Design and Performance Evaluation of a Compact Thermosyphon

Aniruddha Pal & Yogendra Joshi
Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, Georgia

Monem H. Beitelmal & Chandrakant D. Patel
Hewlett-Packard Research Laboratories, Palo Alto, California

Todd Wenger
Thermacore International Inc., Lancaster, Pennsylvania

ABSTRACT: Thermosyphons are a great prospect towards cooling of high heat dissipating electronics. The implementation of a compact thermosyphon for cooling of a Pentium 4 microprocessor in a Hewlett Packard Vectra PC is being presented in this paper. The thermosyphon involves four components in a loop: an evaporator with a boiling enhancement structure, a rising tube, a condenser and a falling tube. Experiments are done to assess the effects of working fluids and of system inclination on the performance of the thermosyphon loop. Numerical simulations are performed to reflect on the efficiency of the condenser in natural convection condition and the effect of the system fan on the performance of the condenser.

1 INTRODUCTION

Personal computers are currently air-cooled by either free or forced convection. Fans attached to the central processing unit (CPU) are the most popular cooling devices for low power dissipation systems. They are popular because of reliability, cost, efficiency and ease of implementation. However, for higher frequency chips (above 1000 MHz), the air-cooled heat sinks have some limitations in the form of bulky size, noise and insufficient cooling performance. As a result, current practice of dense packaging of electronics in compact spaces demands novel ways of heat dissipation, which will be able to dissipate as much as 100 W/cm^2 at chip levels while maintaining the device at acceptable temperatures, typically below 85°C.

One of the proposed solutions for dissipating high heat fluxes is the use of liquid cooling techniques. Two-phase heat transfer is a very attractive liquid cooling process, as high heat fluxes can be removed through vaporization of the fluid in an evaporator attached to or enclosing the heat source, but the added process of condensing the vapor and bringing it back to the evaporator adds complexity in the system.

A thermosyphon is such a device, which successfully implements two-phase liquid cooling by indirect contact with electronics. A two-phase thermosyphon basically consists of an evaporator and a condenser, which are connected through a passage, or a loop. Heat is transferred from the source through an interface to the evaporator, where the fluid vaporizes taking the latent heat of vaporization. The vapor then moves to the condenser, where it condenses. The released heat is dissipated to the ambient from the condenser and the condensed liquid is returned to the evaporator, thus completing a closed loop. The density difference between the liquid and vapor creates a pressure head, which drives the flow through the loop, and as such no driving force is needed. Thermal resistance at the interface between the heat source and the evaporator is a key design parameter. This must be minimized in order to get the benefit of low thermal resistance of the evaporator.

2 BACKGROUND

Tengblad and Palm (1996) tested several prototypes of thermosyphons with single and multiple evaporators. Ramaswamy et al. (2000) studied the combined effects of pressure and sub-cooling on the performance of two-chamber thermosyphons with gradually increasing heat fluxes. Lang et al. (2001) took the concept further and studied the effects of imposed circulation and location of condenser on the performance of a two-phase thermosyphon in a confined space. Garner et al. (2001) looked into the applications of loop thermosyphon (LTS) in high-density packaging. Recently, Webb et al. (2001) demonstrated a system capable of dissipating 100 W from a single CPU system.

   The above-mentioned works established the importance and the prospect of thermosyphon as a compact thermal management device, both in terms of flexibility in size and cooling capacity range. While laboratory demonstrations of thermosyphons exist in the above literature, implementation of an actual device for a current microprocessor has not been presented. This work describes the design, construction and performance assessment of a two-phase compact thermosyphon for a Hewlett Packard Vectra VL800 PC with a peak microprocessor power dissipation of 80 W.

3 DESIGN PROCESS

For designing the thermosyphon from ground up, the following steps are taken:

- Verifying Cooling Requirements
- Developing Compact Loop Thermosyphon Design
- Fabrication of LTS unit

3.1 *Verifying Cooling Requirements*

The first step in designing the thermosyphon is to verify the cooling requirements and the available space for integrating the device with the CPU. The baseline specifications and parameters chosen are shown in Table 1.

Table 1. Design Specifications and Parameters for Vectra Desktop

CPU	Pentium 4, 1400 MHz
Heat Load	80 W
Air Flow	0.00377 m^3/s (8 CFM)
Maximum CPU Case Temperature	70°C
Maximum Internal Ambient Temperature	45°C
Available Condenser Volume	75 (H) $\times$ 82 (W) $\times$ 26 (D) mm
Available Evaporator Volume	32 (L) $\times$ 32 (W) $\times$ 29 (D) mm
Allowed Temperature Rise Above Ambient	25°C
Target Thermal resistance	0.312 °C/W

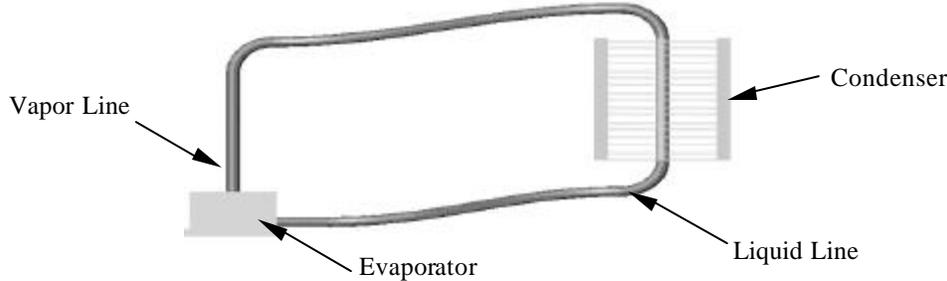## 3.2 Developing Compact Loop Thermosyphon (LTS) Design

Figure 1. Conceptual Design of the Loop Thermosyphon

Based on the design requirements, the conceptual design of the LTS is conceived as shown in Figure 1.

The design for the LTS is to have:

- An evaporator attached to the back of the chip with a boiling enhancement structure inside.
- A vapor outlet line from the top of the evaporator to the entrance of the condenser.
- An Aluminum finned condenser with horizontally oriented fins.
- A liquid line from the condenser outlet to the evaporator inlet.

3.3 Fabrication of LTS Unit

Majority of the design efforts focused on the following issues:

- Sizing of liquid and vapor line.
- Easy manufacturability of the LTS unit.
- Suitable design of the condenser with appropriate number of fins.

Using Methanol as the working fluid at a power input of 240 watts, the liquid and vapor lines are sized. Except for Aluminum fins, Copper is used everywhere, because of its compatibility with the working fluid and the ease at which copper can be joined (brazed or soldered). The copper tubing also easily lends itself for custom bending of vapor and liquid lines in the compact space of a PC. The condenser is designed with 43 fins (38.1×82.5 mm), made of Aluminum 3003-H 14. The fins are pushed onto the condenser tube and attached using epoxy. Space restrictions inside the cabinet allowed only a horizontal orientation of the condenser with vertical fins.

The heat input block attached to the chip is approximately 12.7×12.7 mm. A spring-mounted thermocouple is placed in the center of the block. Using STYCAST TC-4 thermally conductive compound, the LTS evaporator is centered on the heat input block. The thermocouples for measuring temperature are placed as shown in Figure 2.

One critical factor in the design of a thermosyphon is incipience overshoot, which is the increase in temperature needed above the boiling point of the working fluid to initiate boiling. One solution is the use of boiling enhancement structure, which not only increases the surface area of the heat source in the evaporator but also increases the critical heat flux, as bubble generation rate is increased. A three-dimensional boiling enhancement structure (Ramaswamy, 1999) is fabricated and attached to the heat input block. The boiling enhancement structure (Figure 3) is made up of six layers of square ($15 \times 15 \times 0.8$ mm) copper pieces. Microchannels of approximately 0.31 mm width are machined on both sides of each copper strip in mutually perpendicular directions, and then the copper strips are soldered at 231°C (Pb-Sn liquidus temperature).
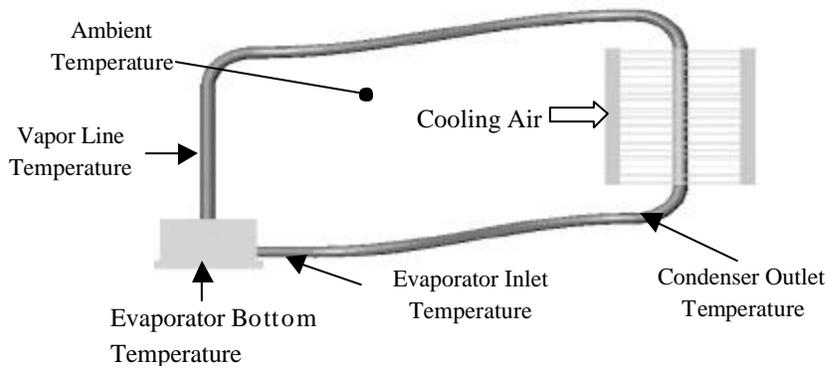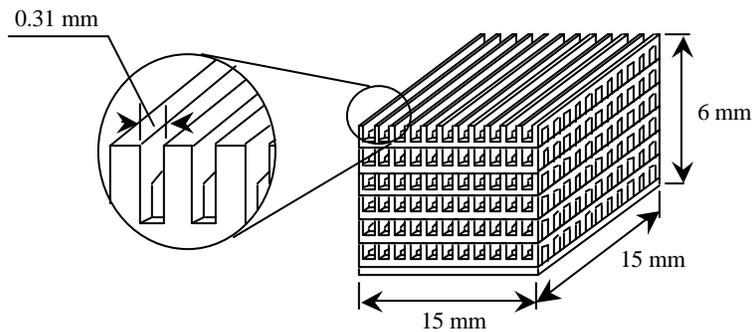
Figure 2. Thermocouple Locations

Figure 3. Magnified View of the Boiling Enhancement Structure

4 STUDY OF THERMOSYPHON PERFORMANCE

The performance of the compact thermosyphon is studied based on the following topics:

- Effect of working fluids (water and dielectric liquid).
- Effect of inclination of the thermosyphon.
- Efficiency of the condenser in natural convection condition.

4.1 *Effect of working fluids*

The thermosyphon is tested with two fluids – deionized ultra filtered (DIUF) water and PF5060, a dielectric liquid. The LTS is mounted on a test stand and 0.0075 m^3/s (16 CFM) of cooling air is passed over the condenser fin stack. The parameters measured at various power inputs for two different working fluids are given in Table 2 and 3.

   The water charged system clearly outperformed the dielectric liquid charged system as seen from the thermal resistance values. Water is a better working fluid, as the thermal properties of water are better than that of PF5060. The high thermal resistance in PF5060 may be attributed to the high amount of dissolved gases in it. It has been shown that the dissolved gases enhance heat transfer coefficient, however, they lead to a decrease in critical heat flux, thus increasing thermal resistance. The pressure of the dissolved gases may also have an adverse effect on the condenser and lead to overall performance degradation. However, with proper degassing procedure before charging the system, it may be possible to improve the performance of the PF5060 charged system.

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Table 2. Thermosyphon charged with 8×10^{-6} m^3 (8 cc) DIUF, Test Data

Power (W)	Temperature (°C)						Thermal Resistance (°C/W)
	Evaporator Bottom	Vapor Line	Condenser Outlet	Evaporator Inlet	Ambient	ΔT (Evaporator to Ambient)	
25	34.6	32.7	31.4	32.0	23.2	11.4	0.46
41	40.5	39.0	38.0	38.3	23.3	17.2	0.42
55	45.3	43.6	42.5	42.3	23.2	22.1	0.40
71	52.3	50.3	49.6	48.8	23.3	29.0	0.41
82	54.5	53.9	52.7	52.2	23.1	31.4	0.38
86	57.4	56.0	55.2	54.5	22.6	34.8	0.41

Table 3. Thermosyphon charged with 8×10^{-6} m^3 (8 cc) PF5060, Test Data

| Power (W)	Temperature (°C)						Thermal Resistance (°C/W)
	Evaporator Bottom	Vapor Line	Condenser Outlet	Evaporator Inlet	Ambient	ΔT (Evaporator to Ambient)	
25	45.6	37.5	36.5	37.1	22.1	23.5	0.94
41	61.6	50.3	48.4	48.7	22.2	39.4	0.96
51	71.5	58.8	56.3	56.3	22.3	49.2	0.97
65	83.3	68.9	65.2	65.1	22.4	60.9	0.94
81	101.0	83.7	79.0	78.7	22.6	78.4	0.97

4.2 *Effect of Inclination on the Performance of the Thermosyphon*

Experiments are performed using a HP custom designed protractor assembly to tilt the loop thermosyphon under a controlled condition to assess the effects of inclination on the loop thermosyphon performance.

4.2.1 *Experimental Method and Apparatus*
A schematic of the experimental apparatus is shown in Figure 4.

The heating is provided by a copper heater block, attached to the bottom of the evaporator with thermal interface material (thermal conductivity = 0.39 W/m-K). Both the evaporator chamber and the heater block assembly are centered inside a bolted Plexiglas/Ultem enclosure to ensure a firm attachment between the evaporator and the surface of the heat source to minimize the interface resistance and to reduce heat loss to the environment. The evaporator dimension is $25 \times 25 \times 25$ mm with a base of $30 \times 30 \times 10$ mm, while the heat source dimension is $30 \times 30 \times 10$ mm. A cartridge heater is inserted into the copper block to provide 60 W of total power. The heat is provided by a variable power supply monitored by a digital wattmeter. A fan powered by an HP E3636A DC power supply cools the condenser. The protractor assembly (Figure 5) allows the tilting of the thermosyphon loop in increments of 2° from the horizontal position in both the positive and negative direction with a ±0.2°. Temperatures of the evaporator, the condenser, the heat source and the inlet air to the condenser are measured using nine type T thermocouples. Steady-state temperature measurements are collected and recorded using an HP75000B data acquisition unit and an HP Vectra XU 6/200 utilizing an HP Visual Engineering Environment (HPVEE) software program.
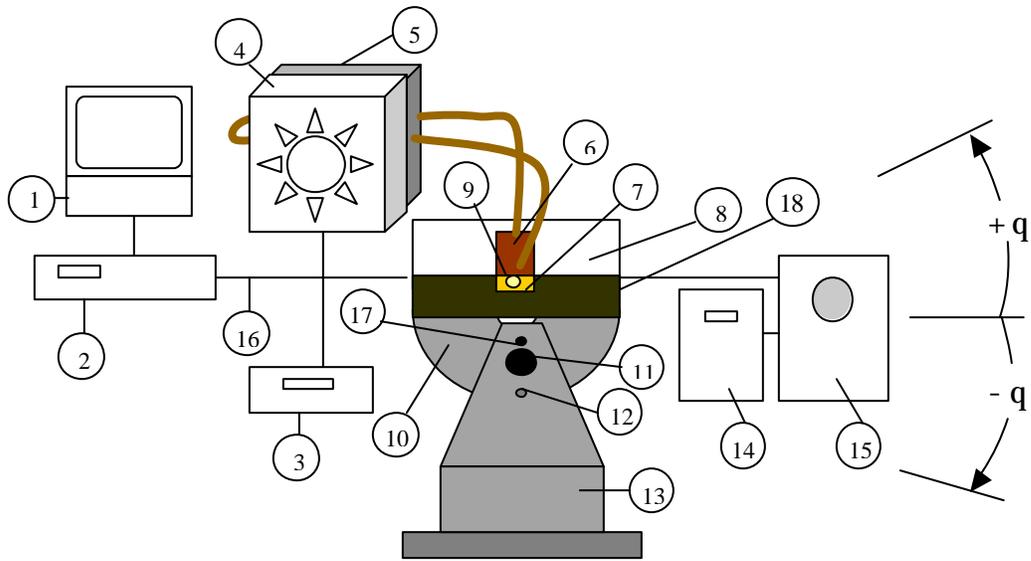
1. HP Vectra XU	10. Protractor
2. HP 75000B Data Acquisition Unit	11. Fine tuning pin
3. HP E3616 DC Power Supply	12. Locking pin
4. Condenser fan	13. Assembly stand
5. Condenser	14. Wattmeter
6. Evaporator chamber	15. Variable Auto Transformer
7. Heater Block	16. Type T thermocouples
8. Plexiglas enclosure	17. Shaft
9. Cartridge heater	18. Insulation

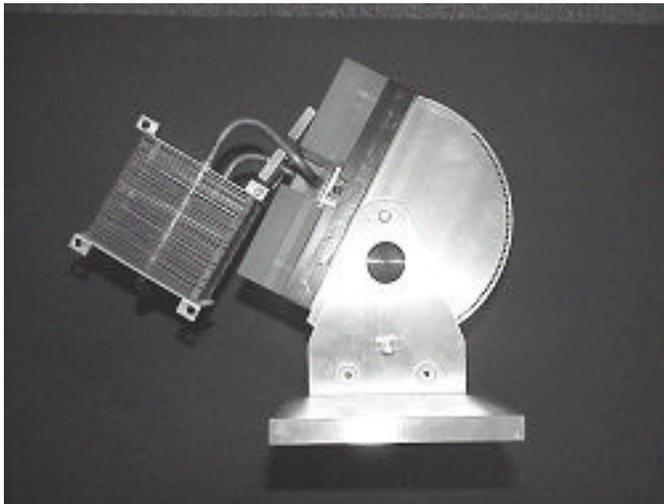Figure 4. Schematic of the Thermosyphon Test Assembly

Figure 5. Protractor Assembly with the Thermosyphon
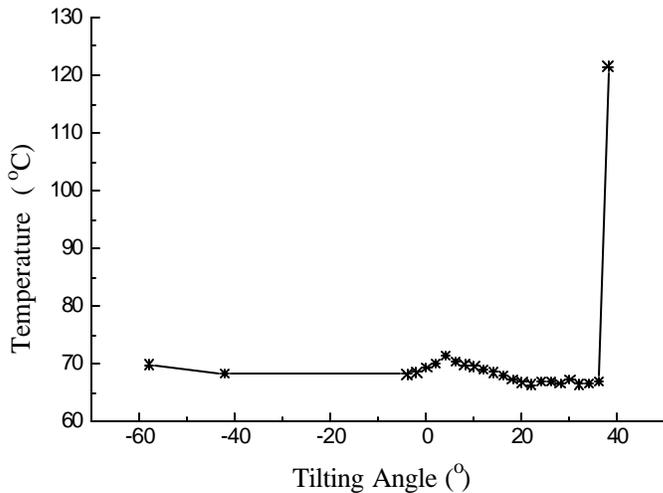
4.2.2 *Results and Discussion*

Figure 6. Evaporator Temperature vs. Tilting Angle

Figure 6 shows that the evaporator maintains a uniform temperature for tilting angle up to -58°, which is the maximum tilting angle allowed by the protractor assembly fixture in the negative direction. In the positive direction the evaporator temperature is maintained between 66-71°C for tilting angles up to 36°, after which the evaporator temperature shoots up over 120°C. At this point, the severe inclination prevents the return of sufficient amount of liquid into the evaporator to replace the vapor leaving the chamber, leading to dry out. Thus the evaporator temperature increases suddenly due to the lack of cooling liquid present inside the evaporator chamber. The processes take place suddenly without any indication, and thus produce a discontinuity in the temperature readings. It is observed that the dry out angle (38°) is at the position when the condenser outlet level is at the same or close to the evaporator inlet level. This is expected since this loop is gravity aided and for liquid to return to the evaporator chamber, a positive gravity level has to be present between the condenser outlet and the evaporator chamber inlet. However, the length of the falling tube and the system charge may also be factors in determining the dry out angle.

An interesting point to observe in Figure 6, is that the evaporator temperature is much higher than 56°C (boiling point of PF5060 at atmospheric pressure), which suggests that the boiling might have occurred at a temperature higher than 56°C. This increase in boiling point may be attributed to higher pressure inside the evaporator, which can be caused by insufficient degassing or overcharging of working fluid in the thermosyphon.

Figure 7 shows temperature difference between the condenser and the evaporator versus the tilting angle. A key finding from this preliminary testing approach is that the thermal performance of the thermosyphon loop remained almost unaltered for a very large tilting angle.
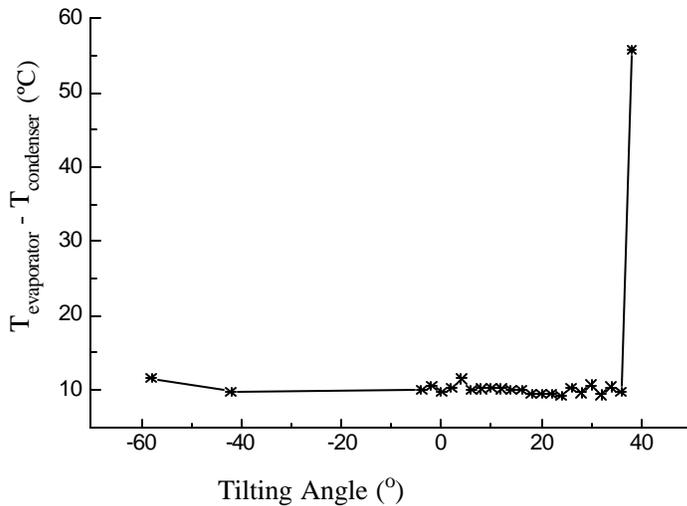
Figure 7. ($T_{evaporator} - T_{condenser}$) vs. Tilting Angle

4.3 *Efficiency of the Condenser in Natural Convection Condition*

4.3.1 *Numerical Simulation*

To assess the performance of the condenser in the absence of a fan, a system based numerical approach is adopted to model the inside of the PC cabinet. The commercial software, ICEPAK is used for numerical modeling. The objective is to see how the condenser works without the aid of a system fan.

Two cases are considered: one with system fan and the other without the fan. For the first case, the system fan served as the major air outlet, while for the later case, the grill present next to the system fan in the chassis served as the air passage. The grill is represented in the numerical model as a vent with 80% opening. The openings at the front of the chassis are represented as vents (rectangular, 30% opening is assumed) in the model for both the cases. Since the power supply unit (PSU) is separated from the condenser by a perforated plate and has its own dedicated cooling fan, it is assumed not to heat up the condenser. Also, the PSU fan is expected to have very little effect through the perforated plate and moreover, since it is an exhaust fan, it can only increase the cooling performance. So the PSU and the PSU fan are not included in the model. The audio card, video card and the CD bays are considered to be the major obstructions to air flow, and as such they are included in the model. In the absence of actual measurement of airflow rate through the fan, the characteristic curve for the exhaust fan used in HP Vectra is given as input to ICEPAK for the forced convection case.

4.3.2 *Results and Discussions*

For the case with fan, it is found that for an average condenser tube wall temperature of 80°C, the net heat dissipation from the condenser is about 66 W. The same boundary condition is used for the case where the system didn't have a fan. The results for both the cases are tabulated in Table 4. It is found that the condenser subjected to natural convection is able to dissipate 60% of the amount of heat dissipated by the condenser subjected to forced convection. So the presence of fan creates a big difference in heat dissipation. However, bigger sized fins may give better heat dissipation with natural convection. The isotherms for the 43rd fin (fin is vertically oriented) for both the cases are shown in Figure 8. In case 1, the forced convection has resulted in a symmetrical temperature profile in the fin (both about horizontal and vertical axes). In case 2, due to natural convection, temperature profile is symmetric only about the vertical axis. Due to heat conduction from the condenser tube, the isotherms are concentric at the vicinity of the tube, however at the upper and lower

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parts of the fin (away from the tube) due to natural convection, there is a difference in boundary layer thickness, which gives difference in temperature contours.

The average heat transfer coefficient from the condenser tube for the natural convection case is also much lower than the case with a system fan. These results show that the system fan improves the performance of the condenser a great deal, however, a better design of condenser may prove sufficient to dissipate heat through natural convection.

Table 4. Results from Numerical Simulation

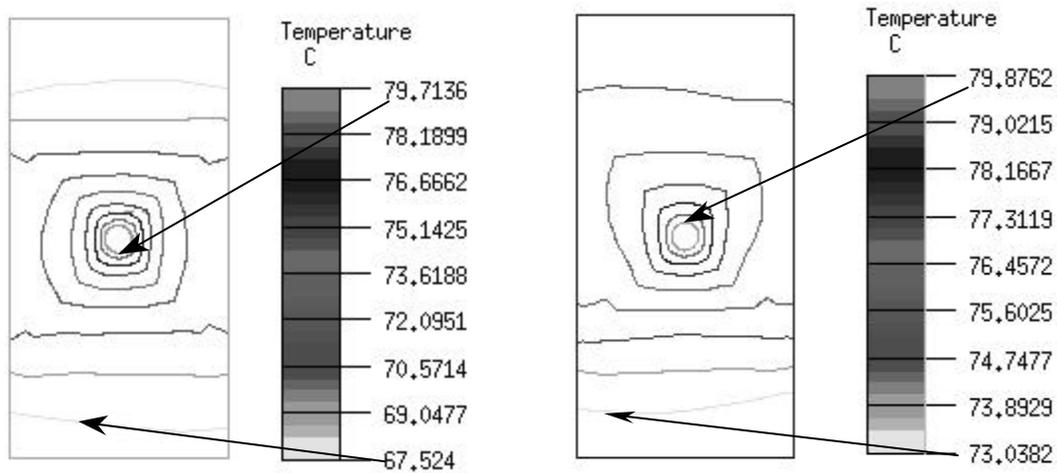| Parameters	With Fan	Without Fan (Natural Convection)
Condenser Tube Temperature	80 °C	80 °C
Ambient Temperature	22.6 °C	22.6 °C
Heat Flow from Condenser Tube	66 Watts	40 Watts
Average Heat Transfer Coefficient from Condenser Tube	515 W/(K-m^2)	315 W/(K-m^2)

Figure 8. Temperature Profiles in the 43rd Fin for Forced Convection System (left) and Natural Convection System (right).
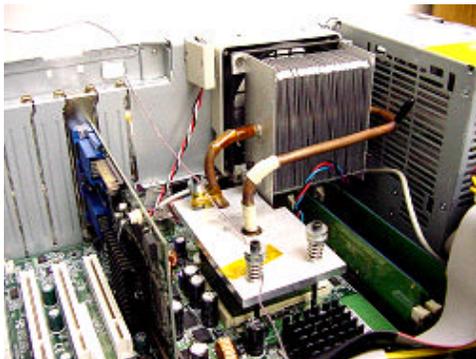

Figure 9. HP Vectra PC with the Thermosyphon Assembly

5 CONCLUSIONS

In this paper a design and performance evaluation approach for a two-phase compact thermosyphon (Figure 9) for cooling a Pentium 4 processor in a HP Vectra PC is presented. This study identifies the working fluid, the orientation and the system fan as important factors affecting the performance of the loop thermosyphon. Water is identified as the better working fluid with respect to dielectric liquid; however, optimum charging and proper degassing may improve the qualities of dielectric liquid as a working fluid. The thermal performance of the thermosyphon is also found to be unaffected for large inclination angles. Natural convection case is found to be inefficient, however a change in design may make the condenser efficient. The presented experimental and numerical results represent a preliminary study on the performance of the loop thermosyphon. Further detailed numerical and experimental studies are needed to accurately predict the behavior of the thermosyphon.

6 ACKNOWLEDGEMENT

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