926 KSME International Journal, Vol. 12, No. 5, pp. 926~941, 1998

Design of a Two-Phase Loop Thermosyphon for Telecommunications System(I)
— Experiments and Visualization —

Won Tae Kim,* Kyu Sub Song and Young Lee*****

(*Received August 11, 1997*)

A two-phase loop thermosyphon system is developed for the B-ISDN telecommunications system and its performance is evaluated both experimentally and by visualization techniques. The design of the thermosyphon system proposed is aimed to cool multichip modules (MCM) upto heat flux of 8 W/cm^2. The results indicate that in the loop thermosyphon system, cooling heat flux is capable of 12 W/cm^2 with two condensers under the forced convection cooling of the condenser section with acetone or FC-87 as the working fluid. The instability of the working fluid flow within the loop is observed using the visualization techniques and temperature fluctuation is stabilized with orifice insertion.

Key Words: Two-Phase Loop TheRmosyphon, B-ISDN Telecommunications, Visualization, Flow Instability, Orifice Inserts

Nomenclature

h_{fg}	: Latent heat of vaporization (J/kg)
L^*	: l_c/l_e
m	: Mass flow rate (kg/s)
q	: Heat flux (W/cm^2)
t	: Time (s)
T_h	: Temperature at the center of evaporator within heater (℃)
TCT	: Two-phase closed thermosyphon
TLT	: Two-phase loop-type thermosyphon
Δt_{h-air}	: Overall temperature difference (℃)
U	: Overall heat transfer coefficient (W/m^2 K)
V	: Volume (m^3)
V^+	: Non-dimenasionlized volume as defined in Eq. (4)

Subscripts

c	: Condenser section
e	: Heated zone
ev	: Evaporator section
f	: Fluid
h	: Hot
T	: Total
WF	: Working fluid

Greek letters

ρ	: Density (kg/m^3)

* Department of Industrial Engineering, Kongju National University, Chungnam, KOREA
** Packaging Technology Section, Electronics and Telecommunications Research Institute(ETRI), Taejon, KOREA
*** Department of Mechanical Engineering, University of Ottawa, Ottawa, Ontario, Canada

1. Introduction

It has been known for a long time that very effective heat transfer could be obtained by means of evaporation and condensation of a fluid. For past two decades heat pipes have been typically used in space applications, where the needs for spacecraft temperature equalization and for transferring heat to radiate surfaces without power-driven pumps. The use of heat pipes is currently expanding into areas with increasing density and higher heat dissipation in electronics, where high heat dissipation causes extreme temperature gradient in the device heat sinks (Adami and Yimer, 1988; Kishimoto and Harada., 1992). Immersion or direct liquid cooling is an alternative way (Michael, 1994), but causing auxiliary problems due to mainly direct contact of the electronic components with liquid. It has been addressed in references (Chi, 1976; Peterson and Ma, 1996; Hewitt et al., 1994; Holman 1996) where heat pipe could serve as an effective heat spreader by dropping the peak temperature of elements. In telecommunications system, the multichip module (MCM) of high density packaging with a function of decreasing signal propagation delay has been implemented to incarnate the asynchronous transfer mode (ATM) switching systems having the capacity of a Broadband Integrated Services Digital Network (B-ISDN). Since MCM need more powers for greater speed and more power densities compared to the conventional electronics components (Kishimoto et al., 1992, 1994; Lee et al., 1996), the thermal management for electronic and electrical elements has become an important and serious issue with the rapid increase of chip power and power densities.

There have been many previous works that has dealt heat pipe the cooling of the high-density heat dissipation, packaged with VLSI and MCM (Nelson et al., 1994; Lee, et al., 1996; Kim et al.,

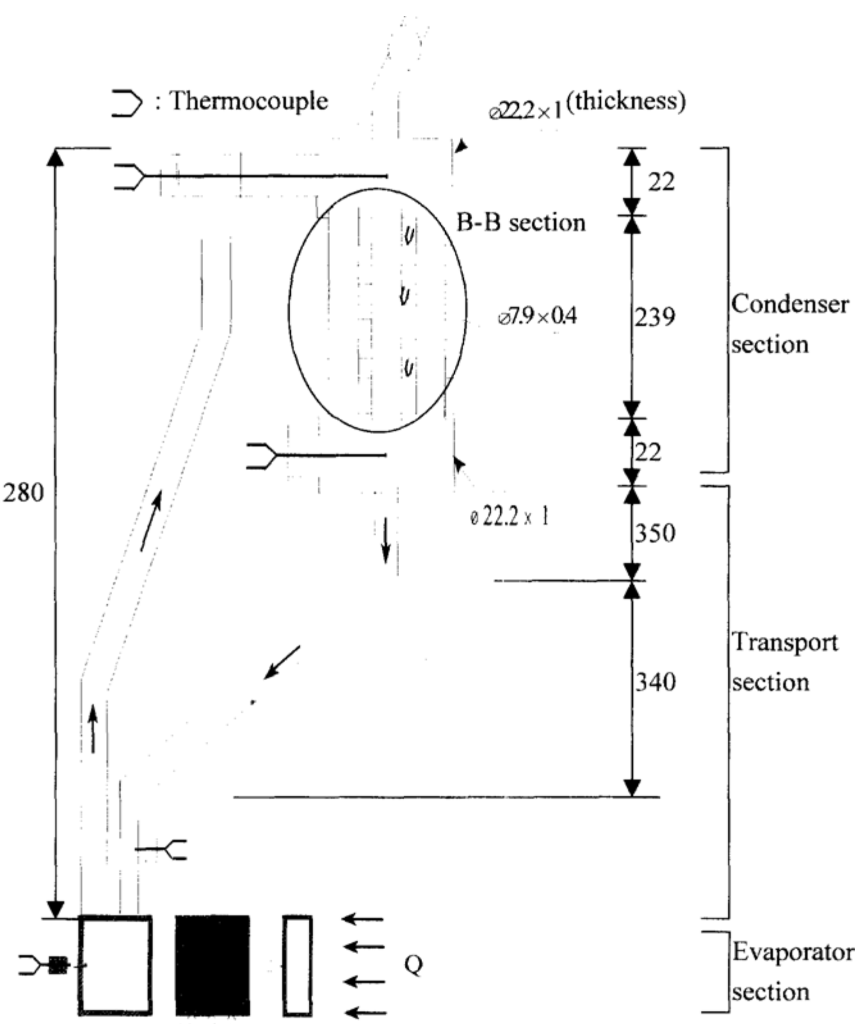


Fig. 1 Schematic of TLT assembly (unit : mm).

1996). Cooling systems, which can deal with such high heat flux are those that employ either closed or loop-type two-phase thermosyphon combine the advantages of the conventional air cooling systems with the advantages of fluid cooling, utilizing the phase change heat transfer of a working fluid (Faghri, 1994; Peterson, 1994). In case of a two-phase closed thermosyphon (TCT, i. e., wickless heat pipe), it is subject to counter-flow of falling liquid and rising vapour of the working fluid within the thermosyphon (Lee and Mital, 1972; Rhi, et al., 1997). On the other hand, the two-phase loop thermosyphon (TLT) is separated in two flows, and thus the efficiency of the thermosyphon cooling system would be increased. In this way, the two-phase flow is not hindered by the counterflow of the falling fluid and rising vapour, and it is expected that there could be a large increase in the performance as a heat transfer element.

In an operating TLT, the overall heat transfer capability is increased because of no flooding caused by counter-flow. The major differences between the two-phase loop thermosyphon used in the present study and the conventional two-phase loop thermosyphon are that the former is relatively very large in size with much larger heat transfer capacity and that the layout and the structure are quite different from those of the latter, as shown in Fig. 1. Consequently, the previous experimental results or analytical studies on the conventional two-phase loop thermosyphon can not be directly applied to the present system. For the present objective, loop-type two-phase thermosyphon under the condition of forced convection mode of the condenser section is used as a higher efficiency autonomous heat transferring device from the MCM.

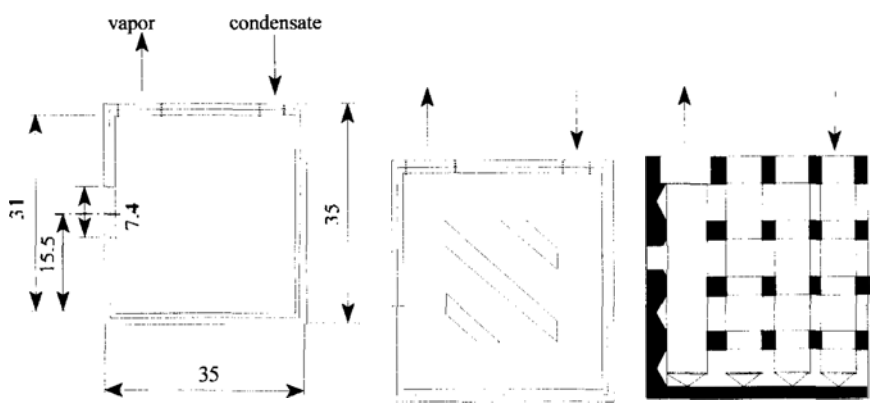
2. Experimental Apparatus and Procedures

The experimental apparatus consists of the main TLT assembly, the cooling system for the condenser section, the heat generation section and the charging system.

2.1 Experimental apparatus

The main TLT assembly is made of three parts: the evaporation section, the long smooth transporting section, and the finned condenser section. The schematic of the test assembly is shown in Fig. 1. One flow tube is connected to the upper end of the evaporator and to the upper end of the condenser, and is used only for moving the mixture of vapor and liquid. And the other is connected to the lower end of the condenser and to the lower part of the evaporator through an extended tube inside, and is for returning of the condensate from the condenser to the evaporator.

All three evaporator sections were made from the same copper plate with the dimensions of 35 mm×35 mm×9.5 mm (thickness). Effective surface area (heat transfer surface) is 35 mm×35



(a) (b) (c)

Fig. 2 Different types of the evaporator section (unit: mm).

mm. Three different designs were adopted for the evaporator sections of the system; i. e., Evaporators A, B and C. For evaporator A, a square cavity (31 mm×31 mm×7.5 mm) was cut out on the back side (non-working surface) of the evaporator as shown in Fig. 2 (a). Evaporator B was designed which had three ribs on the evaporator cavity wall. The ribs incidentally increased the area of the boiling surface. All other parts of the system remained the same. For two evaporators B and C, only the internal cavity structure and physical dimension were different. The details of evaporators B and C are illustrated in Fig. 2 (b) and (c), respectively.

The condenser section was made of brass tube with copper fins. There were two types; which were condenser #1 and #2 with two tubes in line as tabulated in Table 1. Finned tubes were soft soldered into two parallel horizontal headers made of copper cylinders. Condenser #2 was used with evaporators B and C.

The evaporator and condenser are connected with two transporting lines as seen in Fig. 1; one for the two-phase flow from the evaporator to the condenser, and the other for the condensate return from the condenser to the evaporator. Two glass tubes are provided in the inclined section of this line; one (=12.7×2.5 mm and 310 mm in length) is provided in the upper vertical section and the other is in the inclined section (9.5×2.1 mm and 600 mm in length). The glass tubes are for the visualization study of the flow regimes as well as for the fluctuation of the flow in the tubes.

A fan was used for cooling, which the finned parts of the TLT's were installed into the channel made from transparent acrylic. Figure 3 shows the layout of the cooling fan including condenser section. The front channel was installed perpendicularly to the desk board and from the back of the desk board, another channel made from the same material was bonded to the first. A cooling fan (Archer 273−242, 80 mm, 32 cfm, 120 VAC, 12 W) was screwed to another flange. The hot-wire probe (Wallac Inc. -type Ni-125 ANE) was installed from the bottom side of the front channel before the finned tubes at a distance of 180 mm from the air entrance as shown in Fig. 2. The hot-wire probe was connected to a thermoanemometer (Wallac Inc. - type GGA22F), which had the ability to measure not only air velocity but also air temperature. The cooling fan was connected to a variable autotransformer with the output of 0 ~ 140 V and up to 10 A. The special flat plate heater is designed and manufactured for the simulation of the MCM operation (Kim et al., 1997). The heater consisted of a heating element made from BeO on a ceramic sheet plate with metal as an electrical conductor from one side and covered with thin layer of electrical insulating material and a copper box. The heat transferring copper wall of the heater was covered with a CHO-Therm T500 insulator -elastomer (Chomerics Inc.) - the filler for the tight contact between the heater and the evaporator of the TLT. Two nylon boxes, one with the evaporator section and the other, with the heater, were screwed together in such way that the heater with the filler was put on the evaporation section.

Both forced and free convection modes were used for cooling of the condenser. For the free convection mode, the condensers were inclined with respect to the horizontal plane from angles of 90° (vertical position) to 8°. The charging system consisted of a vacuum pump, a McLeod gauge, vacuum valves and a syringe to charge the desired amount of the working fluid into the test. A temporary closure of the loop thermosyphon was made with a NUPRO vacuum valve (B4HK-TW, SWAGELOK Co.). The valve was connected to the upper header through a brass tube.

Table 1 Characteristics of condensers.

Condenser Number	Number of tubes	O. D. mm	I. D. mm	A^{v}_{crsect} cm²	A_{cond} cm²	A^{air}_{crsect} cm²	A_{fin} cm²	A_{conv} cm²
1	2	6.4	5.6	0.49	70	37.1	840	920
2	3	7.9	7.1	1.2	135	18.8	562	712

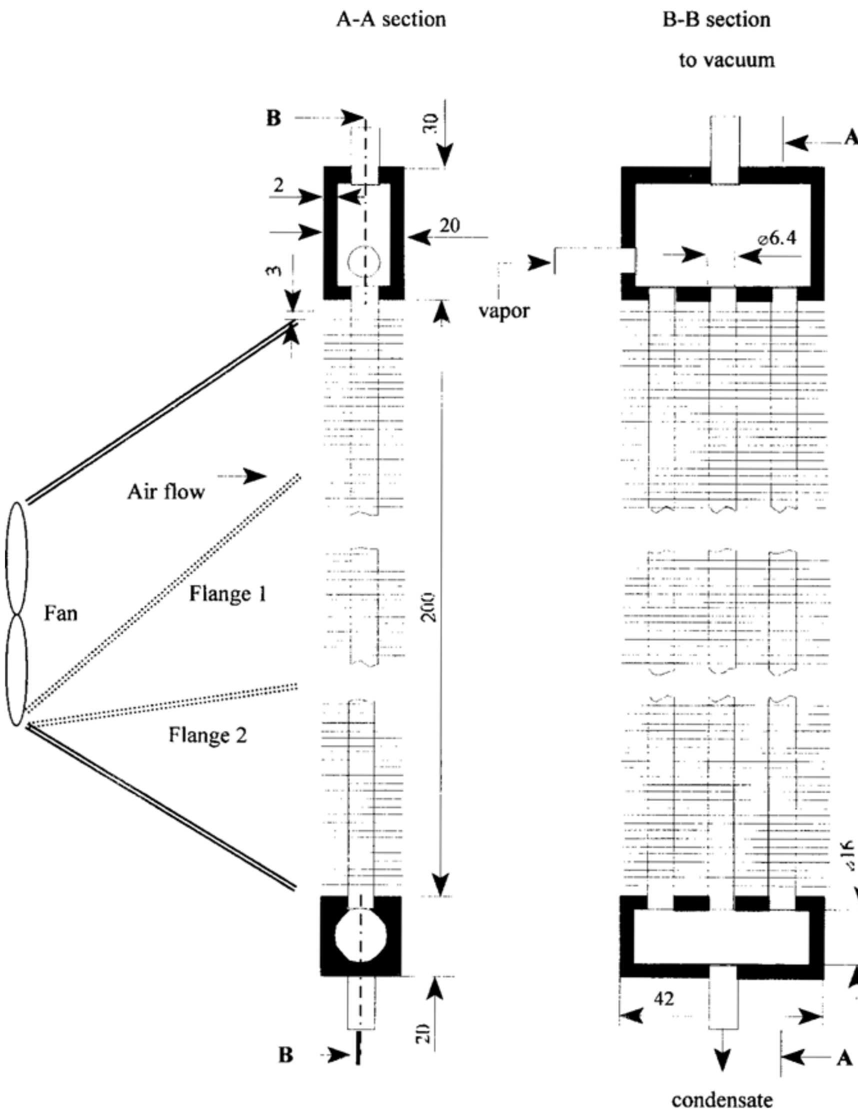


Fig. 3 Condenser section and cooling fan layout.

The selection of a suitable working fluid is one of the most important aspects of the design and manufacturing process, because the basis for the operation of a TLT is the vaporization and condensation of the working fluid. To find the most suitable working fluid, R-11, R-113, ethanol, water, FC-72, FC-87, and acetone were tried under various conditions. The behavior of fluorinert liquids for FC-72 and FC-87 as boiling working fluids was seldom investigated only for the conditions close to the pool boiling of FC-72 (Nelson, et al., 1994; Lee, et al., 1992) and no study was reported in the literature on the heat transfer rates in the TLT except the study by Kishimoto et al. (1994) for FC-87.

To determine the effect of a change in the pressure drop in the flow channel of the TLT on the flow instability and temperature fluctuation, three orifice inserts (cylindrical: d=5.5 mm and 6 mm in height, $d_{orifice}$=0.5, 0.7, 12 mm) were used. These inserts were placed inside the condenser line using Tee connector situated near the evaporator section (140 mm from the upper edge of the evaporator). And for the visualization of the two

-phase flow regimes in the transporting line above the evaporator section, a photographic technique was used.

2.2 Experimental procedure

The thermosyphon tube was connected to a Leybold mechanical vacuum pump (type ADEA 71 N4) to remove the inside air and other non-condensible gases. In the present study, the usual amount of the working fluid charged was about 50 ml for thermosyphon, which was about *10* times larger than the volume of the thermosyphon evaporator. The initial pressure without the working fluid was not higher than (5 ~ 10) × 10−3 Torr.

The power to the evaporator heating section was increased very carefully in steps to the desired heat flux. To reach the steady state, it usually took approximately 1 hour. Voltage was fluctuated within a range of ±0.2 V. By using the thermo-anemometer, the air flow velocity and air temperature were measured. The air flow velocity was carefully controlled by a slip switch because of the sensitivity of the switch itself. After changing the velocity setting, about 10 minutes were required to achieve steady state readings. Once a steady velocity was obtained, various values were recorded. After a steady velocity and heating power were obtained, the temperature was checked for a steady value. To determine these temperatures, the signals from the thermocouple in the heater center were observed until it registered no temperature fluctuation (approximately 40 ~ 60 minutes).

In each experiment, when the steady power and temperature readings were attained, the temperatures of the thermocouple specially attached on the upper end of the tube which connected the upper header with the sealing valve was checked to see if there was any non-condensible gases. Once system became stable, all data was recorded. After each series of test for a given working fluid was conducted, the thermosyphon was thoroughly cleaned by repeated washing and then vacuum dried, tested for vacuum and any leak was checked. For the measurement of temperature fluctuation, three FISHER Recordall Series 5000 devices were used together with a PC for digitizing of the results.

3. Data Reduction

The overall heat transfer coefficient, UT, of the test TLT was defined as:

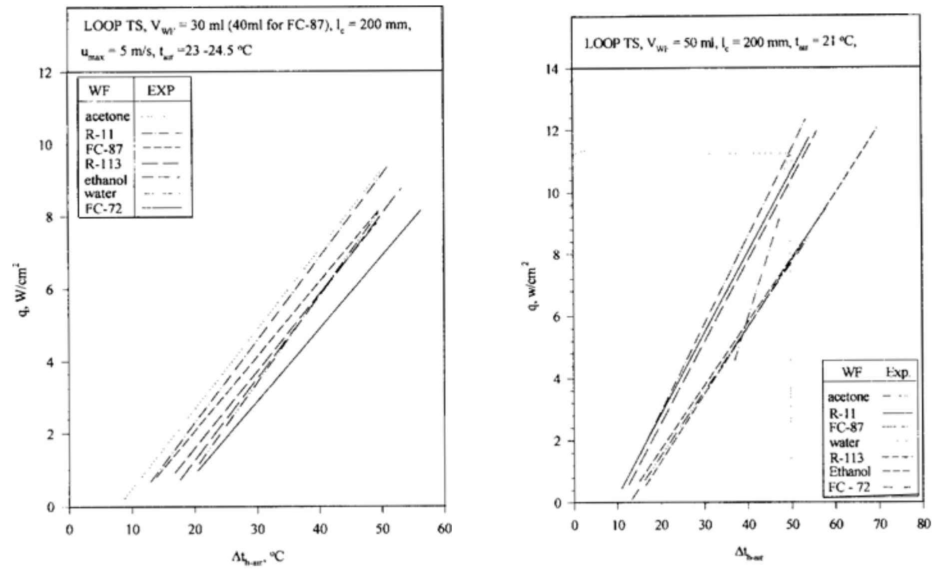
$$U_T = \frac{q}{t_h - t_{air}} = \frac{q}{\Delta t_{h.air}} \quad (1)$$

, where the heat flux per unit area, q, was calculated from the power measurements which was obtained from the voltmeter and ammeter coupled into the power supply circuit of the main heater. The heat transfer characteristics of the present TLT are evaluated in terms of the overall temperature difference of the system, $\Delta t_{h.air}$, which is one of two appropriate parameters to identify the heat transfer phenomena involved, because the heat flux, q, is simply (U_T, $\Delta t_{h.air}$). The other is, of course, the overall heat transfer coefficient, U_T. When the value of heat flux, q, is fixed, ? th-air is proportional to $1/U_T$. The heat losses through the insulation of the heating and evaporation section were negligible because the good insulation of the heating and the evaporator sections. The temperatures in the heated section and condensing section were measured by thermocouples.

The errors involved in the calculation of the thermosyphon heat transfer coefficient were generally due to the inaccuracy of the temperature and the power measurements. Even if the readings of the power and the temperatures were recorded after the steady state has been reached, a small fluctuation was observed (±0.2 V for voltage, ±0.01 A for current and ±0.2 °C for temperature).

4. Results and Discussion

During the phase change process, the working fluid absorbs the heat of vaporization by boiling or evaporation process. Due to the differences in densities between vapor and liquid states of the working fluid, the vapors rise up, which enter into the upper tube. And vapors move up to the condenser section of the system because of the pressure drop created by decreasing the volume



(a) One condenser (b) Two condensers

Fig. 4 Effect of Δt_{h-air} on Q for different working fluids at forced convection mode of $u_{max}=5$ m/s.

near the cold surfaces of the condenser during the condensation process.

In Fig. 4, the results of the TLT cooling system with one condenser are compared with those with two condensers and they show that there is an increase of about 15 to 20% in the performance of the latter. This is expected because one of the major thermal resistance in the system occurs at the interface between the fins of the condenser and the ambient air which is the ultimate heat sink. All figures indicate that the two-phase closed thermosyphon system of the present design would satisfy the design objectives imposed, including the design target cooling heat flux of over 10 W/m^2 with the overall temperature difference of 50 ℃.

In the present study, acetone, FC-87 and water were used as the main reference working fluids and FC-72, R-113, R-11 and ethanol as the supplemental working fluids. With acetone, experiments to investigate all possible effects were carried out in the first series of experiments to find the heat transfer characteristics of the proposed system. In the figures, the choice of working fluid for TLT did not affect the overall heat transfer characteristics greatly, which was not the case for Rhi, et al., (1997). The figures demonstrate that acetone seems to be the best working fluid. The worst seems to be FC-72. All the other working fluids exhibited nearly the same heat transfer performance. However, it was noticed that water as a working fluid for the present application was accompanied by a large pressure fluctuation in the loop, consequently inducing a high temperature fluctuation in the evaporator section. Because of this reason, water is not recommended as a working fluid for the present design and application.

In Fig. 5, the heat transfer performance characteristics with three different evaporators are presented. Evaporator A indicated that the heat transfer wall of the evaporator was not strong enough to maintain its flatness under the changing internal cavity pressure during the experiment. Consequently, the value of the contact thermal resistance between the heater and the evaporator together with the filler varied with the pressure. The figures indicated that the evaporators B and C were superior to evaporator A. However, a close examination of the evaporator A disclosed that it resulted mainly because of an increase in the thermal contact resistance between the heater surface and the evaporator due to the buckling deformation of the relatively thin wall of the heat transfer surface of the evaporator, induced by the vacuum inside. The results indicated that the design of evaporator C be adopted for performance as well as for the easiness of its manufacturing process and at the same time to maintain the same advantages of evaporator B.

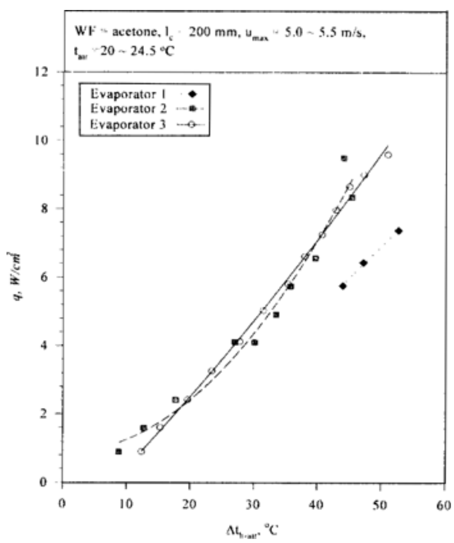


Fig. 5 Effect of Δt_{h-air} on q for different evaporator types.

The velocity of air on the condenser section should be a factor, which affects the heat transfer capability of the system. The effect of the air velocity in the finned condenser section on the heat transfer capacity of the proposed system is shown in Fig. 6 (a). In addition, the filler conductivity, kfiller, is compared to the heat transfer capacity. It was seen that the total thermal resistance was reduced because of the increased convection on the condenser section. However, an increase in the air velocity larger than 2 ~ 3 m/s would not significantly increase the overall heat transfer capacity of the assembly. The reason for this phenomenon is illustrated in Fig. 6 (b). In the figure, it was seen that the air flow velocity on the condenser section has little effect on the forced convection heat transfer coefficient once the air velocity is larger than 2 ~ 3 m/s. Even when the heat transfer capacity of the working fluids was different, a similar tendency could be observed. The effect of air velocity for one and two condensers is shown in Fig. 6 in which the same trend can be noticed.

It is expected that an increase in the heat transfer area of the condenser should reduce the over-

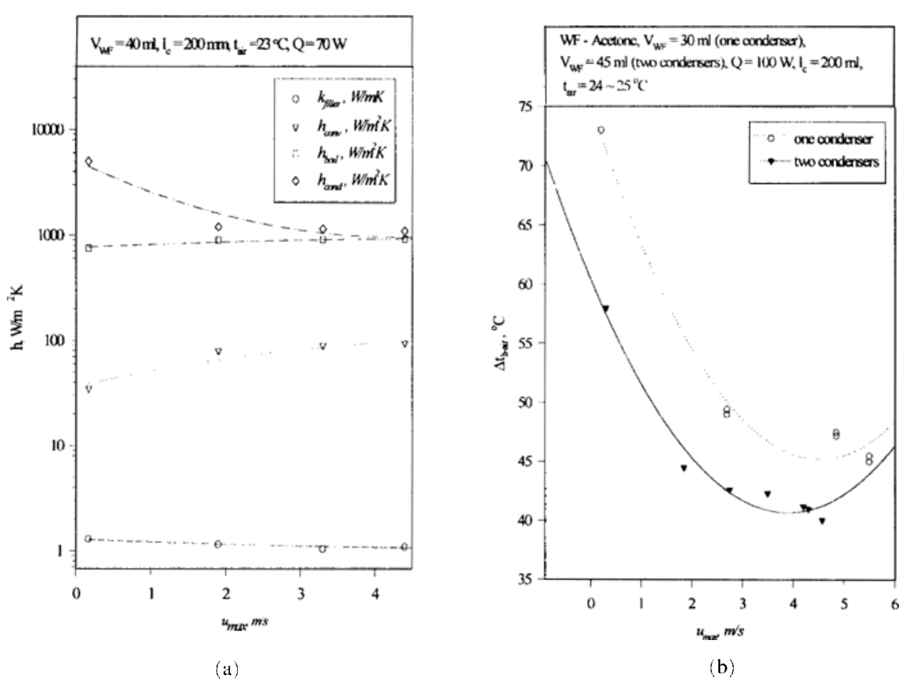


Fig. 6 Effect of air velocity on (a) heat transfer coefficients and (b) Δt_{h-air}.

all thermal resistance of the system, thus increase the overall heat transfer capability. And the experimental apparatus was modified to accommodate two condenser sections to see if an increase in the condenser section area would further improve the heat transfer performance of the proposed system under the conditions of the forced and free convection, respectively, at the outer condenser surfaces. The results are quite encouraging as shown in Fig. 7 at the air velocity of 4.5~5 m/s. However, it can be deduced that there must be a limit to the number of condensers that can be increased for the system.

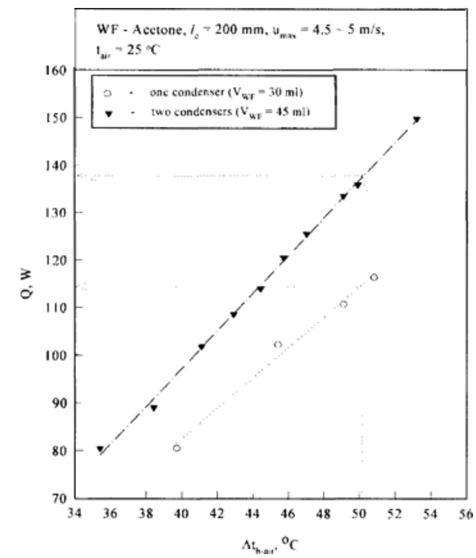


Fig. 7 Effect of number of condensers on Q vs. Δt_{h-air}.

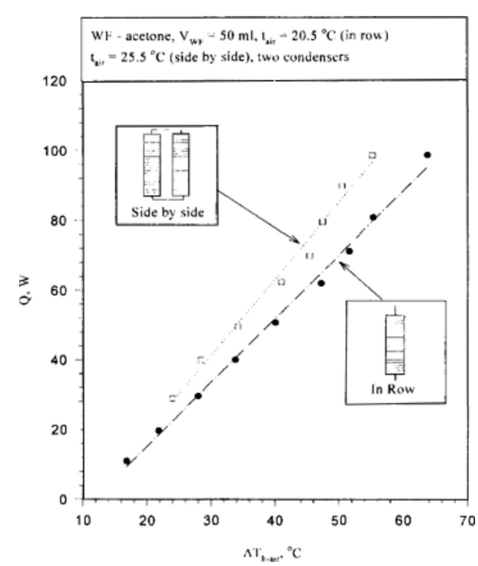


Fig. 8 Effect of Q with different condenser arrangement.

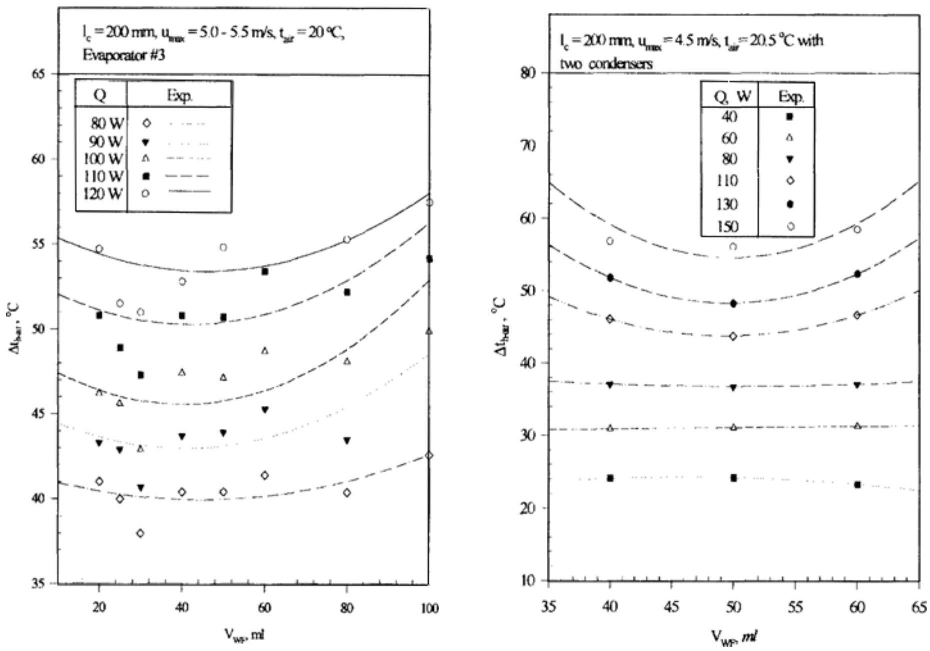
In Fig. 8, the effect of the arrangement of two condensers on the heat transfer rate is shown. As expected, the heat transfer performance of the case for the condensers arranged side-by-side is higher than that of the case for in-row. This is because in side-by-side arrangement, all incoming air faced by the both condensers is at the ambient temperature, whereas in in-row arrangement, the incoming air faced by the second condenser is at a higher temperature because it has already picked up heat from the first condenser.

The relative quantity of the working fluid, V^+, the constraint in the operation of a thermosyphon, was found to vary directly with an increase or decrease of the heat transfer performance of the system. Rhi, et al. (1997) estimated the optimum amount of a working fluid based on an assumption that the boiling heat transfer in a TCT takes place in two modes; the heat transfer in the evaporator is that of the pool boiling, whereas the evaporative heat transfer is taken place from the falling condensate liquid film above the pool of the working fluid.

$$V^+=\frac{V_l}{V_{ev}} \tag{2}$$

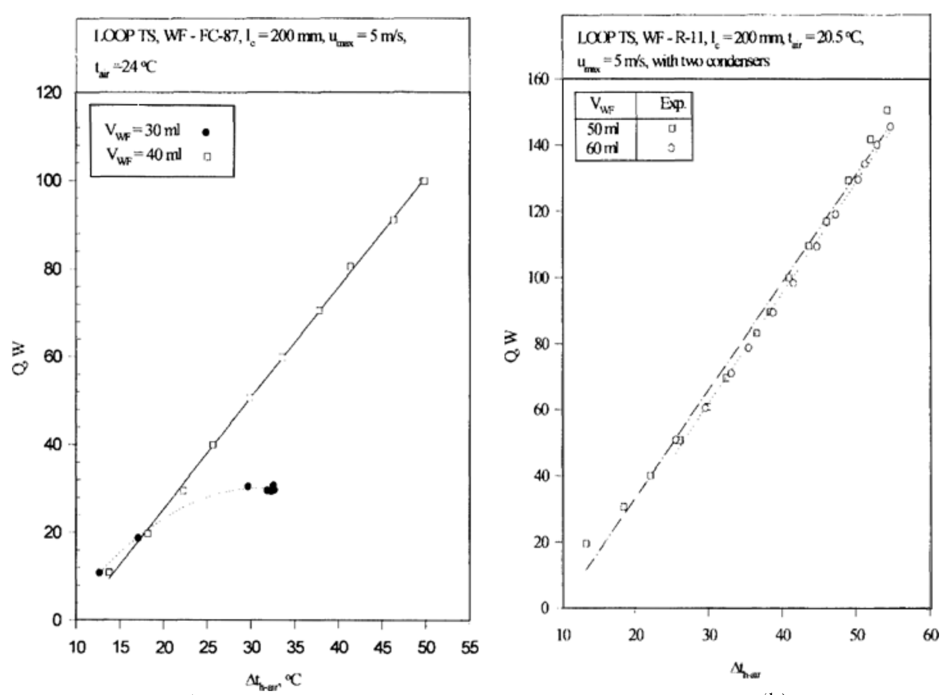
Due to the reason that this issue in the TLT has not well documented, instead of using the dimensionless parameter, V^+, for the volume of working fluids, the dimensional volume of working fluids, VWF, is used because the objective of the present study is very specific. Figs. 9 (a) and (b) illustrate the effect of the amount of the working fluid charged in the system on the total temperature difference or on the heat transfer rate for the working fluids; acetone, FC87 and R11. It showed that for the system with one condenser the optimum value of the filling charge was in the range from 30 to 40 ml and for the system with two condensers, about 50 ml.

It was noticed from Figs. 9 and Figs. 10 that all other parameters have little or no effect. At the



(a) (b)

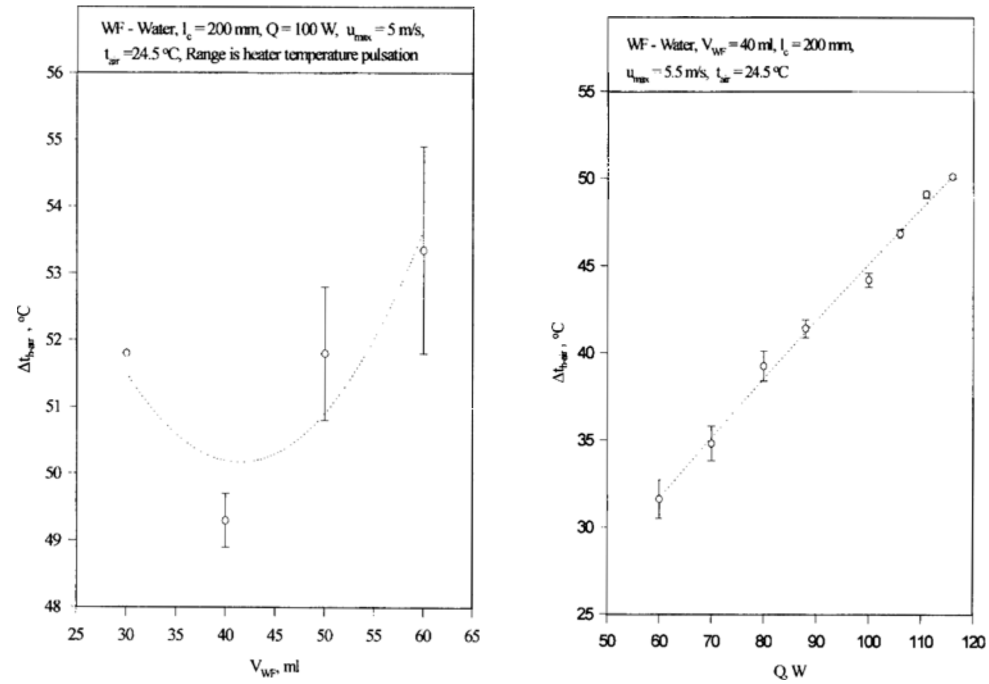
Fig. 9 Effect of WF on Δt_{h-air} for the number of condensers; (a) one (b) two.



(a) (b)

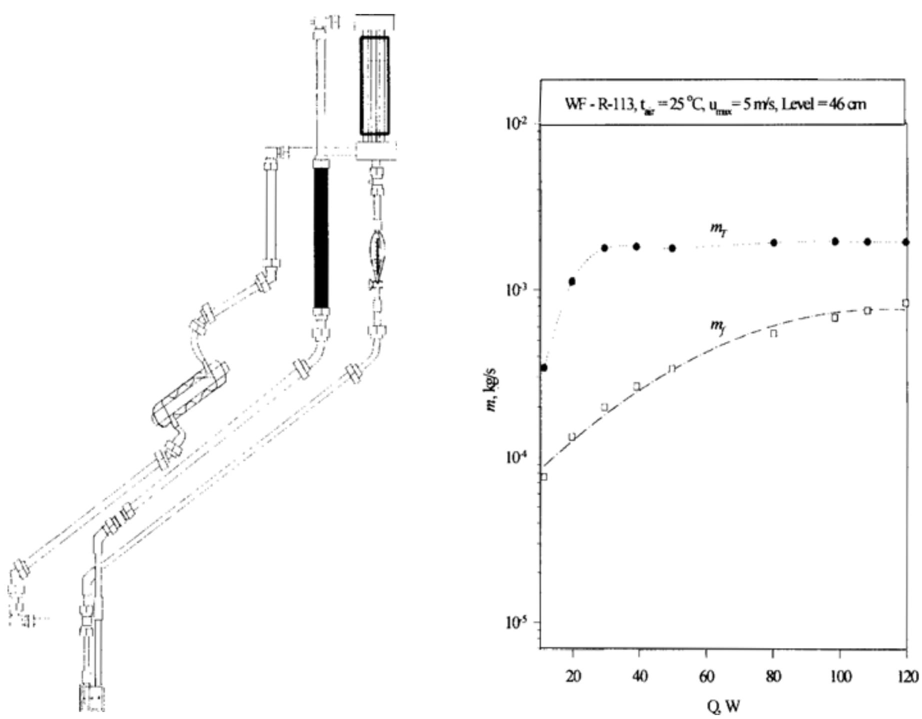
Fig. 10 Effect of the amount of different WFs on Q vs. Δt_{h-air}.

same time, it was seen in Figs. 10 that there existed a minimum value of the filling charge, which depended mainly on heat flux, the type of working fluid and the internal volume of the



(a) (b)

Fig. 11 Temperature pulsation of Δt_{h-air} on the amount of WF and Q.



(a) Modified TLT apparatus (b) Heat transfer rate

Fig. 12 Layout of modified TLT and effects of heat transfer rate on mass flow rate.

system.

The effect of the amount of working fluids, V_{WF}, on the overall temperature difference, Δt_{h-air}, are shown in Figs. 11. It is interesting to note that the larger the amount of working fluid, V_{WF}, and the smaller the heat transfer rate, Q, the larger the fluctuation of the overall temperature difference, Δt_{h-air}.

Since a purely analytical solution for the system within the loop is not possible due to the two-phase fluid flow involved, both measurements of mass flow rate and visualization were conducted. To supplement the experiment, a novel device was introduced to measure the total mass flow rate, m_T, of the working fluid. The apparent mass flow rate, m_f, is the theoretical mass flow rate calculated from the energy equation $Q = m_f h_{fg}$. The total mass flow rate and apparent mass flow rate are respectively defined as:

$$m_T = \frac{Q}{h_{fg}} \tag{3}$$

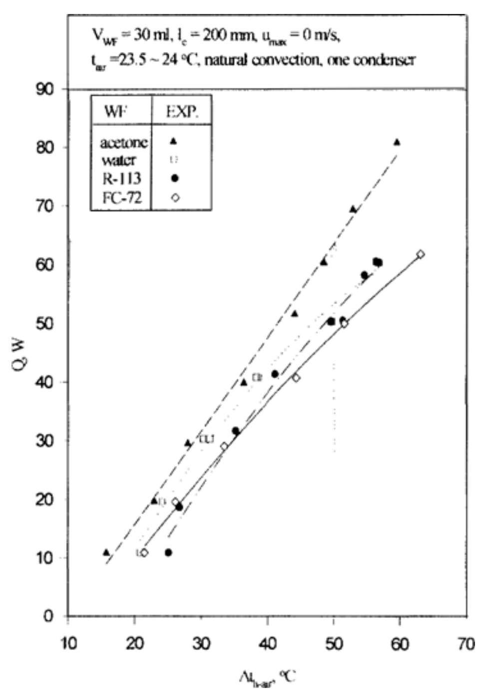
$$m_f = \frac{\rho V}{t} \tag{4}$$

In the loop of a TLT, the working fluid may still have a quality less than 1 and consequently,

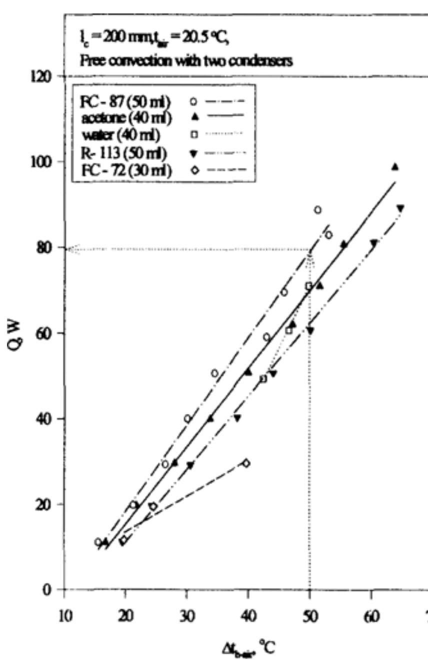
$$m_T > m_f \tag{5}$$

To test this hypothesis, TLT apparatus is slightly modified to be able to measure the value of mT very close to the reality as shown in Fig. 12 (a). In Fig. 12 (b), the ratio between the total and apparent mass flow rate strongly depended on the total heat transfer rate. From the result, it was concluded that a great attention must be paid in the analysis how to compute the mass flow rate which satisfied both the mass and energy balance equations. Knowing the total mass flow rate, the average quality of the working fluid in the two-phase region can be obtained.

A series of experiment was carried out to establish the maximum cooling heat flux of the proposed TLT system under the free convection mode and the results are presented in Fig. 13. It was seen in Fig. 4 and Figs. 13 that with acetone as the working fluid, the present TLT can provide, with the overall temperature difference of 50℃, the cooling heat fluxes of over 10 W/cm2



(a) One condenser



(b) Two condensers

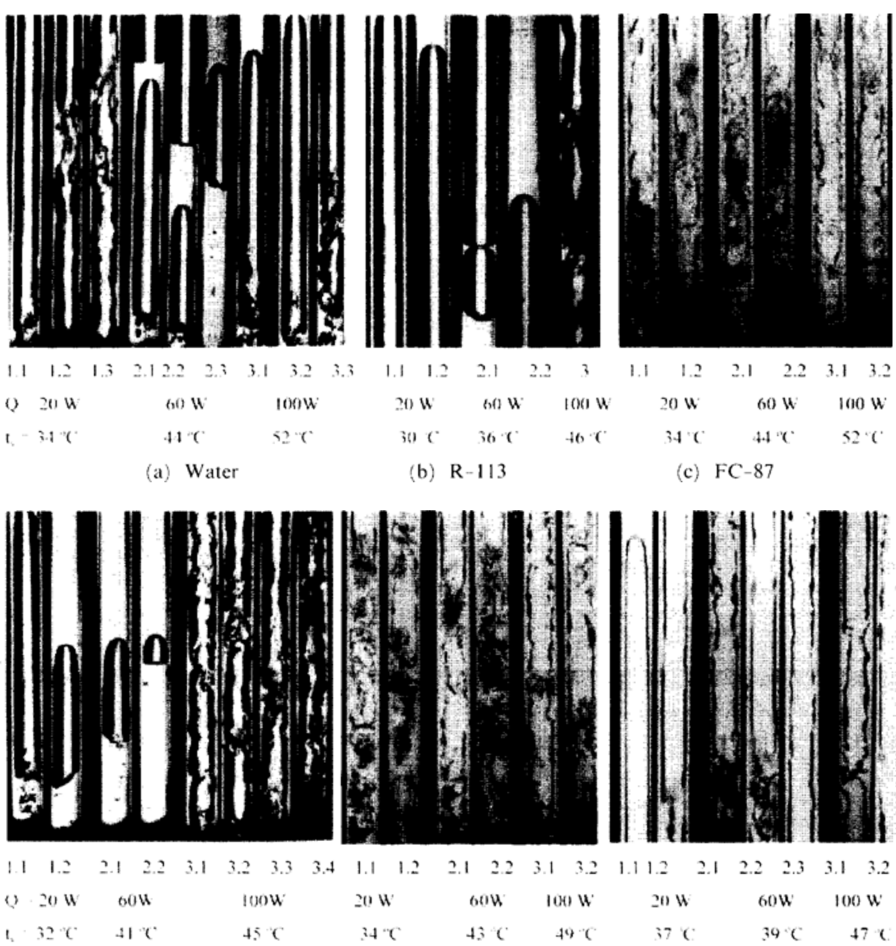
Fig. 13 Effect of Δt_{h-air} on Q at free convection mode.

under the forced convection mode and about 5 W/cm^2 under the natural convection mode.

From the physical processes involved in the operation of a TLT, several limits can be anticipated (Lee and Mital, 1972; Hewitt, 1994); evaporator dry-out due to the lack of the working fluid, burn-out (critical) heat flux in the evaporator section, flow instabilities associated with two phase flow heat transfer. Flow oscillations make the apparatus vibrate and cause a premature critical heat flux (Larkin, 1984). However, a literature survey reveals that none of the previous researches covers the method able to minimize the thermal resistance occurred between the evaporator and heating device. In experimental studies, only two limits were encountered; the minimum amount of the working fluid and the flow instability, especially with water as the working fluid. The instability of the working fluid flow within the loop is one of the major concerns for the

Table 2 Test conditions; Flow patterns, Fig. 14(a) to (f).

WF	V_{WF}, ml	u_{max}, m/s	Q, W	t_V, ℃
Acetone	40	5	20 ; 60 ; 100	32 ; 41 ; 45
Ethanol	40	5	20 ; 60 ; 100	37 ; 39 ; 47
FC-87	40	5	20 ; 60 ; 100	34 ; 44 ; 52
R-11	40	5	20 ; 60 ; 100	34 ; 43 ; 49
R-113	40	5	20 ; 60 ; 100	30 ; 36 ; 46
Water	40	5	20 ; 60 ; 100	34 ; 39 ; 41



(a) Water (b) R-113 (c) FC-87

(d) Accetone (e) R 11 (f) Ethanol

Fig. 14 Flow patterns for different WF's; $V_{WF}=40$ ml and $u_{max}=5$ m/s.

design of any loop-type two-phase thermosyphon.

Photographic observation of flow regimes and temperature fluctuation was obtained. In Fig. 14 to Fig. 15, photographic observation of flow regimes and the temperature fluctuations were obtained. From an attempt for the flow visualization as shown in Fig. 14, in which the flow pattern is denoted by numerical scripts, represented the boiling process and was made to correlate with the temperature fluctuation of the evaporator section. For the given heat load condition, the differences in the boiling characteristics of (a), (c), and (d) are well established. It was supposed that the acetone was boiling first because its saturation was lower than those of others. It was noted that a large pulsation of the bubble appears at high heat load of $Q=100$ W.

In Fig. 15, the transient temperature fluctuations as a function of time are illustrated for three different working fluids. It can be seen that even without any insert to damp the flow fluctuation, acetone and FC-87 hardly showed any fluctuation, whereas water has shown very strong fluctuation, especially at the evaporator and at the inlet of the condenser section. The reason for this may be that a change in the system vapor pressure at the range of the saturation temperature of water is strongly amplified. This result indicates that water as the working fluid shows the highest instability for the range of the present study. One of the reasons seems to be the low vapor pressure range associated with in the temperature range of the present application. For this reason, water is

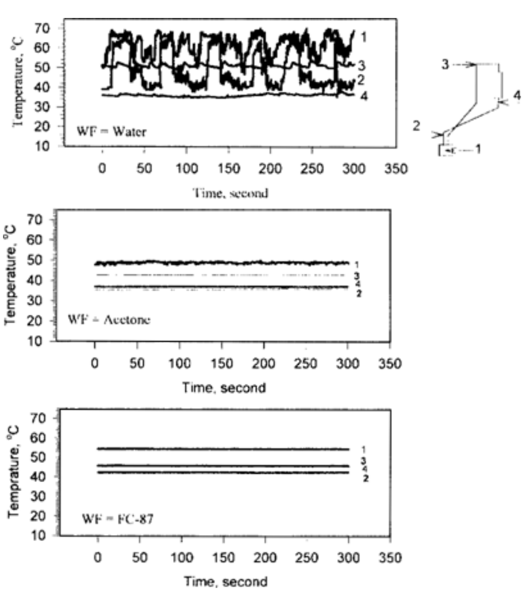


Fig. 15 Transient temperature fluctuation on heat load of $Q=100\,W$; $V_{WF}=40$ml, $l_c=200$ ml, $u_{max}=5$ m/sec.

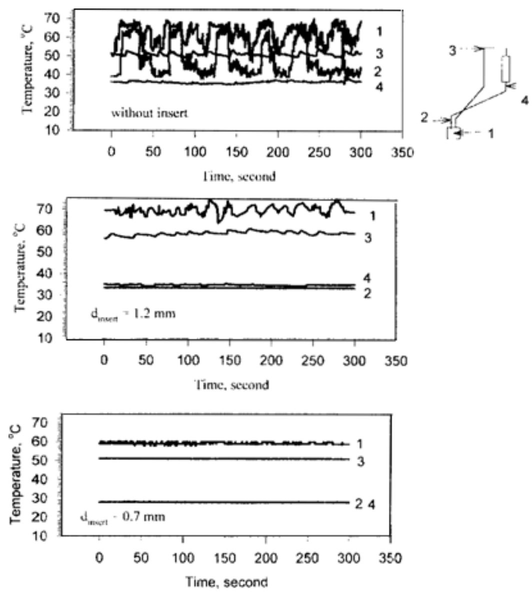


Fig. 16 Effect of insert diameter on temperature fluctuation; WF = water $Q=100$ W.

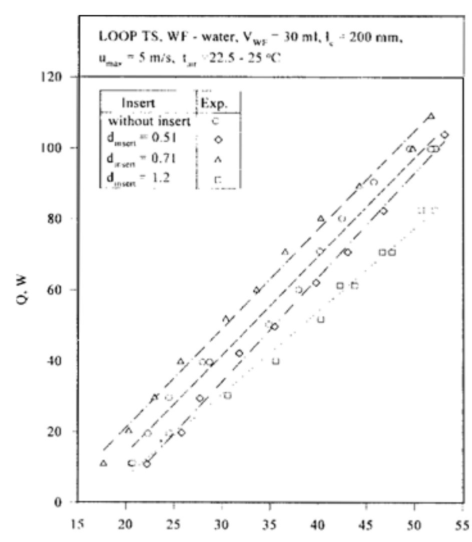


Fig. 17 Effect of Δt_{h-air} on Q with different insert.

not recommended as a possible choice of working fluid of the proposed system.

The results effected by insert orifices are shown in Figs. 16 and 17. These orifice inserts were used to investigate the effect of a pressure drop change in the flow channel of the TLT on the flow instability and temperature fluctuation. It can be seen in Fig. 16 that decreasing the size of orifices from 3.7 mm (no insert) to 0.7 mm drastically reduced the fluctuation of the temperature, especially that of the evaporator. It is indicated that with the orifice of 0.7 mm, the overall temperature fluctuation can be almost eliminated as the power is increased. It was expected that the smaller the size of the orifice, the larger the overall pressure drop of the system, thus reducing the total mass flow rate which would have resulted in low heat transfer performance. And Fig. 17 shows that at the orifice diameter of 0.7 mm in the present TLT system, heat transfer rate is attained to its peak. It is predicted that the reason for decreasing the TLT performance at the orifice diameter of 0.51 mm and 1.2 mm is due to high fluctuations and blocked flow circulation along the TLT loop respectively, caused by improper orifice inserts, as shown in Figs. 15 and 16.

5. Conclusions

From the experimental and visualization study, the following conclusions are made:

(1) The experimental results indicate that the loop type thermosyphon system of the TLT design would satisfy the all initial design.

- The overall heat transfer coefficient of the system was found to increase gradually with saturation temperature.

- Cooling heat fluxes of a little over 9 W/cm^2 with one condenser and 12 W/cm^2 with two condensers with the overall temperature difference of 50°C were attained under the forced convection cooling of the condenser section, with acetone or FC-87 as the working fluid.

- Under the natural convection cooling of the condenser section, a cooling heat flux about 5 W/cm^2 was attained.

(2) For the condenser section, numbers of condenser, its arrangement and convection mode in the finned condenser section were identified.

- For all working fluids, increasing the air velocity to the condenser section led to a reduced heater temperature and to an increased heat transfer capacity.

- The cooling air velocity to the condenser section affected strongly at 4.5~5 m/s the heat transfer capacity of the system.

(3) Effects of working fluids of water, acetone, ethanol, R11, R113, FC-72 and FC-87 were clarified, including the quantity of the working fluid and the fluctuation of the heights of the working fluid.

- Water as the working fluid was not recommended in view of high fluctuations as well as high instability for the range of the system.

- The quantity of working fluid, VWF, optimized. It showed that 30~40 ml was reasonable for water and acetone and for FC-87 40~50 ml was good. Increasing VWF would decrease the thermal performance of the system.

(4) From the different type of design for the evaporator, the contact resistance was seen as one of the major thermal resistance of the system and Evaporator C was recommended.

(5) From a measurement for the mass flow rate, the present experiment was identified instead of performing the purely analytical solution.

(6) From the photography of flow regimes and temperature fluctuation, flow instability and boiling process was visualized.

(7) In the temperature fluctuations as a function of working without any insert to damp the flow fluctuation, acetone and FC-87 hardly showed any fluctuation, whereas water has shown very strong fluctuation resulting not recommended as the working fluid

References

Adami, M. and Yimer, B., 1988, "Development and Evaluation of an Planar Heat Pipe for Cooling Electronic System," *Proc. of ASME Nat. Heat Transfer Conf.*, pp. 499~507.

Chi, S. W., 1976, Heat Pipe Theory and Practice: A Sourcebook, McGraw-Hill, New York.

Chu, R. C. and Simons R. E., 1994, "Cooling Technology for High Performance Computers: Design Applications, Cooling of Electronic Systems," NATO ASI Series E: Applied Science - Vol. 258, Edited by Kakac et al., pp. 71~95, Kluwer Academic Publishers.

Faghri, A., 1995, *Heat Pipe Science and Technology,* Taylor & Francis, Washington, DC.

Hewitt, G. F., Shiress, G. L. and Bott, T. R., 1994, *Process Heat Transfer,* CRC Press, London.

Holamn, J. P., 1996, *Heat Transfer,* 8th ed., McGraw-Hill, New York.

Kim, W. T., Lee, Y. P. and Yoon, S. Y., 1996, "The Thermal Performance of Open-type Two-phase Loop Thermosyphons for Multichip Module," *Proc. KSME-JSME Conf.,* pp. 77~82.

Kishimoto, T. and Harada, A., 1992, "Two-Phase Thermosyphon Cooling for Telecom Multichip Modules," *ASME Advances in Electronics Packaging,* pp. 135~141.

Kishimoto, T., Harada, A., Sasaki, S. and Iwata, T., 1994, "Heat Pipe Cooling Technologies for Telecom JMCMs," *Proceedings of 4th Int. Heat Pipe Symposium,* pp. 132~141.

Larkin, B. S., 1984, "Temperature Control of Electronic Systems Using Inexpensive Heat Pipes," *5th IHPC,* Preprints IV, pp. 136~141.

Lee, Y. and Mital, U., 1972, "A Two-Phase Closed Thermosyphon," *Int. J. Heat and Mass Transfer,* Vol. 15, pp. 1695~1707.

Lee, Y., Rhi, S. H., Pioro, I. and Kim, W. T., 1996, "An Experimental Study on a Cooling System using Two-Phase Closed Thermosyphons for Telecommunication MCM," *9th ISTP,* pp. 480~485.

Michael, P., 1991, *Handbook of Electronic Package Design,* Marcel Dekker, Inc., pp. 385~451.

Nelson, R. D., Sommerfeldt, S. and Bar-Cohen, A., 1994, "Thermal Performance of An Integral Immersion Cooled Multichip Module Package," *IEEE Tr. on Components, Packaging,* and Manufacturing Technology, Part A, Vol. 17, No. 3, pp. 405~412.

Peterson, G. P., 1994, *Heat Pipes,* John Wiley & Sons, Inc., pp. 270~322.

Peterson, G. P. and Ma, H. B., 1996, "Theoretical Analysis of the Maximum Heat Transport in Triangular Grooves: A Study of Localized Micro Heat Pipes," *ASME J. Heat Transfer,* Vol. 118, pp. 731~739.

Rhi, S. H., Kim, W. T. and Lee, Y., 1997, "A Cooling System Using Wickless Heat Pipes for Multichip Modules: Experiment and Analysis," *KSME Int. J.,* Vol. 11, No. 2, pp. 208~220.