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PERFORMANCE STUDY OF THERMOLOOP: EFFECT OF EVAPORATOR FILL RATIO AND CONDENSER CONDITION

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ABSTRACT

Thermoloop, a special kind of two-phase loop thermosyphon, is a highly promising micro-electronics cooling technology which has already received much attention of the researchers. In the present study, an experimental investigation has been conducted to determine how the variation of liquid fill ratio of the evaporator and convection condition of the condenser influence the performance of the thermoloop. The thermoloop was tested at a heat input of 75 watt by varying the liquid fill ratio from 20% to 80% for cases where the condenser dissipated heat by natural convection and forced convection. The experiments were carried out keeping the thermoloop device horizontal. It was found that the time for the completion of a heat transport cycle (the return of the liquid back into the evaporator after first being evaporated in the evaporator and then being condensed in the condenser, which occurs in a periodic manner) increases with evaporator fill ratio and that the maximum temperature of the evaporator does not vary considerably. The maximum pressure developed within the system increases with increasing evaporator fill ratio for both type of condenser convection conditions. Variation in the convection condition of the condenser seems to have very insignificant influence on the heat transfer performance such as maximum evaporator temperature, pressure developed within the system and cycle frequency.

Keywords: Thermoloop, liquid fill ratio, heat transport cycle, phase change heat transfer

1. INTRODUCTION

Over the past few years, one of the major trends in the electronic industry has been the miniaturization of the electronic devices which leads to the electronic components with greater power demand and higher packaging density. Among the liquid cooling processes, indirect liquid cooling techniques are drawing more attention of the researcher's to avoid the design complexities associated with the direct liquid cooling. Two-phase indirect liquid heat transfer is a very appealing cooling process as it has been long recognized that high heat flux removal can be achieved through vaporization of the liquid in an evaporator attached to the heat source. The most common passive, two-phase heat transfer devices today, capable of operating against gravity are the different types of heat pipes [1-3] all of which require very sophisticated wick structure. The high cost associated with these devices place limit on their applicability at this moment for micro-electronics cooling in spite of the excellent thermal performance shown by them. But thermoloop [4], a pulsated two phase loop thermosyphon, can be a very useful and commercially effective thermal solution for the modern electronics cooling which can overcome the limitation of gravity dependence of simple loop thermosyphons.

Thermoloop is regarded as a very promising solution for the high end electronics cooling because of its ability to meet the requirement of very high heat flux dissipation and also because of its promise to represent a low cost solution. The thermoloop device consists of an evaporator, a condenser, the vapor and liquid lines and differs from the loop thermosyphon by the addition of two flow controllers and a flexible reservoir/ accumulator placed between them.

Very few experimental tests have been performed on the realization and feasibility of loop thermosyphons as a cooling device for high end electronics cooling. The idea of a loop thermosyphon, a particular pump-less heat transfer device operating against gravity without any capillary structure was first realized by Sasin et al. [5]. Filippeschi [6] analyzed all the periodic two phase heat transport devices that operate against gravity and given these particular devices the generic name of periodic two phase thermosyphon (PTPT). Fantozzi et al. [7] reported the gravity independence periodic heat transfer regime of a mini PTPT, by placing the condenser above and below the evaporator. They also reported the experimental results on the feasibility of two different kinds of PTPT to cool desktop computer processor [8].

Web et al. [9] reported the tests on a loop thermosyphon which is capable of dissipating a heat flux of 100W from a desktop computer CPU. Pal et al. [10]

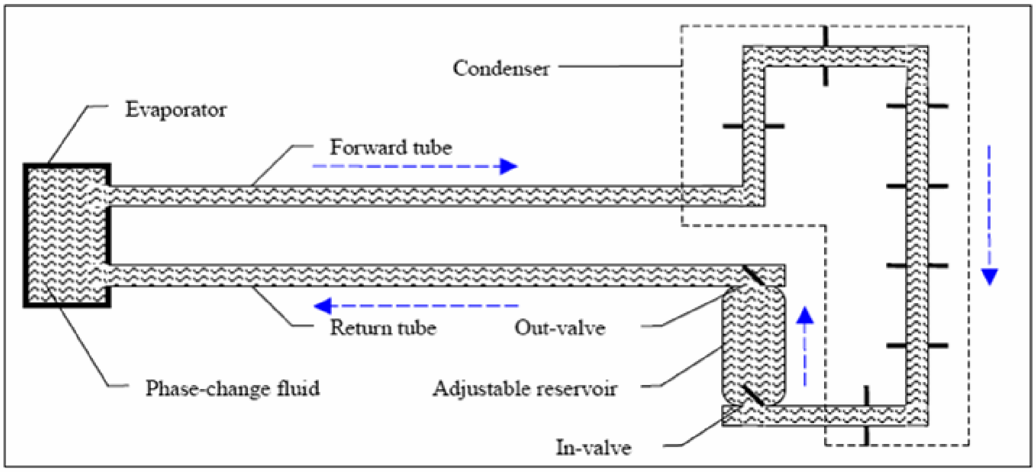

Fig 1: Schematic of thermoloop (Courtesy: Alam Thermal Solutions, Inc., USA)

experimentally investigated the effect of working fluids and the inclination of the thermosyphons for the cooling of a Pentium-4 microprocessor in a Hewlett Packard Vetra PC with a peak heat dissipation of 80W. The result of these works showed the prospect of loop thermosyphon as the future solution of high density electronics cooling in the respect of cost competitiveness, flexibility of design and heat dissipation capacity.

Alam [4] came up with a design alternative to name it the thermoloop and performed few experimental tests identifying the key performance parameters. But no further investigation has been found so far on the realization and feasibility of thermoloops as the cooling device for electronic component and on the optimization of its design.

The working principle of the thermoloop device is shown in Fig. 1. While in idle, the thermoloop device is completely filled with working fluid and the bellow remains flaccid. The thermoloop device sets in operation automatically when the evaporator temperature slightly exceeds the boiling point of the working fluid. The device operates periodically in the following two sequential and repeating states –Vapor Transfer State and Condensate Return State.

Vapor Transfer State: This state begins when boiling initiates inside the evaporator. As heat is supplied into the evaporator, vapor bubbles form, coalesce, and push the liquid out from the forward tube and condenser tube into the bellow trough the In-Valve keeping the Out-Valve closed. At this stage, the rate of evaporation is much more than the rate of condensation in the condenser. Starting from flaccid, the bellow expands and provides room for incoming liquid. Thus, the forward tube and part of condenser tube get filled in with saturated vapor replacing liquid. Pushing out liquid and replacing liquid with vapor continue as long as evaporation rate is slightly higher than condensation rate and this phenomena end up when these rates become equal.

Condensate Return State: At the end of Vapor Transfer State, all liquid in the evaporator gets vaporized and the steady increase in pressure suddenly comes to an end. At this stage, the evaporation rate is much lower than the condensation rate. As a result, the pressure in the evaporator to condenser suddenly drops and stored liquid in the bellow quickly enters into the evaporator via Out-Valve and repeat the cycle.

Both the states continue to repeat as long as the device is in operation. The device adjusts its functionality (cycle frequency, volume occupied by vapor, rate of heat absorption and rate of heat dissipation) depending on the thermal load at the evaporator and the heat removal rate from the condenser.

In the present study, experiments were performed placing the thermoloop horizontally for a constant power input and the results were examined to determine the influence of evaporator liquid fill ratio on the heat transfer performance such as temperature of the evaporator, temperature range of condenser inlet and outlet, cycle speed and the pressure developed within the system etc. The condenser convection condition of the thermoloop was also varied and its effect on the heat transfer performance was analyzed.

2. EXPERIMENTS

For the present study, the experimental set-up consisting of the test specimen and other equipments were mounted on two vertical column frames placed at a distance apart (Fig. 2). For the thermoloop, a rectangular evaporator was used which was made of copper and had the outside dimension of (72.5× 60× 20) mm^3. The inside volume of the evaporator was 75 cc. The internal surface of the evaporator was grooved to facilitate the boiling enhancement characteristics. A spiral shape condenser was used which was made of copper also, having 6 turns. The diameter of the condenser coil was 6.35 mm. There were two flow control valves placed on either side of the adjustable reservoir. The forward and return tubes both had an outside diameter of 6.35 mm (1/4 inch) and were 0.04 inch thick. The forward and return lines had the same total length, each being 56 cm long. A flexible reservoir with the size (6×2×6.4) cm^3 was connected in

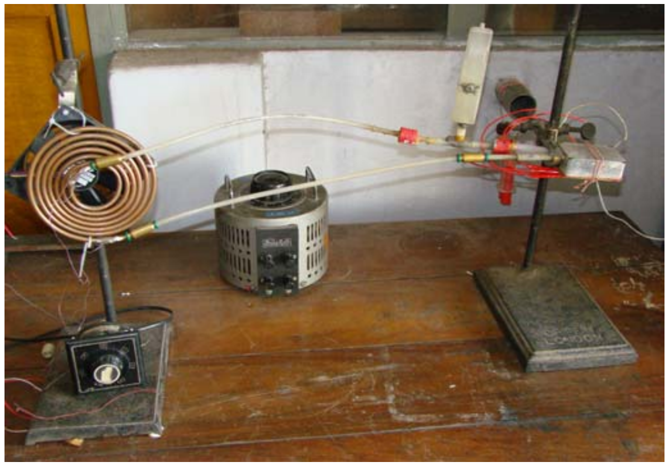

Fig 2: Front view of the complete test section

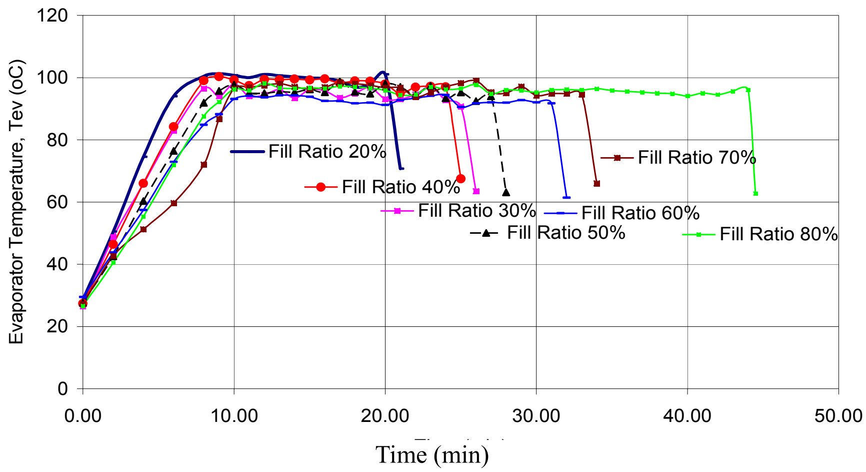

Fig 3: Variation of evaporator temperature for different fill ratios with time under natural condenser convection

the return line at a distance of 46.5 cm from the condenser. Two one-way check valves were placed on either side of the reservoir at a distance of 5 cm each.

The experiments were performed using water as the working fluid. The water was injected using a syringe and the forward and return lines, and the condenser was completely filled. The fill ratio of the evaporator was varied from 20% (15 cc) to 80% (60 cc). Considerable care was taken to eliminate the formation of bubble or gas pockets in the forward and return lines and also in the condenser. Two flat AC heaters (1.5 kΩ, RH-50), each of 50 watts were connected in parallel to the bottom of the evaporator. The heaters were connected using thermal interface material (Thermolite®) to improve the thermal contact. All the tests were carried out by placing the device horizontally (i.e. keeping the evaporator and condenser at the same level of elevation). A forced draft axial fan (average air velocity 4.75 m/s) was attached to the condenser for the forced convection condition. All the tests were performed at an ambient temperature of 28 ± 2 ^{0}C.

Thermocouples (K-type) were connected to evaporator and condenser inlet and outlet to measure the temperature. Height of the accumulated liquid in the reservoir was measured which served as an indicator of the pressure in the system. During the experiment, the heat input was kept constant and the temperature of the evaporator and condenser and the height of the accumulated water in the reservoir were recorded at a regular interval of one/two minutes.

3. RESULT AND DISCUSSION

In the present investigation, the liquid fill ratio of the evaporator was varied from 20% to 80% for both natural and forced convection condition of the condenser. Evaporator temperature, outlet temperatures of condenser inlet and outlet and height of the water column accumulated in the reservoir was measured at a constant heat input by varying evaporator fill ratio and condenser convection condition.

Fig. 3 shows the variation of evaporator temperature (T_{ev}) with time (t) for all the fill ratios under natural convection condition of the condenser. The temperature of the evaporator was observed to increase linearly with time up to the boiling temperature of the liquid, then became steady as boiling started. This temperature remained constant until the pressure in the evaporator reaches a lower value than that of the reservoir. Then liquid flows back into the evaporator from the reservoir and temperature of the evaporator drops down. The evaporator temperature varies in a similar fashion in the following cycles. The same event occurs for all the fill ratios, but the cycle time increases with increase in the fill ratio. It is seen from the graph that the cycle time increased from 21 min for 20% fill ratio to 28 min for 50% fill ratio and 45 min for 80% fill ratio. This experimental behavior is quite predicted and in congruence with the theoretical consideration. The greater the amount of liquid inside the evaporator (fill ratio), the greater time it takes to completely evaporate all the liquid inside it. Greater time is also required to condense all the vapor produced and to produced and to develop a sufficient pressure gradient so that the accumulated liquid in the reservoir can exert sufficient pressure to open the 2nd check valve and enter into the evaporator, i.e. complete the cycle. The maximum temperature of the evaporator for all the fill ratios remains nearly constant, the variation being less than 5^0C.

It was also observed that the time required for completion on one cycle decreases after the 1st cycle and the cycle time becomes nearly constant after two or three cycles. From Fig. 4 it can be seen that for the 20% fill ratio, while the time required for the 1st cycle is 20.5 min, it is 20.5 min and 19 min for the 2nd and 3rd cycles and then becomes constant for the later cycles. The same thing is true for the other fill ratios. The underlying reason is that, for the 2nd and later cycles, the time required for evaporating is reduced significantly as the evaporator is already at an elevated temperature. The temperature of the evaporator at the beginning of the 1st cycle is room temperature (28.8^0C), while its temperature is about 45^0C higher (about 72^0C) for the 2nd and successive cycles. Due to evaporator remaining at a higher temperature, the time for evaporating all the liquid inside the evaporator decreases and the cycle time decreases as a result. The same trend is observed for both kind of convection condition of the condenser.

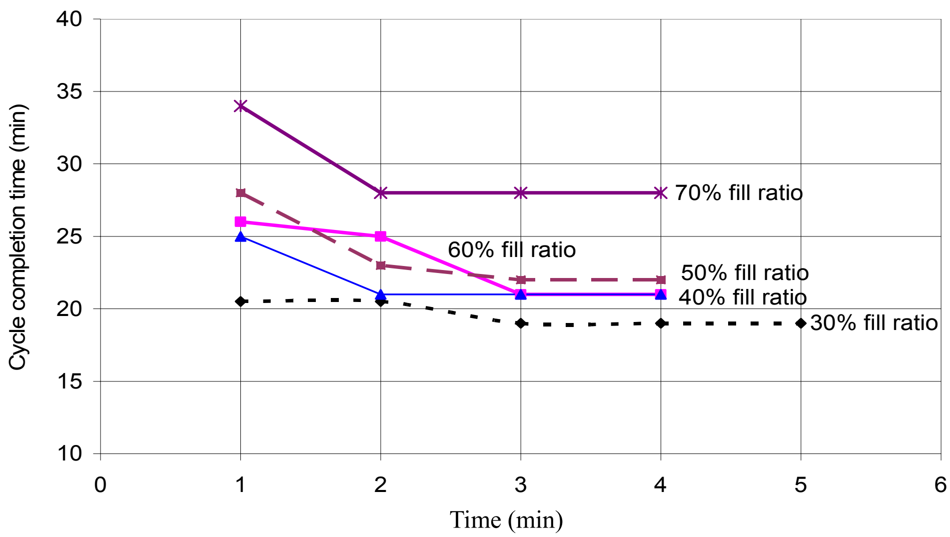

Fig 4: Variation of cycle time with no. of cycles for various fill ratios

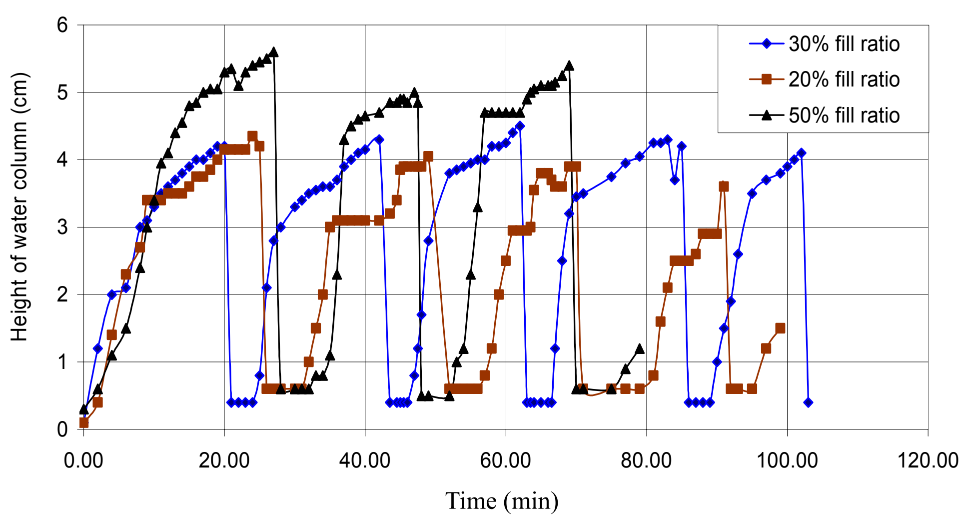

Fig 5: Variation of pressure as a function of time for different fill ratios

The pressure development in the system (as indicated by the height of the accumulated water in the reservoir, h) is also shown as a function of time for different fill ratios (Fig. 5). It is observed that the pressure rises rapidly as long as the temperature in the evaporator increases, attains a steady value as boiling ceases, and then the pressure drops when the cycle is completed. The value of the maximum pressure developed in the system increases with increase in the fill ratio due to greater amount of water vapor produced. As seen from the graph, the maximum height of the water column in the reservoir is 5.6 cm for 50% fill ratio while it is 4.05 cm for 20% fill ratio.

Inlet temperature of the condenser (T_1) varies in a similar manner as that of evaporator temperature for all fill ratios under natural convection of the condenser (Fig. 6). Maximum temperature of the condenser inlet was practically same for all fill ratios tested, the value being nearly around 100^0C. Outlet temperature of the condenser (T_2) was also showed similar trend for all fill ratios tested.

3.1 Effect of Convection Condition of the Condenser

The variation of all the system parameters with time when a fan is used to cool the condenser is shown in Fig.7. The addition of fan showed very little effect on the heat transfer performance of the thermoloop. The cycle completion time or cycle time reduced slightly for one or two fill ratios, but the overall effect on the cycle speed is very insignificant.

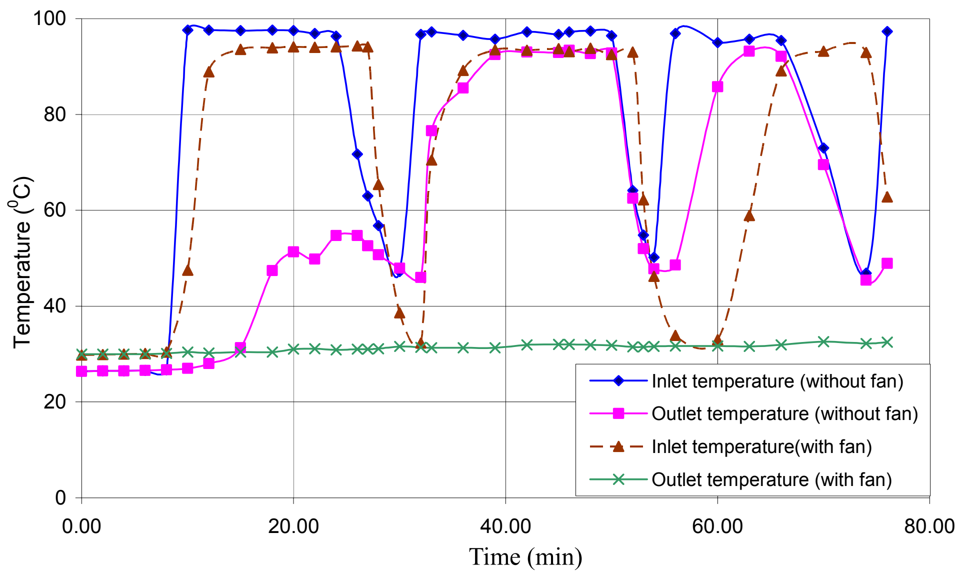

Fig 6: Condenser temperature variation with time for 30% fill ratio (with and without fan

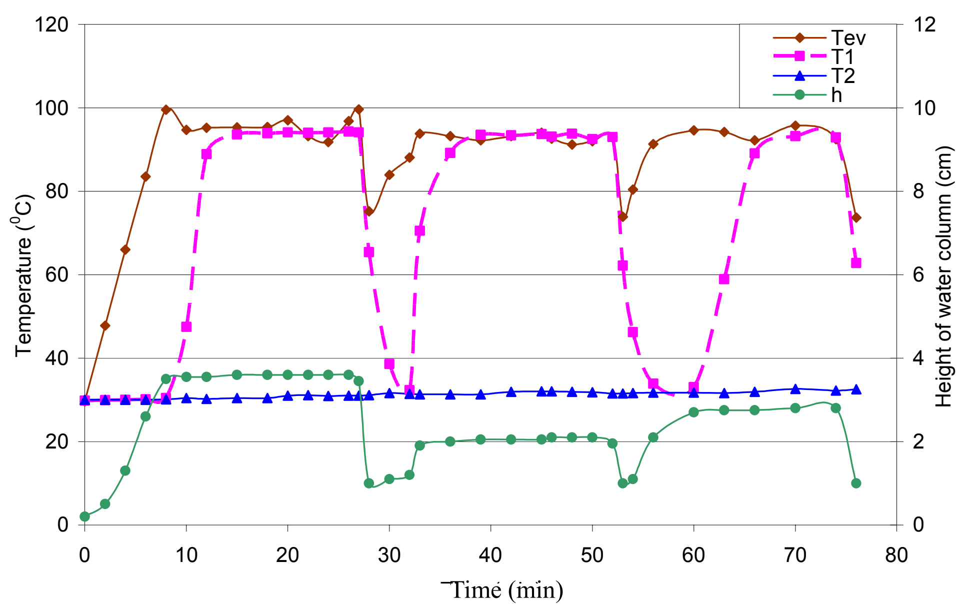

Fig 7: Variation of system parameters for 30% fill ratio (with fan)

The condenser inlet temperature varied similarly as that of naturally cooled condition when a fan is used, and the maximum condenser inlet temperature was nearly equal for both convection types (fig. 6). Condenser outlet temperature changed significantly by the addition of the fan. Condenser outlet temperature remained nearly constant throughout all the cycles and the value was about 31^0C for all the fill ratios tested. Fig. 7 shows variation of temperatures with time for the fill ratio of 30% when a fan is used. The pressure development in the system seemed to be independent of the convection condition of the condenser. It showed similar nature for. all the fill ratios under both type of convection condition.

Although the convection condition of the condenser did not influence the heat transfer performance of the thermoloop (cycle frequency, maximum temperature of the evaporator, pressure development in the system etc.) for low heat input for which the tests were performed, it is likely to have more prominent effect on the heat transfer performance for greater heat input

4. CONCLUSIONS

The experimental results performed on a thermoloop device to study the influence of variation in the evaporator fill ratio and condenser convection condition on the heat transfer performance have been presented in this paper. The cycle completion time was observed to increase continuously with the increase in the fill ratio and maximum temperature of the evaporator (which simulates the microprocessor temperature of the computer, for example) remained nearly same for all the cases tested both when the condenser was cooled naturally and by fan. The cycle time was less for the 2nd and later cycles than the 1st cycle and became constant and steady after two or three cycles for all the fill ratios examined. The maximum condenser inlet temperature was also almost constant for all liquid fill ratios tested. Condenser outlet temperature varied significantly under different convection condition. The condenser convection condition had very insignificant effect on the maximum temperature of the evaporator and also on the cycle time. Maximum pressure developed in the system also increased with increase in the fill ratio and was not affected significantly by the condenser convection condition.

5. REFERENCES

1. Maydanik, Y.F., “Loop heat pipes- theory, experimental developments and application”, KN-19, 13th International Heat Transfer Conference, 13-18 August, 2006, Sydney, Australia.
2. Kawaji, M., Nikkanen, K. and Lu, C.G., “Experimental study of performance of looped and unlooped pulsating heat pipes”, EQP-34, 13th International Heat Transfer Conference, 13-18 August, 2006, Sydney, Australia.
3. Wang, S. and Nishio, S., “Heat transfer performance of a closed oscillating heat pipe”, EQP-38, 13th International Heat Transfer Conference, 13-18 August, 2006, Sydney, Australia.
4. Alam, M., “Thermoloop heat transfer technology”, 9th AIAA/ASME Joint Thermophysics and Heat Transfer Conference, 5-8 June 2006, San Francisco, California.
5. Sasin, V. J., Borodkin, A. A. and Feodorov, V. N., “Experimental Investigation and analytical modelling of autoscillation two-phase loop”, 9th IHPC, Los Alamos, U.S.A. June 1995.
6. Filippeschi, S., “On periodic two-phase thermosyphons operating against gravity”, International Journal of Thermal Sciences, Vol-45, pp 124-137, 2006.
7. Fantozzi, F., Filippeschi, S. and Latrofa, E. M., “Upward and downward heat and mass transfer with periodically operating loop thermosyphons”, Superlattices and Microstructures, Vol-35, pp 339-351, 2004.
8. Fantozzi, F., Filippeschi, S. and Latrofa, E. M., “Miniature pulsated loop thermosyphon for desktop computer cooling: feasibility study and first experimental tests”, 5th Minsk International Seminar on Heat pipes, Heat Pumps, Refrigerators, 8-11 September, 2003.
9. Webb, R.L et al. “Thermosyphon Concept to cool Desktop computers and Servers”, Proceedings of the International Electronic Packaging Technical Conference, 2001.
10. Pal A., Joshi Y., Beitemal A.H., Patel C.D., and Wenger T., “Design and Performance Evaluation of a Compact Thermosyphon”, Proceedings of the United Engineering Education, Thermas santa Fe, New Mexico, January 2002.