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Experimental investigation of vapor chamber with different working fluids at different charge ratios

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KEYWORDS

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Abstract Vapor chamber is one of highly effective thermally spread techniques in electronic cooling. In the following a work is conducted to evaluate thermal performance of 2.0 mm high and 50 mm diameter vapor chamber with water and methyl alcohol at different charge ratios. Also, a solution of water and Propylene Glycol at two concentration 50%, 15% were tested to study the effect of using surfactant as enhancement agent for working fluid. Also total thermal resistance of the chamber is divided into three types (junction resistance, internal resistance, and condenser resistance) to investigate and determine which type of thermal resistance has a major effect on chamber total thermal resistance.

1. Introduction

A problem commonly encountered in thermal management of electronic packages is thermal spreading resistance which occurs as heat flows by conduction between source and sink with different cross-sectional areas. Typical applications include cooling of electronic devices, both at package and system level, and cooling of power semi-conductors using heat sinks. Chen et al. [11] state that heat pipes have superior performance in heat transfer because they transfer energy via the liquid–vapor phase change of the inner working fluid. They can effectively remove the waste heat generated by highly concentrated heat sources, such as CPUs or LEDs. As a consequence, the heat pipes and the vapor chambers (flat plate heat pipes) have been becoming popular for the thermal solutions of the electronic elements. To reduce the temperature rise of a concentrated heat source, an external finned heat sink providing a lower

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Nomenclature

R_T	overall thermal resistance (°C/(W/cm^2))	C_P	specific heat constant (J/kg °K)
R_J	junction thermal resistance (°C/(W/cm^2))	T_o	outlet cooling water temperature (°C)
R_I	inside thermal resistance of the vapor chamber (°C/(W/cm^2))	m	cooling water mass flow rate (kg/**s**)
R_c	condensation thermal resistance of the vapor chamber (°C/(W/cm^2))	T_i	inlet cooling water temperature (°C)
Q	heat load removed from the chamber through cooling water (W)	T_s	working fluid saturation temperature (°C)
Q_L	heat losses from the system to the surrounding (W)	T_{ei}	temperature of inner surface of the evaporator plate (°C)
Q_{in}	heat load input through nickel chromium heater (W)	T_{ci}	temperature of inner surface of the condenser plate (°C)
I	current (A)	T_{co}	temperature of outer surface of the condenser plate (°C)
V	Voltage (V)	T_J	temperature of outer surface of evaporator plate (Junction temperature) (°C)

thermal resistance to the ambient is usually applied. Generally, the thermal resistance of a heat sink is reduced by increasing its heat-transfer area. However, this is not suitable when the heat sources become highly concentrated. This is because the spreading resistance of a spreader plate may dominate the overall thermal resistance. Note that a concentrated temperature profile gives rise to a local hotspot and may fail the whole chip entirely. Therefore, it is crucial to flatten the temperature distribution and reduce the associated spreading resistance accordingly. To effectively reduce this undesired effect, the vapor chambers have been proven to be better than the metal spreaders.

Generally the vapor chamber is a vacuum container with/without wick structure lining the inside walls that is saturated with working fluid (typically water). As heat is supplied, the liquid at that location immediately vaporizes, and the vapor rushes to fill the vacuum. Where ever the vapor comes into other cooled surface the vapor condensed and returned again to the evaporation surface.

Hsieh et al. [1] performed an experiment to examine the spreading thermal resistance of centrally positioned heat sources and the thermal performance of water charged, gravity assisted flat vapor chamber to be used for electronic cooling. Also parametric studies including different heat fluxes and operating temperatures were conducted, and the effect of the relevant parameters on the cooling performance in terms of spreading resistance was presented and discussed. They also stated that there are a very limited number of experimental studies available regarding the flat vapor chamber heat sinks. Moreover, little has been done to quantify the spreading resistance of a vapor chamber heat spreader due to phase change and its comparison with using solid metal as base materials.

Go [2] evaluates thermal performance of an acetone-charged vapor chamber heat sink containing new micro wick structures for cooling microprocessors in PC desktop applications. Cooling performance was examined by measuring the working temperatures and thermal resistances for various heat inputs and for three different tilt angles. The assembled vapor chamber heat sink showed a heat removal capacity of 80 W/cm^2 at the junction temperature of 85 °C and an ambient temperature of 24 °C, indicating an overall thermal resistance of 0.76 °C/W.

Koito et al. [3] performed numerical analysis for a flat-plate heat pipe "vapor chamber". The mathematical model of the vapor chamber formulated in this study is a two-phase closed disk-shaped chamber and is placed between a small heat source and a large heat sink. Wick sheets and a wick column are provided inside the vapor chamber to circulate the working fluid. Experimental investigation is also carried out, and fairly good agreement is obtained with the numerical results,

Ming et al. [4] designed a novel grooved vapor chamber. The grooved structure of the vapor chamber can improve its axial and radial heat transfer and also can form the capillary loop between condensation and evaporation surfaces. The effect of heat flux, filling amount and gravity to the performance of this vapor chamber is studied by experiment. From experiment, they also obtained the best filling amount of this grooved vapor chamber. By comparing the thermal resistance of a solid copper plate with that of the vapor chamber, it is suggested that the critical heat flux condition should be maintained to use vapor chamber as efficient thermal spreaders for electronics cooling. They also developed a two-dimensional heat and mass transfer model for the grooved vapor chamber. The numerical simulation results show the thickness distribution of liquid film in the grooves is not uniform. The temperature and velocity field in vapor chamber are obtained. The thickness of the liquid film in groove is mainly influenced by pressure of vapor and liquid beside liquid–vapor interface. The thin liquid film in heat source region can enhance the performance of vapor chamber, but if the starting point of liquid film is backward beyond the heat source region, the vapor chamber will dry out easily. The optimal filling ratio should maintain steady thin liquid film in heat source region of vapor chamber. The vapor condenses on whole condensation surface, so that the condensation surface achieves great uniform temperature distribution. By comparing the experimental results with numerical simulation results, the reliability of the numerical model can be verified.

Hsieh et al. [5] developed three dimensional analytical solution using product solutions via the separation of variables for spreading thermal resistances of centrally positioned heat sources of a vapor chamber heat sink with and without a partition for electronic cooling is presented. Parametric study including partition thickness and height was performed, and the effect of the relevant parameters on the heat transfer performance in terms of the base spreading resistance was examined.

Chen et al. [6] developed and test a simplified transient three-dimensional linear model. In the proposed model, the vapor is assumed as a single interface between the evaporator and condenser wicks, and this assumption enables the vapor chamber to be analyzed by being split into small control volumes. Comparing with the previous available results, the calculated transient responses have shown good agreements with the existing results. For further validation of the proposed model, a water-cooling experiment was conducted. It is found that the transient response of the vapor chamber is affected by the heat capacitance of heating block. The associated influence is hence considered in the present model. The calculated transient behavior is close to the measurements but still shows some discrepancies without considering the heat capacitances of the insulation block and the cold plate. The thermal resistance calculated from the model is 8.6% lower than the average of the experimental results.

Zhang et al. [7] performed thermal analysis to compare thermal performance of a board-level high performance flip-chip ball grid array package equipped with solid Cu or vapor chamber (VC) as the heat spreader and Al-filler gel or in solder as thermal interface material (TIM). The effect of different heat source sizes was also examined. Numerical results indicate that for the particular test vehicle under a power dissipation of 160 W, the thermal performance is remarkably enhanced by switching TIM from Al-filler gel to In solder while the enhancement by using VC instead of solid Cu heat spreader is only observable when solder is incorporated. Moreover, the performance of VC gradually enhances then retards as heat source size decreases. The retardation can be attributed to more dominant role of die in heat dissipation when heat source size gradually shrinks.

Wang et al. [8] measured evaporation resistances of loosely-sintered copper-powder evaporators in operating flat-plate heat pipes. They also visualized the evaporation processes through a top glass plate. Irregular or spherical powders of different size distributions were investigated. Uniform heating of 16–170 W/cm^2 was applied to the base plate near one end with a heated surface of 1.1 cm^2. At the other end was a cooling water jacket. The evaporation performance was first examined with the effect of liquid flow resistance minimized, i.e., the copper powders covered only the heated area with the remaining region covered with sintered copper wire screens. Similar to multi-layer mesh wicks, quiescent surface evaporation without nucleate boiling was observed for all test conditions, in spite of the abundant nucleation sites. The water film receded and the evaporation resistance reduced with increasing heat flux. Once partial dry out occurred, evaporation resistance starts re-rise. The minimum evaporation resistances were about 0.08–0.09 K Cm^2/W for wicks containing fine powders. These values are similar with those for multi-layer-mesh wicks having a fine bottom screen. In the absence of fine powders, the minimum evaporation resistances were significantly larger. In the second part for homogeneous sintered-powder wicks, the large flow resistance tended to retard the condensed water from returning to the evaporator. However, this can be compensated by a larger charge and/or a thicker wick.

Wong et al. [9] construct a low thermal resistance, multi-artery heat pipe spreader vapor chamber by designing a thin (monolayer) evaporator wick and distributed permeable columnar arteries supplying liquid (water) to highly concentrated heat source region. The condenser wick is layered copper screens in intimate contact with the columnar arteries. The measured evaporator thermal resistance is less than 0.05 K Cm^2/W using a 1 cm^2 heat source, and the critical heat flux is about 380 W/cm^2. The resistance is dominated by the small effective thermal conductivity of the evaporator wick and by the small conduction path through the receding meniscus within it. This resistance decreases nonlinearly with the heat flux, due to a decrease in the radius of the receding meniscus. The evaporator wick resistance is the smallest at high heat fluxes. This is much better (smaller thermal resistance and higher CHF) than the uniform artery heat spreader.

Hwang et al. [10] develop a network model for MAHPS to optimize its three-dimensional heat and liquid flow. The baseline design uses water as external coolant, and air cooling is also considered. The results show while large number of columns is needed to make the most of the maximum capillary pressure by removing the most heat, the columns also limit the evaporation area. For a 1 cm diameter heater in a 5 cm diameter VC, the optimized number of columns is about 37. In addition to the viscous-capillary limit, we also use the evaporator wick-superheat limit set at 10 °C. MAHPS shows superior performance compared to the uniform artery, by reducing its overall thermal resistance.

In the following experimental testes performed to examine performance of 2-mm high and 50-mm diameter vapor chamber using pure water, methyl alcohol at different fill charge ratios. Also solution of water with two percentage of propylene glycol as surfactant is tested to study effect of using surfactant as enhancement agent for water working fluid. Also total thermal resistance of vapor chamber is divided into three types (junction (R_J), internal (R_I), and condenser resistance (R_C)) to investigate and determine which type of thermal process that happened in chamber has major value of thermal resistance.

2. Experiment setup

The tested vapor chamber consists of three main parts top plate, bottom plate and space ring. The top and bottom plates are made from copper with thickness 1 mm. The space ring was made from copper, with inner, outer diameter 50 mm, 70 mm and 2 mm thickness respectively. The above three parts are collected together through eight circumferential bolts. A suitable sealant agent used to completely prevent leakage.

Two holes with 1 mm diameter radial drilled through spacing ring to allow two thermocouples fixation inside vapor chamber cavity on inner surface of evaporator and the second one fixed on inner surface of condenser. A charging valve with path to chamber cavity used to make charging process easy and simple see Fig. 1. A cooling path constructed from transparent acrylic cover with oil seal to prevent leakage, fixed on outer condenser surface of chamber using suitable glue. Inlet and outlet cooling water tubes fixed on this cover.

Thermocouples were fixed on four locations as shown in Fig. 2. At location (1) (top surface of heat distributing plate)

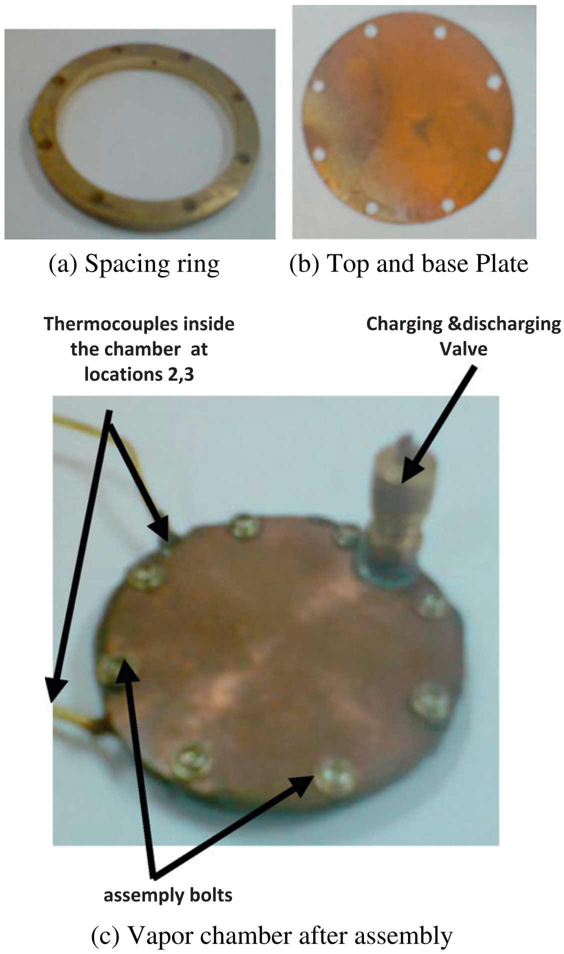

Figure 1 Photo of assembled vapor camber and parts before and after assembly.

two thermocouples are inserted with spacing about 5 mm between their probes to read contact surface temperature. Average of these two readings is referred to as junction temperature (T_J). At location (2) single thermocouple is used. This point is referred as evaporator inside temperature (T_{ei}) Also at location (3) single thermocouple is used to measure temperature of inner surface of condenser (T_{ci}). Temperature of location (4) is measured by two thermocouples and referred to as (T_{co}). Inlet and outlet temperature of cooling water measured using two thermocouples and referred as T_i, T_o All above eight T-type thermocouples are calibrated before experiment installation on data acquisition system, which will be used to record the reading for each experiment. Cooling water flow rate measurement is done using scaled container (50 ml) combined with a stop watch which is used to measure collecting time of certain volume of cooling water.

Nickel chromium wire with diameter 0.2 mm is coiled around disk shape mica paper and then covered with suitable thermal cover forming 150 W film heater with 50 mm diameter and 2.5 mm thickness. The heater is connected to a voltage regulator power supply where input power controlled by varying input voltage. Input current and voltage is measured through two precise multi-meters.

After vapor chamber was assembled leakage test done to avoid leakage during experiment. The test was done by raising the pressure inside chamber up to 2 bars while it was immersed in water pool using suitable air compressor. If vapor chamber properly passed over test, a distributing plate is then connected to the heater and then fixed on outer surface of the evaporator. Finally after finishing all above, the chamber then charged and installed with insulating blocks designed to reduce losses from it to surrounding. See Fig. 3.

The experiments were done with following procedure. The chamber is completely evacuated up to 0.025 Pa. For each working fluid at specified charge ratio a specified amount of working fluid charged through the charging valve. Then cooling water with a fixed amount flows. Next the tested power

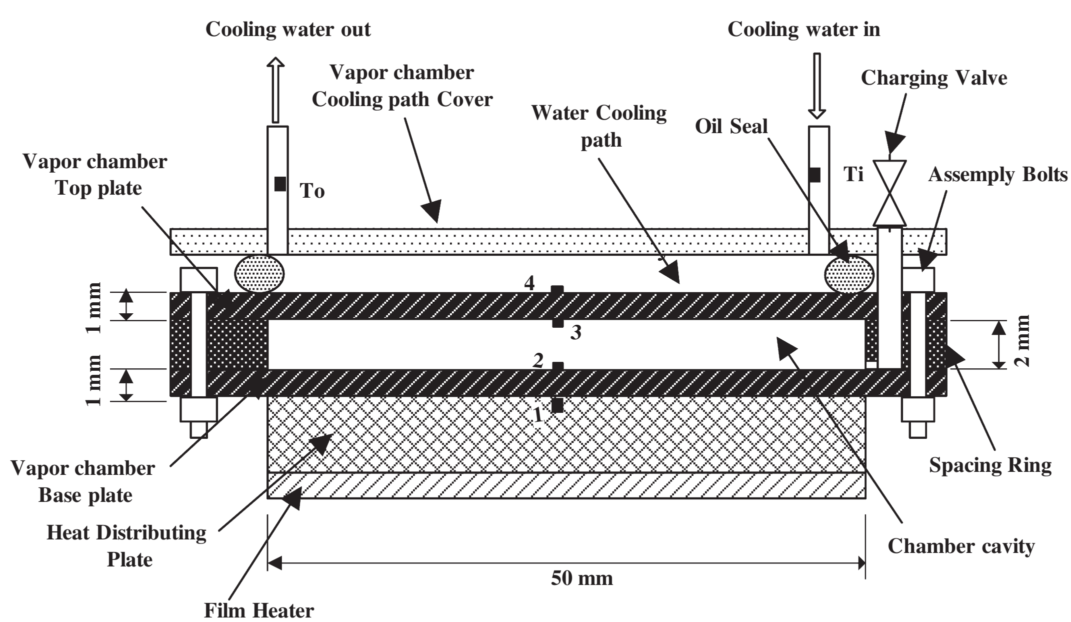

Figure 2 Tested vapor chamber details.

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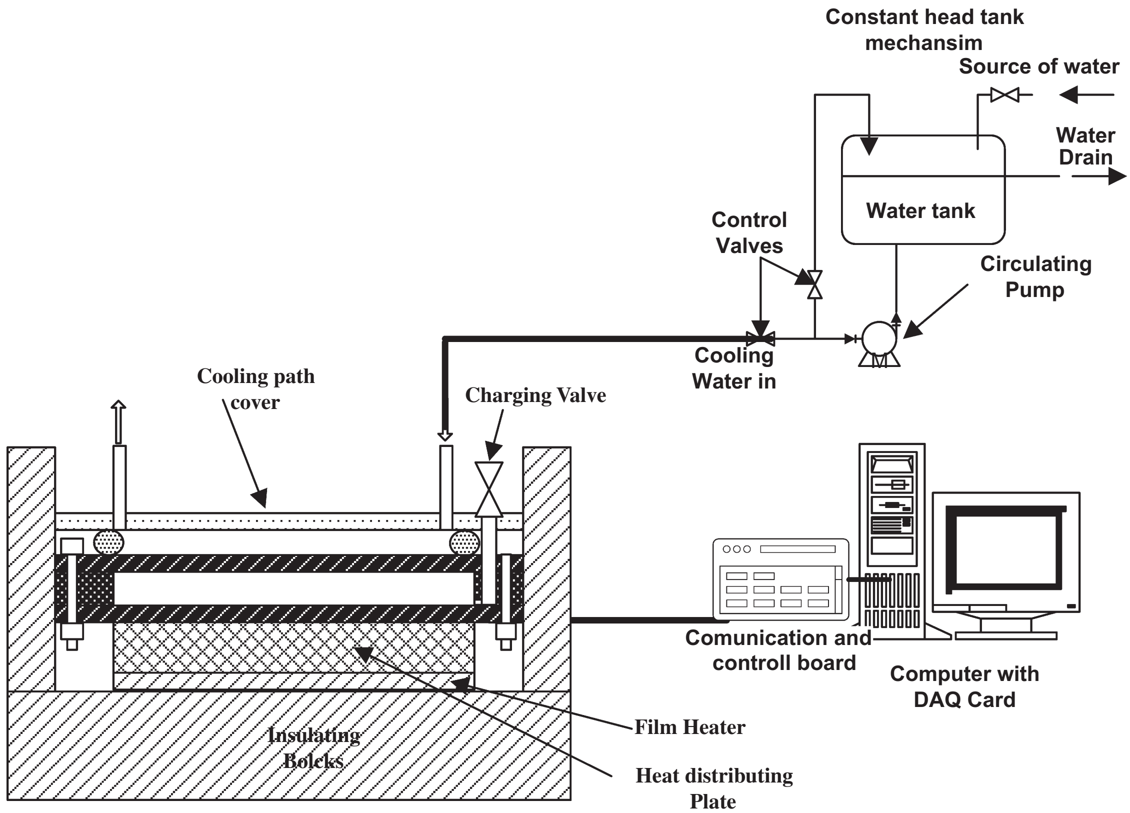

Figure 3 Test rig layout.

level is applied. Finally the data acquisition starts to record the thermocouple reading.

3. Data reduction and processing

As stated by Ref. [11] the total thermal resistance of a vapor chamber comprises a conduction resistance and a spreading resistance. The conduction resistance is the one-dimensional resistance provided that the heat source is of the same size as the vapor chamber. The spreading resistance is resulted from a concentrated heat source. So that the Thermal resistances of the vapor chamber in our case divided into three types of resistance for purpose of comparing effect of thermal process type (evaporation, vapor movement and condensation) on it. The thermal subdivision called junction thermal resistance (R_J), inside thermal resistance (R_I), and condensation thermal resistance (R_c). These different thermal resistances can be evaluated based on the following relations:

$$R_T = \frac{(T_j - T_{co})}{Q} \quad (1)$$

$$R_J = \frac{(T_j - T_{ei})}{Q} \quad (2)$$

$$R_I = \frac{(T_{ei} - T_{ci})}{Q} \quad (3)$$

$$R_c = \frac{(T_{ci} - T_{co})}{Q} \quad (4)$$

where Q is rate of heat removed from chamber through cooling water calculated from following equation

$$Q = m * C_P(T_o - T_i) \quad (5)$$

The working fluid saturation temperature is the average temperature at location 2 and 3.

$$T_S = (T_{ei} + T_{ci})/2 \quad (6)$$

The heat lost from the test rig system could be evaluated as follow.

$$Q_L = Q_{in} - Q \quad (7)$$

where

$$Q_{in} = I * V \quad (8)$$

Uncertainty estimation was made considering instruments errors, measurement variance, and calibrations errors for water flow rate and temperature measurements. The error of thermocouple reading was 0.1 °C for thermocouple reading 25 °C with an error percentage ±0.4%. The scaled container used for water flow measurements accuracy is 0.05 ml. The error in cooling water mass flow rate is ±5.1%. The encountered error for removed heat from condenser is ±6%. Thermal resistance error counted to be ±6.8%.

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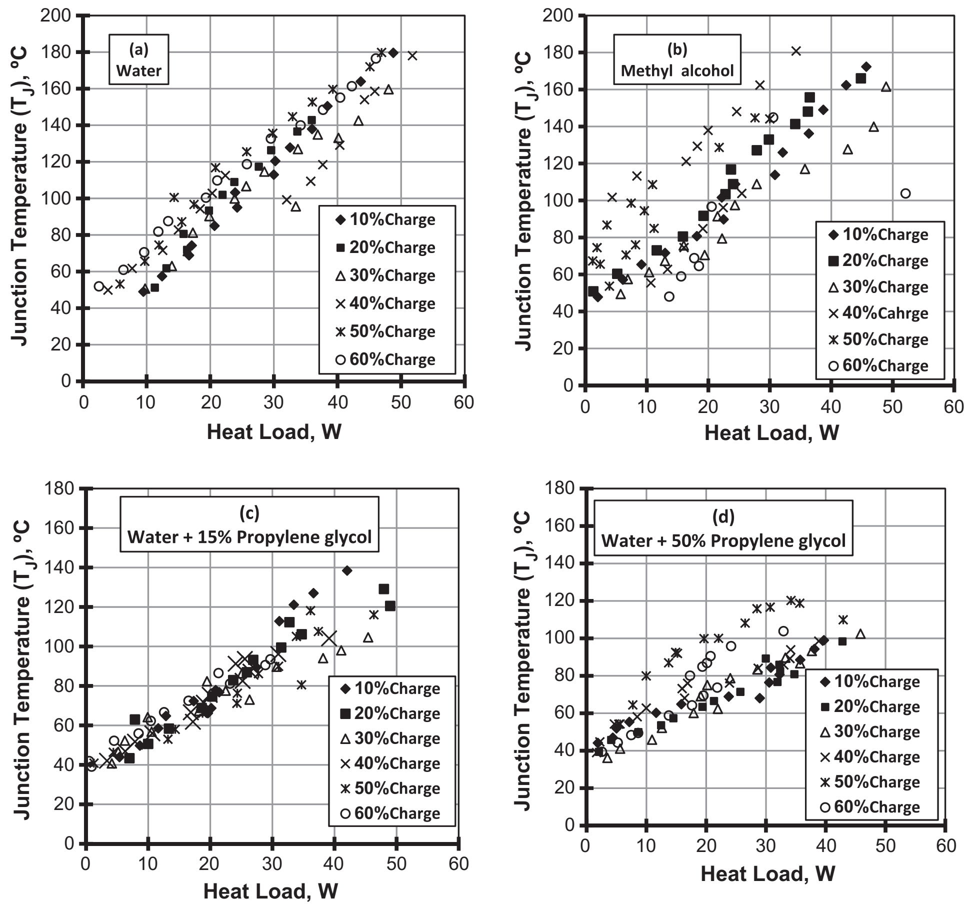

Figure 4 Effect of heat load on base temperature (location (1)) using (a) water, (b) methyl, alcohol, (c) solution of water + 15% propylene glycol and (d) solution of water + 50% propylene glycol as working fluid at different charge ratios.

4. Results and discussion

4.1. Effect of heat load

Effect of input heat load variation on outside base temperature T_J (temperature at locations (1)) is shown in Fig. 4. At low heat load variation of charge ratio have no significant effect on T_J. Water is more efficient than methyl alcohol for all ranges of charge ratio at all heat loads. This due to that the value of latent heat of water is much more than that for methyl alcohol. Using solutions of water and propylene glycol as surfactant has a good effect where temperature decrease at same power more than using pure water only. There is no effect of charge ratio on junction temperature at all heat loads. For solution of water with 50% propylene glycol as a working fluid, significant effect of charge ratio on T_J start to appear at all ranges of heat load. The charge ratio 50% and 60% has highest value of T_J rather than other charge ratios. Totally the value of T_J for a solution of water and 50% propylene glycol is lower than that for solution of water and 15% propylene glycol. It could be conclude that increase of propylene glycol percentage decreases outside based temperature due to increases surfactant effect on surface tension and consequently on wet ability of water with the base plate of vapor chamber which prevents forming of hotspots.

Effect of heat load on vapor chamber total thermal resistance for different working fluid at different tested charge ratio is shown in Fig. 5. For all working fluid at different charge ratios thermal resistance is high at low heat load and starts to decrease with heat load increases. This is due to that heat transferred through thin layer of fluid happened by conduction mode only where no boiling or free convection occurs at this low level of heat flux. This could be referred as the effect of working fluid resistance to evaporation. With the increase of heat load plate temperature and heat flux reached to sufficient values to enhancement and pushes the evaporation process through hot plate of vapor chamber. So that evaporation resistance decreases and consequently total thermal resistance decreases. With adding Propylene glycol, surface tension of water decreases and consequently rewetting rate for hot plate of chamber increases, which reduces the evaporation resistance. At low heat load chamber total thermal resistance for solution of water and propylene glycol decreases more than thermal resistance of pure water by about 30%.

For a solution of water and 15% propylene glycol thermal resistance at low power is high and starts to decrease with increase of heat load. Also for a solution of water and 50% propylene glycol the thermal resistance at low power is high and starts to decrease with increase of power. The two solutions

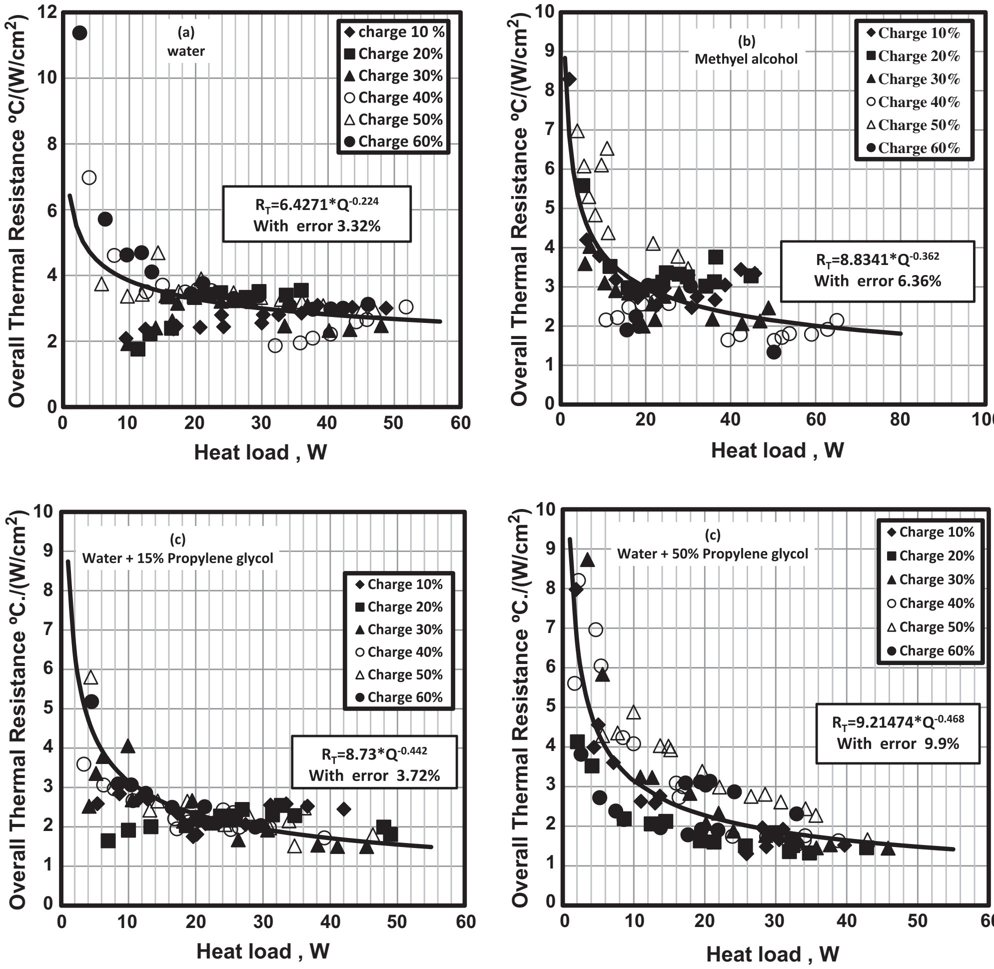

Figure 5 Effect of heat load on vapor chamber total thermal resistance using water, (b) methyl alcohol, (c) solution of water + 15% propylene glycol and solution of, (d) water + 50% propylene glycol as working fluid at different charge ratios.

Table 1 Constant *A* and *B* for Eq. (9).

Type of working fluid	*A*	*B*	Error (%)
Pure water	6.4271	−0.224	3.32
Methyl alcohol	8.8341	−0.362	6.36
Solution of water + 15% propylene glycol	8.7300	−0.442	3.72
Solution of water + 15% propylene glycol	9.2147	−0.468	10

of water and 15% and 50% propylene glycol have not much different in total thermal resistance which means that any value in enhancement (lowering) in surface tension for working fluid have a great effect on chamber performance. This is due to that the difference in evaporation rate between surfactant and water which allow the smallest value of the surfactant to be effective as large quantity.

A trail to build a correlation to investigate chamber total thermal resistance with variation of working fluid is done and trend line on figure is drawing according to this correlations. This correlation takes the following form.

$$R_T = A * Q^B \tag{9}$$

Constant *A* and *B* varied with working fluid and are listed in Table 1 with declaration of error percentage.

4.2. Effect of charge ratio

Total average thermal resistances for vapor chamber at different tested working fluids represented at Fig. 6. Average thermal resistance calculated as mathematical average for all tested power levels at each charge ratio. The chamber total thermal resistance that used water as working fluid is lower than that used methyl alcohol as working fluid. When using propylene glycol as water surfactant the chamber total thermal resistance decreased by about 50% of total thermal resistance of that using pure water. For a solution of water with 50% propylene glycol total thermal resistance is

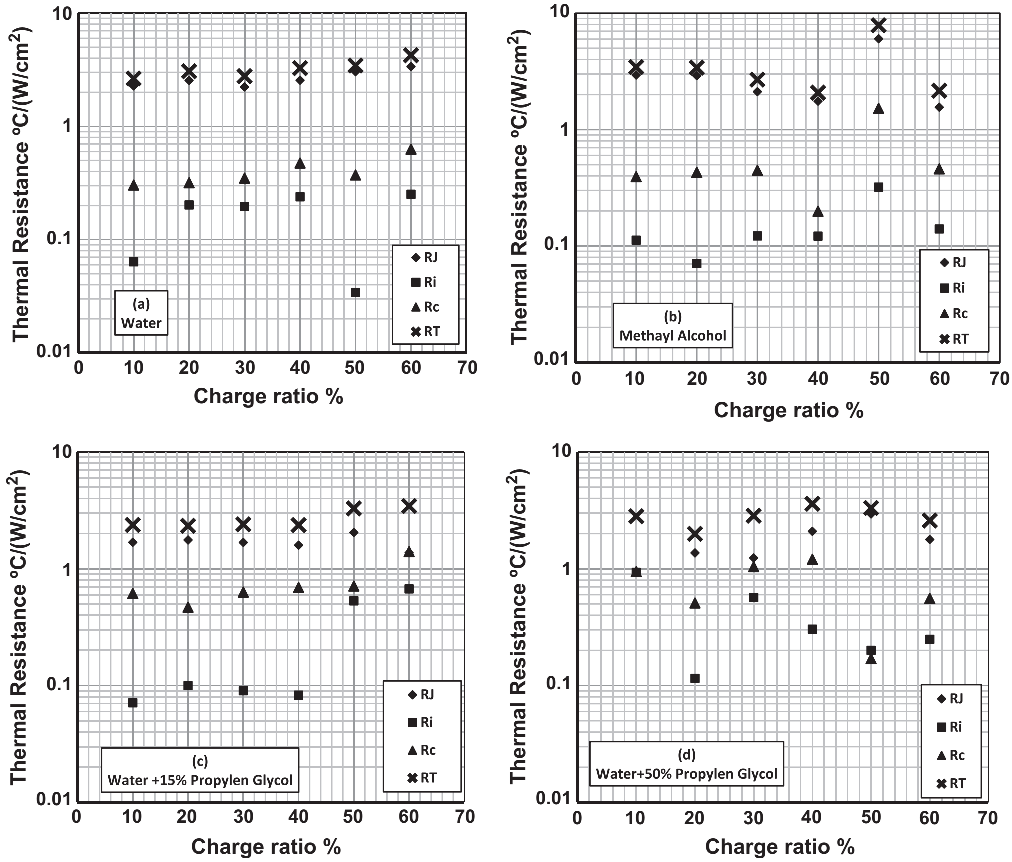

Figure 6 (a) Effect of charge ratio on average different types of vapor chamber thermal, resistance for (a) water, (b) methyl alcohol, (c) water + 15% propylene glycol, and (d) water + 50% propylene glycol as working fluids.

lower than total thermal resistance of solution of water with 15% propylene glycol. For all working fluid junction thermal resistance is a major chamber thermal resistance.

The other two types of resistance, internal resistance and condensation resistance have a lower percentage of total thermal resistance. For water as working fluid charge ratio 10% and 30% are best charge ratios. This is because the more the charge ratio increases the more the thickness of liquid water layer increases and with the fact that the heat transferred through the layer firstly by conduction especially at this very thin layer so that the thermal resistance increases with charge ratio increases.

For methyl alcohol charge ratio 40% is best charge ratio. The methyl alcohol has lower latent heat of vaporization and low density and viscosity than water at the same temperature. From figure the more charge ratio increases the more thermal resistance decreases. This due to that at certain value of the heat load the amount of working fluid is not sufficient so that dray out may occurs at some places in chamber which disappeared with the increase of charge ratio. With the more increase of the charge ratio the effect of liquid layer thickness starts to appear again so that after 40% charge ratio the total thermal resistance increases again.

For a solution of water with 15% propylene glycol the thermal resistance did not change greatly at charge ratios 10%, 20%, 30% and 40%. At these values of charge ratio total thermal resistance is approximately the same. This is almost due to the existence of propylene glycol which increases wet ability of

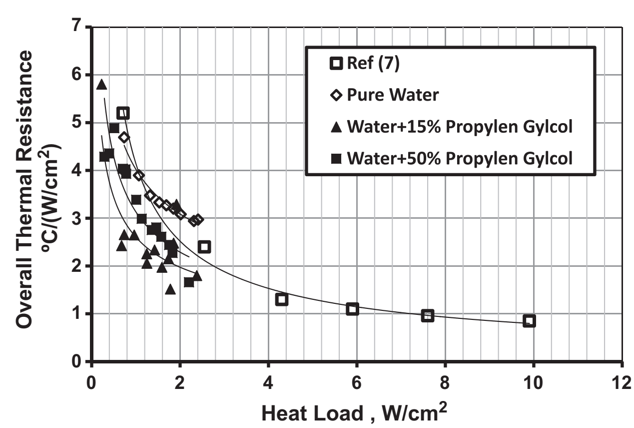

Figure 7 Comparing between total thermal resistances for present work with data extracted, from Ref. [7] (for charge ratio 50% at different input heat loads).

water. After that with increase of charge ratio the effect of layer thickness started to appear. For a solution of water with 50% propylene glycol charge ratio 20% is the best one. After that the effect of thickness of the layer is appeared again.

A comparing between present work and data extracted from Ref. [7] is shown in Fig. 7. The comparing is done at charge ratio 50% and for water and when using water with surfactant. The results show that using propylene glycol will enhance vapor chamber performance than using pure water.

5. Conclusion

Vapor chamber is one of highly effective thermally spread techniques in electronic cooling. In the following a work is conducted to evaluate thermal performance of **2.0** mm high and **50** mm diameter vapor chamber with water and methyl alcohol at different charge ratios. Also, a solution of water and Propylene Glycol at two concentration 50%, 15% tested to study the effect of using surfactant as enhancement agent for vapor chamber working fluid. The chamber total thermal resistance is divided into three types (junction, internal, and condenser resistance) to investigate and determine which type of thermal resistance has a major effect on chamber total thermal resistance.

The results show that using water as working fluid is much better than using methyl alcohol. Charge ratio 30% is best for most tested working fluids. Using surfactant with water is much better than using water only. The more surfactant concentration increases (propylene glycol) the more total thermal resistance of the vapor chamber decreases. The junction resistance has the greatest value of thermal resistance of the vapor chamber about 90% of the total thermal resistance. So that the total thermal resistance reduction process should gets a great attention to reduce junction resistance through enhancement evaporation process.

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