Wall Temperature (° C)
Axial Distance (mm)
0
20
40
60
80
100
120
140
160



(b) Water
d = 6 mm, L= 150 mm
θ = 5°, v = 0 m/s
Q = 0.5 W
Q = 1.0 W
Q = 1.5 W
Q = 2.0 W
Q = 2.5 W
Q = 3.0 W
Q = 3.5 W
Q = 4.0 W
Wall Temperature (° C)
Axial Distance (mm)
120
100
80
60
40
20
0
20
40
60
80
100
120
140
160


(c) Water
d = 6 mm, L= 150 mm
$\theta = 10^{o}$, v = 0 m/s
Q = 0.5 W
Q = 1.0 W
Q = 1.5 W
Q = 2.0 W
Q = 2.5 W
Q = 3.0 W
Q = 3.5 W
Q = 4.0 W
Wall Temperature (° C)
Axial Distance (mm)
0
20
40
60
80
100
120
140
160
120
100
80
60
40
20

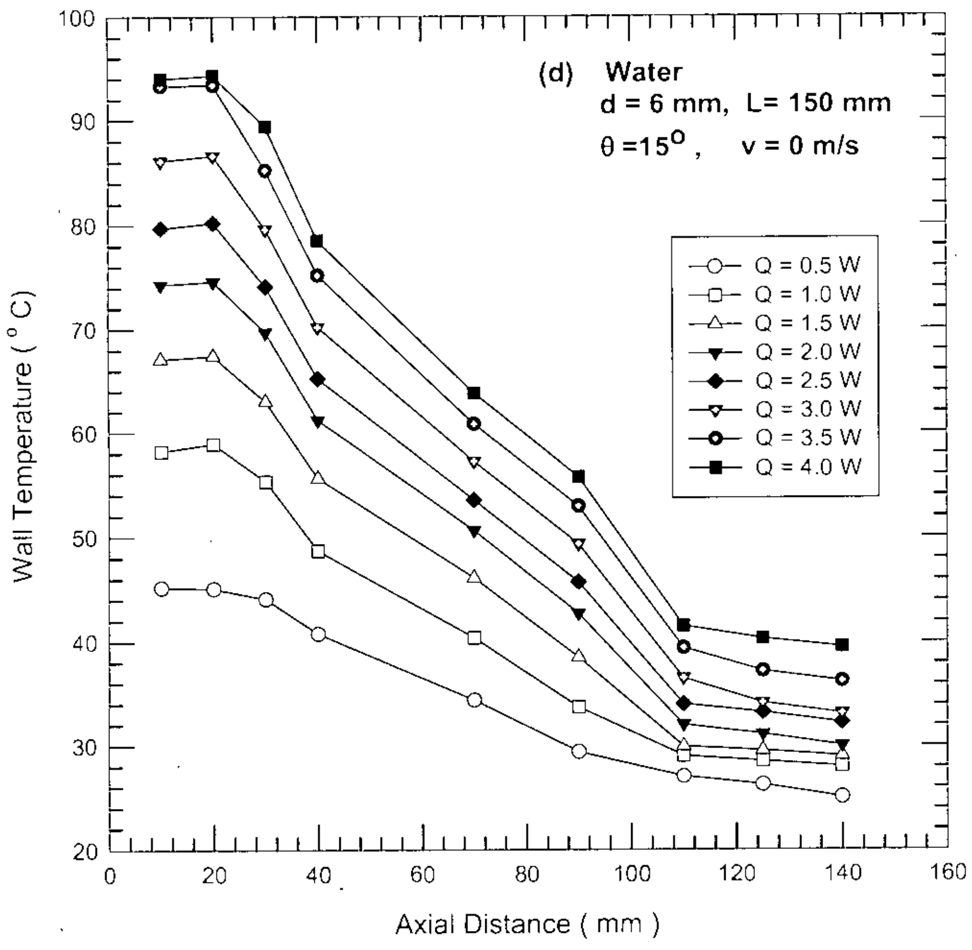


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

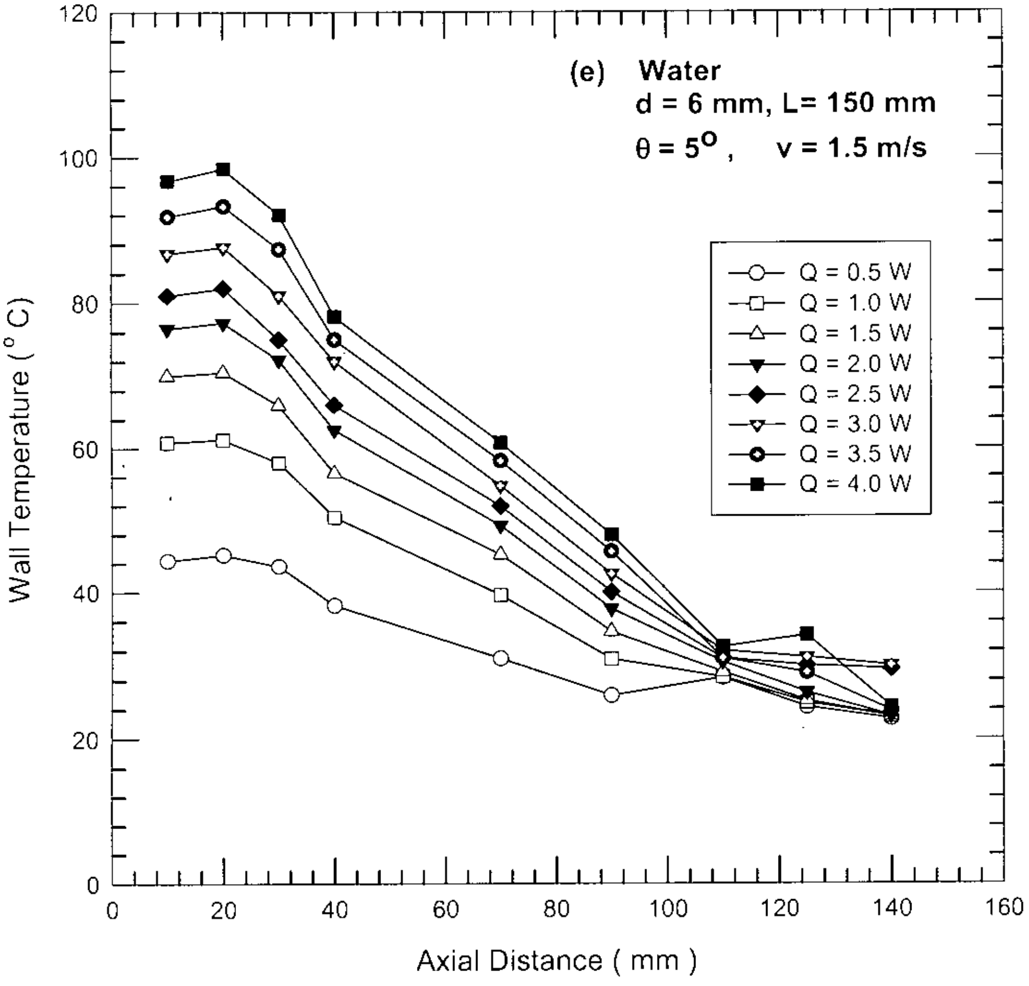


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

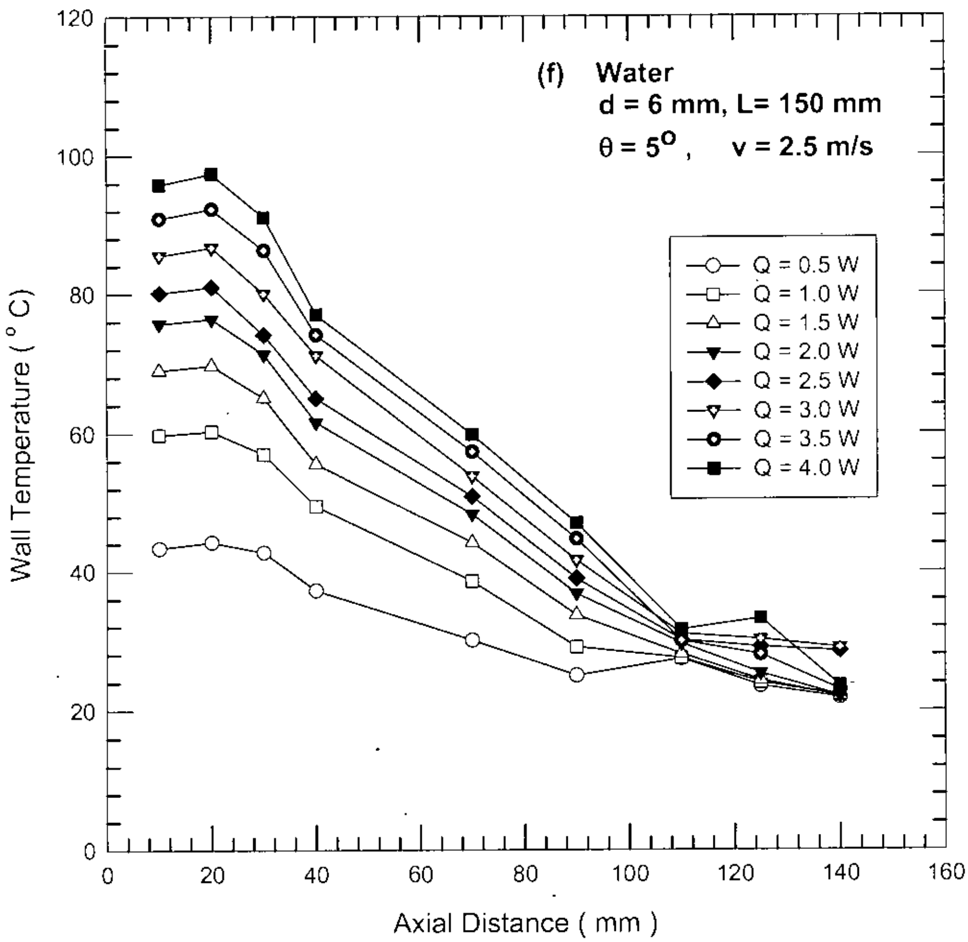


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

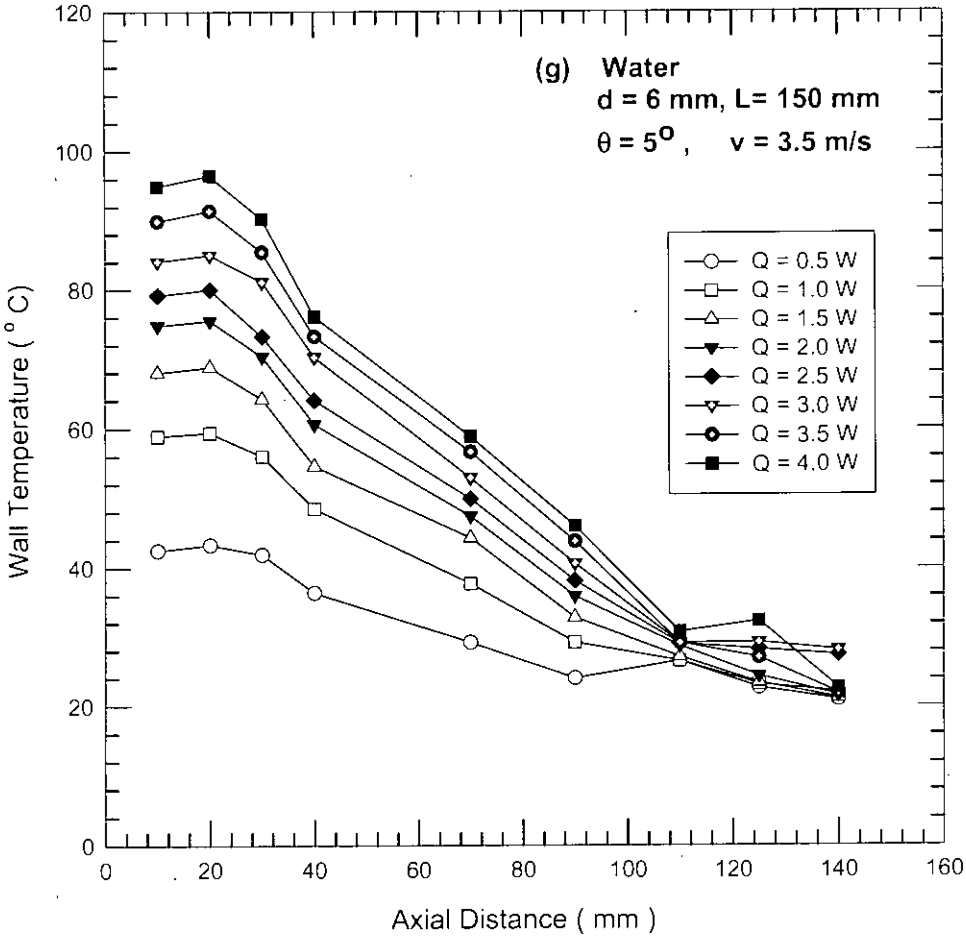


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

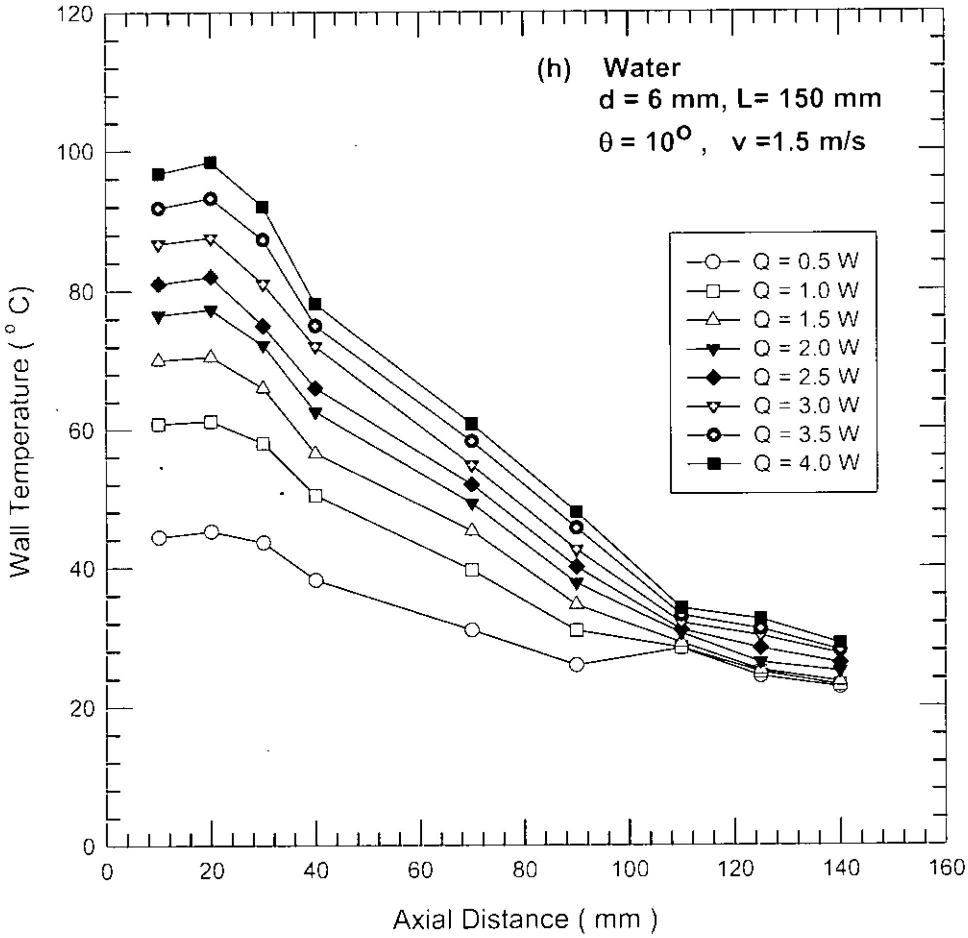


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

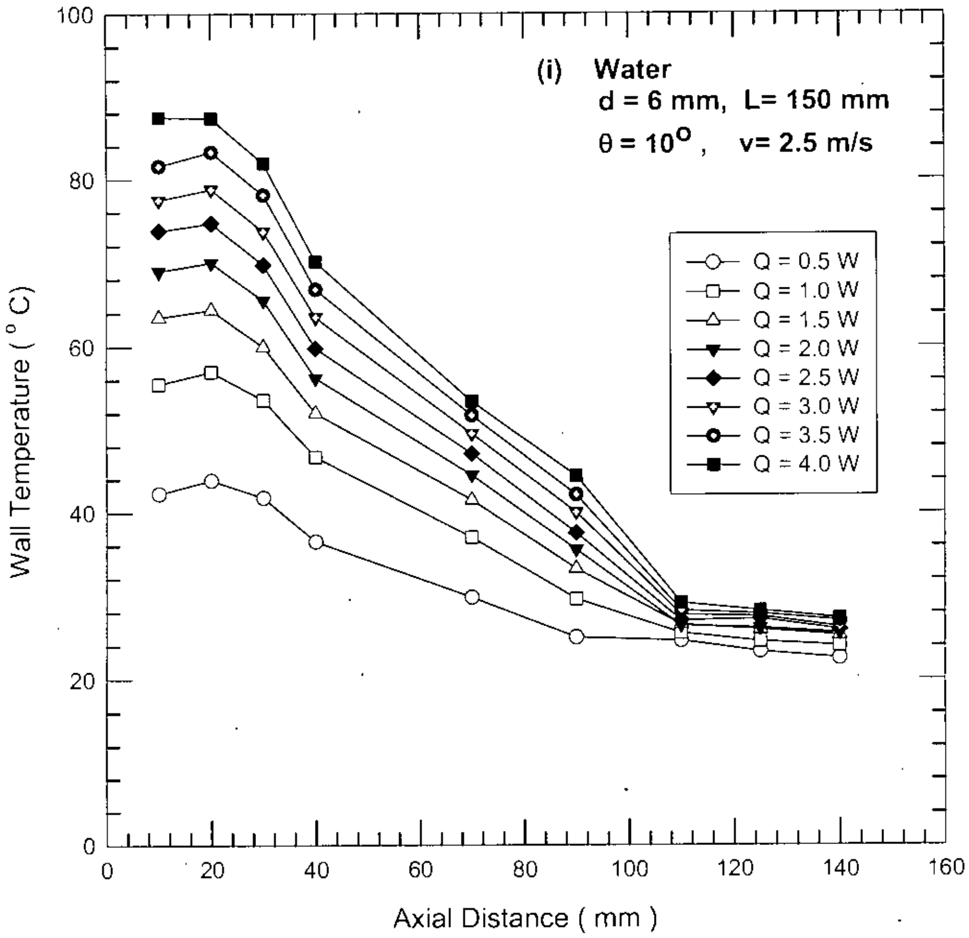


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

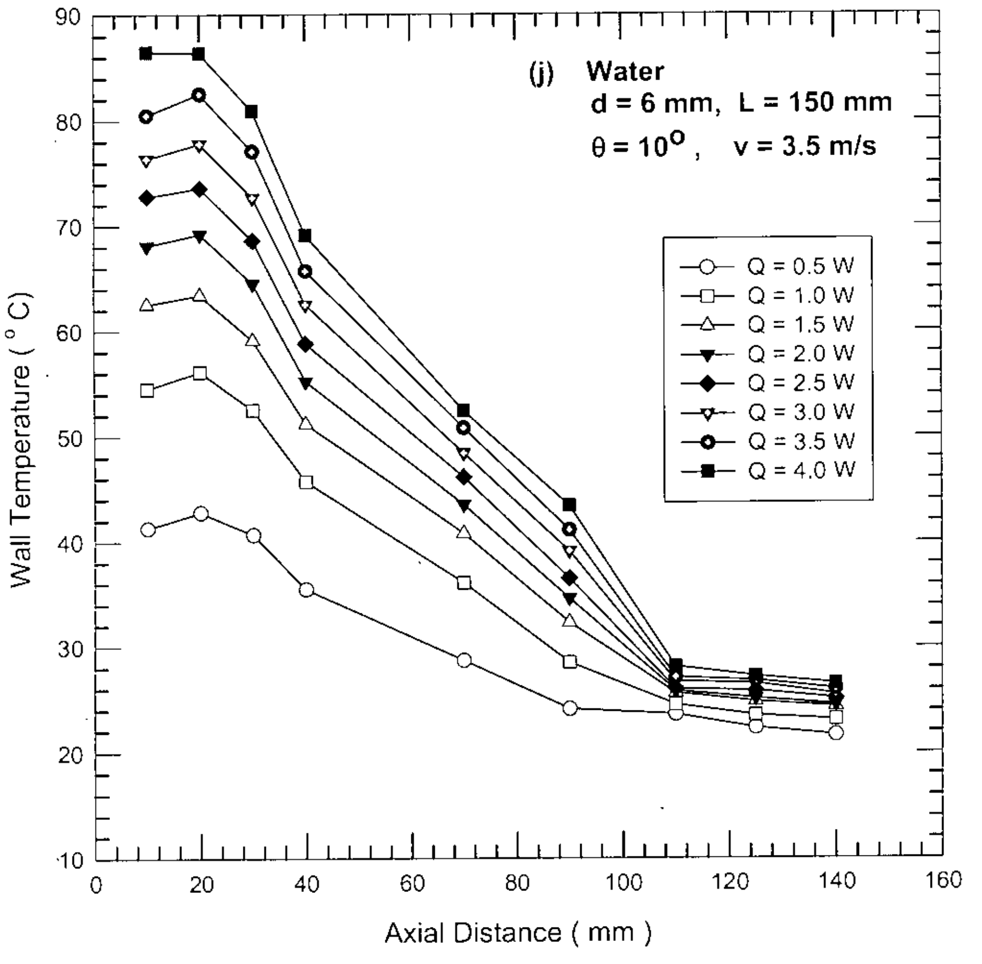


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

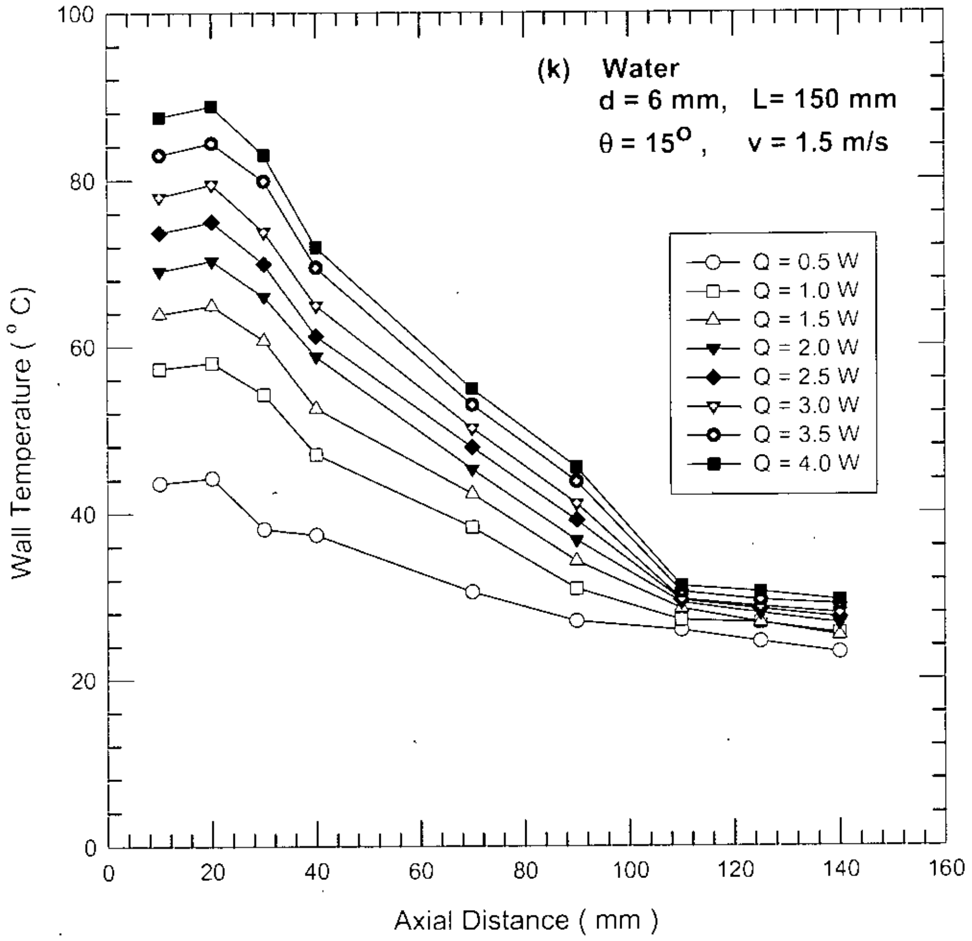


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

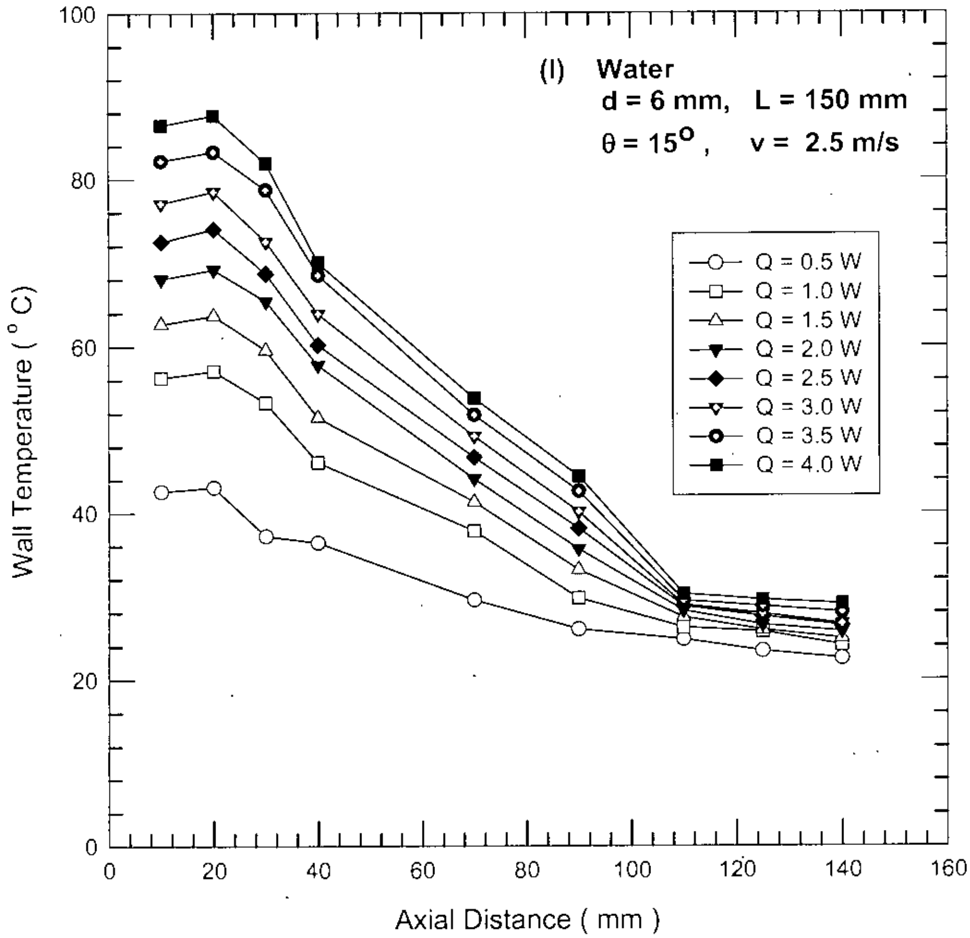


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

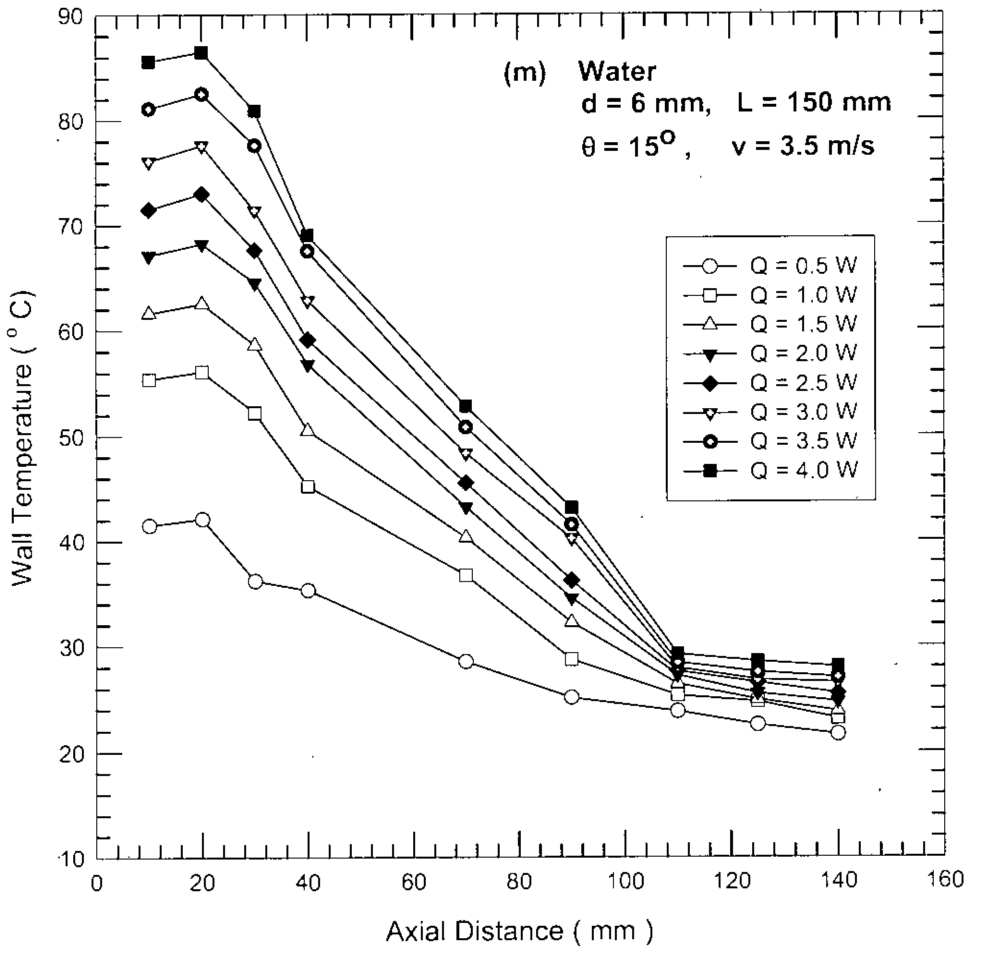


Fig.6.2 Axial temperature distribution along the miniature heat pipe for various powers supplied [L_e = 50 mm, L_c = 50 mm, L_{ad} = 50 mm] (continued)

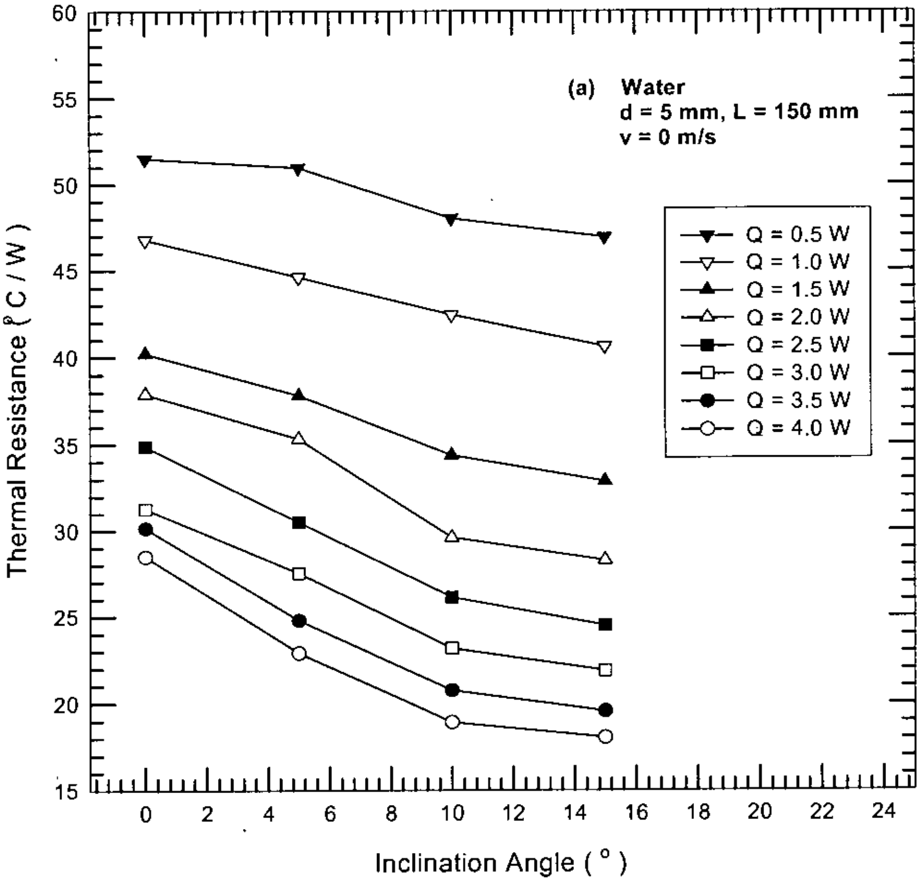


Fig. 6.3 Effect of inclination angle on thermal resistance

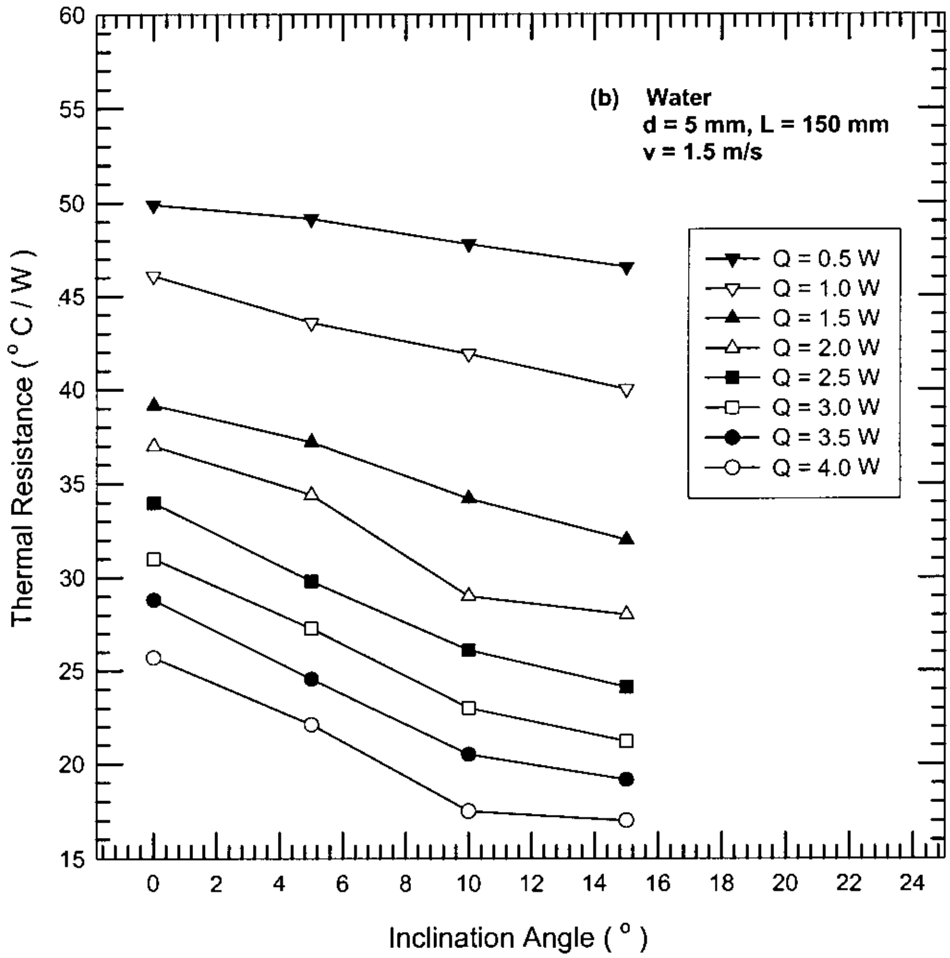


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

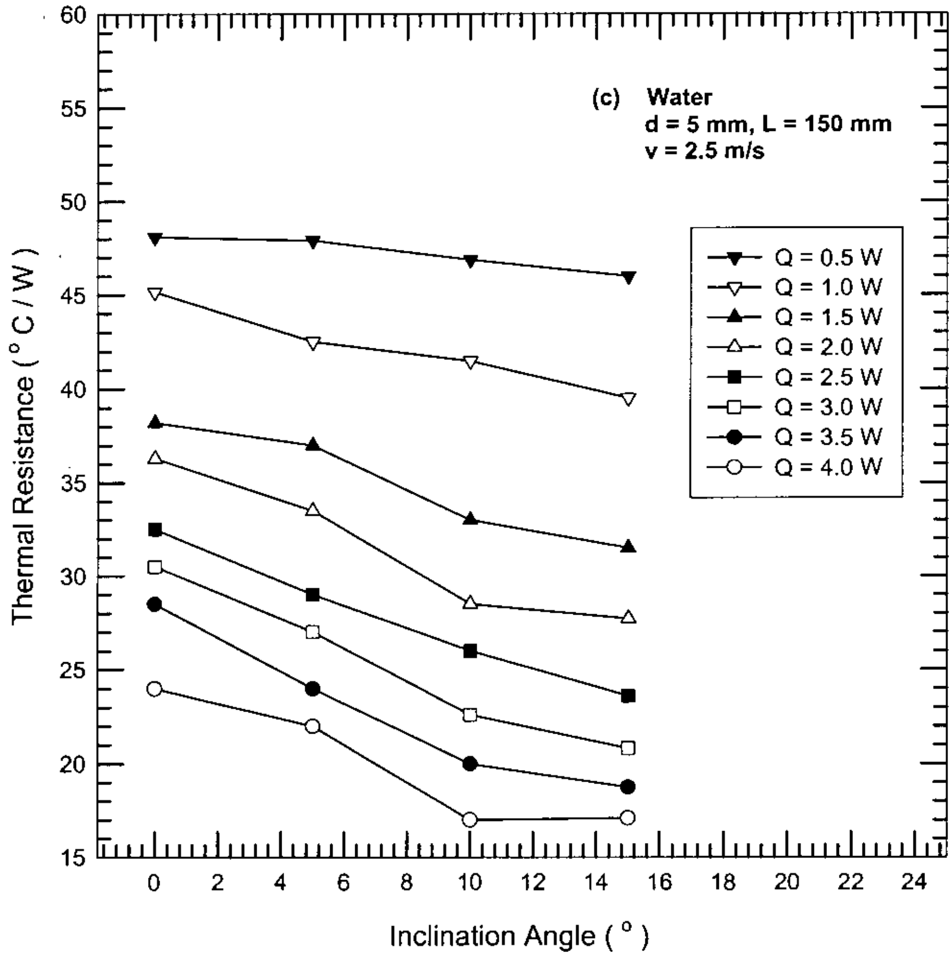


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

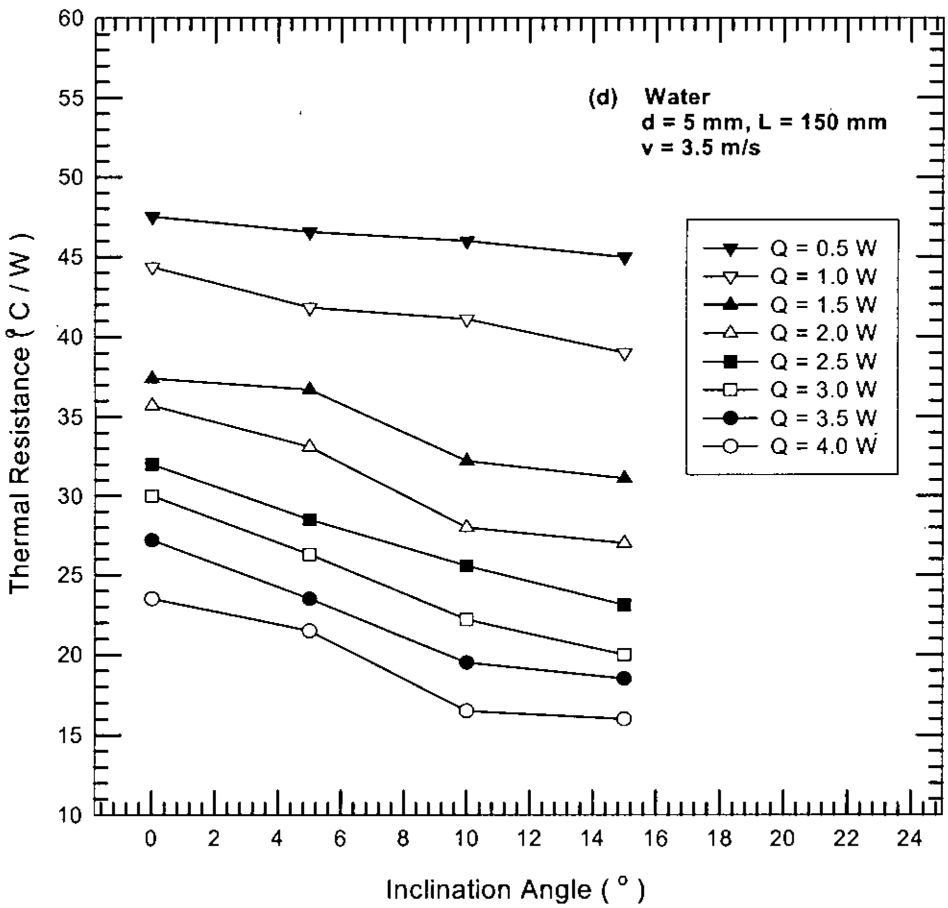


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

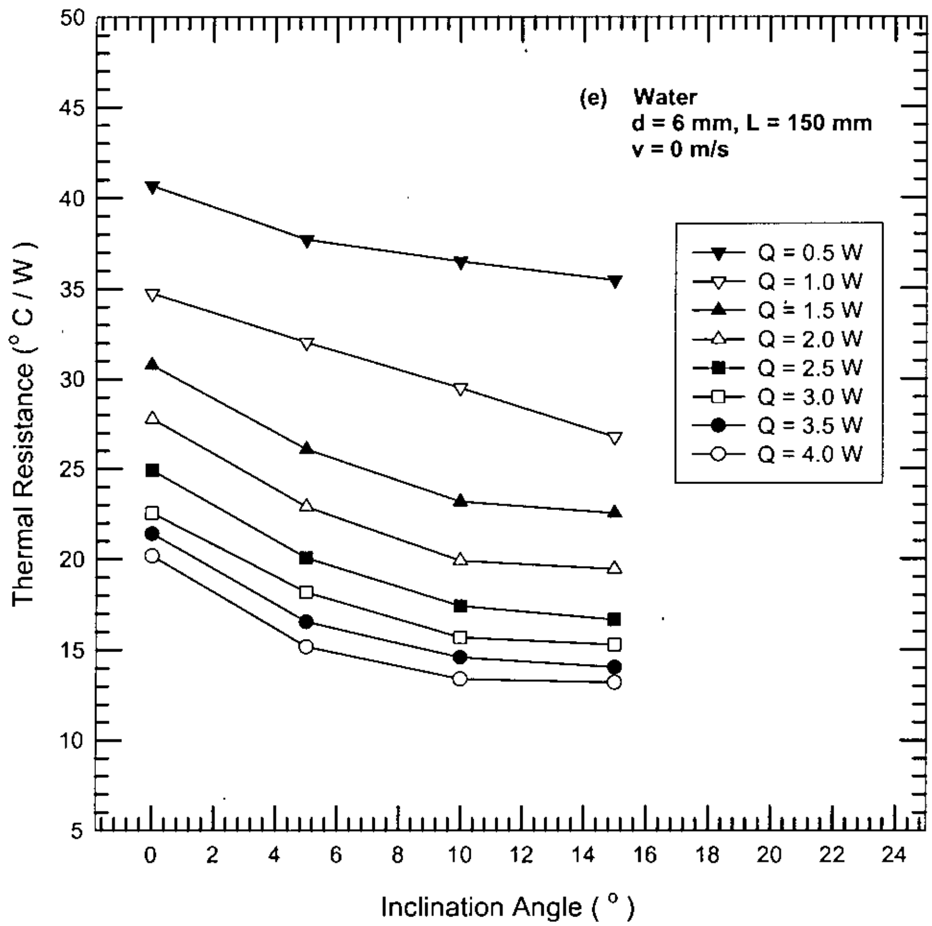


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

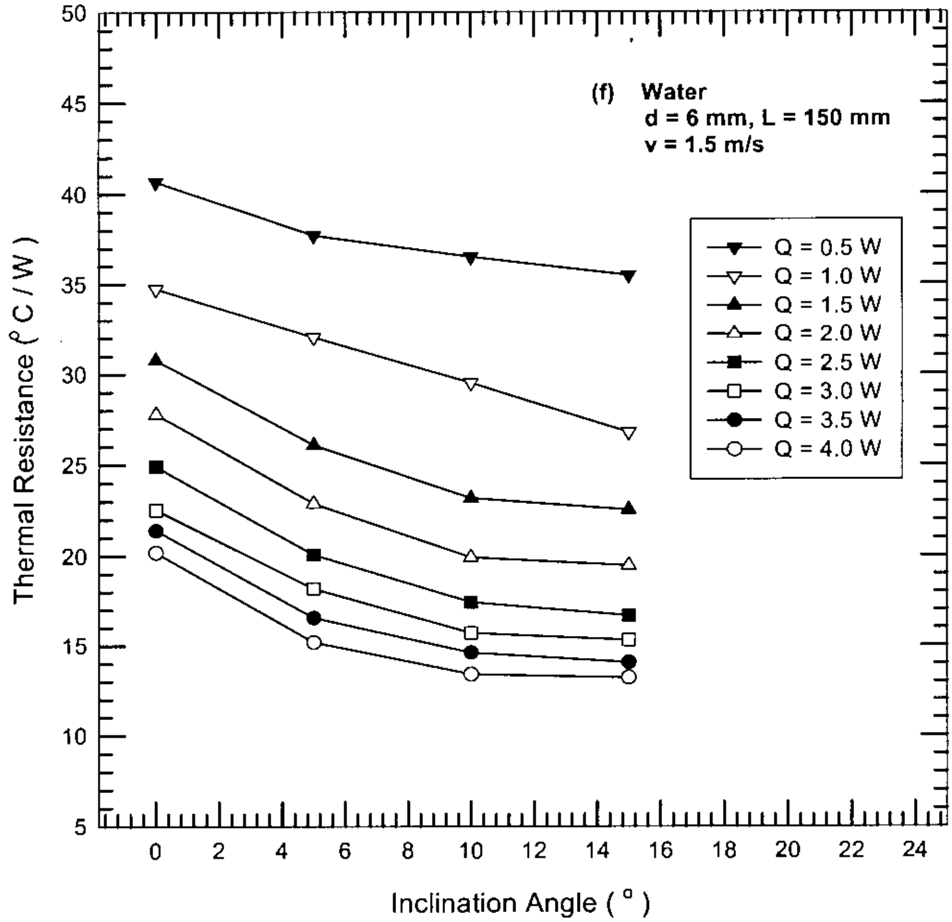


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

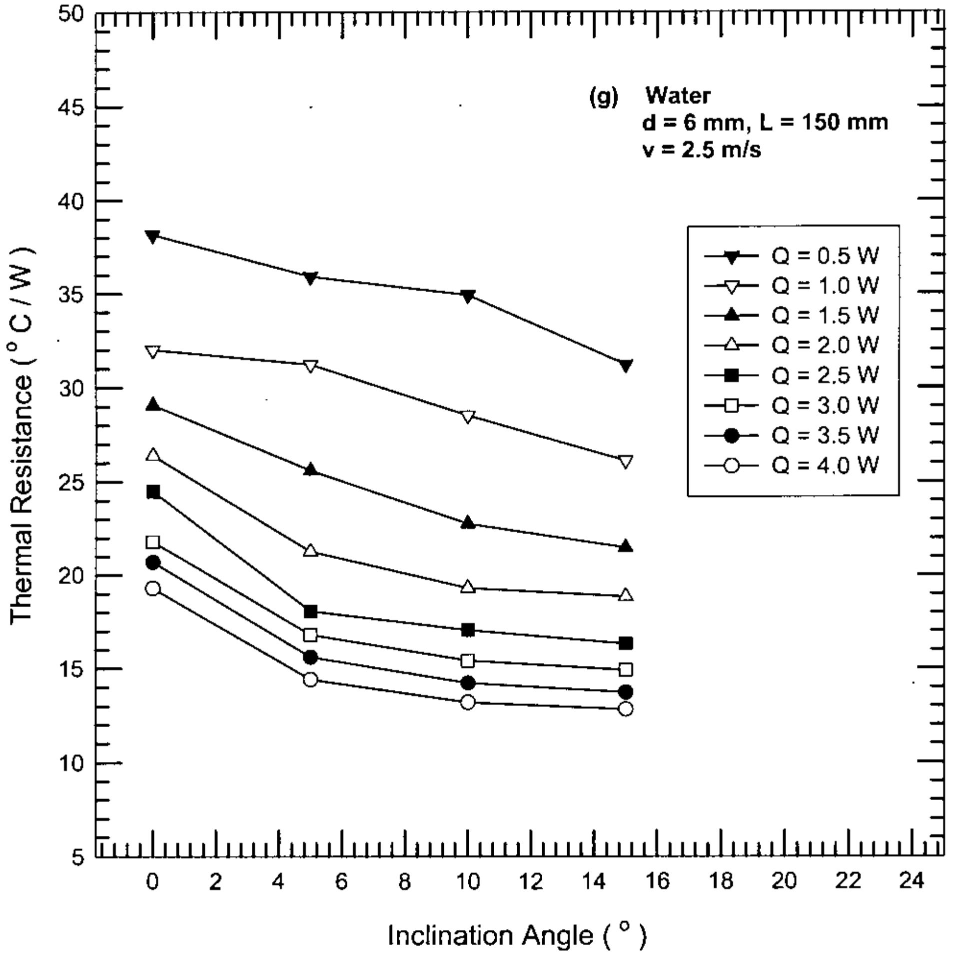


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

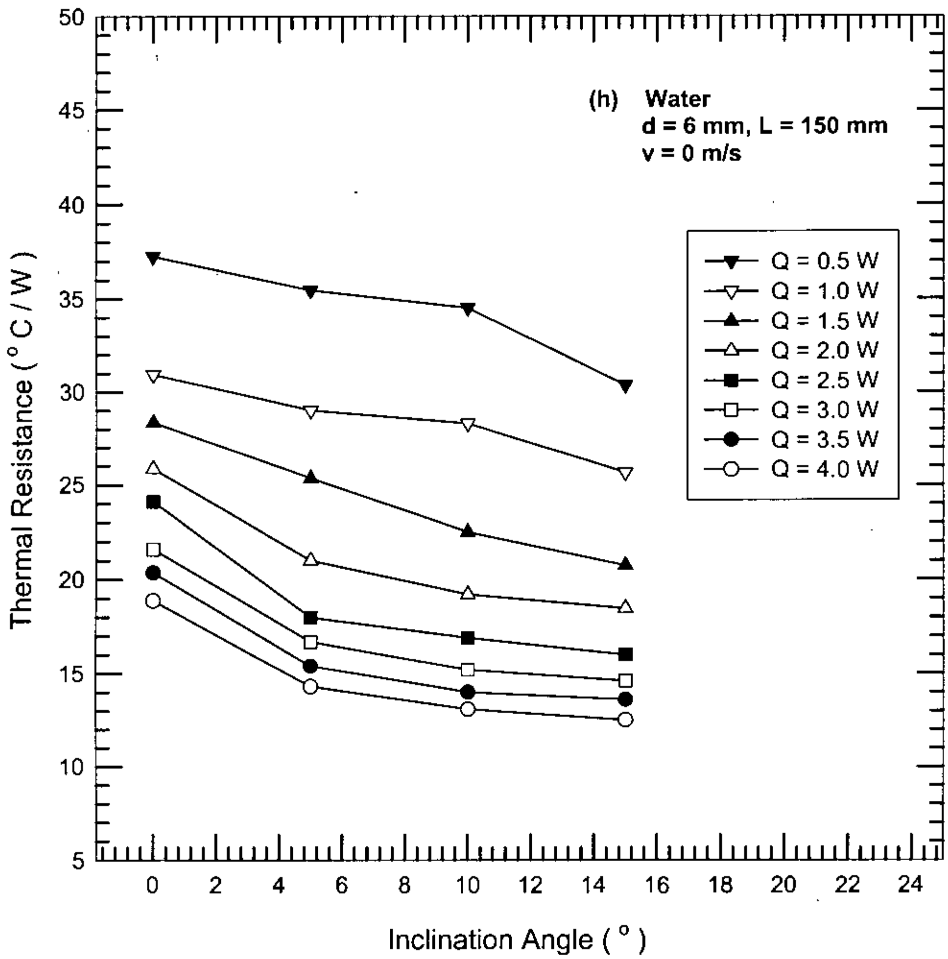


Fig. 6.3 Effect of inclination angle on thermal resistance (Continued)

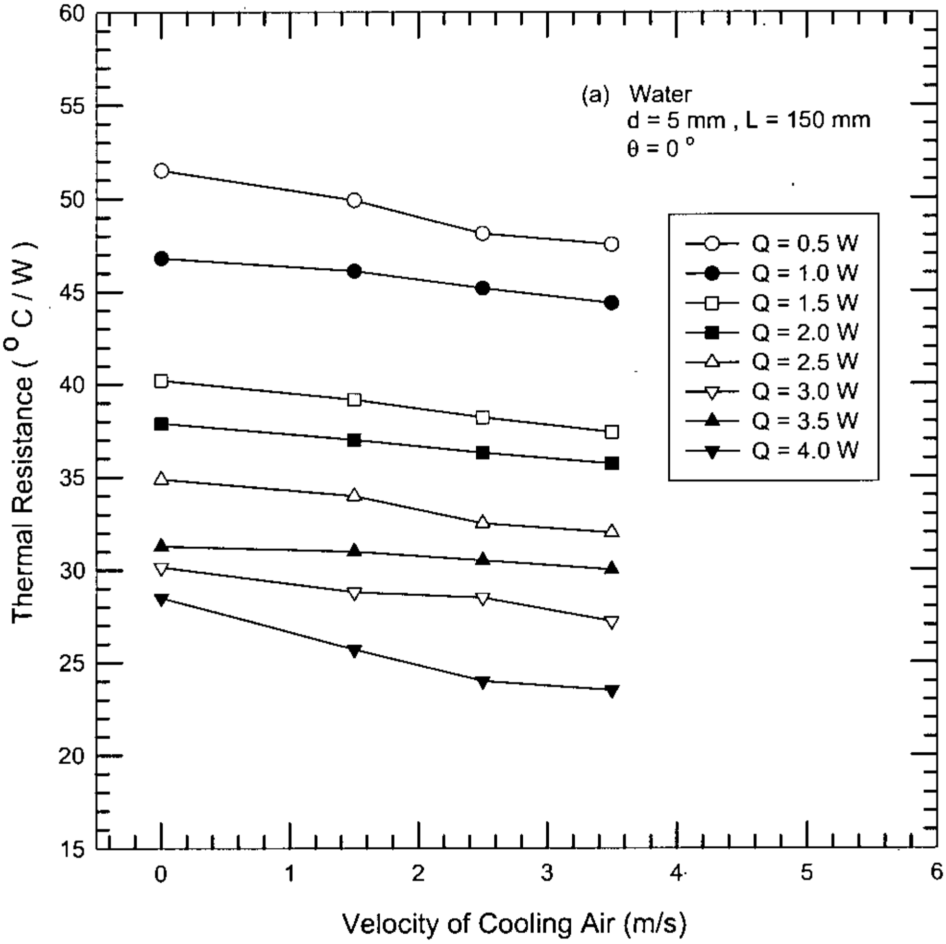


Fig. 6.4 Thermal resistance vs. velocity of cooling air

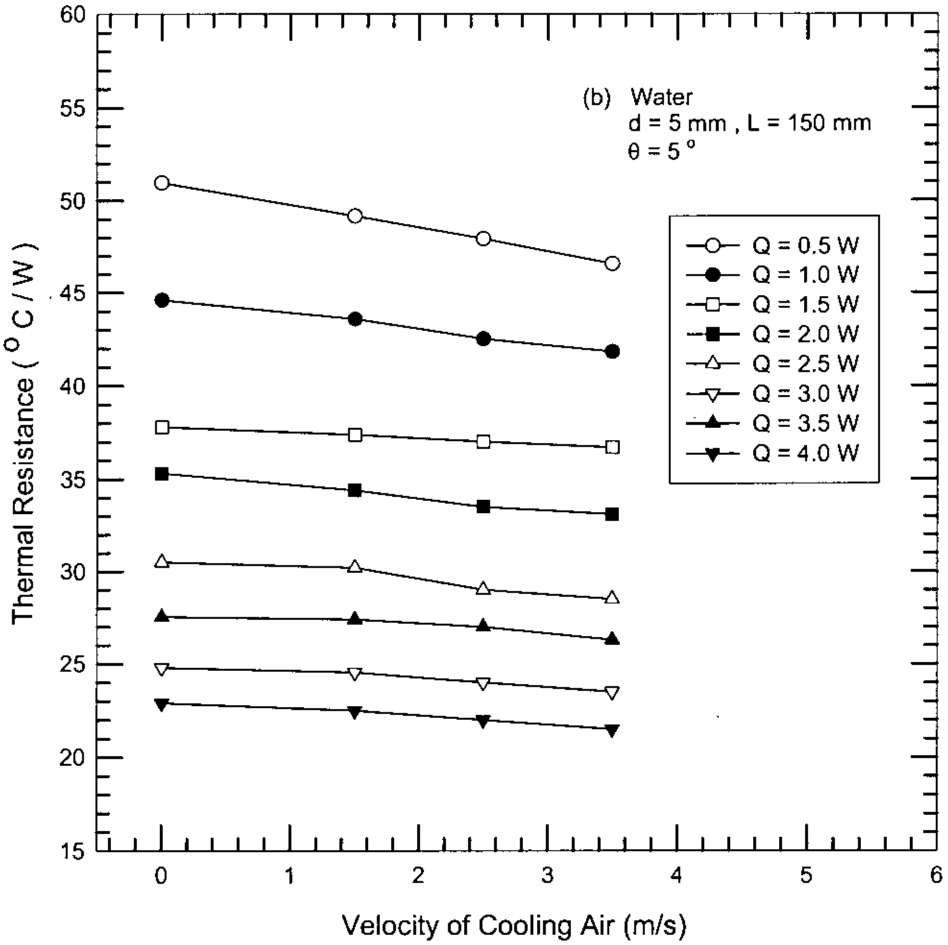
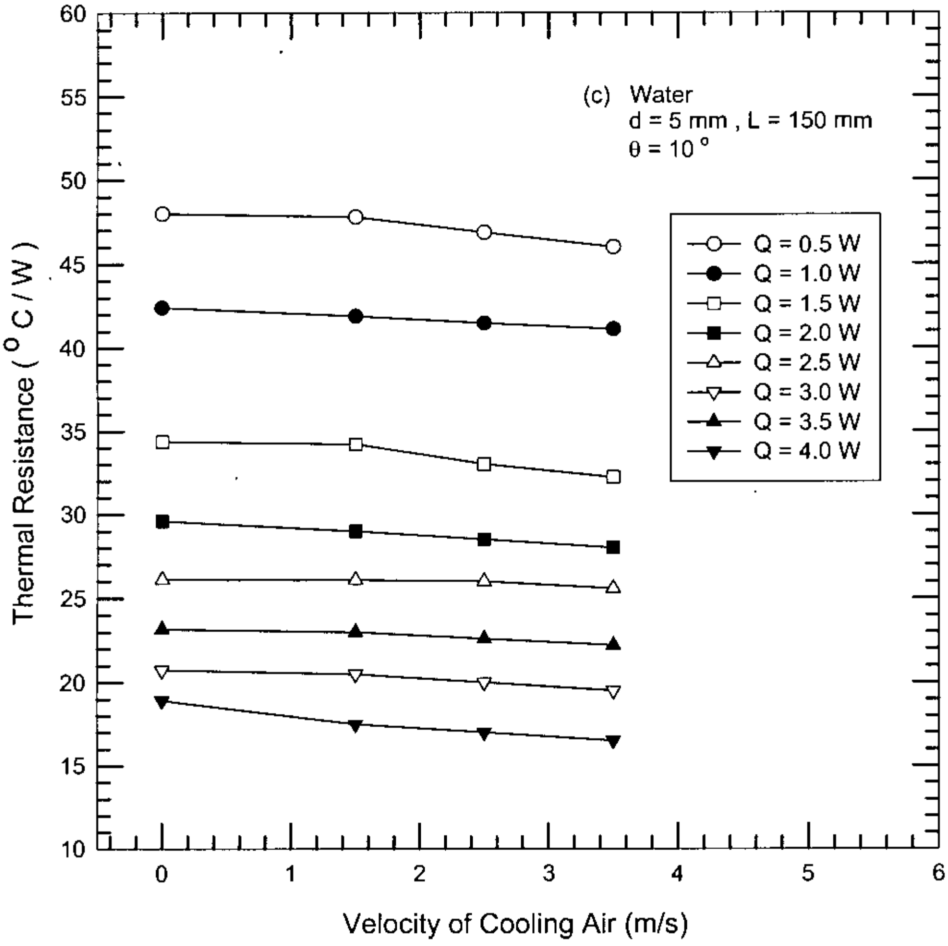


Fig. 6.4 Thermal resistance vs. velocity of cooling air (continued)



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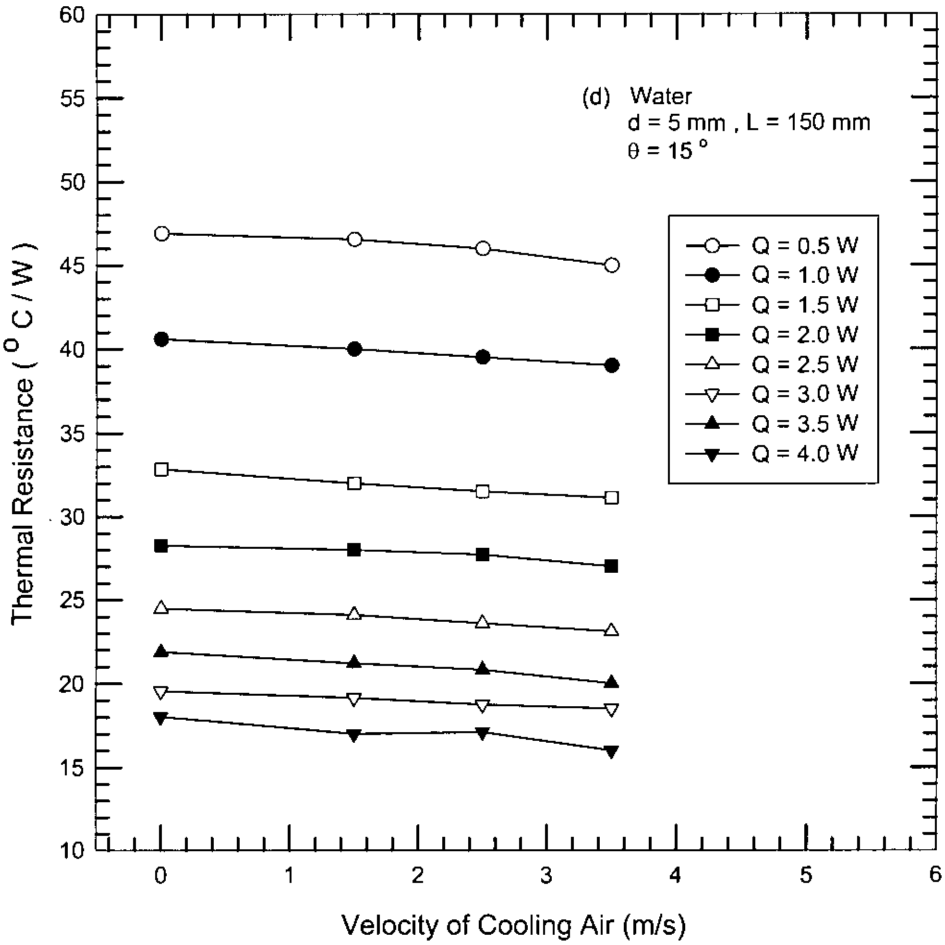

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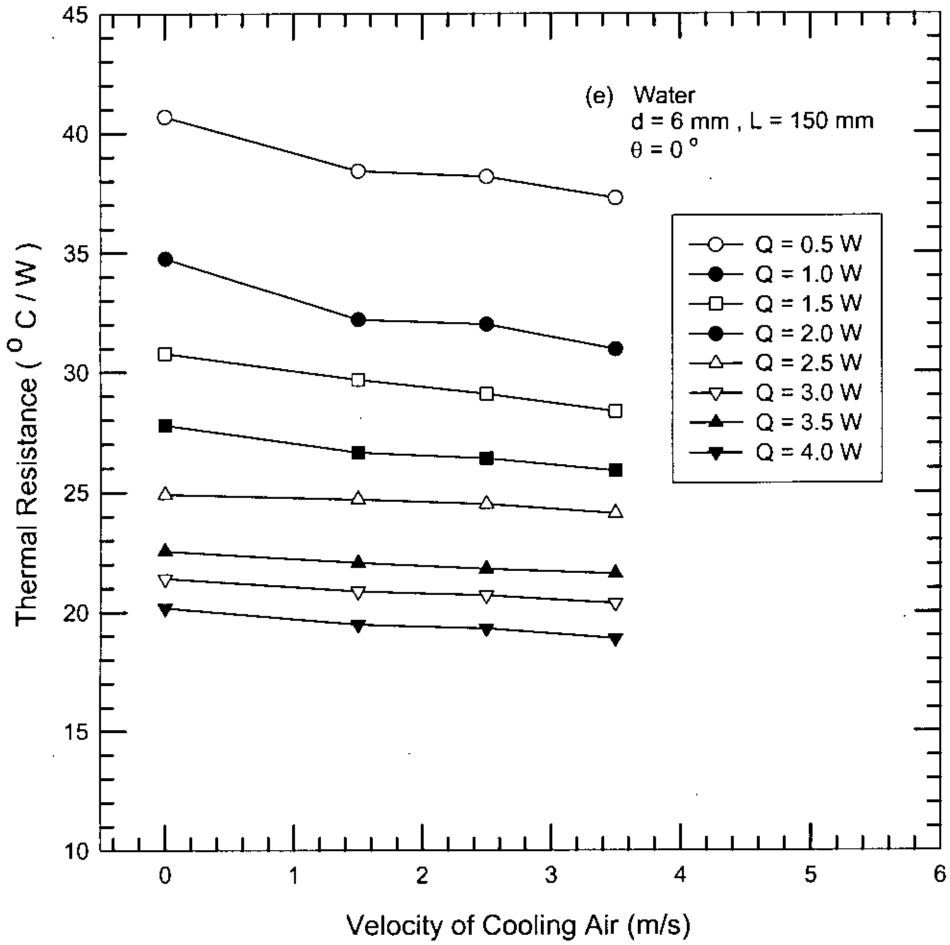

Fig. 6.4 Thermal resistance vs. velocity of cooling air (continued)

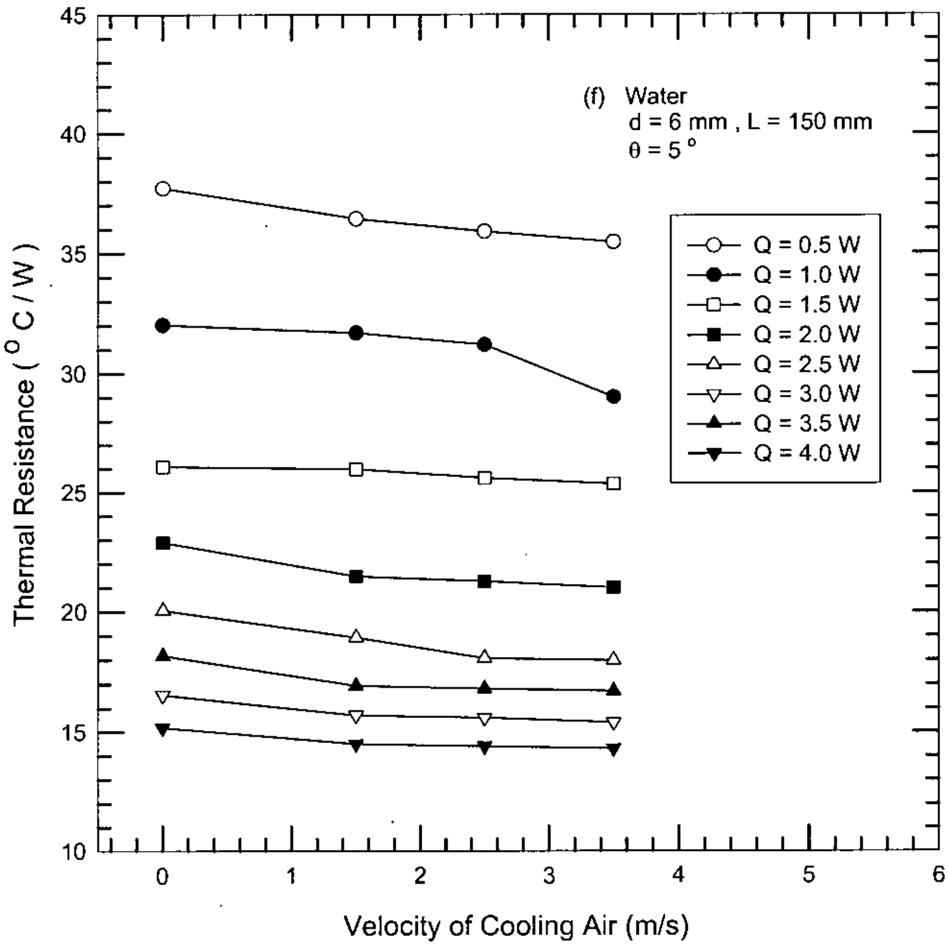

Fig. 6.4 Thermal resistance vs. velocity of cooling air (continued)

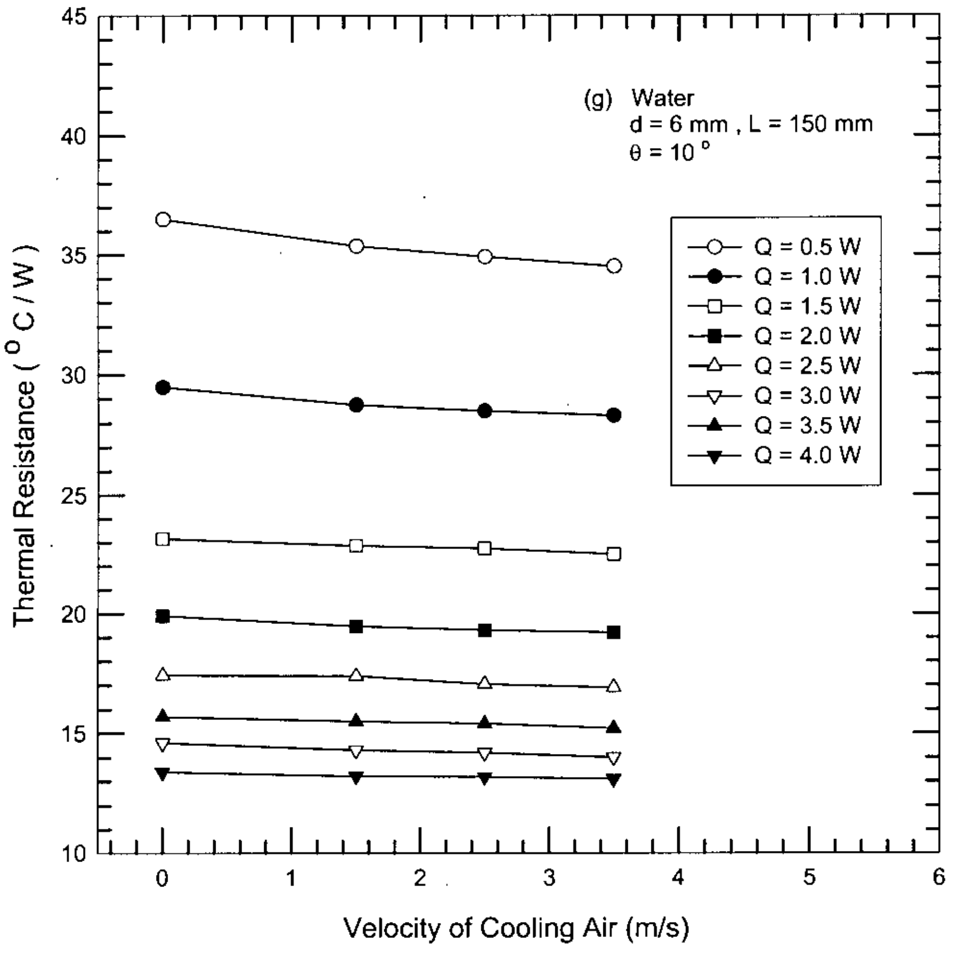

Fig. 6.4 Thermal resistance vs. velocity of cooling air (continued)

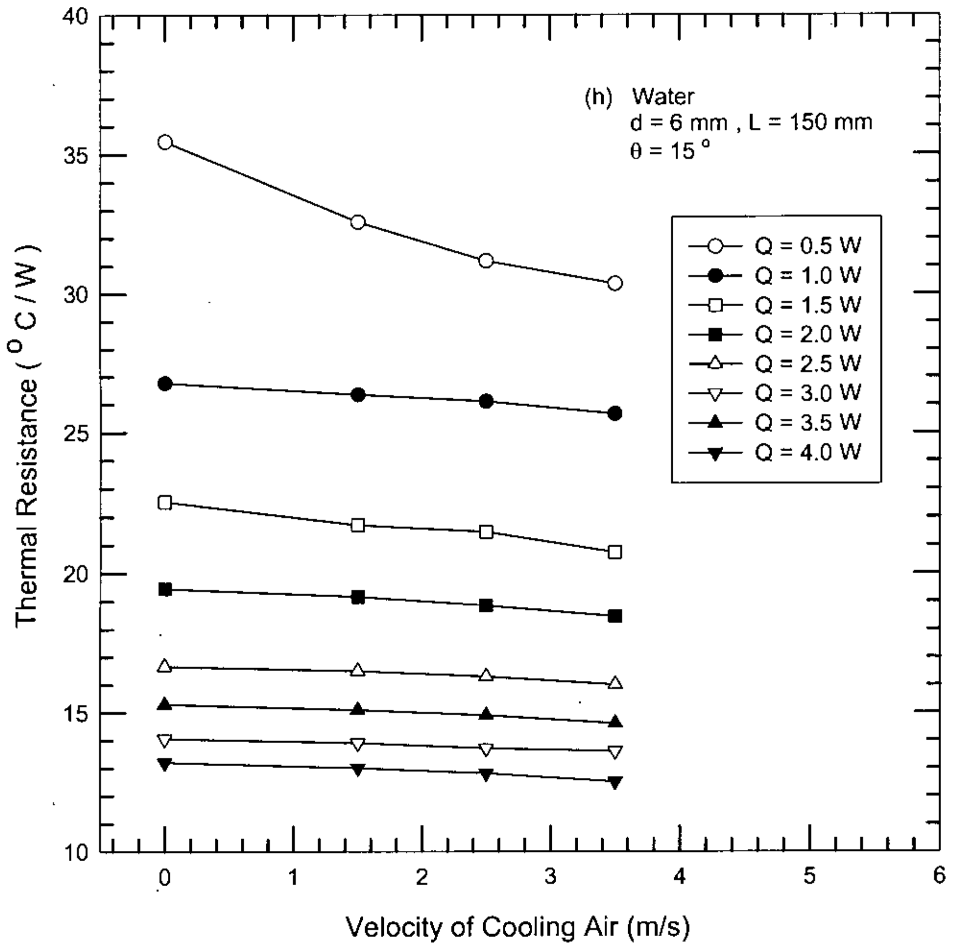

Fig. 6.4 Thermal resistance vs. velocity of cooling air (continued)

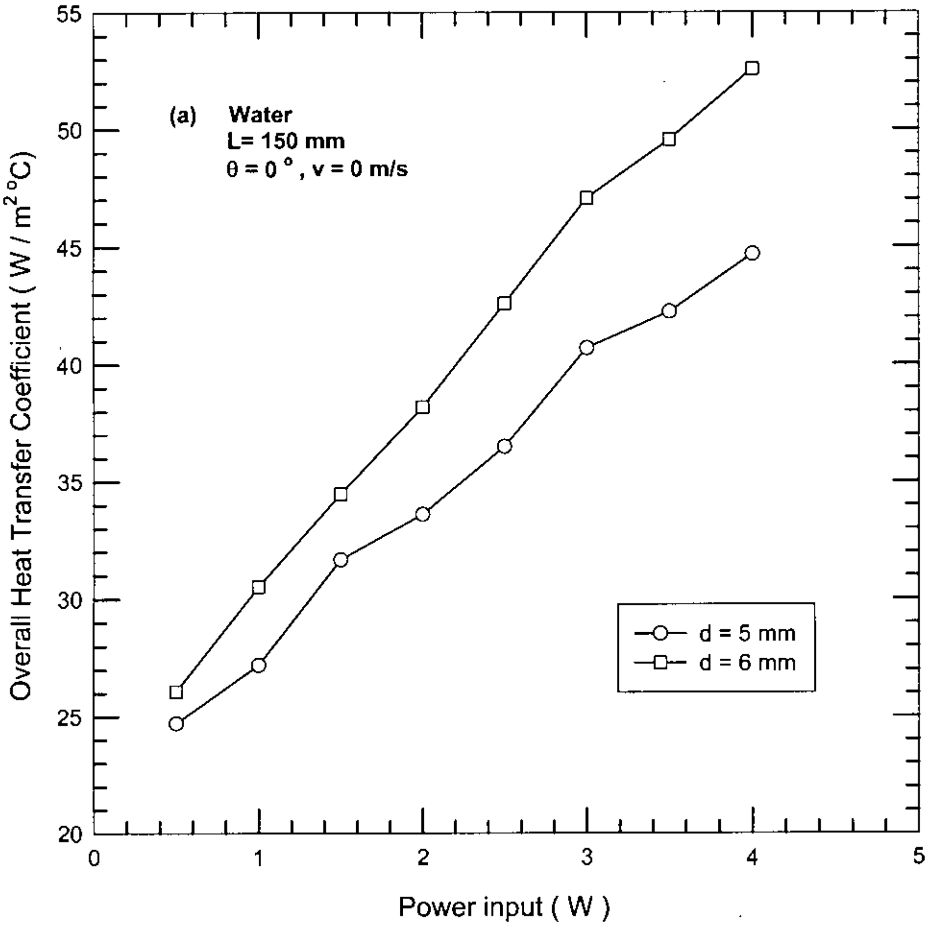

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient

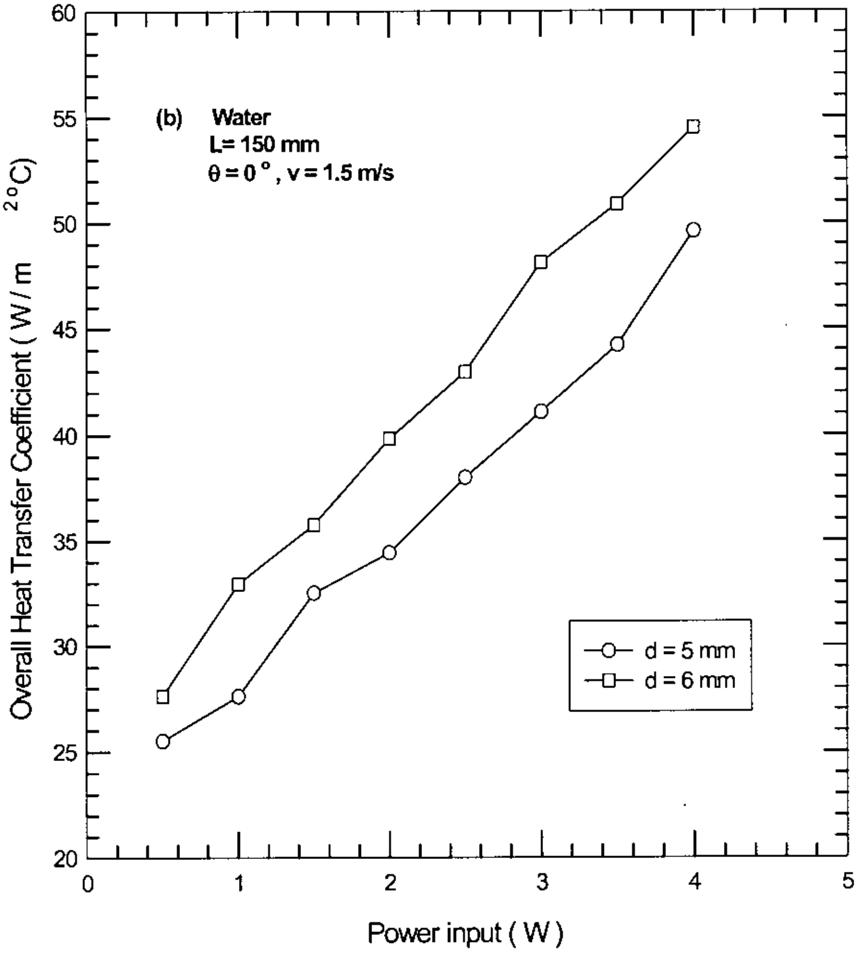

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

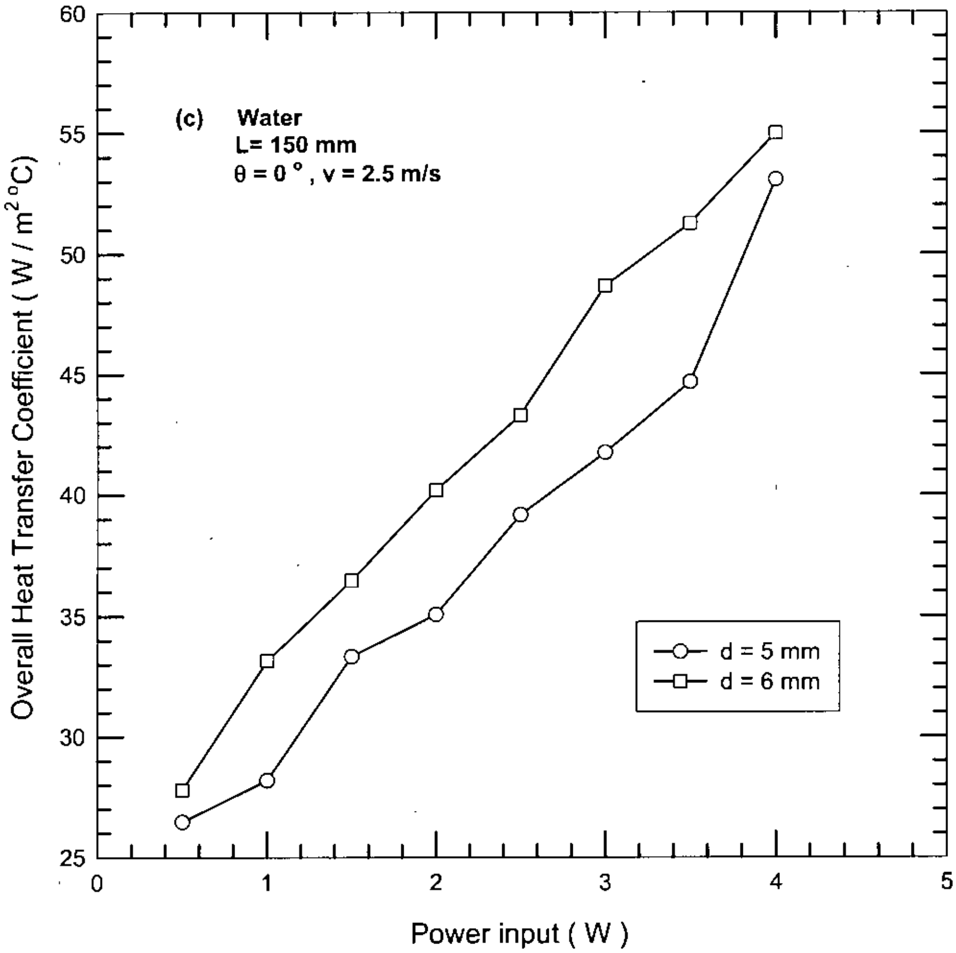

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

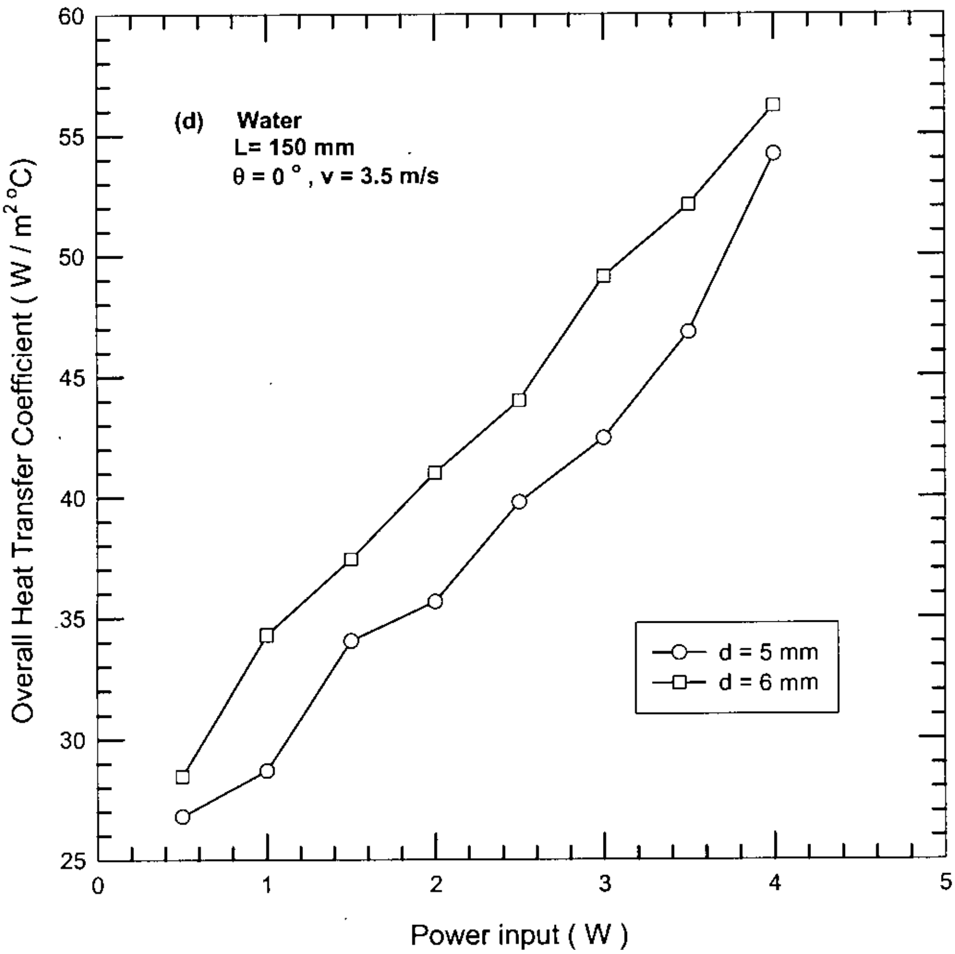

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

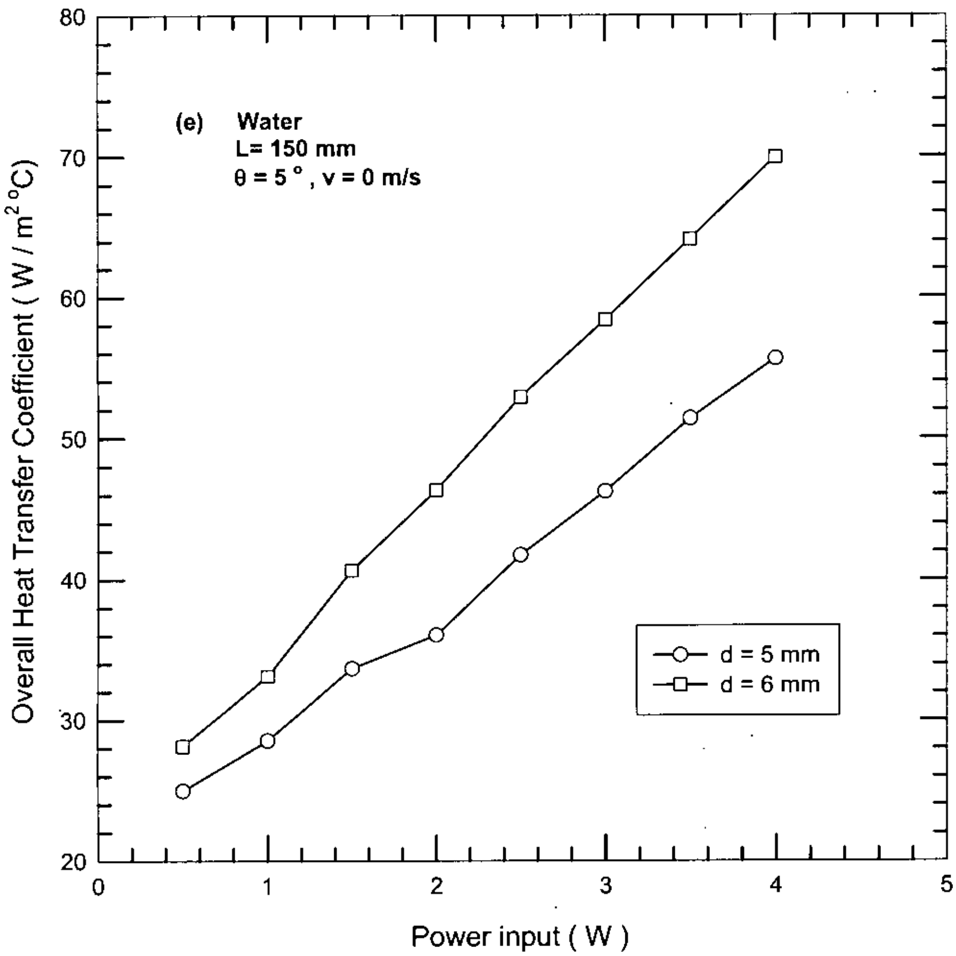

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

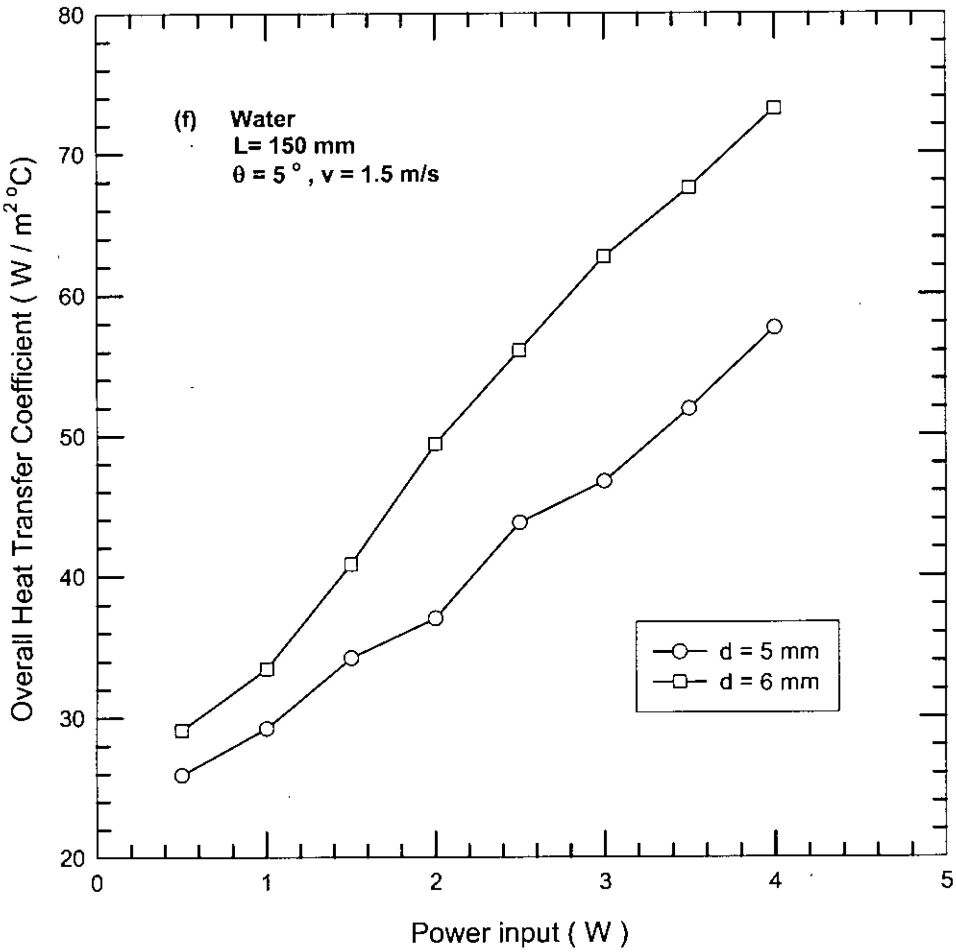

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

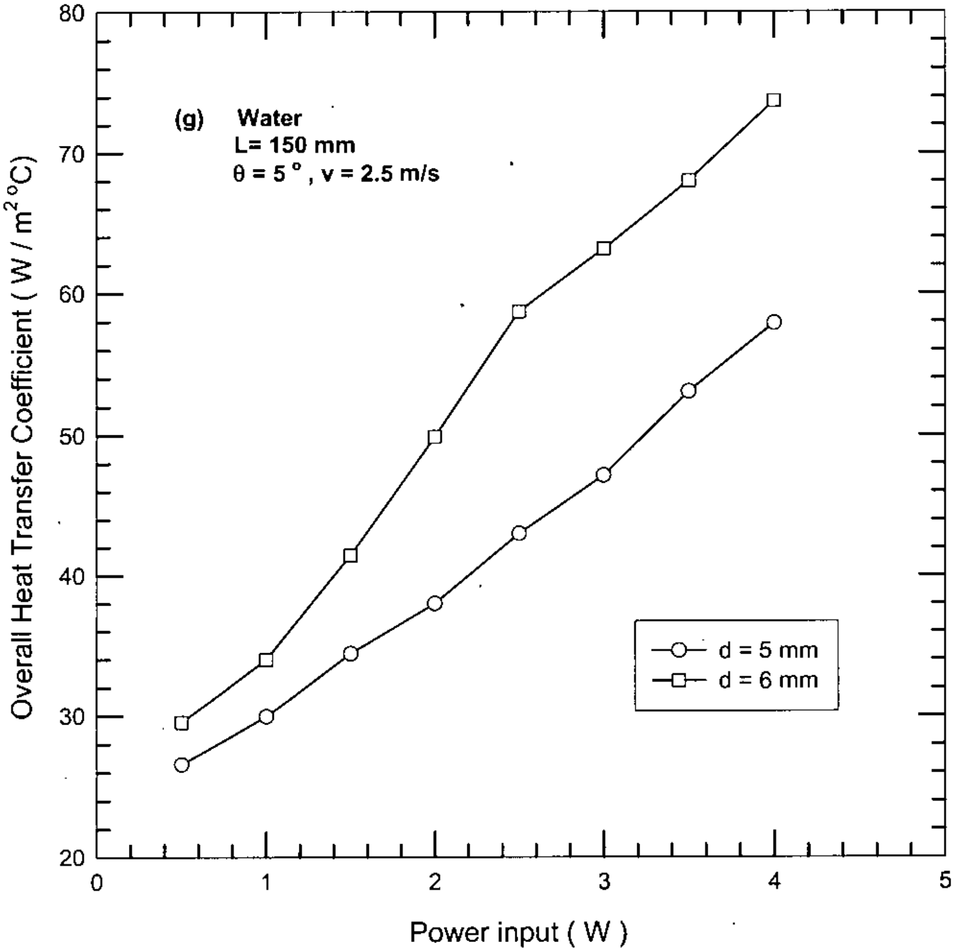

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

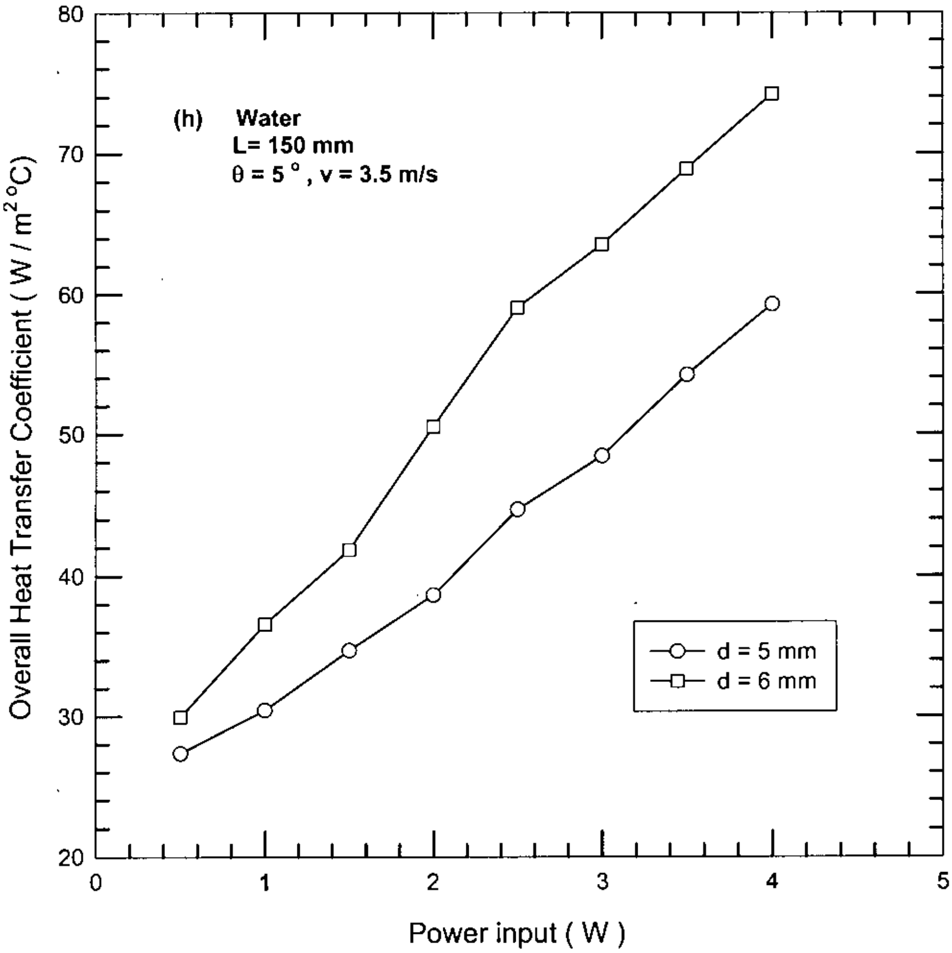

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

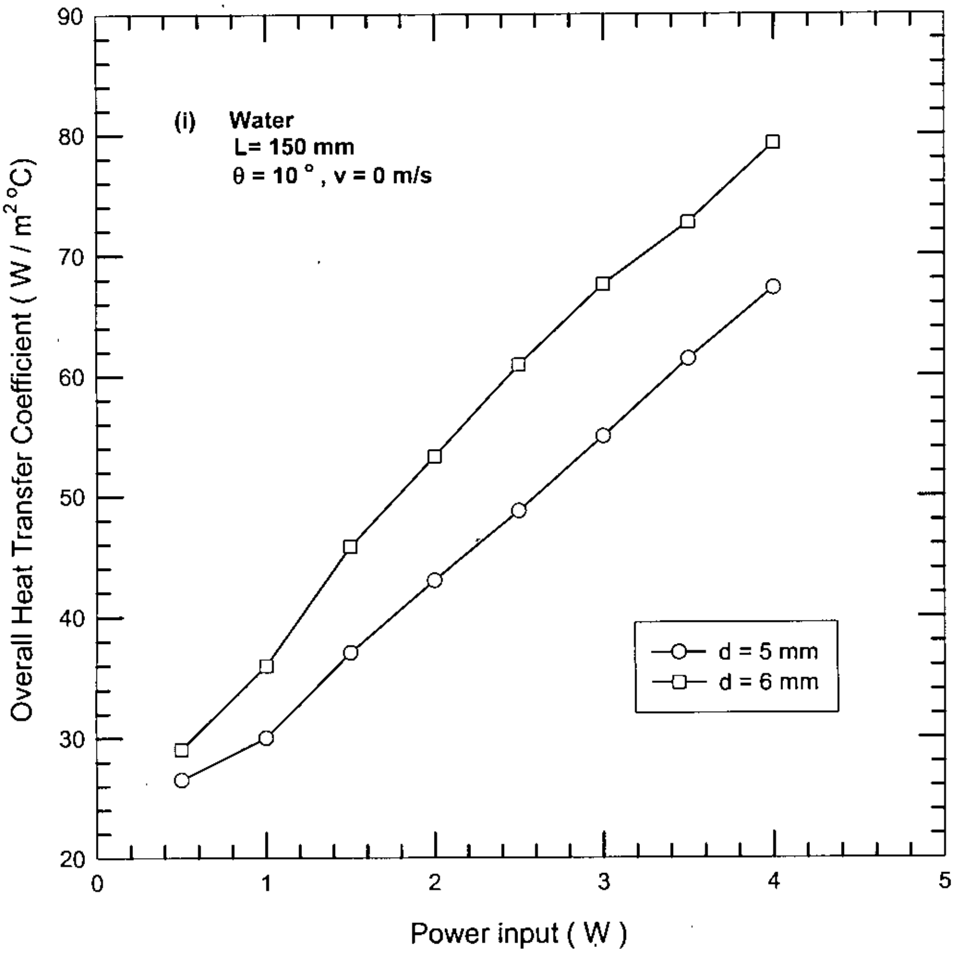

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

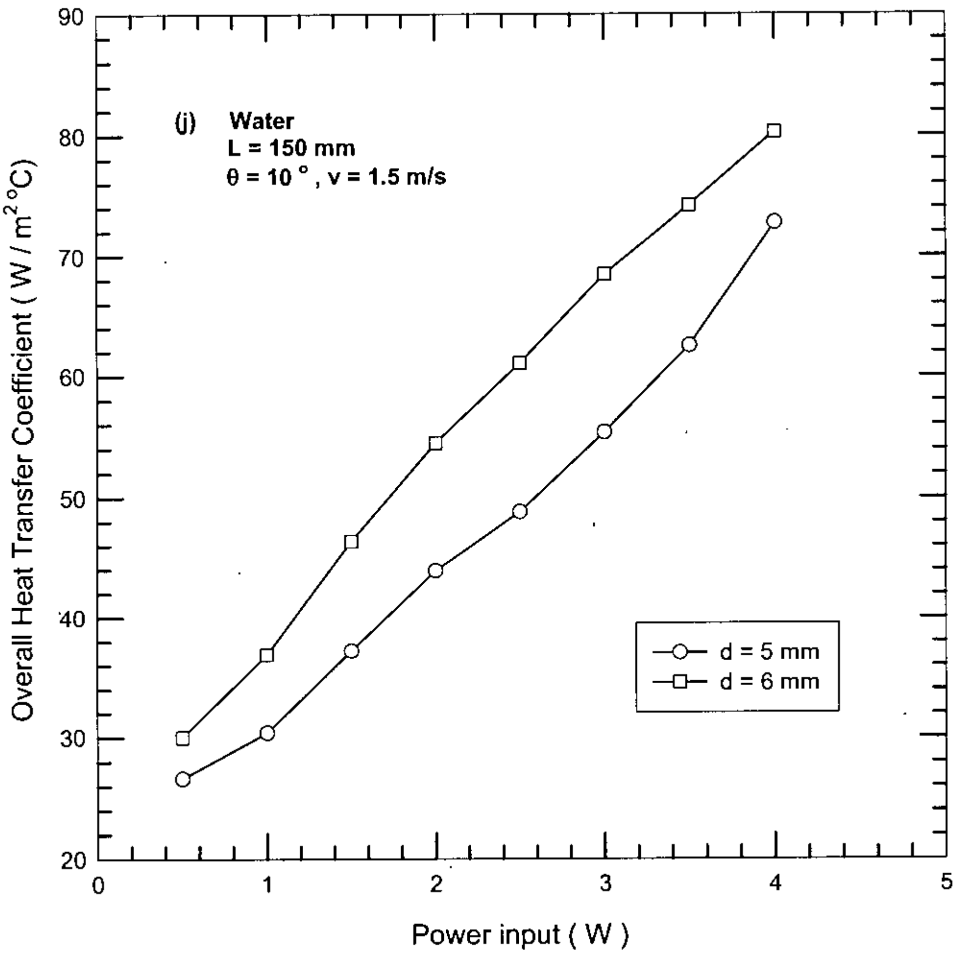

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

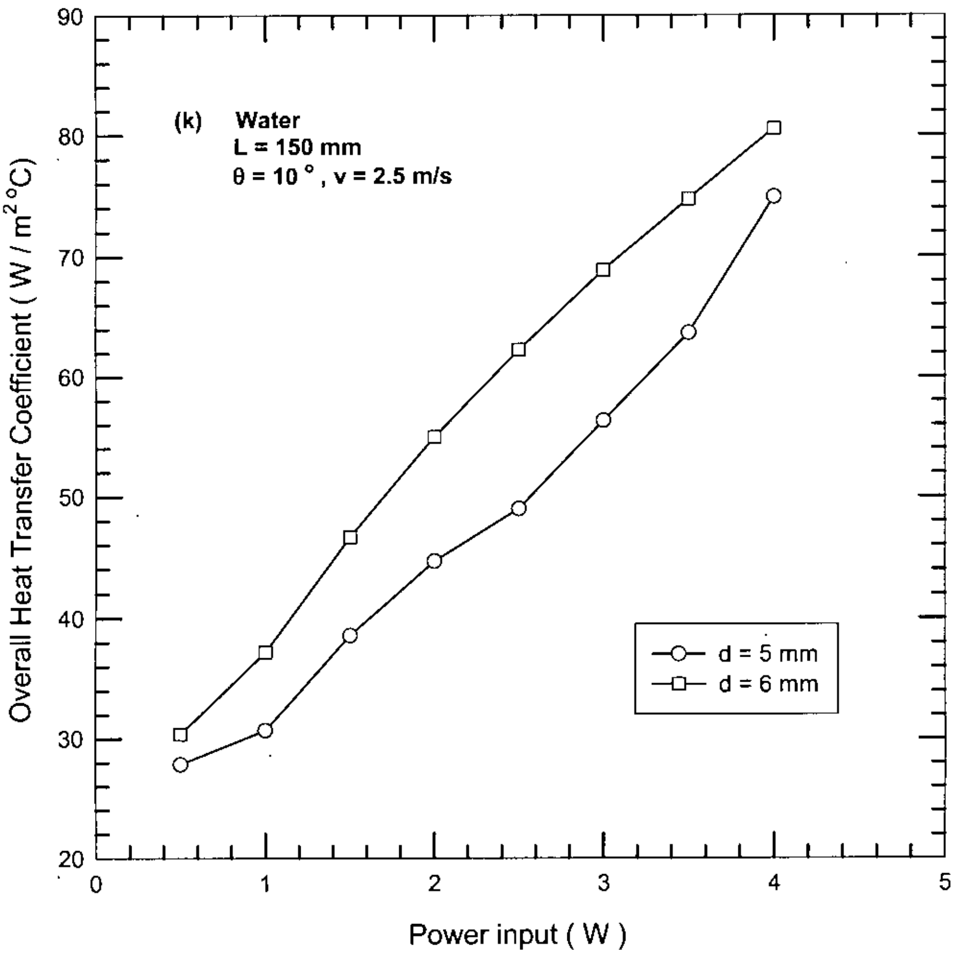

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

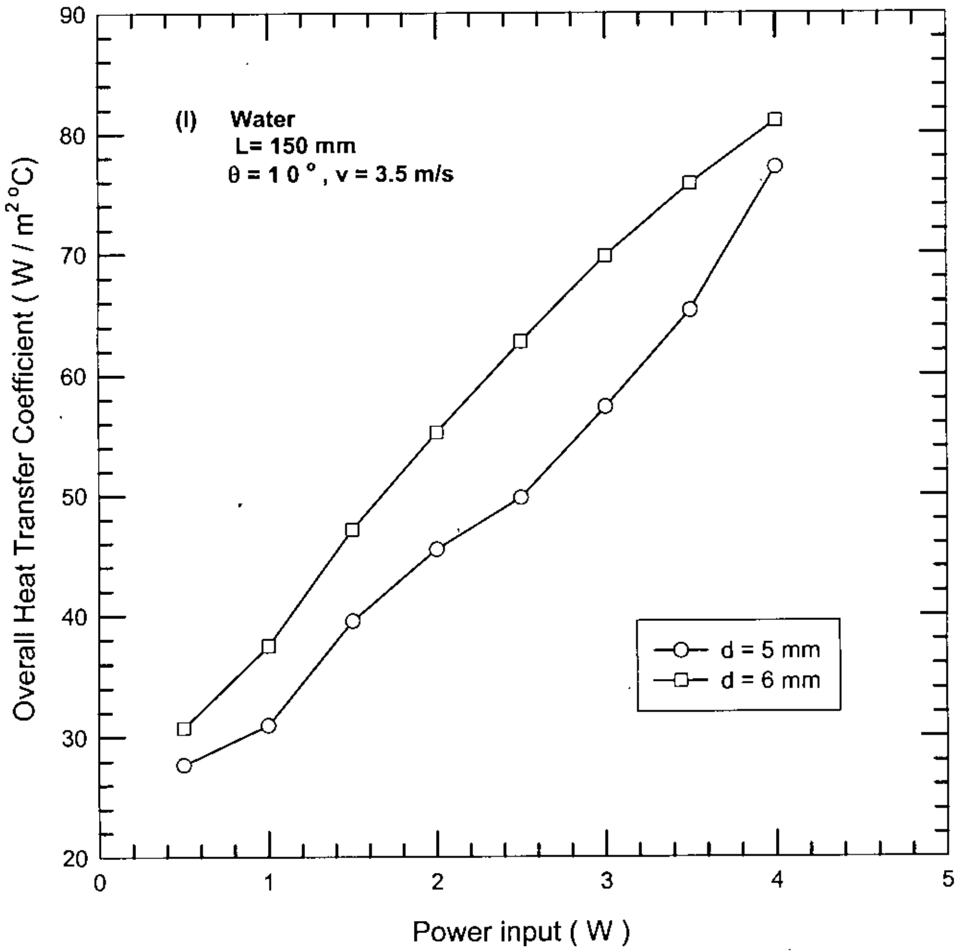

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

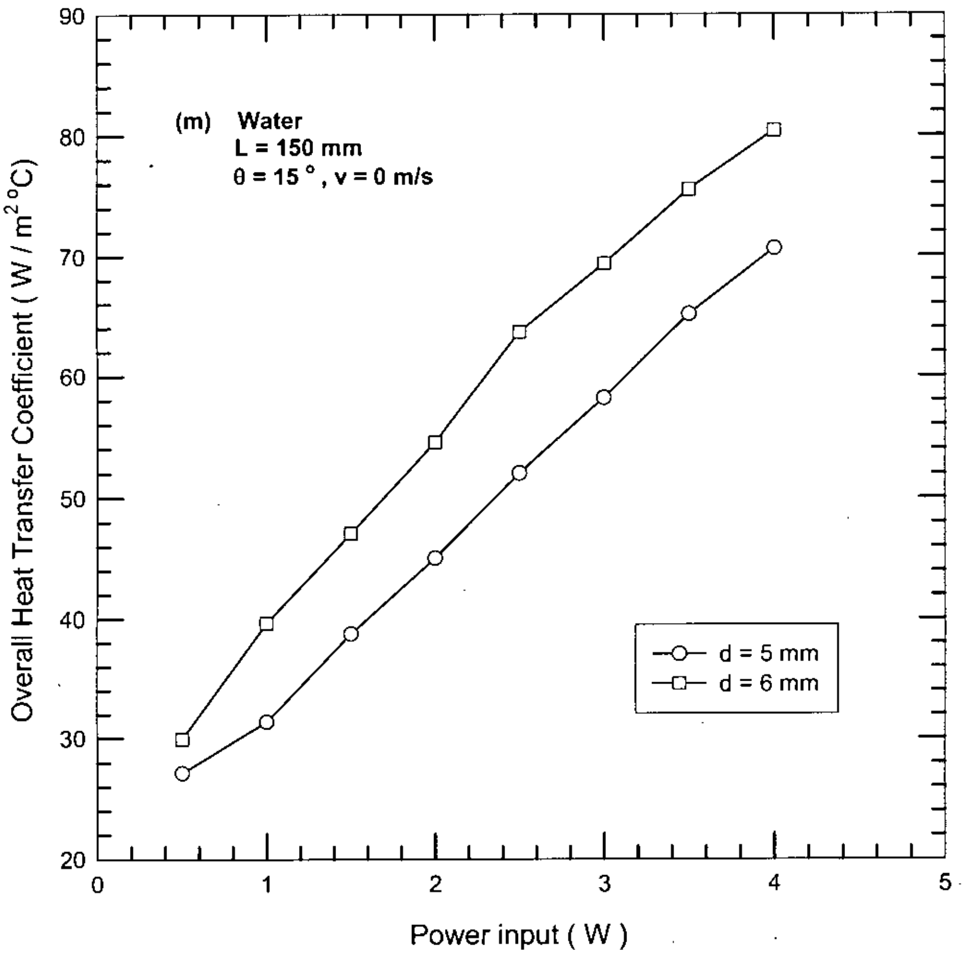

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

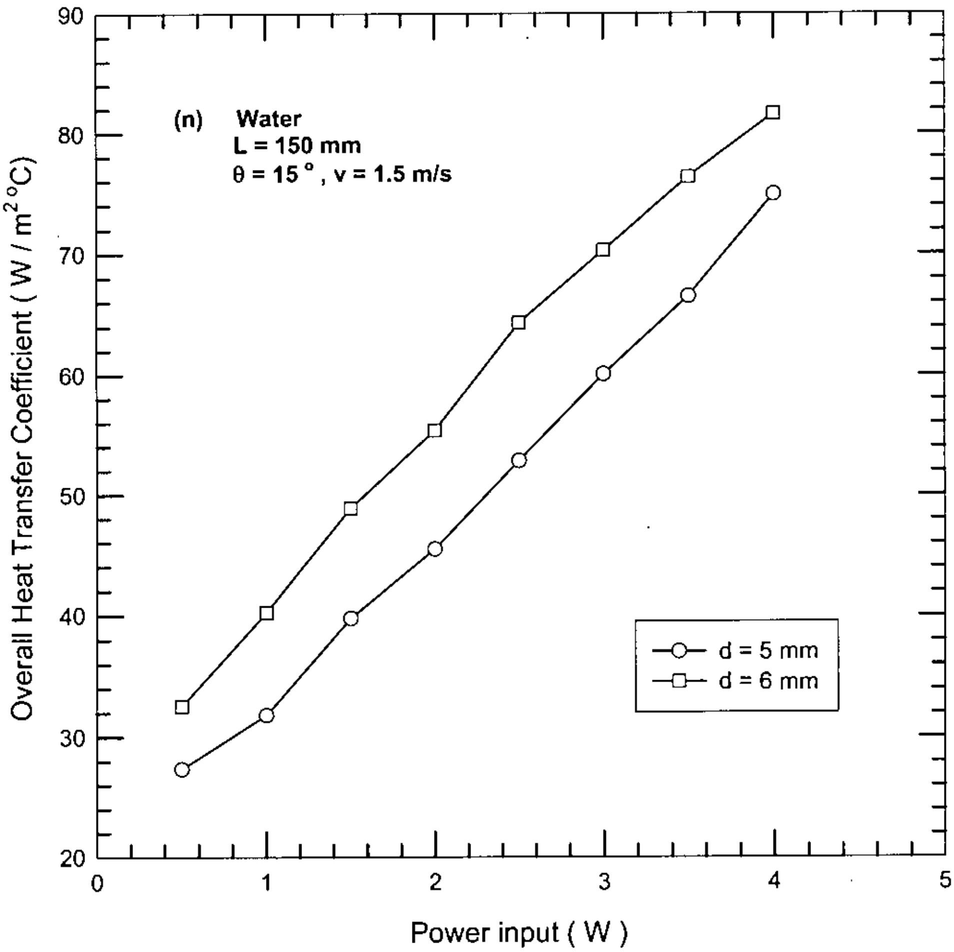

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

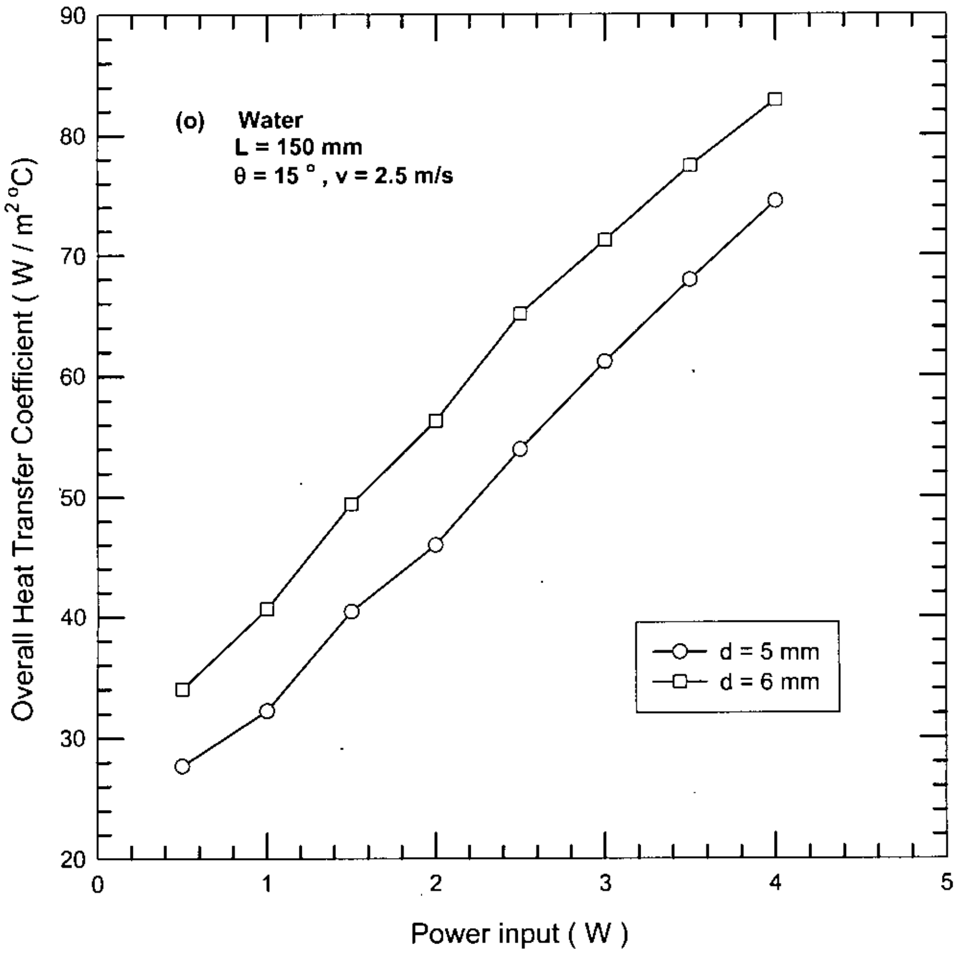

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

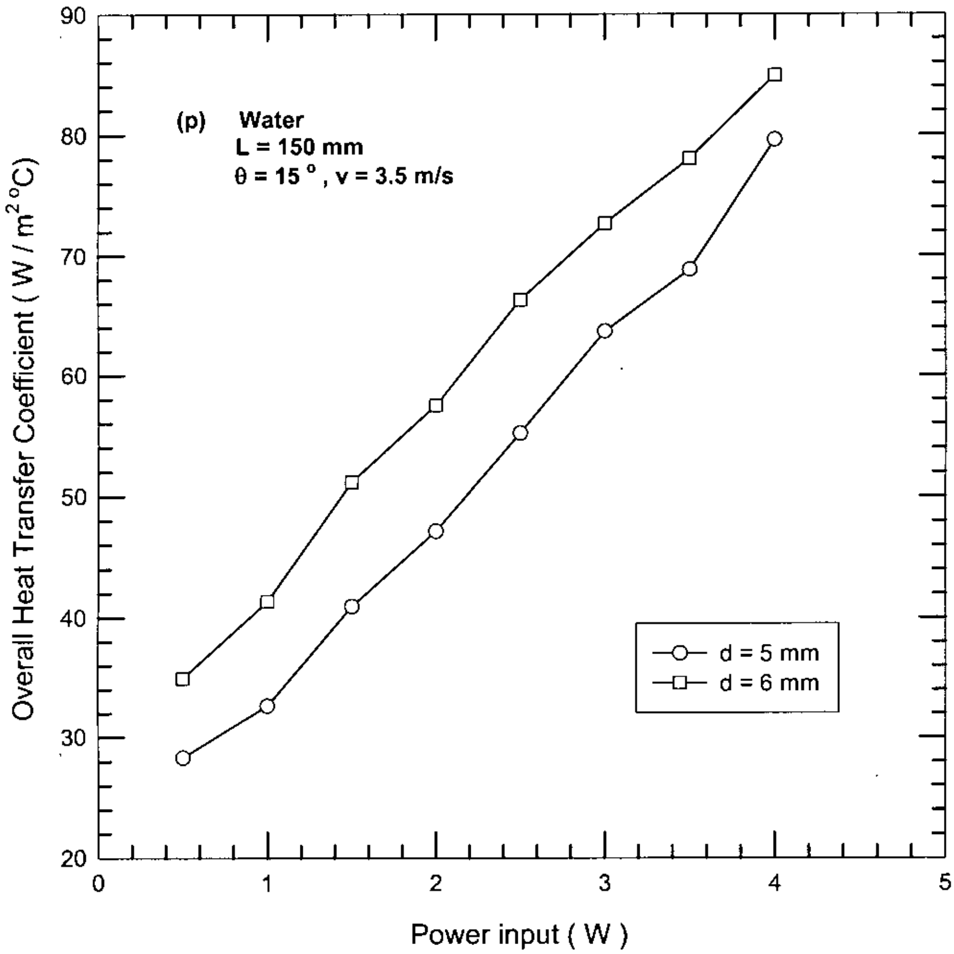

Fig. 6.5 Effect of diameter of heat pipe on overall heat transfer coefficient (continued)

CHAPTER 7

7. CONCLUSIONS

The results of the performance test for a miniature heat pipe having the diameter of 5 and 6 mm and length of 150 mm for the purpose of applying to cooling of miniature electronic parts are summarised as follows:

a. At a particular heat input the wall temperature of the evaporator section is higher when the heat pipe is placed horizontally and the wall temperature of the evaporator section decreases as the inclination angle increases. This implies that the action of gravity, which serves to speed up the flow of liquid from condenser to evaporator, increases with increase in inclination angle.

b. The thermal resistance decreases with the increase in airflow for cooling at the condenser. No significant change in thermal resistance during forced convection cooling of v_{air} = 1.5 m/s or faster within the stable zone where no dry out occurs. Thermal resistance also decreases with the increase in thermal load.

c. The inclination angle and diameter of the heat pipe has definite influence on the overall heat transfer coefficient. The experimental results show that the overall heat transfer coefficient increases as the diameter of the heat pipe and the inclination angle increases. As the diameter of the heat pipe is increased from 5 mm to 6 mm the overall heat transfer coefficient is increased 2 to 6 W/m^2 °C with increase in heat input.

CHAPTER 8

8. RECOMMENDATIONS

- Further investigation can be carried out on miniature heat pipe by using different fluids e.g methanol, ammonia etc.
- An experimental investigation can be carried out on miniature heat pipe by increasing angle of inclination up to 90°.
- A comprehensive investigation can be carried out on miniature heat pipe by reducing charge ratio and using wick.

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