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EXPERIMENTAL STUDY OF THERMAL RESISTANCES IN A VAPOUR CHAMBER HEAT SPREADER

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***Abstract*—Today's microchip circuits are almost ten times smaller and denser than a decade ago. Modern CPUs feature several processing cores as well as an integrated graphics processor. This results in very high power and heat concentrations which make it an ever growing challenge to cool them. The same cooler that cools a large 100W heat source just fine, will struggle to remove the same 100W from a heat source half the size. The vapour chamber heat spreader evenly distributes heat to all heat pipes and improves cooling efficiency. A vapour chamber is a vacuum sealed, flat, metal structure that contains a working fluid that changes liquid to gas when heat is applied. Performance of a new vapour chamber of size 60 mm × 60 mm is studied. In the top plate the conventional wick structure is replaced by parallel grooves with inter groove opening. The inner surface of the bottom plate is sintered copper wick. Tip of the groove touches the wick directly so that the grooves function as vapour path, and gives more mechanical strength. The thermal resistances were measured for heat load increasing from 5W to 30W. Low thermal resistance is obtained under different heat load. The spreading resistance plays important role in the thermal resistance.**

***Index Terms*—Vapour chamber, Sintered Wick, Thermal resistance**

I. INTRODUCTION

The reduction in size, integration of functionality, and the increase of clock speed in electronic systems causes increase in power dissipation which is leading to higher heat fluxes and higher temperatures. For the reliable operation of the system, it is important to keep all the components like CPU, GPU operating within safe temperature limit. Thermal management and high heat flux removal technologies will be widely used in system design for this. Few characteristics required for a good heat transfer technology are low cost, high power input, multiple heat sources and adaptability to a wide range of heat flux. Now a days there are number of methods to overcome the overheat problem. Liquid cooling system like, cooling in micro channel, spray cooling, jet impingement cooling, heat pipes, thermosyphons etc. Among these technologies, heat pipes may offer high heat removal capability without any complicated cabin design. But there are some difficulties when using a heat pipe. They transfer heat only in one direction. Its round shape means that they are physically difficult to get close to a very small heat source. There is couple of solutions such as embedding them in a solid chunk of metal to spread the heat up to the connected pipes. But then we are relaying on the relatively inefficient heat transfer of copper. We can get heat directly to the heat pipes by flattening them out and playing the right up against something. But now due to the sheer size of heat pipe and the size of heat source, we can only connect a few heat pipes. Here comes the use of Vapour chamber heat spreader. Vapour chamber offers balance in high heat removal capability, spreading hot spot problems. It maintains an isothermal heat sink base. Vapour chamber can spread the thermal energy out incredibly efficiently. So we can use traditional fins or even heat pipes to spread the heat even further. That would enable efficient operation up larger heat sink than before.

In recent years, several studies have been conducted on the performance, or application of vapour chambers. In the year 2004 Shung-Wen Kang conducted an experiment on Micro heat pipe heat spreaders. Two wick designs, one using 200 µm wide etched radial grooves and the other with 100 mesh

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copper screens was investigated [1]. Jeung Sang Go evaluates the thermal performance of an acetone-charged VC heat sink containing new micro wick structures for cooling microprocessors in PC desktop applications [2]. K.N. Shukla fabricated and tested a wickless vapour chamber for two different working fluids, copper-water and aluminium-water Nano fluids with the filling ratio of 30% of the vapour chamber volume [3]. Yong Tang a and Dong Yuana multi-artery vapour chamber was designed, built and tested in this study. In this vapour chamber, the wick structure consists of sintered copper powder layers on the top and bottom plates, operating as evaporator and condenser wick respectively [4]. Yi Peng a and Wangyu Liu manufactured leaf-vein-like structure by chemical etching, and two different vapour chambers (diameter is 90 mm, the evaporator and condenser have the same wick structures) are compared concerning their cooling performances [5]. In the year 2010, G.S. Hwang Constructed a vapour chamber by designing a thin (monolayer) evaporator wick and distributed permeable columnar arteries supplying liquid (water) to highly concentrated heat source region [6]. A prototype vapour chamber was built by Meng-Chang Tsai and tested for its thermal performance under five different orientations. Spreading resistance exhibited the dominating factor in determining the overall thermal resistance of a vapour chamber [7].

From the literature it is clear that the wick structure plays an important role in the performance of vapour chamber. There are different wick materials like screen mesh, wire mesh, carbon nano tubes, etched metal, sintered copper powder are used. Among these types sintered copper powder wick shows good performance.

In this work a new design of vapour chamber is proposed and tested. It has a simple and efficient design. It can be easily manufactured with low cost and has a good anti dry out features.

II. THEMODIFIEDVAPOURCHAMBER

A vapour chamber much like a heat pipe is made up the vacuum sealed, flat metal structure that contains a working fluid that changes from liquid to gas when heat is applied. A conventional vapour chamber is shown in fig 1. Fig 2 shows the modified vapour chamber. In the modified vapour chamber, the conventional wick laid in the upper plate is replaced by parallel grooved plate, having an internal opening for spreading the vapour. The tip of the grooves directly touches the wick structure in the evaporator section. This leads to some advantages.

- Simple design and easy to manufacture at low coast.
- The flow resistance is less in grooves than the wick.
- More surface area for condensing the vapour.
- Good compression strength without supporting studs.

Because of these we can thinning the container wall of the vapour chamber .In this study we present the test results for a 60mm × 60mm vapour chamber with thickness 5mm. Sintered copper powder wick for evaporator section and parallel grooves for condenser section. The working fluid is Deionized water.

III. EXPERIMENTAL METHOD

Seventeen type K thermocouples were installed along the test section and all temperatures outputs were connected to a data logger and continuously logged. The temperature measurement uncertainty was ±0.10C. For thermal insulation Glass wool (k < 0.04 W/mK) and gypsum board (k < 0.17) were used to cover the whole device. The experimental methode was taken from literature [7].

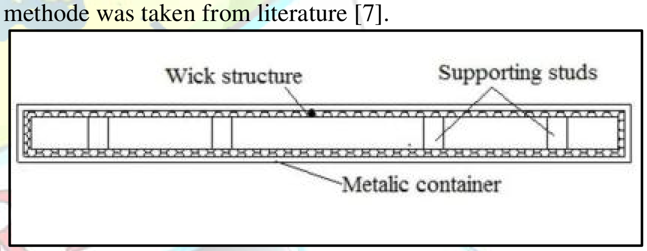

Fig 1: Schematic drawing of a conventional vapour chamber

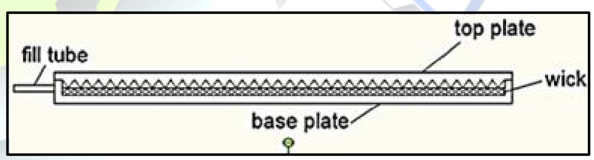

Fig 2: Schematic drawing of a modified vapour chamber

a.*Schematic of VCHS test section and measure point*

The test section consists of a vapour chamber heat spreader, Aluminum cold block, heater block and a cartridge heater. The Photographs of the vapour chamber heat spreader shown in Fig 3. It consists of sintered copper wick of 1mm

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thickness and the parallel grooves are made with 1mm depth and 1.2 mm width. Fig 4: shows the schematic representation of test section. The cartridge heater inserted into the heater block. The vapour chamber is sandwiched between the cold block and heater block. Fig 5: shows the positions of thermocouples.

The thermal performance of vapour chamber heat spreader evaluates in terms of its thermal resistance. The total thermal resistance can be expressed as

$$R_{TOTAL} = R_{1D} + R_S + R_{C.} \qquad (1)$$

Where R_{1D} is the one dimensional resistance, R_S is the spreading resistance, and R_C is the condensing resistance. They can be expressed as

$$R_{1D} = \frac{t}{hA} = \frac{\bar{T}_b - \bar{T}_t}{q} = \frac{\frac{1}{4}\sum_6^9 Ti - \frac{1}{5}\sum_{10}^{14} Ti}{q} \qquad (2)$$

$$R_S = \frac{\bar{T}_e - \bar{T}_b}{q} = \frac{\frac{1}{4}\sum_1^4 Ti - \frac{1}{4}\sum_5^9 Ti}{q} \qquad (3)$$

$$R_C = \frac{\bar{T}_t - \bar{T}_c}{q} = \frac{\frac{1}{5}\sum_{10}^{14} Ti - \frac{1}{3}\sum_{15}^{17} Ti}{q} \qquad (4)$$

The resistance of the vapor chamber can be derived as

$$R_{VC} = R_S + R_{1D} = \frac{\bar{T}_e - \bar{T}_t}{q} = \frac{\frac{1}{4}\sum_1^4 Ti - \frac{1}{5}\sum_{10}^{15} Ti}{q} \qquad (5)$$

$$R_{TOTAL} = R_{1D} + R_S + R_C = \frac{\bar{T}_e - \bar{T}_c}{q} = \frac{\frac{1}{4}\sum_1^4 Ti - \frac{1}{3}\sum_{15}^{17} Ti}{q} \qquad (6)$$

where A is the base area of vapour chamber, h is the heat convection coefficient, t is the thickness of vapour chamber, T_e is average bulk temperature of heater, T_b is average temperature of evaporator section of vapour chamber, T_t is average temperature of condenser section of vapour chamber, T_c is average temperature of cold block, and q is the heat transfer rate through the heater surface. The related resistance of a vapor chamber can be calculated using equations (4)-(8) respectively. Average temperatures at each particular surface were calculated by taking the arithmetic mean of the related probes when they reach steady state. The total resistance in this case was the sum of one dimensional, spreading, and condensing resistance.

Fig 3: Photograph of the vapor chamber heat spreader

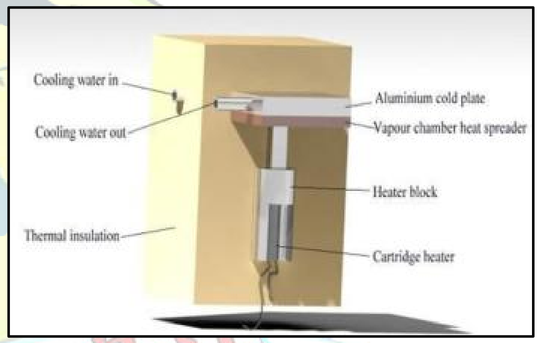

Fig 4: Schematic representation of test section

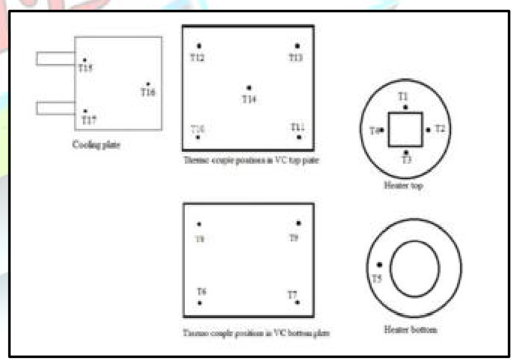

Fig 5: Thermocouple positions on the test section

b. *Experimental setup*

The schematic experimental setup was shown in fig 6. The chamber was placed in between the cold plate and heater block. A cartridge heater inserted into the heater block to add

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heat to the vapour chamber. The heater block had an effective heating area 10mm × 10 mm. The vapour chamber top surface was attached to an aluminium cold block. The heat was removed by the cooling water circulated through the cold block. The whole system is thermally insulated using glass wool and gypsum board. The electrical power varied from 5W to 50W, and the flow rate of cooling water was 12 L / min. For the proper contact between the heating area and the bottom surface of the vapour chamber, thermal paste was introduced between them. The experimental setup was shown in fig 7.

Experiments were performed to record all the temperatures when input power was applied from 0 W to 50 W in a step of 5W. The steady state temperatures were recorded to evaluate the thermal performance of the vapour chamber.

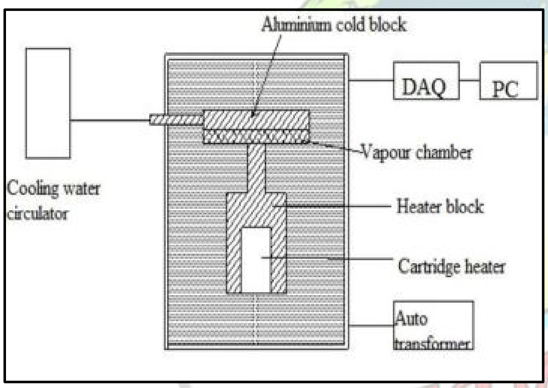

Fig 6: Schematic experimental set up

Fig 7: The experimental setup

IV. TEST RESULTS

Fig 8 shows the electrical power input Q compared with the heat transfer rate (q) through the heater surface. The q is calculated by equation

$$q = \frac{T_e - T_{etop}}{R_A}, \text{ Where } T_e = \frac{1}{5}\sum_1^5 Ti, \; T_{etop} = \frac{1}{4}\sum_1^4 Ti \qquad (7)$$

Where R_A is the thermal resistance of the heater block. T_e and T_{etop} are the average bulk temperature of the heater block and average temperature of the heater surface respectively.The heat transfer rate „q" is less than that of input electrical power „Q". This is because of the heat loss from the heater. When the input power is 50W, the difference in heat load and input power is about 13%. „q" is applied in equations 2 – 6 to have more accurate evaluation on thermal resistance.

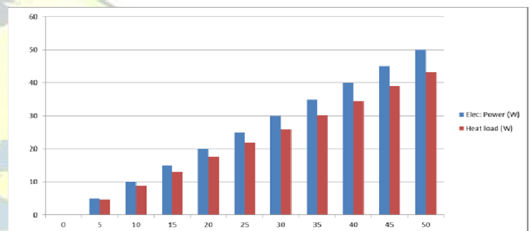

Fig 8: Electrical power compared with heat transfer rate

a.*Thermal resistance of modified vapour*

The thermal resistances were calculated from the measured temperatures. The one dimensional, spreading, condensing and total thermal resistance are plotted against time in fig 9. In the first 900 seconds of the experiment the condensing resistance shows negative values. This is because, at this time rang the average temperature of the cold plate was higher than the average temperature of the vapour chamber surface. During this period of time all the resistances showed unstable variations compared to others. The total resistance and spreading resistance show same behavior. The spreading resistance dominates all other resistances.

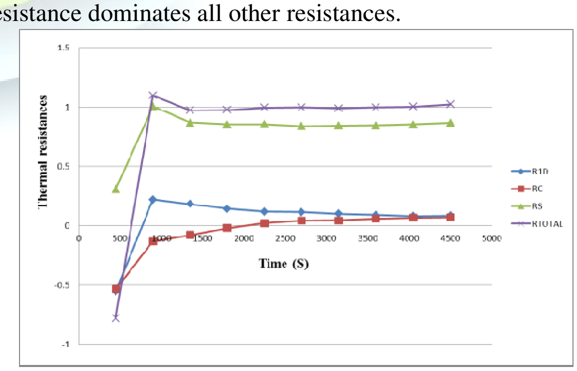

Fig 9: Thermal resistances v/s time

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b.*Relation between thermal resistances and heat transfer rate*

The thermal resistances are plotted as a function of heat transfer in fig 10. In fig 10, there is a steep decrease in the resistance when heat transfer rate was applied from 7 W to 13W. The situation may be explained as a star up operating in the vapour chamber during that time. After that, there are slight variations in the resistance showed. From fig 10, it is observed that the thermal resistances reduces and become stable when heat transfer rate increases.

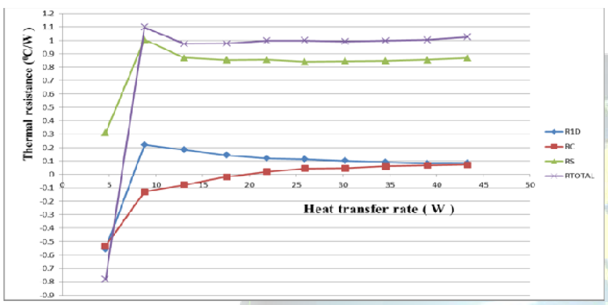

Fig 10: Effect of thermal resistances on heat transfer rate

c.*The heat spreading capacity of the modified vapour chamber*

The temperature distributions on the vapour chamber heat spreaders bottom surface and top surface are tabulated on table1 and table 2. In this study the difference between the maximum and minimum temperatures (ΔT) calculates as shown in tables. It can be indicated that a slight difference in ΔT showed on both bottom and top surfaces. This means that the modified vapour chamber heat spreader has good spreading capacity

TABLE NO 1: TEMPERATURE DISTRIBUTION ON THE VCHS BOTTOM SURFACE

Heat input (W)	T6	T7	T8	T9	$\Delta T(T_{max}-T_{min})$
10	40.42	40.20	40.84	40.70	0.64
20	43.46	43.28	43.60	43.72	0.44
30	46.20	45.96	46.18	46.42	0.46
40	48.84	48.60	49.04	48.96	0.46
50	51.54	51.38	51.70	51.62	0.32

TABLENO2:TEMPERATUREDISTRIBUTION ON THE VCHS TOP SURFACE

Heat input (W)	T10	T11	T12	T13	T14	$\Delta T(T_{max}-T_{min})$
10	38.70	38.77	38.82	38.39	38.72	0.43
20	40.84	40.91	40.98	40.20	40.63	0.78
30	43.24	43.87	43.80	43.10	43.02	0.85
40	45.80	45.88	45.18	45.78	45.90	0.72
50	47.80	47.62	47.92	47.98	47.82	0.36

V. CONCLUSION

A modified vapor chamber heat spreader with sintered wick on evaporator side and parallel grooves on condenser side were fabricated and tested. Experiments were performed to examine the thermal resistance with different power inputs. The results showed that thermal resistance decreased as the input heat load increased and spreading resistance shows the dominating factor in determining the overall thermal resistance of a vapor chamber. Because of the grooves in the condenser side, the modified vapour chamber can be easily manufactured at low cost.

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