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Air-Cooling System for Metal Oxide Semiconductor Controlled Thyristors Employing Miniature Heat Pipes

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Different cooling methods for metal oxide simeiconductor controlled thyristors (MCTs) are discussed, and an air-cooling system employing miniature heat pipes as thermal spreaders is proposed. Theoretical analyses show that the proposed air-cooling system outperforms many single-phase liquid-cooling systems. A cooling scheme involving installation of miniature heat pipes in the mounting surface of MCTs for single-phase liquid cooling is also described. For the fabrication of miniature heat pipes, an electric-discharge-machining (EDM) wire-cutting method is introduced, which can be used to directly fabricate miniature heat pipes into the heat sink as an integral part of the cooling system. A miniature heat pipe sample is fabricated employing the proposed EDM wire-cutting method and is tested under different working conditions. Experimental results show that the miniature heat pipe works well, with an effective thermal conductance more than 100 times that of copper. The EDM wire-cutting method developed in this article provides a new heat pipe fabrication method that can be used not only for MCT cooling, but for many other applications as well.

Nomenclature

A = heat transfer area, m^2
A_{ct} = total cross-sectional area of the heat pipe, m^2
A_c = cross-sectional area of the heat pipe vapor space, m^2
h = heat transfer coefficient, W/m^2-K
k = thermal conductivity, W/m-K
k_{cu} = thermal conductivity of copper, W/m-K
k_{eff} = effective thermal conductance of the heat pipe, W/m-K
L = length of the heat sink, m
n = total number of fins
P = perimeter of the fin, m
Q = heat transfer rate, W
q = heat flux, W/m^2
R = thermal resistance, K/W
s = pitch of miniature heat pipes in the heat sink block, m
T = temperature, K
T_c = liquid coolant temperature, K
T_∞ = cooling air temperature, K
t = thickness of the miniature heat pipe, m
W = width of the heat sink, m
w = width of the miniature heat pipe, m
μ = dynamic viscosity, N-s/m^2
ν = kinetic viscosity, m^2/s
ρ = density, kg/m^3
ω = angular frequency, rad/s

Subscripts
b = base
c = cross section, or case
conv = convection
f = fin or junction
fb = fin base
hp = heat pipe
jc = connection to case
ms = MCT mounting surface
sp = spreading

Introduction

UNDER the U.S. Air Force More-Electric Aircraft Initiative, metal oxide semiconductor (MOS) controlled thyristors (MCTs) have been developed. The MCT is a solid-state switching device that can handle high currents at reasonably fast speeds. The potential for MCT technology applications is enormous, which includes improving aircraft power system density, survivability and reliability, enabling variable speed motor drives to be reliable and cost effective, and improving the efficiency of air conditioners and other applications that require variable speed control.[1–5] Since the MCT operates at current densities of 100–200 A/cm^2 over a die area of about 1 cm^2, the heat flux generated is on the order of 100 W/cm^2 for low-voltage applications and up to 300 W/cm^2 for 270-V applications. While a single MCT could generate a heat load of 100 W or more, the junction temperature is restricted to a maximum of 150°C.

Because of the local high heat flux, special cooling techniques are needed. Because of their high cooling capacity, liquid-cooling techniques, such as direct immersion cooling, flow-through two-phase cooling, spray and jet impingement cooling, and venturi flow cooling, are under consideration. However, liquid cooling techniques, especially two-phase liquid cooling, are relatively expensive and their applications are not without problems. Common problems associated with two-phase cooling techniques include the thermal instabilities of the coolant and possible leakage of the fluid in the loop, which may present a problem for cooling system reliability and discourage electronic package designers. Spray and jet impingement cooling are effective, but relatively difficult to use. In addition, for a liquid cooling system, a closed circulating loop

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is needed, and the heat absorbed from the electronic devices by liquid must be removed either by a refrigeration or radiator system.

Air cooling is reliable and convenient, but it is usually difficult to sustain high heat flux. However, with a specially designed cooling scheme, a large heat transfer rate is achievable.[6,7] An advanced air cooling system was presented by Biskeborn et al.,[7] in which air was delivered to a plenum containing flow nozzles that supplied air to each of the 22 modules mounted on a circuit board. Up to 3.8 W per chip, 90 W per module, and 1.3 kW per board could be dissipated.

In this article, an air-cooling system using miniature heat pipes to reduce the local heat flux for the MCT cooling is proposed. The miniature heat pipes are embedded or directly manufactured in the heat sink, mounting surface, or electronic devices. Since the heat pipe has a very high thermal conductance,[8-10] the heat generated by the MCT is effectively spread to a larger area. As a result, the heat flux at the heat transfer surface is substantially reduced. A reduction of the heat flux level from an order of 100 W/cm^2 to an order of 1 W/cm^2 at the cooling surface may be possible. Therefore, air cooling in conjunction with cooling fins may be employed for MCT cooling. To directly fabricate miniature heat pipes into the cooling system, an electric-discharge-machining (EDM) wire-cutting method is introduced, and a miniature heat pipe sample employing the EDM wire-cutting method is fabricated and tested. Experimental testing results show that the miniature heat pipe works very well under various working conditions, which proves the feasibility of the proposed air-cooling system.

Heat Sink with Miniature Heat Pipes for Air Cooling

An air-cooling design for MCT employing miniature heat pipes and air impingement is shown schematically in Fig. 1. The MCT is mounted on a copper heat sink, and a number of miniature heat pipes are directly fabricated in the lower part of the heat sink. The miniature heat pipes extend longitudinally in the heat sink and spread the heat from the MCT to a larger rectangular area, $L \times W$. In the upper portion of the heat sink, small fins are machined to enhance the heat transfer between the heat sink and cooling air. The cooling air is supplied through the air duct, distributed to a row of nozzles, and then impinged over the heat sink fins.

To evaluate the heat transfer capacity of the cooling system, a thermal circuit is drawn in Fig. 2 based on the one-dimensional and steady-state assumptions. In the circuit, T_j is the junction temperature of the MCT, T_{ms} is the average temperature at the interface of MCT and the heat sink, T_{fb} is the average temperature at the fin base, and T_∞ is the cooling air temperature at the exit of the nozzle. The junction-to-case thermal resistance is R_{jc}, and R_{sp} is the thermal resistance associated with the thermal spreading function of miniature heat pipes from the MCT to the much larger surface area of the fin base. The thermal resistances from the finned surface to the cooling air and from the unfinned surface to the cooling air are R_f and R_b, respectively. To evaluate R_f and R_b, the impingement heat transfer coefficient between the cooling air and heat sink should be obtained. A literature survey shows that correlations for a single jet and an array of jets impinging normally on a smooth surface are readily available.[11,12] The present jet arrangement with a single row should have an average heat transfer coefficient between those of a single jet and an array of jets. In this analysis, typical values of geometrical parameters related to this cooling application are taken in the calculation, which are 4 mm for the diameter of the round nozzle, 10 mm for the separation distance between the nozzle and heat sink, and 0.016 for the ratio of the nozzle exit cross-sectional area to the corresponding cooled surface area of the heat sink. If the air velocity at the nozzle exit is assumed to be 20 m/s, and the correlation for an array of round jets impinging normally on a smooth surface[11] is used, the average heat transfer coefficient is calculated to be h = 162.0 W/m^2-

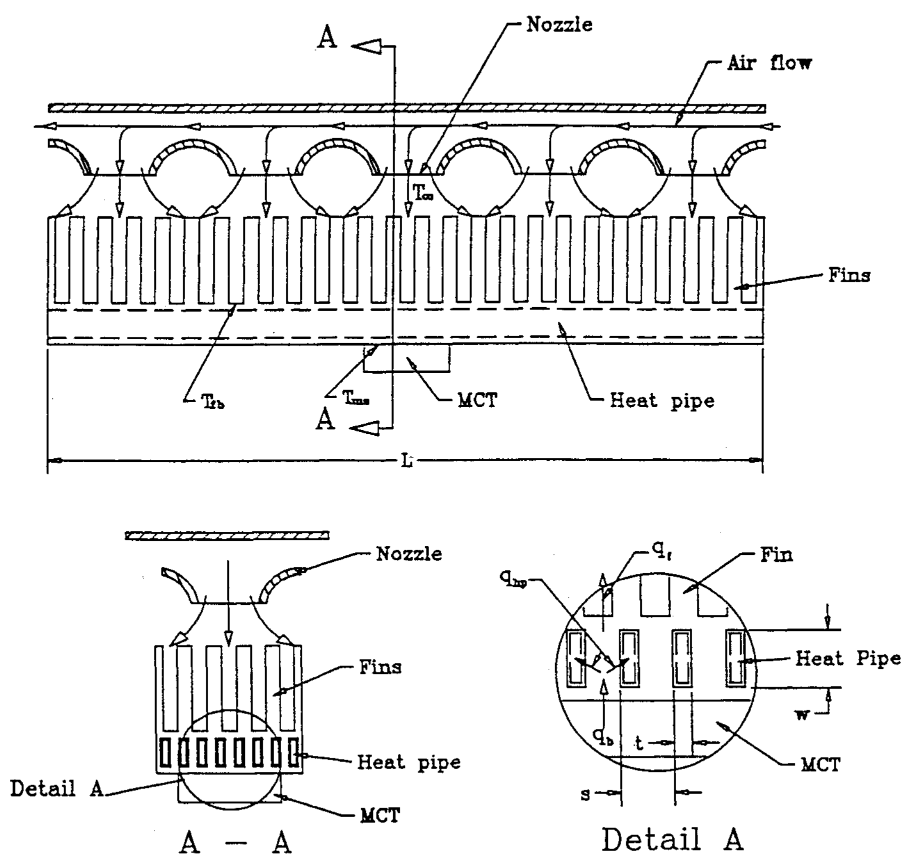

Fig. 1 Schematic of air-cooling system for the MCT employing miniature heat pipes.

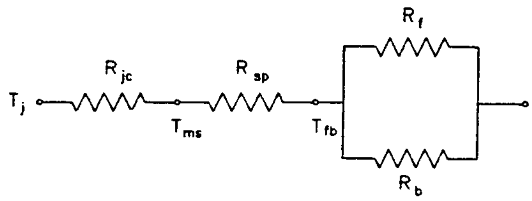

Fig. 2 Thermal circuit for the air-cooling system based on one-dimensional and steady-state assumptions.

°C. Once the heat transfer coefficient is obtained, the thermal resistances R_f and R_b can be evaluated based on the following equations[11]:

$$R_f = \frac{\cosh(mL_f) + (h/mk)\sinh(mL_f)}{n(hPkA_{c,f})^{1/2}[\sinh(mL_f) + (h/mk)\cosh(mL_f)]} \tag{1}$$

where $m = (hP/kA_{c,f})^{1/2}$, and

$$R_b = 1/hA_b \tag{2}$$

where $A_{c,f}$ is the fin cross-sectional area, L_f is the fin length, and A_b is the total area of unfinned surface. Considering a square fin with 2 mm on a side, a length of 15 mm, and a pitch of 4 mm, the total number of fins, based on an overall heat sink dimension of 160 mm (L) $\times$ 20 mm (W), is 200. Following the previous equations, one obtains:

$$R_f = 0.26°\text{C/W} \tag{3}$$

$$R_b = 3.85°\text{C/W} \tag{4}$$

Therefore, the overall thermal resistance between T_{fb} and T_∞ is

$$R_{\text{conv}} = \frac{1}{1/R_f + 1/R_b} = 0.25°\text{C/W} \tag{5}$$

The evaluation of the spreading thermal resistance R_{sp} is somewhat more involved, as it includes heat transfer processes in miniature heat pipes and three-dimensional heat conduction in the adjacent solid. It is also highly dependent on the heat pipe design. To simplify the analysis, an empirical approach for R_{sp} is used here. The spreading thermal resistance is attributed to the longitudinal temperature drop from the center of the MCT to the periphery of the heat sink block, and the transverse temperature drop from the MCT/heat sink interface to the fin base from the conduction in the solid. Since most of the heat from the MCT is absorbed by the heat pipe embedded in the heat sink block and spread in the longitudinal directions, and the distance between the MCT/heat sink interface and the fin base is small in the present case, the transverse conductive thermal resistance should be very small compared to the thermal resistance because of the longitudinal temperature drop. The longitudinal temperature drop is, however, highly dependent on the design details of the heat pipe, and in many cases, should be determined experimentally. Plesch et al.[13] tested a miniature heat pipe with $t = 2$ mm, $w = 7$ mm, and $L = 120$ mm. A typical test result showed that the temperature drop at a horizontal condition with a heat input of 14 W was about 10°C. If eight of these heat pipes are installed in parallel in the heat sink, the total power input would be 112 W, with an average transverse temperature drop of about 5.0°C from the MCT/heat sink interface to the fin base, which corresponds to a thermal resistance of about 0.045°C/W. In the present design, eight miniature heat pipes are considered with an effective length of only about 80 mm (the evaporator is in the middle section of the heat pipe), compared to the 120 mm of heat pipe tested by Plesch et al.[13] If the wick structure is the same, the thermal resistance caused from the thermal spreading function in the present application should be smaller than the 0.045°C/W based on Plesch's heat pipe. After taking into consideration the uncertainties in the analysis, including the interfacial thermal resistance of the conductive grease, the spreading thermal resistance is considered to be

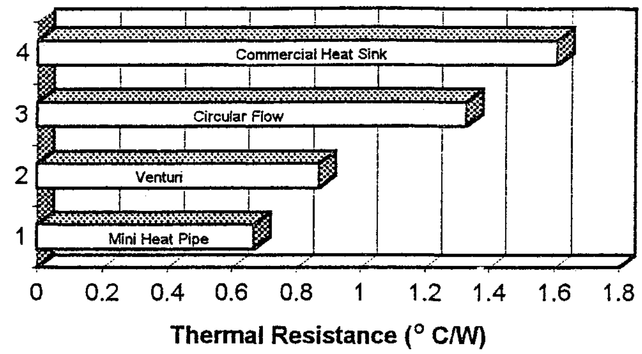

Fig. 3 Thermal resistances for different cooling schemes.

$$R_{sp} = 0.10°\text{C/W} \tag{6}$$

Finally, the junction-to-case thermal resistance R_{jc} should be determined to conclude this analysis. Ponnappan et al.[3,4] conducted MCT cooling tests using three different single-phase liquid-cooling configurations. They found that the values of R_{jc} varied from 0.213°C/W for venturi cooling to 0.421°C/W for commercial liquid heat sink. If an average value of 0.317°C/W is taken and added to the other thermal resistances, the total thermal resistance for the present cooling system would be 0.667°C/W. Compared to the total thermal resistances of single-phase liquid-cooling schemes, 0.87°C/W for venturi cooling, 1.329°C/W for circular flow, and 1.607°C/W for the commercial heat sink, the present air-cooling system outperforms all three liquid-cooling systems (Fig. 3). It should be pointed out that the present result is based on a very simplified analysis and is qualitative in nature. However, it does provide an order of magnitude that shows that direct air cooling with miniature heat pipes has the potential to reach the level of single-phase liquid-cooling systems. Furthermore, it should be pointed out that the air impingement employed in the present cooling scheme is effective but not mandatory. Forced convective cooling across the pin fin surface, for instance, may be used to obtain the same level of cooling capacity.

Like many other heat pipes, miniature heat pipes are subject to some heat transfer limitations and it is preferable to utilize them under a lower heat flux level. Since the heat flux from the MCT is very high, special efforts must be made to reduce the heat flux level into the miniature heat pipes. For this purpose, the miniature heat pipe arrangement shown in Detail A of Fig. 1 is adopted. If the heat flux around the heat pipe perimeter is assumed to be a constant q_{hp}, an energy balance results in

$$2(t + w)q_{hp} + q_f s = q_b s \tag{7}$$

or

$$q_{hp} = \frac{q_b - q_f}{2(t + w)/s} \tag{8}$$

where q_b is the heat flux at the MCT/heat sink interface, q_f is the average heat flux at the fin base, w is the heat pipe width, and t is the heat pipe thickness. It can be seen from the previous relationship that densely packed miniature heat pipes with a slender cross section are preferable for heat flux reduction. For instance, for a heat pipe arrangement with $t = 1.25$ mm, $w = 4$ mm, and $s = 2.5$ mm, the previous equation gives

$$q_{hp} = (q_b - q_f)/4.2 < q_b/4.2 \tag{9}$$

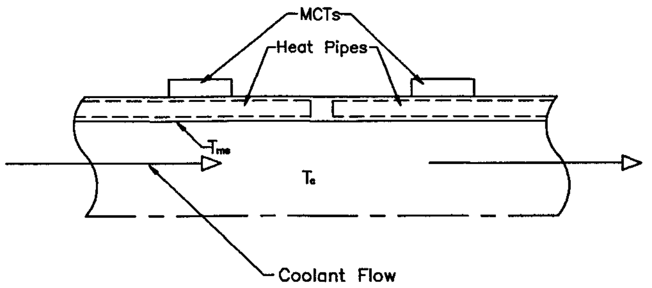

Fig. 4 Liquid cooling employing miniature heat pipes as thermal spreaders.

This indicates that the heat flux can be reduced by at least a factor of 4. It is also important to note from Eq. (8) that the wick structure in the heat pipe should be manufactured entirely around the heat pipe perimeter.

When MCTs work at an even higher heat flux, or an air-cooling arrangement is difficult to realize because of space constraints or air availability, an advanced liquid-cooling scheme may be desired. However, the single-phase liquid-cooling scheme mentioned in the introduction could be improved by employing miniature heat pipes. One of the major reasons that the single-phase liquid-cooling scheme may be less effective than the air-cooling system mentioned earlier can be explained by examining Newton's law of cooling:

$$Q = A_{ms}q_{ms} = A_{ms}h(T_{ms} - T_f) \tag{10}$$

Although h of the liquid-cooling system may be higher than that of the air-cooling system, the actual heat transfer area A_{ms}, for the liquid-cooling system is typically much smaller. For the same heat dissipation rate Q, this results in a higher heat flux and a larger temperature drop between the mounting surface and the coolant. Miniature heat pipes can be used to increase this heat transfer area. Figure 4 shows a single-phase liquid-cooling scheme employing miniature heat pipes, in which the heat pipes are installed in the MCT mounting surface. Because of the employment of miniature heat pipes, the heat generated by the MCT is spread to a much larger area in the mounting surface, and the maximum heat flux at the heat transfer surface between the mounting surface and coolant is significantly reduced. Therefore, under the same liquid-cooling conditions, the maximum MCT temperature can be significantly reduced, and boiling may be avoided. On the other hand, with the same MCT temperature and heat dissipation rate, the requirement for the heat transfer coefficient can be reduced. Since the coolant flow rate and pressure drop are directly related to the heat transfer requirement, both of them can be reduced. Also, as shown in Eq. (10), the requirement for a low coolant inlet temperature can be alleviated.

Fabrication of Miniature Heat Pipes and Experimental Testing Results

In the foregoing analysis for the air-cooling system employing miniature heat pipes, the possible contact resistances between the heat pipe and heat sink block have not been taken into consideration. The most effective way to avoid this contact resistance is to directly fabricate heat pipes into the heat sink as an integral part of the heat sink system. This is very important not only for thermal resistance consideration, but also for cost reduction, when multiple heat pipes are needed in the heat sink. For this purpose, the EDM wire-cutting method was introduced for the fabrication of miniature heat pipes in the heat sink. The advantage of the EDM wire-cutting method is that an array of heat pipes can be fabricated in the heat sink block as an integral part of the heat sink system. The manufacturing costs of the heat sink system can be significantly reduced, and the structural problem associated with the installation of the heat pipes is also substantially alleviated. The EDM method can also fabricate miniature heat pipes with various shapes and small longitudinal grooves for the heat pipe wick structure.

To demonstrate the feasibility of the EDM wire-cutting method, a flat miniature heat pipe was fabricated using the EDM wire-cutting method, shown in Fig. 5, for the overall dimensions and cross-sectional configurations. Small longitudinal grooves are provided around the entire perimeter of the heat pipe, which is one of the requirements for the cooling sink design in Fig. 1. The total length of the heat pipe is 82 mm, with an evaporator length of 19 mm, an adiabatic length of 43 mm, and a condenser length of 20 mm. The cross-sectional dimensions of the vapor space are 0.8 mm (t) $\times$ 5.0 mm (w), and the wall thickness of the heat pipe is 1 mm. Since

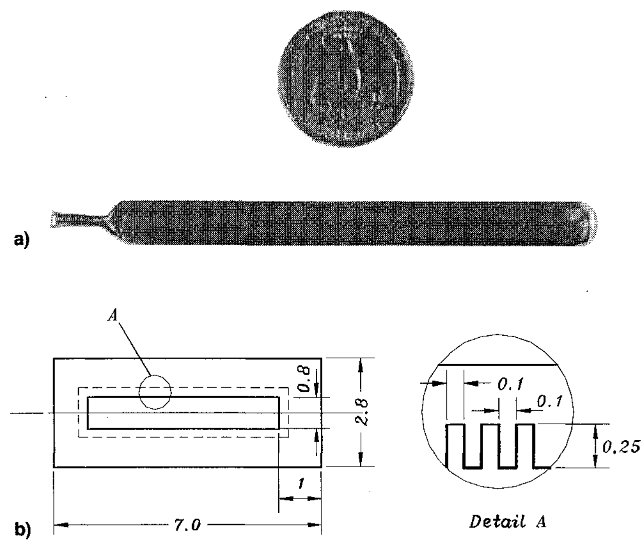

Fig. 5 Sample of miniature heat pipes fabricated using the EDM wire-cutting method: a) overall dimensions and b) cross-sectional dimensions in millimeters.

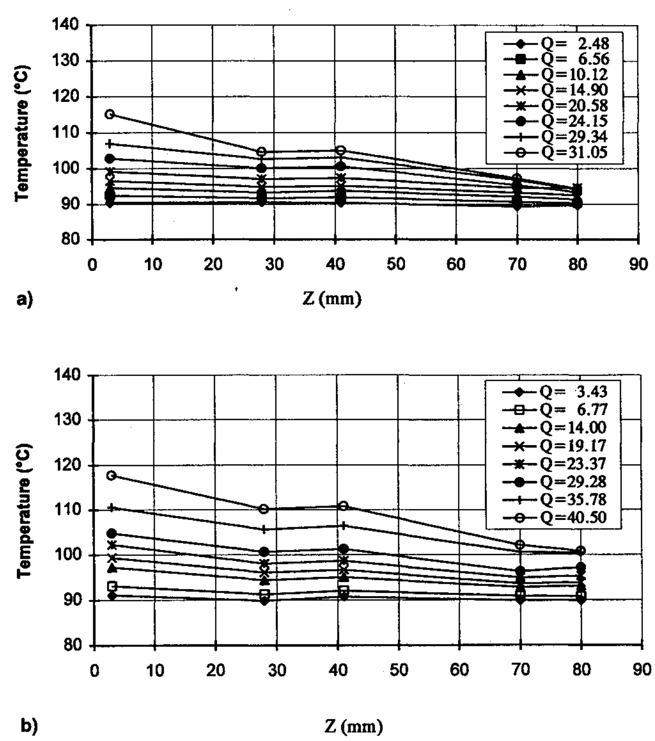

Fig. 6 Experimental results in terms of temperature distributions along the heat pipe length at different heat transfer rates (W): a) horizontal and b) tilt angle of 20 deg.

the miniature heat pipe is to be fabricated directly into the heat sink block, the most crucial geometric parameters in this application are the thickness of the vapor space and the width of longitudinal grooves. The relatively thick wall of the fabricated heat pipe sample shown in Fig. 5 is for the considerations of heat pipe sealing and mechanical strength. When a miniature heat pipe is directly fabricated into the heat sink block, the constraints associated with the heat pipe sample will be significantly reduced.

The experimental apparatus for heat pipe testing consisted of a constant-temperature water circulator, a transformer, a scanning thermocouple thermometer, and a computer. A heater was provided at the evaporator by winding 0.1-mm-diam stainless-steel wires around the evaporator section. The ac power applied to the heater was controlled through a variable transformer and measured with a $4\frac{1}{2}$ digital multimeter. For cooling purposes a cooling jacket was attached to the condenser section, which was then connected with the constant-temperature circulator through plastic tubings to form a coolant loop. The temperatures and flow rates of the circulator could be adjusted for desired testing conditions and heat pipe operating temperatures. Both the evaporator and adiabatic sections were thermally insulated with asbestos papers to reduce heat loss. Heat loss was determined by measuring the temperature difference between the insulation outer surface and the ambient air at different working temperatures. The measured temperature differences were then used to evaluate the heat loss because of natural convection. It was found that within the test range, the maximum heat loss was less than 0.3 W, which was negligible compared to most heat inputs in the test.

Five type-*k* fine wire thermocouples (NiCr–NiAl) were placed along the heat pipe length to measure the surface temperature distributions. The distances of five thermocouples from the evaporator end cap were 3, 28, 41, 70, and 80 mm, respectively. The data acquisition system included a 12-channel scanning thermocouple thermometer, which had a maximum uncertainty of ±0.4°C, and a computer to store the data. The thermometer featured a field calibration for each channel to eliminate the errors of the thermocouple probe and a system for accurate measurement. Within the test range, the errors of measurements were estimated to be less than 1% for temperature and 2% for power input. The test procedures included setting the cooling water temperature and changing the power inputs from a lower level to a higher level. When the temperature in the evaporator section of the heat pipe increased considerably, it was believed that the heat transfer limit was being reached, and the testing was terminated.

Figure 6 presents some typical testing results in terms of temperature distributions along the heat pipe length for different heat inputs in the evaporator section, with the heat pipe in a horizontal orientation (Fig. 6a), and at a tilt angle of 20 deg (Fig. 6b). The maximum heat transfer rate of the heat pipe is about 40 W for the testing cases shown in the figure. Although an even higher heat transfer rate was obtainable, a considerably large temperature drop along the heat pipe length occurred, indicating that the capillary limit was being reached. The maximum heat flux in the evaporator was about 18.3 W/cm^2. This heat flux level could be even higher if a shorter evaporator length is considered, since the maximum heat transfer rate is limited by the capillary capacity. The tilt angle also had a significant effect on the heat pipe performance. For the test results in Fig. 1b, the positive tilt angle resulted in a reflux operation mode, which helped to increase the maximum heat transfer rate, compared to the results for the horizontal position. The heat pipe performance in this study was generally better than the heat pipe performance level adopted in evaluating the value of R_{sp} in Eq. (6).

One important criterion for judging heat pipe performance is the effective thermal conductance of the heat pipe. Figure 7

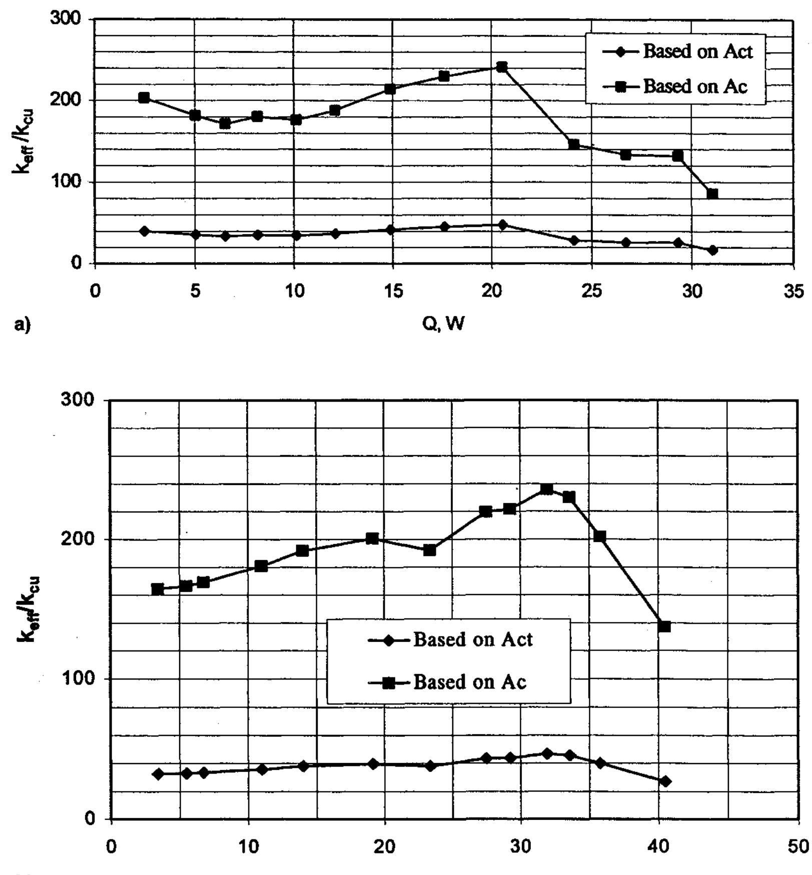

Fig. 7 Ratio of heat pipe effective thermal conductance to copper thermal conductivity as a function of total heat inputs: a) horizontal and b) tilt angle of 20 deg.

shows the corresponding effective thermal conductance for the testing results shown in Fig. 6, in terms of the ratio of heat pipe effective thermal conductance to the thermal conductivity of copper. The thermal conductivity of copper in Fig. 7 was taken to be 380 W/m-K. The effective thermal conductance, based on A_{ct} was very large, considering the small length of the heat pipe tested. As mentioned earlier, the miniature heat pipe is intended to be fabricated directly into the heat sink block. Therefore, the effects of the heat pipe wall on the heat pipe performance should be excluded when evaluating the performance of the tested heat pipe sample. The effective thermal conductance of the heat pipe is recalculated based on A_c, and the results are also plotted in Fig. 7. As can be seen, the effective thermal conductance reaches more than 200 times that of copper, indicating the heat pipe works very well. The present testing results prove that the proposed EDM wire-cutting method is feasible for the fabrication of the miniature heat pipe in the heat sink. In addition, the EDM wire-cutting method developed in this research provides a new heat pipe fabrication method that can be used for many other applications as well. The performance of the miniature heat pipe could be further improved by optimizing the groove structure and vapor space dimensions, which is one of the objectives of future study.

Conclusions

1) For effective cooling of MCTs, an air-cooling system employing miniature heat pipes has been proposed. A theoretical analysis showed that the proposed air-cooling system outperforms many single-phase liquid-cooling systems; therefore, some technical difficulties associated with liquid cooling can be avoided while maintaining a high cooling capacity.

2) For liquid-cooling schemes, miniature heat pipes can be installed in the mounting surface of the MCTs to spread heat to a larger area, allowing the liquid-cooling capacity to be significantly improved.

3) An EDM wire-cutting method can be used to fabricate miniature heat pipes directly into the heat sink as an integral part of the cooling system. As a result, the thermal resistance between the heat pipe and heat sink can be eliminated, and the manufacturing costs of the cooling system can be reduced.

4) A miniature heat pipe sample was successfully fabricated employing the proposed EDM wire-cutting method. The testing results indicate that the heat pipe works very well under various working conditions. Therefore, the EDM wire-cutting technique developed in this research provides a new heat pipe fabrication method that can be used for various applications, including the cooling of MCTs.

Acknowledgments

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References

[1]Nguyen, B. T., ''MOS Controlled Thyristor (MCT) Characteristics Under Advanced Liquid Cooling,'' Aerotech '93 SAE Conf., Costa Mesa, CA, Sept. 1993.

[2]Reyes, A. S., Nguyen, B. T., Weimer, J. A., Beam, J. E., and Chow, L., ''Operation Characteristics of a MOS Controlled Thyristor (MCT) Using a Liquid Cooling Approach,'' Intersociety Energy Conversion Engineering Conf. Paper 929477, 1992.

[3]Ponnappan, R., Leland, J. E., Chang, W. S., Beam, J. E., Nguyen, B. T., and Weimer, J. A., ''Active Cooling of MCT Using Venturi Flow,'' Intersociety Energy Conversion Engineering Conf. Paper 93457, 1993.

[4]Ponnappan, R., Leland, J. E., Beam, J. E., Fronista, G., and Weimer, J. A., ''Effective Cooling of MCT and IGBT Using Venturi Flow,'' Intersociety Energy Conversion Engineering Conf. Paper AP-87, 1995.

[5]Mahefkey, T., ''Thermal Management of High Temperature Power Electronics Limitations and Technology Issues,'' High Temperature Electronics Conf., Charlotte, NC, June 1994.

[6]Aung, W., *Cooling Technology for Electronic Equipment*, Hemisphere, New York, 1986.

[7]Biskeborn, R. G., Horvath, J. L., and Hultmark, E. B., ''Integral Cap Heat Sink Assembly for the IBM 4381 Processor,'' *Proceedings of the 4th International Electronic Packaging Society Conference*, 1984, pp. 468–474.

[8]Cao, Y., Faghri, A., and Mahefkey, T., ''Micro/Miniature Heat Pipes and Operating Limitations,'' American Society of Mechanical Engineers HTD-Vol. 236, 1993, pp. 55–62.

[9]Peterson, G. P., *Heat Pipes*, Wiley, New York, 1994.

[10]Faghri, A., *Heat Pipe Science and Technology*, Taylor and Francis, Washington, DC, 1995.

[11]Incropera, F. P., and Dewitt, D. P., *Introduction to Heat Transfer*, 2nd ed., Wiley, New York, 1990.

[12]Martin, H., ''Heat and Mass Transfer Between Impinging Gas Jets and Solid Surfaces,'' edited by J. P. Hartnett and T. F. Irvine Jr., *Advances in Heat Transfer*, Vol. 13, Academic, New York, 1977.

[13]Plesch, D., Bier, W., Seidel, D., and Schubert, K., ''Miniature Heat Pipes for Heat Removal from Microelectronic Circuits,'' American Society of Mechanical Engineers DSC-Vol. 32, 1991, pp. 303–313.