Design and Development of innovative system of heat extraction using sintered copper heat pipes

#1Mr. Suhas Ramchandra Pawar, #2Prof. C. Shriramshatri, #3Prof Vivekanand Navadagi

[1]PG Student, Department of Mechanical Engineering, DPCOE, Wagholi, Pune
[2]Associate Professor, Department of Mechanical Engineering, DPCOE, Wagholi, Pune
[3]Assistant Professor, Department of Mechanical Engineering, DPCOE, Wagholi, Pune

Abstract:-Now a day's research in heat transfer area has reached so high, that even heat pipe can be used as a heat exchanger medium. In most of the application design modern age uses heat pipe as a heat exchanging medium. Uses of heat pipe as a heat exchanging medium has been made possible with uses of capillary action and most importantly phase change heat transfer principle. There are lot of researchers are working on this and many recent techniques have come for the heat pipe e.gnano fluids etc. In this paper author has presented a new technique where sintered cu used as a heat exchanging medium in heat pipes. Day by day use of heat pipes are increasing in extensively. Major application of heat pipes electronic devices reason behind this is high performance. The effect of feeling a heat pipe with sintered to study thermal efficiency increases experimentally. The nano fluid while performing this studies was copper nano particles or sintered copper of various sizes. Experiment done here keeping two aims in mind to calculate the thermal efficiency and temperature distribution of heat pipe along the surface of sintered copper using nano fluid under different levels.

Heat pipes have gain lot of importance in today's world. As it is very efficient and reduces the size of the equipment, which is today's need. There is various numbers of applications, where heat pipes can be used in between temperature range of 450 degree Kelvin to 750 degree Kelvin, this application may include nuclear power system radiator, geothermal energy, fuel cells, electronic gadgets and waste heat recovery systems. Since last decade, many researchers are working on the extension of the temperature range to 550 degrees Kelvin with water has been used as a coolant medium.

This paper discusses the heat pipe applications in brief and also identifies the research deficiencies in construction of typical heat pipes. The paper basically is result of intensive efforts to design an efficient heat pipe, its proto typing and analysis by experimentation is also carried out which fulfills the needs of development of new heat pipe design. First part of paper reviews the important works carried out by various scholars. Then the next part discusses the research gaps which gives us motivation to work on geometry, design and material which are most important factors to consider.

Finally paper discusses the test rig developed for the same and observations and results obtained are discussed.

KEYWORDS: Heat pipes, sintered copper heat pipes, Variable Conductance Heat Pipes, thermal conductivity, nano particles, sintered copper etc.

I. INTRODUCTION

The involution of heat pipes is been especially for space craft's applications done by NASA in the early 1960. Major problem of any space craft is to convection of the heat from space craft to outside world, the major difficulty is heat transfer in vacuum is very less. Hence the need of innovation has arise for transfer of heat from inside of space craft to outside and this was the main reason for innovation of heat pipes, and another major issue is this task is to be accomplished without any specific gravity. The major research was transferred towards the transfer of heat by convection method rather than conduction method as it found to be very effective type of heat transfer. In today's world heat pipes used in several application like cell phone, personal computers, laptops and solar collectors etc. Scientists are working towards a development of micro heat pipes, so that it can be used in small electronics gadgets or size of the electronic gadgets can come down by usage of effective heat transfer media.

II. LITERATURE REVIEW

William G. Anderson, John R. Hartenstine, and Christopher J. Peters: Variable conductance heat pipes (VCHPs) is used for spacecraft thermal control. A typical another application of VCHPs is as variable thermal link for lunar rovers and landers advantage of this is, it also minimizes the required electrical power[1]. In the long lunar day, VHCP are used for removing extra heat from electronics gadgets and batteries. A VCHP was introduced in aircraft system to develop an platform for heat transfer for lunar rovers and landers, by doing this we can reduce the heat losses during the lunar night.

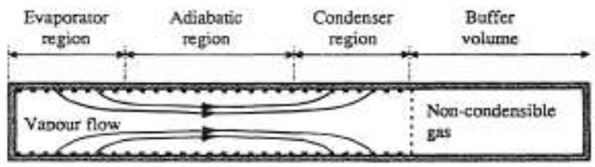


Fig. No.1. Typical Variable Conductance Heat Pipes

In fig no.1.a typical Variable Conductance Heat Pipe (VCHP), has an evaporator, a single condenser, and an electrically heated reservoir at the end of the condenser. This type of system is mostly used for temperature control in spacecraft and it gives a precision of ±1-2°C temperature control over varying powers and temperature of this sink. VCHP was developed to avail thermal link between lunar rovers and landers, for reducing the heat losses in the lunar night without shutting off electrical power. There are few differences in space craft, if VHCP included over conventional method that are as follows

1. This allows operation of evaporator in adverse condition of up to 14°.
2. The usage of VHCP reduces the need of electrical heaters as reservoir located next to evaporator.
3. This helps reducing leakage of heat during the lunar night.

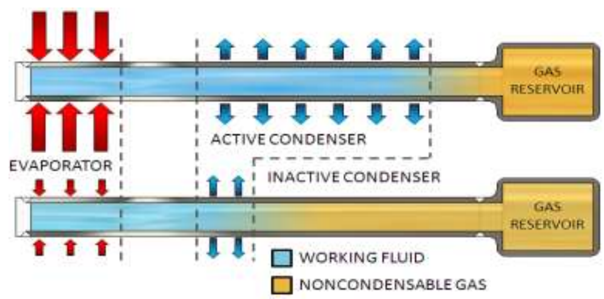


Fig. No.2. Detailed structure of VHCP with location of each component

In this paper author simulated lunar performance testing and proved VCHP shut off as the temperature of the condenser reaches defined lower value.

Dan Pounds1, Richard W. Bonner : Today we all know the efficiency of LED, and LED will evolve one of the most powerful practice to me implemented for energy saving all over the world. Heat management of LED lights is becoming major issue, mainly because reduction of size. This issue is very important while designing high power LED system. Where the size and weight of the systems is increases due to size of the heat sink. The forced cooling methods such as forced air cooling may increase an heat dissipation but at the cost of extra energy consumption [2]. After application of heat pipes solid state lighting for extra efficiency, for these adapting generated heat pipes technologies is implemented.

	Incandescent (%)	**Fluorescent (%)**	**Metal halide (%)**	**LED lights (%)**
Visible Light	8	21	27	20-30
IR	73	37	17	0
UV	0	0	19	0
Total radiant energy	81	58	63	20-30
Non radiant energy	19	42	37	70-80
Total Energy	100	100	100	100

Table No.1. Power conversion percentage of white light sources

In this paper feasibility of high heat flux capable heat pipe embedded in a metal fixture concept in adopted for high power LED lights and it is demonstrated using numerical method as well as experimental study. The MCPCB was made up of aluminum and copper material is used for heat pipes and they are combined together using solder. The copper heat pipes used while carrying out this experiment was using water as a cooling agent. In this paper, author implemented three different designs for heat dissipation of

LED lights. Out of these three designs only two were selected for thermal evaluation based on fabrication method easiness. For prediction of thermal resistance CFD is used and it was verified experimentally. Based on experiment MCPCB 35-45% lower thermal resistance as compared to conventional method.

Darren Campo, Jens Weyant, BryanMuzyka: Military application needs an very high rate of heat dissipation especially in avionics and if heat transfer is not good then it may affect the performance of aircrafts. This leads to innovation of heat pipes in aviation industry for warfighter planes. In most of the applications researchers have molded their view from liquid cooling to air cooling as it is beneficial to chassis for both environmental and thermal benefits. There is advantage of using liquid cooling in electronic instrument but it introduces the risk of leakage of liquid, particularly with replacable type of units[3]. Only because of this disadvantage liquid cooling is avoided in most of electronic applications.

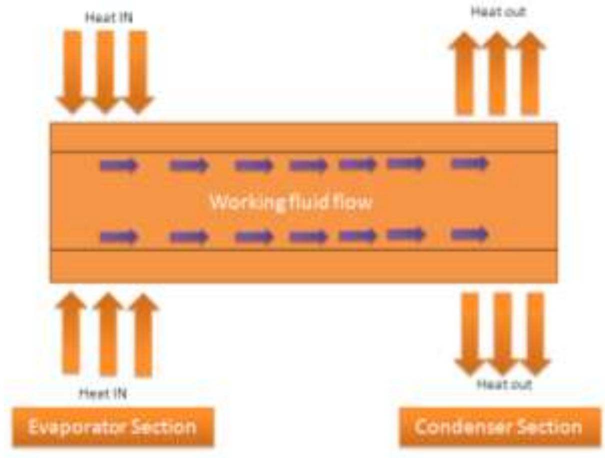


Fig.3. Heat Pipe Operation

In the aircraft or warfighter first area for improving heat dissipation is the heat conduction card. Heat pipes can be in the conventional type of conduction card to improve the heat dissipation and reducing thermal gradients and thermal transfer has been taking place between electronics to wedge lock connection at the end of card cage. At the conclusion of this paper it is written that, heat pipes give multiple methods for increasing thermal performance.

CalinTarau, Carl Schwendeman, William G. Anderson, Peggy A. Cornell,Nicholas A. Schifer: In this paper stirling radioisotope power system(RPS) is taken for experimental analysis, in this case heat must be taken out continuously from the general purpose heat source (GPHS) modules cool and provide insulation at acceptable temperatures. Providing heat exchanging media to stirling converter, if proper cooling is not provided to stirling converter then insulation may get spoil. The incorporation of VCHP will facilitate the refrigeration system to

1. If temperature lower than nominal it will take rest.
2. Pre cooling of modules is possible, it facilitates lower temperature before entering atmosphere of venus.
3. It works on nominal temperature on venus

In this paper advanced stirling radioisotope generator (ASRG's) are used, as this are very attractive energy function for very few space missions and if VCHP is incorporated they become versatile. In this paper conceptual VCHP is implemented for a stirling convertor has four critical components, first condenser, evaporator , second condenser and a non-condensable gas reservoir. NCG is connected between starting of the front and ending in the reservoir. The work of VCHP evaporator is to interface between GPHS modules and transfer heat energy to any of the condenser. In this case especially of aircraft stirling heater head is connected to first condenser and it is active only if stirling converter is operating. The second condenser connected between heater head and NCG reservoir and it will be active when stirling converter is off. The NCG reservoir is located after assembly of second condenser [4]. In this paper VCHP was tested for two different modules, the ASRG and used backup cooling concept and in this paper venus lander concept has been implemented for four feature concept.

William G. Anderson, SanjidaTamanna, CalinTarau, and John R. Hartenstine: In various application we use heat pipes or loop heat pipes (LHPs) in the temperature range of 450degree Kelvin to 750 degree Kelvin, including space nuclear power system radiators, geothermal power ,fuel cells waste heat recovery systems for high temperature electronics cooling. The intermediate temperature cooling is normally is between 450degree Kelvin to 750 degree Kelvin and above 700 degree Kelvin, alkali metal(cesium) heat pipes are used as they give effective cooling. Many researcher has proved that water can be used with titanium or monel envelops, and they have found effective for temperature of around 500 degree Kelvin. As on today there is no practically viable method for complete intermediate temperature range. Various researchers have given list of probable material can be used for intermediate temperature range such as organic compounds, sulphur and halide[5].

Research gap

Heat exchanger are device that transfer heat in order to achieve desired heating or cooling & important design aspect of heat exchanger technology is the selection of appropriate materials to conduct & transfer heat fast & efficiently.

Copper tube heat exchanger technology developed specially for applications that need to withstand harsh conditions. The technology is particularly amenable for higher temperature & pressure environments required in cleaner diesel engines that are being mandated by global environmental regulations.

Microgroove-It is small diameter coil technology smaller diameter coils have better rates of heat transfer than conventional sized coils & they can withstand higher pressure required by new generation of environmental friendlier refrigerant.

Smaller diameter have low material cost because they require less refrigerant fin & coil materials & they enable the design of smaller & lighter high efficiency air conditioners & refrigerators because evaporators & condensers coils are smaller & lighter.

2.1 Characterization of sintered wick

The sintered wick of heat pipe is fabricated by single-component system loose solid-state sintering method. The sintering principle for sintered wick of miniature cylindrical heat pipe is shown in Fig. 1. A pipe with one shrunk end is used as the outside wall, a stainless steel bar is insert into the pipe as the inner mandrel. Powders are then filled into the gap between wall and mandrel. The pipes with full filled powders are placed vertically in the furnace and sintered at appropriate temperature. Finally, the powders are sintered together as the sintered wick on inner pipe wall. During the sintering process, the atoms migrate between the contact powders and metal powders are then fused, the sintering necks are formed, and the bonding strength of sintered wick is improved. The effective thermal conductivity and volume shrinkage of sintered wick increase, while the porosity of sintered wick decreases. In order to better analyze sintering process of sintered heat pipe, the parameters like sintering neck, shrinkage and porosity should be firstly characterized.

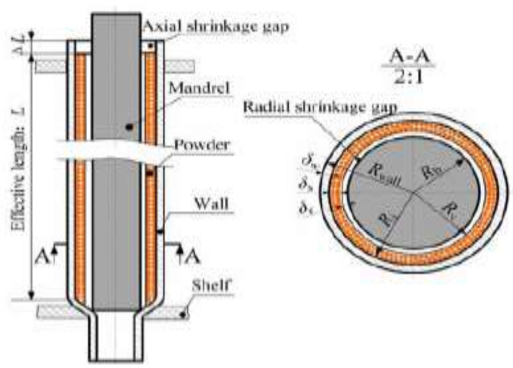


Fig3. Sintering principle of sintered wick

2.1.1 Sintering neck growth rate

Sintering neck formation can be analyzed by the ball–ball sintering model as shown in Fig.The sintering neck growth equation can be expressed by

$$(\eta)^n = \left(\frac{x}{r_{Cu}}\right)^n = \frac{F(T_S)\mathrm{t_S}}{r_{Cu}^m} \qquad \ldots\ldots\ldots (1)$$

Where n and m are the sintering mechanism characteristic parameters;
Ts is the sintering temperature;
$F(T$s) is the sintering temperature function;
ts is the sintering time;
x is the length of sintering neck and
r cu is the copper powder radius.

Neck growth ratio η is defined as a neck dimension divided by the powder diameter. According to Eq. (1), the sintering growth rate is a function of sintering time, sintering temperature and sintering powder radius. The experimental powder radius r Cu always fluctuates in a considerable range and can be calculated by the mean value of two bonded powders radius. Thus, the sintering growth rate can be expressed as

$$\eta = \frac{x}{r_{ave}} = \frac{2x}{r_{Cu1}+r_{Cu2}} \qquad \ldots\ldots\ldots (2)$$

where $r$ave is the average radius of copper powders.

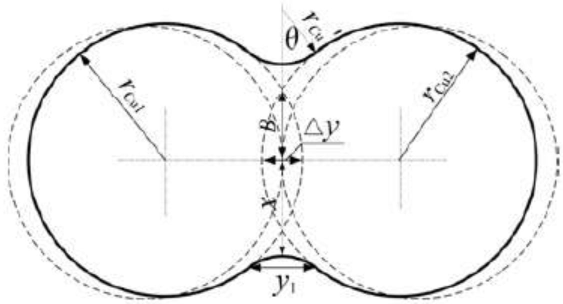


Fig. 4 Ball−ball sintering model2.1.2 Shrinkage

Wick shrinkage greatly reduces wick porosity which affects the thermal performance of sintered heat pipe. According to shrinkage direction, the wick shrinkage can be classified into axial shrinkage and radial shrinkage.
The axial wick shrinkage gap can be observed at the upper end of pipe and can be defined as

$$\lambda_a = \frac{\Delta L}{L+\Delta L} \qquad \text{…………………} \qquad (3)$$

where λ_a is the axial shrinkage rate of sintered wick;
ΔL and L are the axial shrinkage length and total length of sintered wick, respectively. The powders are easily sintered on the inner surface of pipe wall and difficultly bonded on the mandrel. Therefore, the radial shrinkage gap always occurs between mandrel and sintered wick. The radial shrinkage rate λr can be defined as

$$\lambda_r = \frac{\delta_r}{\delta_r+\delta_s} \qquad \text{…………} \qquad (4)$$

where δr and δs are the shrinkage gap thickness and sintered wick thickness, respectively.

2.1.3 Porosity

The porosity of sintered wick has a profound influence on the maximum heat transfer rate according to the working principle of heat pipe. A lot of measuring methods, such as Archimedes method based on imbibitions principle , density method ,and soaking method ,can be used to test the porosity of sintered wick. In the present work, density method is conducted to calculate porosity of sintered wick ε due to its simplicity and effectiveness as follows:

$$\varepsilon = 1 - \frac{\rho_w}{\rho_{Cu}} = 1 - \frac{1}{\rho_{Cu}}\left[\frac{m_w}{V_W}\right] \quad \text{………………} (5)$$

where ρ_{Cu} is the density of pure copper which is equal to
8993 kg/m3; ρ_ws the density of sintered wick; m_w mass of sintered wick; Vw is the total volume of sintered wick.
The mass lose during the sintering process is minimal and ignored. The relationship between porosity and shrinkage rate of sintered wick can be expressed as

$$\varepsilon = 1 - \frac{(R_i^2 - R_b^2)(1-\varepsilon_0)}{(1-\lambda_a)\left[R_i^2 - (R_b - \lambda_r R_b + \lambda_r)^2\right]} \quad \textbf{.............} (6)$$

Ri is the radius of inner wall; Rb is the radius of mandrel; and εis the porosity of filled powder before sintering.

2.2 Heat transfer limit of sintered heat pipe

The heat transfer capability of heat pipe can determine by several limits, such as capillary limit, Boiling limit, entrainment limit, sonic limit, viscous limit . For copper−water cylindrical heat pipes generally operating from 30 to 100 °C, the boiling limit and capillary limit govern the thermal performance of heat pipe . Therefore, the heat transfer limit of sintered heat pipe is determined by the minimum value of boiling limit *Q*b,max and capillary limit *Q*cap,max.
Thus, the maximum heat transfer capability of sintered heat pipe *Q*max can be given as follows:
Qmax = min{Qb,max ,Qcap,max} **……….**(7)

2.2.1 Capillary limit

Wick structure provides necessary flow path and capillary pumping force to return the liquid from the
condenser to the evaporator. For a given wick structure, it has a maximum value of capillary pumping force. If the sum of pressure drop along the fluid circulation in the sintered heat pipe is larger than the maximum value of capillary pumping force, the liquid−vapor interface would recede to reach a new pressure balance. If the new balance cannot be reached, heat pipe would be dry

out due to no sufficient liquid returning to evaporate. So the sintered heat pipe should have enough capillary pumping force to maintain the continuity of interfacial evaporation. The maximum capillary limit can be calculated as the sintered heat pipe works along the horizontal direction by the following equation:

$$Q_{cap,max} = \frac{2\rho_1 \sigma_1 h_{fg} A_w K}{r_h L_{eff}\, \mu_1} \quad \ldots\ldots\ldots\ldots\ldots (8)$$

Where ρ_1 is the liquid density of working fluid; σ_1 is the surface tension of working fluid; h_{fg} is the fluid latent heat of vaporization; g is the gravitational acceleration; μl is the liquid viscosity; Aw is the cross area of sintered wick; rh is the effective pore radius of wick; Leff is the effective length of heat pipe; and K is the wick permeability.

2.2.2 Boiling limit

Heat transfer mechanism at the evaporator is heat conduction and evaporation. When the heat flux is sufficiently high, the liquid in the porous wick would be super heat and nucleate boiling may occur. The bubbles which are formed by nucleate boiling may be trapped in porous wick. The pressure of the bubbles would obstruct the working liquid circulation which causes hot spots on heated wall. The heat flux at which marks the onset of nucleation is defined as the boiling limit of sintered heat pipe. The boiling limit model can be expressed as follows:

$$Q_{b,max} = \frac{4\rho_1 \pi Le\, k_{1eff} T_v}{h_{fg} \rho_v r_b \ln_{(R_i/R_v)}} \quad \ldots\ldots\ldots\ldots\ldots (9)$$

where Le is the length of evaporator section;
$k$eff is the effective thermal conductivity of sintered wick;
Tv is the vapor temperature;
ρv is the density of vapor;
Ri is the outer radius of sintered wick which is equal to the inner radius of wall;
Rv is the vapor chamber radius of heat pipe.
Conservatively suggested the critical radius of the bubbles rb to be a value on the order of 25.4−254 nm for the copper water heat pipes. The theory of LORENZ et al [27] predicted that the first active nucleation sites were larger, so the bubble radius rb was taken the value 254 nm in this study.

2.3 Optimization design of sintered wick

In order to optimize the design of sintered wick structure, heat pipe is assumed to work at the vapor temperature of about 60 °C based on its general operating temperature applied in electronics cooling. The experimental and calculation parameters of sintered wick heat pipe are shown in Table 1. According to the calculation results of Eqs. (8) and (9), the heat transfer limit of the copper water sintered heat pipe is determined by the capillary limit.

2.3.1 Experimental and calculation parameters Cu sintered heat pipe

Table 1

Parameter	Value
Total length, L/mm	200
Evaporator length, Le/mm	35
Condenser length, Lc/mm	65
Outer radius of wall, Rwall/mm	4
Inner radius of wall, Ri/mm	3.7
Material	TP1
TP1thermal conductivity, ks/(W·m^2K^1)	407
Working fluid	Purified water
Operating temperature /K	333
Water thermal conductivity,$k_1/(W.m^2k^1)$	0.62
Latentheatof water, $h$fg/(J·kg−1)	2.358×10^{-6}
Visc. of purified water /$(N.s\,m^{-2})$	406.1×10^{-6}
Porosity of sintered wick	30%−60%
Thickness of sintered wick, δw/mm	0.3−1
Spherical copper powder size, d/μm	40−180

The effects of powder diameter, porosity and sintered wick thickness on heat transfer limit of heat pipe are vividly shown in Fig. 3. Heat transfer limit of heat pipe increases with sintered wick thickness in the range from 0.3 to 1 mm at a given porosity and powder diameter. However, fabrication cost, mass and thermal resistance of sintered heat pipe would also increase due to the increase of sintered wick thickness. Therefore, an optimum sintered wick thickness design should be as thin as possible under the condition of meeting the heat transfer capacity. As heat transfer limit of heat pipe also increases with porosity of sintered wick and powder diameter. The porosity of sintered wick is determined by the original porosity of filled copper powder $\varepsilon 0$, the radial shrinkage rate λr and the axial shrinkage rate λa after sintering. In a word, heat transfer limit of heat pipe is affected by powder diameter and shrinkage. Radial shrinkage rate increases with the sintering process which leads to the decrease of wick porosity.

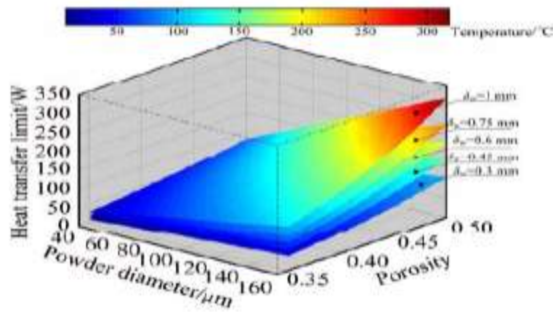


Fig.5 Effects of powder diameter, porosity and wick thickness on heat transfer limit

III. EXPERIMENTAL

3.1 Preparation

100 mesh, 200 mesh and 300 mesh spherical copper powders (Material: TP1, supplied by a Cu powder International, LLC, USA) made by gas atomization were screened as raw materials for sintered wick. The influence of powder diameter distribution on porosity of sintered wick should be considered. 100 mesh, 200 mesh and 300 copper powders were observed by SEM. Powder diameter was analyzed and powder diameter distribution was calculated by statistical method. The normal distributions of different size of powder diameter are vividly shown in Fig. 4. The mean diameters of 100 mesh, 200 mesh and 300 mesh copper powders are 159 μm, 81 μm and 38 μm respectively. The material of pipe wall was also copper (Material: TP1) due to its high thermal conductivity. Its geometric parameters are shown in Table 1.

The material of mandrel was stainless steel (Material: 310S) due to its high temperature resistance, high mechanical strength and non-sticking with copper powder at high temperature.

The diameter of mandrel mainly determines the sintered wick thickness at a given wall. Diameters of 6.8, 6.5, 6.2 and 5.9 mm and length of 250 mm stainless steel bars were fabricated as the mandrels.

Before the beginning of sintering process, a series of preparation processes as follows had to be done:

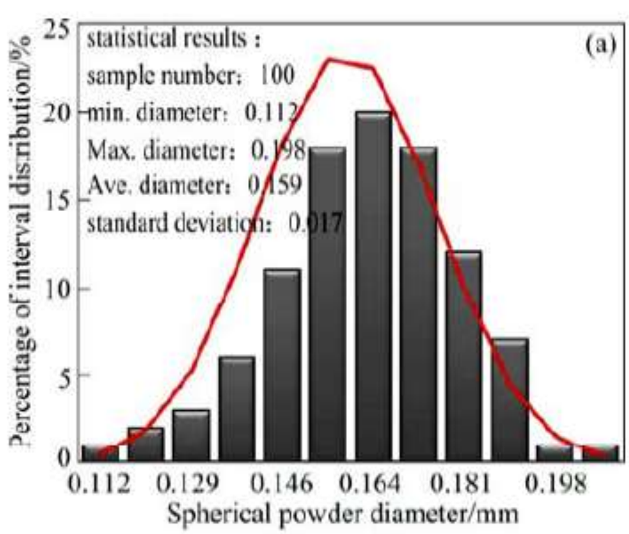
1) cut pipes with the length of 250 mm;

2) shrink one end of pipe to *d*5 mm with the length of 50 mm by the radial forging and swaging method;

3) clean pipe and stainless steel mandrel by ultrasonic cleaning machine;

4) fix mandrel in center of the pipe and fill copper powder into the gap between pipe and mandrel;

5) vertically place the filled pipe into graphite shelf and move the shelf in a sintering furnace



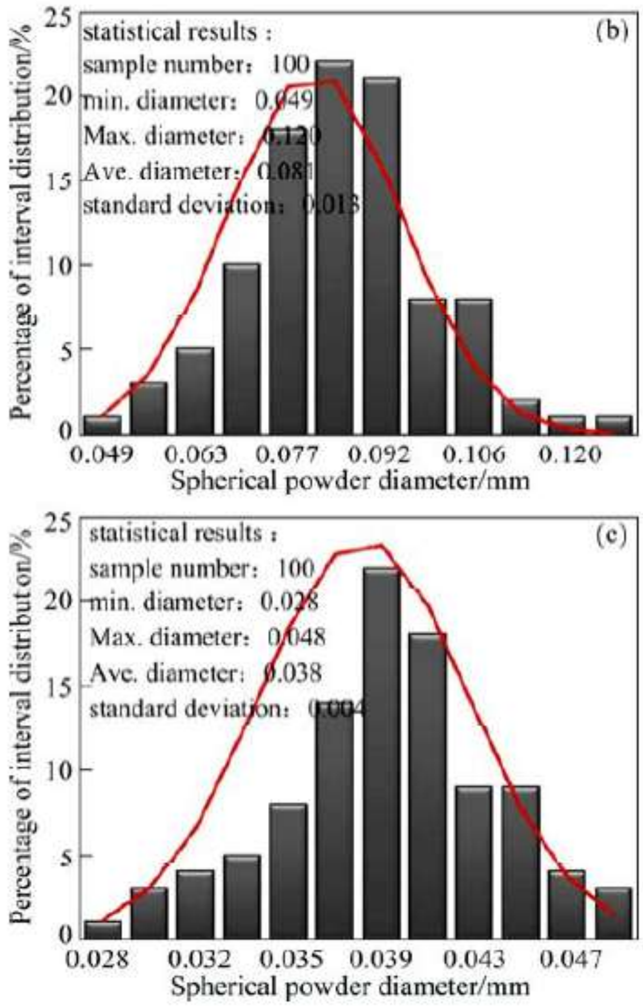


Fig. 4 Normal distribution of spherical copper powder:
(a) 100 mesh; (b) 200 mesh; (c) 300 mesh

3.2 Sintering process

A sintering process for sintered wick of heat pipe was proposed based on the powder sintering theory shown in Fig. 5.

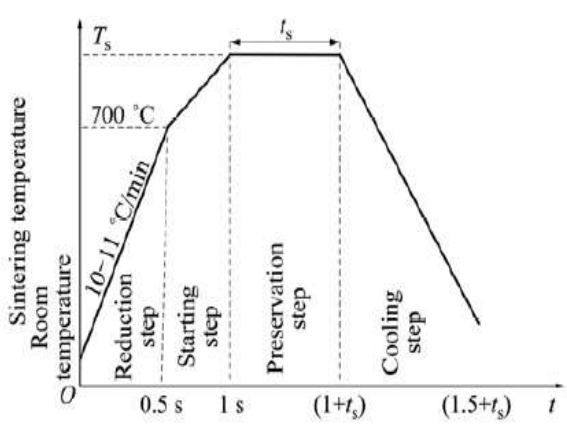


Fig. 5 Sintering process for sintered wick

The sintering process was carried out in the sintering furnace under gas protection atmosphere. The sintering process can be divided into 4 stages: reduction step, starting step, preservation step and cooling step

In reduction step, the pipes filled with copper powders placed in the furnace began to be sintered. Firstly, seal and pump the furnace at the vacuum degree of 0.1−1 Pa. Secondly, blow 90% protecting gas N2 and 10% reducing gas H2 into the furnace till the pressure of atmosphere reached about 0.3 MPa. When the furnace

temperature was about 350 °C, exhaust the atmosphere and pump in 100% protecting gas N2. The temperature increase rate was set at 1000 °C/h till the furnace temperature reached 700 °C. During the reduction process, the oxide layer of copper powders and pipe was reduced into steam. Steam and impurity gases released from organic compounds were exhausted. The reduction process guaranteed the golden yellow surface quality of sintered wick. In starting step, thermal inertia of furnace increased with heating rate which could lead to furnace temperature far over the sintering temperature. Thus, the furnace heating rate decreased from 1000 to 500 °C/h when the furnace temperature reached 700 °C. When the furnace temperature was beyond 700 °C, the copper powder began to stick together, the sintering neck formed, the strength of the sintered wick increased, but
the morphology of the sintered wick almost remained unchanged.
In preservation step, the furnace was heated at a fixed temperature for a holding time. The holding temperature and holding time were defined as the sintering temperature and sintering time, respectively.
During the preservation phase, the sintered necks grew, the center distance of adjacent powders decreased, and the shrinkage rate, the effective capillary pore radius and the bonding strength of sintered wick increased. It is the key step of sintering process which decides property of sintered wick.
In cooling step, the furnace temperature was cooled
by its internal recycling N2 gas at the cooling rate of 600°C/h. The hardness of sintered pipe was affected by the cooling rate and the tapping temperature of sintered pipe. If the tapping temperature was high, the sintered pipe may be oxidized. The appropriate tapping temperature was about 120 °C.

IV. WORKING PRINCIPLE

A heat pipe is essentially a passive heat transfer device with an extremely high effective thermal conductivity. The two-phase heat transfer mechanism results in heat transfer capabilities from one hundred to several thousand times that of an equivalent piece of copper.

The heat pipe in its simplest configuration is a closed, evacuated cylindrical vessel with the internal walls lined with a capillary structure or wick that is saturated with a working fluid. Since the heat pipe is evacuated and then charged with the working fluid prior to being sealed, the internal pressure is set by the vapor pressure of the fluid.

As heat is input at the evaporator, fluid is vaporized, creating a pressure gradient in the pipe. This pressure gradient forces the vapor to flow along to a cooler section where it condenses giving up its latent heat of vaporization. The working fluid is then returned to the evaporator by the capillary forces developed in the wick structure.

Heat pipes can be designed to operate over a very broad range of temperatures from cryogenic (< -243°C) applications utilizing titaniumalloy/nitrogen heat pipes, to high temperature applications (>2000°C)using tungsten/silver heat pipes. In electronic cooling applications where it is desirable to maintain junction temperatures below 125-150°C, copper/water heat pipes are typically used. Copper/methanol heat pipes are used if the application requires heat pipe operation below 0°C.

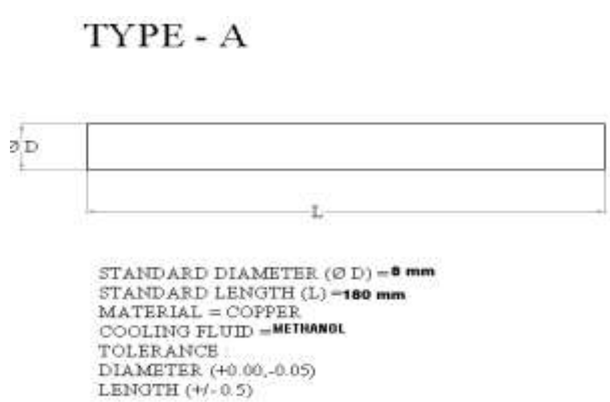
V. DESIGN & DEVELOPMENT

There are many factors to consider when designing a heat pipe: compatibility of materials, operating temperature range, diameter, power limitations, thermal resistances, and operating orientation. However, the design issues are reduced to two major considerations by limiting the selection to copper/water heat pipes for cooling electronics. These considerations are the amount of power the heat pipe is capable of carrying and its effective thermal resistance. These two major heat pipe design criteria are discussed below.

Heat pipe geometry- size selection

Heat pipes are available in standard diameters from 3 to 12mm and in lengths from 50mm to 250 mm, shape be as shown in figure below:

Heat transfer capability for above heat pipe



Maximum watts at different temperature

DIAMETER	40^0C	60^0C
8 mm	67.6 WATT	135

- The power handling figures are for heat pipe working in horizontal position. –Straight pipe
- Length 180 mm long
- evaporator length 104 mm
- condenser length 76 mm
- Sintered copper powder---wick structure

VI. JUSTIFICATION FOR SELECTION OF VARIOUS PARAMETERS OF HEAT PIPE SELECTION:

Material of body of heat pipe

Copper is selected as material of body for three main reasons:
a) Copper gives best thermal conductivity as compared to other heat pipe materials such as aluminium or stainless steel.

b) Copper material can be sealed easily using brazing technology …which is reliable and cost effective as compared to fusion weld in aluminum and stainless steel.

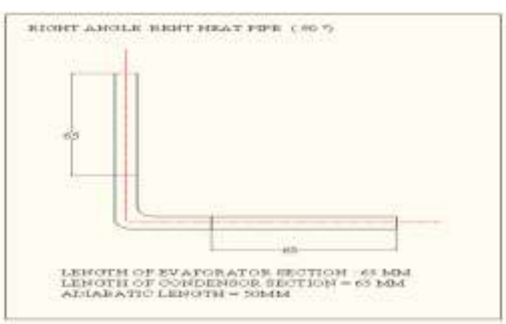
c. In case of bent heat pipes the technology of bending the heat pipe is done only after the production of straight heat pipe …i.e. straight length of heat pipe , in our case 180 mm is made first and then the pipes are bent to required degree using bending machine ---three roller adjustable mechanism to give desired shape and profile…..hence the tube material has to be ductile for easy rolling without any cracks developed at the surfaces ,…thereby preventing any leakages and loss of thermal conductivity properties.

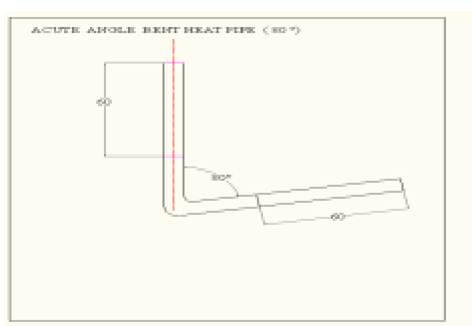
VII. PROCESS SHEET

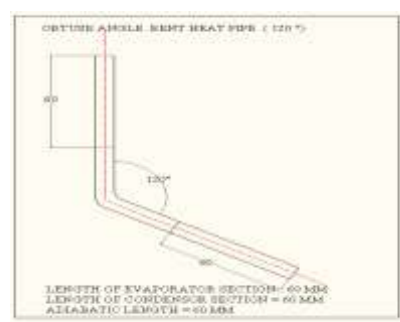
Geometry selection of heat pipes

With reference to IC349MU08 ---Application of heat pipes for cooling in automobiles The angle of bend selected were 30 degrees „ 45 degrees and 60 degrees …these are specific angles for given applications….here study was done for automotive roof cooling…more over the applications that we propose are related to cooling of electronic equipment such as portables , printing machinery ..etc here the angle of bend normally required after market survey were found to be 90 degrees …120 degrees and acute angles in minimal cases though just between 80 to 70 degrees ---for printing machines adjust roller circuit cooling….hence according to requirement the angle of bends have been selected and more over the selection of bend as it depends upon the geometrical shape of component to be cooled so, also it is governed by the machinery available for bending.

Bending of heat pipes below 70 degree is not possible because if three roller design ie, commonly used for bending …it is not possible to retract the form roller after bend…hence 70 degree acute angle is selected….as far 90 degrees and 120 degrees ..They are the most popular configuration in market due to easy accessibility and replaceabitlity and availability in local market







VIII. TEST RIG SET UP

Description of test rig:

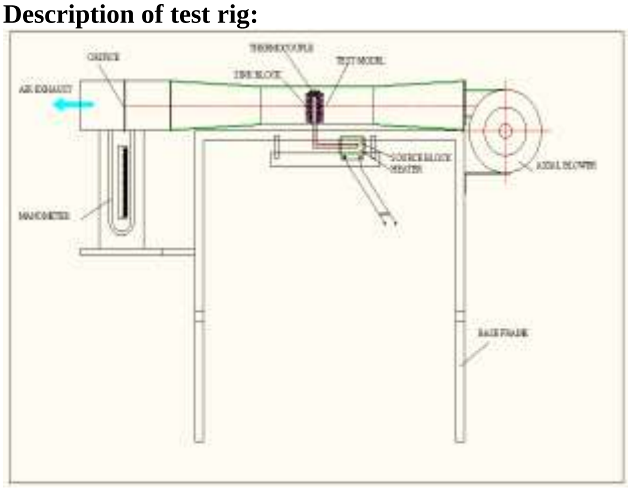


Fig. 6 Test Rig Setup

The test rig comprises of the following arrangements:
a) Axial blower.
b) Air duct with venturi action
c) Air discharge measurement set-up
d) Temperature measurement & display J Type Probe
e) Heater arrangements as heat source at bottom
f) Interchangeable heat pipe fixtures:

Blower specification:

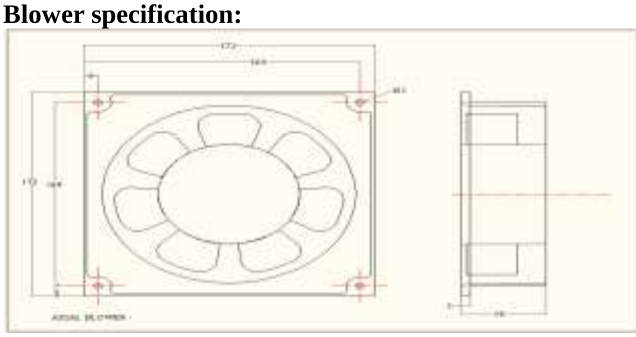


Fig.7 Blower specification

SINGLE PHASE AC- AXIAL BLOWER
VOLTAGE: 230 V
CURRENT : 2 Amps
SPEED = 0 TO 3000 VARIABLE
Speed of the blower and thereby the discharge can be controlled using an electronic speed regulator.

IX. METHODOLOGY

a) Procedure of trial
1. Start the heater by switching on the power
a) Measure voltage (V- volts) using voltmeter
b) Measure current (I –amps) using Ammeter
Calculate Power input (P) = V x I
2. Start the blower and adjust and maintain speed at level-1
a) Take manometer reading (h) mm of water
Using this reading it is possible to find velocity of air and discharge at level-1 of speed of blower.
3. Note down thermocouple temperature reading (Tf)
Gradient of temperature (ΔT) = Tf- Tambient)
4. Repeat the same set of reading for different levels of manometer readings i.e., level-2.
Level-3. Level-4, level-5 and level-6

OBSERVATION TABLE2

SR .No.	VOLTAGE (VOLT)	Time T min	CURRENT (Amps)	MANOMETER READING (h2) (CM)	DUCT AIR TEMP. Tf^0 C	AMBIENT TEMP. Ta ^{0} C
Level-1	220	3	1.0	0.5	58	29
Level-2	225	3	1.1	0.56	61.3	28
Level-3	224	3	1.1	0.61	63.2	29
Level-4	226	3	1.0	0.67	64.8	28
Level-5	221	3	1.2	0.73	66.2	29
Level-6	220	3	1.1	0.79	67.2	28

9.1 calculations:
Sample Calculation:
Mass flow rate =0.0148 kg/sec,
Specific heat of water =4180 J/kg*k
QIN =107.8 W,
TE =390C,
TC= 350C,
TIN =250C, TOUT = 26 0C,
QOUT =m*cp*(TOUT-TIN)
= 0.0148*4180*(26-21) = 61.864 W,
%ŋ = QOUT÷ QIN = (61.864÷107.8) ,
Thermal resistance
RTH= ((TE- TC) ÷ QIN) in (0C / W)
= ((39-35) ÷ 107.8) = 0.03710C / W

Discharge –

$$Q = Cd\frac{\pi}{4}X\sqrt{2g\left(H.\frac{Q_W}{Q_a}\right)} \quad \frac{\mathrm{m}^3}{\mathrm{sec}}$$

Where H= Difference of levels in manometer in m

$$Q_W = Density\ of\ water = 1000\ Kg/m^3$$
$$Q_a = Density\ of\ air = 1.275\ Kg/m^3$$

Q = 0.6 x (π /4) x 0.0165^2 x sq. rt (2 x 9.81 x 0.005 x (1000/1.275))

Q=0.00112 m3/sec

Q = A x V
V= Q/ A = 5.262 m/sec

2. Heat transfer through het pipe:
Q = h A (ΔT)
Where;
Q = Heat transferred (watt)
U = Overall heat transfer coefficient of the fin material
H= heat transfer coefficient
A=Area of fins =$0.28m^2$----as per geometry of fin
For laminar flow

h = 0.64 Re $^{1/2}$(μCp/K) $^{1/3}$ (K/L)
Where,
Re = V/v …. *(ASSUMING FIN LENGTH (L) =0.96 = 1M)*
Where V= velocity of air m/sec
v= kinematic viscosity of air = 20 x 10^{-6} m^2/sec
Re = 5.226 / 20 x10^{-6}
Re =261300
Cp = specific heat of air = 1.005 kJ/kgk
K=0.027 W/m-k
μ-=7.344 x 10^{-10}kg.m/sec----calculated from Prandtl number—0.701
Pr= μCp/K
L=length of fin=length of fiv x no of fins) = 0.96m

h = 0.64 x $261300^{1/2}$ (7.344 x 10^{-10} x 1.005 / 0.027) $^{1/3}$ x (0.027 / 0.96)
h= 0.49

Q = h A (ΔT)
= 0.495 x 0.34 x (58-29)
= 0.495 x 0.34 x (29)
Q= 4.887 watt

Effectiveness of fin = Heat output / Heat input =

Heat input = (V x I / 60) x time (min) or(Heater rating / 60) x time (min)
= 220 x 1/60 x 3 =11watt
Effectiveness (ε) = 4.887 /11

X. RESULTS AND DISCUSSION

10.1Limits to Heat Transport

The most important heat pipe design consideration is the amount of power the heat pipe is capable of transferring. Heat pipes can be designed to carry a few watts or several kilowatts, depending on the application. Heat pipes can transfer much higher powers for a given temperature gradient than even the best metallic conductors. If driven beyond its capacity, however, the effective thermal conductivity of the heat pipe will be significantly reduced.

Heat Transport Limit	Description	Cause	Potential Solution
Viscous	Viscous forces prevent vapor flow in the heat pipe	Heat pipe operating below recommended operating temperature	Increase heat pipe operating temperature or find alternative working fluid
Sonic	Vapor flow reaches sonic velocity when exiting heat pipe evaporator resulting in a constant heat pipe transport power and large temperature gradients	Power/temperature combination, too much power at low operating temperature	This is typically only a problem at start-up. The heat pipe will carry a set power and the large ^T will self correct as the heat pipe warms up
Entrainment/ Flooding	High velocity vapor flow prevents condensate from returning to evaporator	Heat pipe operating above designed power input or at too low an operating temperature	Increase vapor space diameter or operating temperature
Capillary	Sum of gravitational, liquid and vapor flow pressure drop and the capillary pumping head of the heat pipe wick structure	Heat pipe input power exceeds the design heat transport capacity of the heat pipe	Modify heat pipe wick structure design or reduce power input
Boiling	Film boiling in heat pipe evaporator typically initiates at5-10 W/cm^2 for screen wicks and 20-30 W/cm^2 for powder metal wicks	High radial heat flux causes film boiling resulting in heat pipe dry out and large thermal resistances	Use a wick with a higher heat flux capacity or spread out the heat load

Therefore, it is important to assure that the heat pipe is designed to safely transport the required heat load. The maximum heat transport capability of the heat pipe is governed by several limiting factors which must be addressed when designing a heat pipe. There are five primary heat pipe heat transport limitations. These heat transport limits, which are a function of the heat pipe operating temperature, include: viscous, sonic, capillary pumping, entrainment or flooding, and boiling. Figures 8 and Fig.9 show graphs of the axial heat transport limits as a function of operating temperature for typical powder metal and screen wick heat pipes.

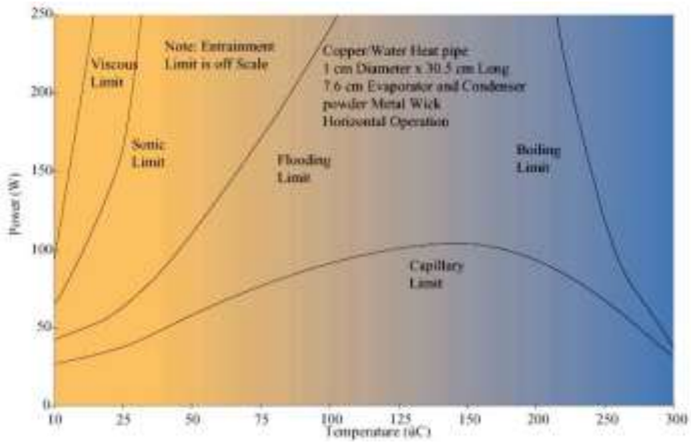


Figure 8: Predicted heat pipe limitations

As shown in Figures 8 and 9, the capillary limit is usually the limiting factor in a heat pipe design.

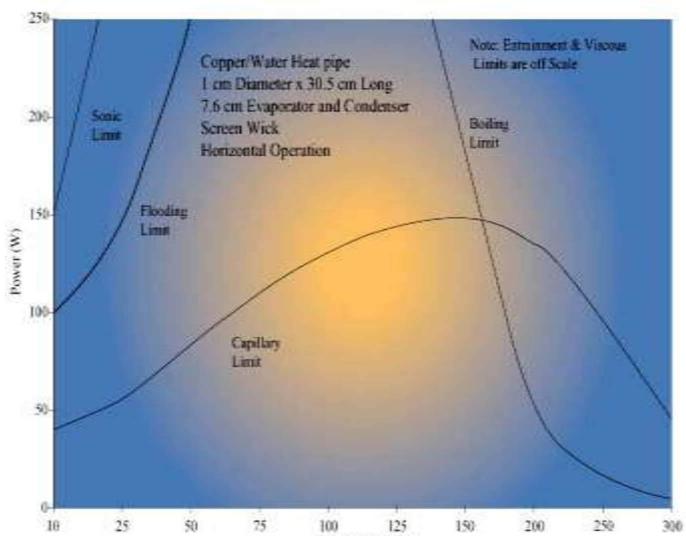


Figure 9: Predicted heat pipe limits

The capillary limit is set by the pumping capacity of the wick structure. As shown in Figure 4, the capillary limit is a strong function of the operating orientation and the type of wick structure.

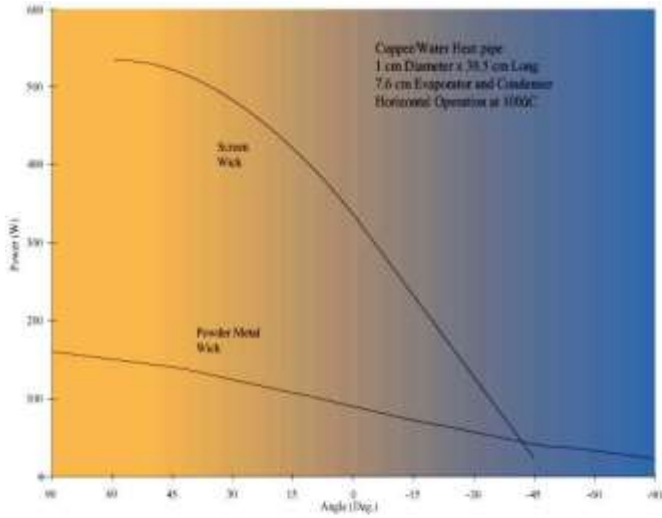


Figure 10: Capillary limits vs. operating angle

The two most important properties of a wick are the pore radius and the permeability. The pore radius determines the pumping pressure the wick can develop. The permeability determines the frictional losses of the fluid as it flows through the wick. There are several types of wick structures available including: grooves, screen, cables/fibers, and sintered powder metal. Figure 5shows several heat pipe wick structures .It is important to select the proper wick structure for your application. The above list is in order of decreasing permeability and decreasing pore radius. Grooved wicks have a large pore radius and a high permeability, as a result the pressure losses are low but the pumping head is also low. Grooved wicks can transfer high heat loads in a horizontal or gravity aided position, but cannot transfer large loads against gravity. The powder metal wicks on the opposite end of the list have small pore radii and relatively low permeability. Powder metal wicks are limited by pressure drops in the horizontal position but can transfer large loads against gravity.

Effective Heat Pipe Thermal Resistance

The other primary heat pipe design consideration is the effe. heat pipe thermal resistance or overall heat pipe ΔT at a given design power. As the heat pipe is a two-phase heat transfer device, a constant effective thermal resistance value cannot be assigned. The effective thermal resistance is not constant but a function of a large number of variables, such as heat pipe geometry, evaporator length, condenser length, wick structure, and working fluid.

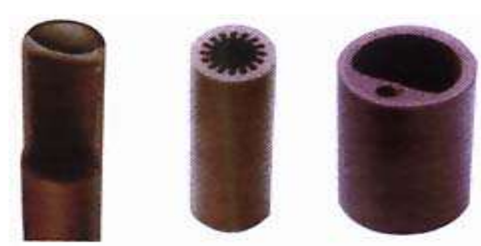


Figure 11: Wick structures

The total thermal resistance of a heat pipe is the sum of the resistances due to conduction through the wall, conduction through the wick, evaporation or boiling, axial vapor flow, condensation, and conduction losses back through the condenser section wick and wall.

Figure 12 shows a power versus ΔT curve for a typical copper/water heat pipe.

The detailed thermal analysis of heat pipes is rather complex. There are ,however, a few rules of thumb that can be used for first pass design considerations. A rough guide for a copper/water heat pipe with a powder metal wick structure is to use 0.2°C/W/cm2 for thermal resistance at the evaporator and condenser, and 0.02°C/W/cm2 for axial resistance.

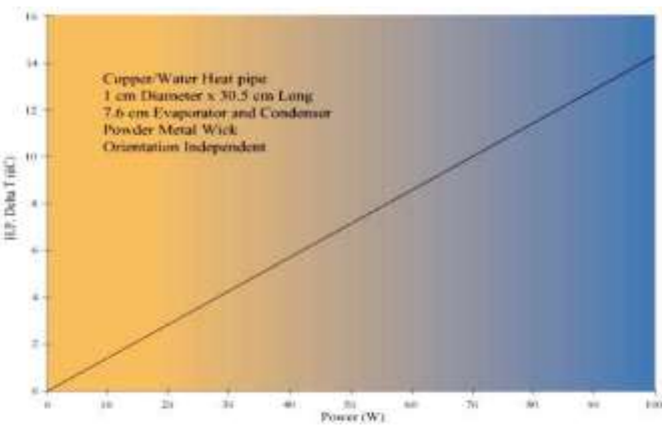


Figure 12: Predicted heat pipe Delta-TT

The evaporator and condenser resistances are based on the outer surface area of the heat pipe. The axial resistance is based on the cross-sectional area of the vapor space. This design guide is only useful for powers at or below the design power for the given heat pipe. For example, to calculate the effective thermal resistance for a 1.27 cm diameter copper/water heat pipe 30.5 cm long with a 1 cm diameter vapor space, the following assumptions are made. Assume the heat pipe is dissipating 75 watts with a 5 cm evaporator and a 5 cm condenser length. The evaporator heat flux(q) equals the power divided by the heat input area ($q = Q/A_{evap}$; q= 3.8 W/cm^2). The axial heat flux equals the power divided by the ross sectional area of the vapor space ($q=Q/A_{vapor}$; q = 95.5 W/cm^2).

The temperature gradient equals the heat flux times the thermal resistance.

$$\Delta T = q_{evap} * R_{evap} + q_{axial} * R_{axial} + q_{cond} * R_{cond}$$

$$\Delta T = 3.8\ \mathrm{W/cm^2} * 0.2\mathrm{°C/W/cm^2} + 95.5\ \mathrm{W/cm^2} * 0.02\mathrm{°C/W/cm^2}$$

$$+ 3.8\ \mathrm{W/cm^2} * 0.2\mathrm{°C/W/cm^2}$$

$$\Delta T = 3.4\mathrm{°C}$$

It is important to note that the equations given above for thermal performance are only rule of thumb guidelines. These guidelines should only be used to help determine if heat pipes will meet your cooling requirements, not as final design criteria

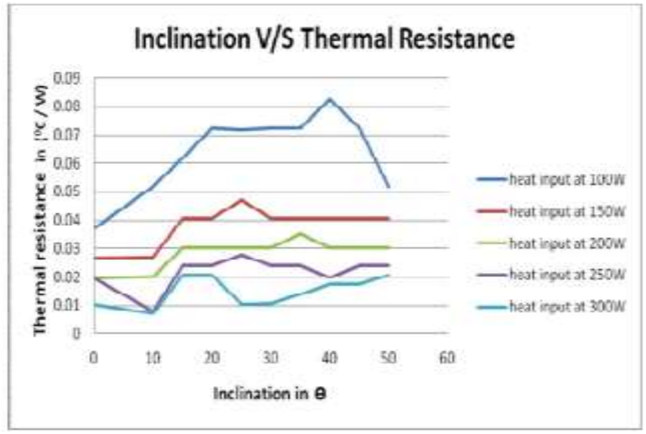


Fig. 13 Variation of thermal resistance with inclination

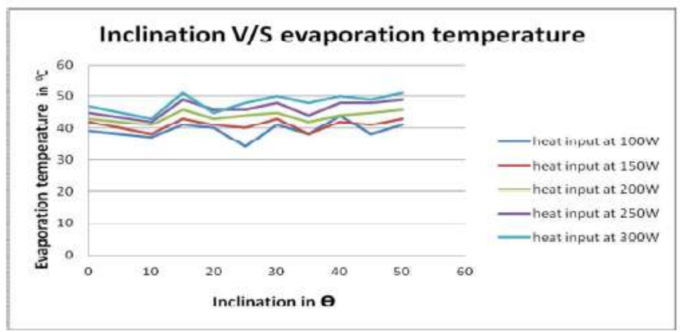


Fig.14 Variation of evaporation temperature with inclination

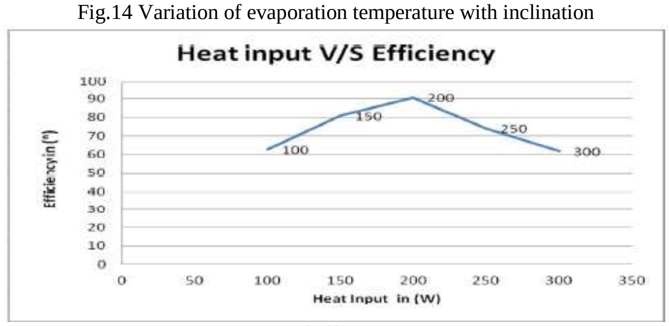


Fig .15 Variation of efficiency with heat input

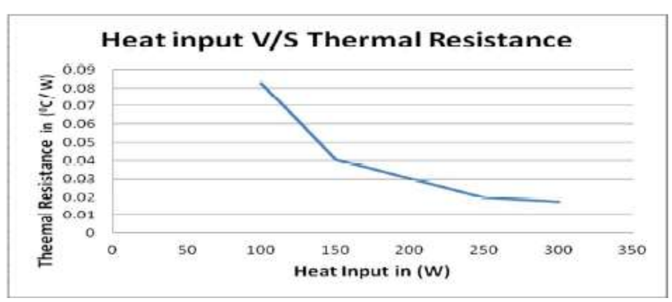


Fig.16 Variation of thermal resistance with heat input

XI. RESULTS AND CONCLUSONS

A study on performance of sintered copper wicked heat pipe is done by varying heat input and having capacity Heat load of 250 Watts Maximum. From this experiment we conclude:

1. The efficiency of heat pipe is decreasing after crossing 250W of input because the working fluid (water) is crossing its burn out temperature and the working fluid capacity to absorb latent heat of vaporization decreases.
2. The variation of efficiency of heat pipe with angle of orientation is very less because of the strong capillary action of sintered copper wick.
3. The increase in the condensation temperature and evaporation temperature is due to increase in latent heat of vaporization as the heat input increases.
4. The average efficiency if sintered copper heat pipe is 74.28% when worked at a heat input range of 100W -300W.

In this paper author has summarized different application of heat transfer in great detail. It is observed that, there is no specific material for intermediate temperature range and it is found that water can be used as a heat exchanging medium. but maximum care must be taken to avoid leakage. The VCHP discussed in great detail and it is implemented to withstand multiple freeze/thaw cycles without any affecting the performance. Use of VCHP is very good innovation for space craft and it acts as a gas diode. Further design and development of the new heat pipe has been carried out to validate through experiments.

REFERENCES

1. W. G. Anderson, M. C. Ellis, and K. Walker, “Variable Conductance Heat Pipe Radiators for Lunar and Martian Environments,” SPESIF 2009, Huntsville, AL, February 24 - 27, 2009.
2. Birur, G., Tsuyuki, G., “JPL Advanced Thermal Control Roadmap – 2009”, presented at the Spacecraft Thermal Control Workshop, March 10-12, 2009.
3. M. C. Ellis and W. G. Anderson, “Variable Conductance Heat Pipe Performance after Extended Periods of Freezing,” SPESIF 2009, Huntsville, AL, February 24 - 27, 2009.
4. Hartenstine, J. R., Walker, K. L., and Anderson, W. G., “Loop Heat Pipe with Thermal Control Valve for Variable Thermal Conductance,” 41st International Conference on Environmental Systems (ICES 2011), Portland, OR, July 17-21, 2011.
5. Marcus, B. D., “Theory and Design of Variable Conductance Heat Pipes: Hydrodynamics and Heat Transfer,” NASA Report No. NASA-CR-146195, April 1971.
6. W. G. Anderson, S. Tamanna, C. Tarau, J. R. Hartenstine, and D. Ellis, “Intermediate Temperature Heat Pipe Life Tests”, 16th International Heat Pipe Conference, Lyon, France, May 20-24, 2012.
7. W. G. Anderson, J. R. Hartenstine, D. B. Sarraf, and C. Tarau, “Intermediate Temperature Fluids for Heat Pipes and Loop Heat Pipes,” 15th International Heat Pipe Conference, Clemson, SC, April 25-30, 2010.
8. Anderson, W. G., “Intermediate Temperature Fluids for Heat Pipes and LHPs,” W.G. Anderson, Proceedings of the 2007 IECEC, AIAA, St. Louis, MO, June 25-27, 2007a.
9. Anderson, W.G., Bonner, R.W., Dussinger, P.M., Hartenstine, J.R., Sarraf, D.B., and Locci, I.E., “Intermediate Temperature Fluids Life Tests – Experiments” Proceedings of the 2007 IECEC, AIAA, St. Louis, MO, June 25-27, 2007b. Anderson, W.G., Dussinger, P.M., Bonner, R.W., and Sarraf, D.B., “High Temperature Titanium-Water and Monel-Water Heat Pipes,” Proceedings of the 2006 IECEC, AIAA, San Diego, CA, June 26-29, 2006a.
10. Anderson, W.G., Dussinger, P.M., and Sarraf, D.B., “High Temperature Water Heat Pipe Life Tests,” STAIF 2006, pp. 100-107, American Institute of Physics, Melville, New York, 2006b.
11. Anderson, W.G., “Evaluation of Heat Pipes in the Temperature Range of 450 to 700 K,” STAIF 2005, Albuquerque, NM, February 13-17, 2005.
12. Anderson, W.G., Rosenfeld, J.R., Angirasa, D., and Mi, Y., “The Evaluation of Heat Pipe Working Fluids In The Temperature Range of 450 to 750 K,” Proceedings, STAIF-2004, pp. 20-27, Albuquerque, NM, February 8-12, 2004.
13. Anderson, W.G., “Sodium-Potassium (NaK) Heat Pipe,” Heat Pipes and Capillary Pumped Loops, Ed. A Faghri, A. J. Juhasz, and T. Mahefky, ASME HTD, 236, pp. 47-53, 29th National Heat Transfer Conference, Atlanta, Georgia, August 1993.
14. William G. Anderson, John R. Hartenstine, and Christopher J. Peters “Variable Conductance Heat Pipes for Variable Thermal Links” . 16th International Heat Pipe Conference Lyon, France, May 20-24, 2012.
15. Dan Pounds, Richard W. Bonner “High Heat Flux Heat Pipes Embedded in Metal Core Printed Circuit Boards for LED Thermal Management” 14TH IEEE ITHERM Conference.
16. Darren Campo, Jens Weyant, Bryan Muzyka “Enhancing Thermal Performance in Embedded Computing for Ruggedized Military and Avionics Applications” 14th IEEE ITHERM Conference
17. CalinTarau, Carl Schwendeman, William G. Anderson, Peggy A. Cornell, “Variable Conductance Heat Pipe Operated with a Stirling Convertor” 11th International Energy Conversion Engineering Conference San Jose, CA, July 15-17, 2013.

