한국동력기계공학회지 제5권 제1호 pp. 27~34 2001년 2월

Experimental Studies on the Heat Transfer Performance of Plain and Low Finned Thermosyphons

평관 및 낮은 핀관으로 제작한 열사이폰의 열전달 성능에 관한 실험적 연구

S. S. Ye, K. I. Han, S. H. Park and D. H. Cho

예석수 · 한규일 · 박성현 · 조동현

Key Words : Boiling(비등), Condensation(응축), Working Fluid(작동유체), Inclination Angle(경사각), Two-phase(이상), Thermosyphon(열사이폰)

요 약 : 관 외벽에 낮은 핀을 가진 수직 및 경사 열사이폰의 열전달 성능에 관한 실험적인 연구를 하였다. 관 외벽에 낮은 핀을 가진 이상밀폐 열사이폰의 열전달 성능을 비교 분석하기 위하여 동일한 규격의 평관에서도 실험적인 연구를 하였다. 작동유체는 증류수와 CFC-30을 사용하였다. 열사이폰의 경사각과 작동온도를 변화시키면서 실험한 결과 경사각의 변화에 따라 열사이폰의 열전달 성능은 큰 변화를 나타내었다. 그리고 평관으로 제작한 열사이폰보다 관 외벽에 낮은 핀관을 가진 동관으로 제작한 열사이폰의 열전달 성능이 높게 나타났다. 그리고 열사이폰의 경사각이 20~50° 범위에서 열전달 성능이 높게 나타났다.

Nomenclature

A : Heat transfer surface area (m^2)
D : Tube diameter (m)
e : Fin height (m)
h_c : Condensation heat transfer coefficient (W/m^2K)
h_{fg} : Heat of vaporization (kJ/kg)
k : Thermal conductivity (W/mK)
L : Lenght (m)
P_f : Distance between fins at fin top (m)
P_s : Pressure (N/m^2)
Q : Heat flow rate (W)
R : Radius (m)
S_b : Distance between fins (m)
t : Fin thickness (m)
T : Temperature (K)
U : Overall heat transfer coefficient (W/m^2K)

Greek Symbols

ρ : Density of condensate (kg/m^3)
μ : Dynamic viscosity of condensate (Ns/m^2)
θ : Angle formed by the horizontal (°)
ψ : Liquid filling as the ratio of working fluid volume to total volume of a thermosypon (%)

Subscript

ai : Adiabatic zone
c : Cooling fluid
ci : Condenser
ei : Evaporator
h : Heating fluid
Nu : Nusselt number
l : Liquid

접수일 : 2000년 12월 1일
한규일 : 부경대학교 기계공학부
조동현 : 대진대학교 기계설계공학과
예석수, 박성현 : 부경대학교 대학원

1. INTRODUCTION

It has been known that high rates of the heat transfer can be obtained by means of the evaporation - condensation. There is closed system that utilizes this process, namely, two-phase closed thermosyphons. It can be used for a great variety of terrestrial applications, such as solar energy utilization, heat recovery, and energy conservation. In these applications two-phase closed thermosyphon can be operated with the condenser on the upper part, so that the gravitational force can help the liquid return to the heated zone. In this way high heat flow rates through two-phase closed thermosyphon may be obtained. In some cases, the two-phase closed thermosyphon heat exchangers are required to be inclined to the direction of gravity. Research on the heat transfer in the two-phase closed thermosyphon was first performed by Cohen and Bayley[1] and followed by Larkin[2], Lee and Mital[3], Stret'tsov[4], Savchenkov[5] and Andros[6], so that considerable information on the heat transfer and flow phenomena has been obtained so far.

Many investigators agree that there exists a range of inclination angle at which the heat transfer coefficient of the thermosyphon compared to vertical position. Because the thermosyphon works with the assistance of gravity, its transport capability is highly dependent on the direction of gravity. Tu et al.[7] have reported the effect of inclination angle on the thermosyphon using a carbon steel tube with water as the working fluid. They suggest an operation angle between 50° and 55° from horizontal position. Negishi and Sawada[8] carried out an investigation with a copper thermosyphon of 330mm in length and 13mm in diameter. The high heat transfer coefficient was found between 20° and 40° for water and between 40° and 5 0° for Freon 113 in a glass tube. It is the intention here to gain better insight into the transport processes of a thermosyphon by observing local phenomena in different parts of the device. In this paper, the experimental studies on the vertical and the inclined thermosyphon with low integral - fins have been carried out. Especially the effects of inclination of thermosyphon and the heat flow rate upon the heat transfer coefficient within a heating zone and a cooling zone are investigated and analyzed. The plain thermosyphon having the same inner diameter and outer diameter as those of the finned thermosyphon is also tested for comparison.

2. EXPERIMENTAL THERMOSYPHONS DESCRIPTION

The main characteristics of the experimental thermosyphon are shown in following Table 1. The cross section and photograph of finned thermosyphon are shown in Figs. 1 and 2.

Table 1 Geometric specifications of plain and low integral-fin thermosyphons

Tube No.	Tube specification		Fin specification				Tube length
	Do	Di	fin Density	pf	t	ϕ	
	mm	mm	fpm	mm	mm	deg	mm
	<Plain thermosyphon>						
1	15.7	12.7					1200
	<Low integral-fin thermosyphon>						
2	15.7	12.7	649	1.54	1.5	2	1200

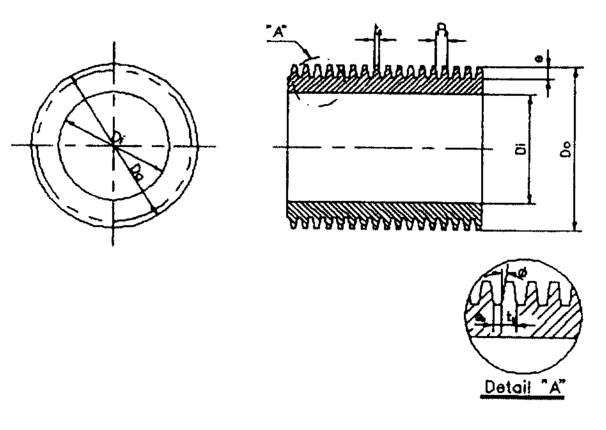


Fig. 1 Cross section of finned tube

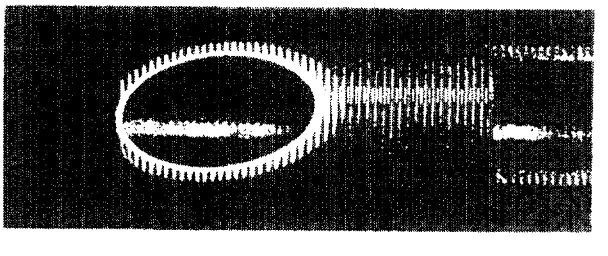
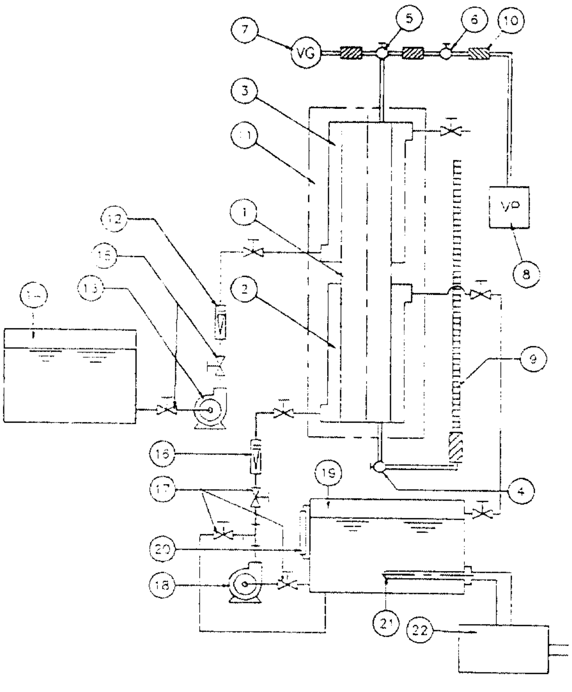


Fig. 2 Photograph of finned tube

3. EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus is illustrated in Fig 3. The main heat transfer system consists of the test thermosyphon, a hot water boiler, a cooling and heating jacket, a cooling and heating system and related instrumentation. The tested thermosyphon is illustrated in Fig 4. The tested thermosyphon used was made of a 15.9mm O.D. standard copper tube. The tube was 1200mm long and had a 12.7mm I. D., with a wall thickness of 1.6mm. Two 550mm long water jackets were set on the tube. One was used as a heating jacket for an evaporator and the other was used as a cooling jacket for a condenser. An inlet small tube for heating or cooling water flow into each jacket was directed at a tangent to the inside surface of the jacket so as to prevent the thermosyphon from direct exposure to the water flow. This thermosyphon can be positioned with any inclination, from 0° to 90° with respect to the horizontal position. The inside surface of the thermosyphon was cleaned thoroughly by a neutral cleanser, ethanol and distilled water. A plug was placed in the tube wall near the end of the condenser in order to allow the inner space of the thermosyphon to be evacuated and the working fluid to be injected. Nine thermocouples were soldered on the outside surface of the tube along its length and thermocouples were uniformly spaced on the circumference at the center of the evaporator and condenser. Four more thermocouples were placed at the inlets and the exits of two water jackets. The outputs of these thermocouples were recorded on a digital thermometer. A mechanical vacuum pump (Edwards EM80) with a rating of 10^{-4} Torr was used to remove air and other non- condensable gases. In the present study, distilled water and methylene chloride were chosen as the working fluids, since are compatible with copper and safe materials to work with. The working fluid was injected into the tube after evacuating air. After injecting the working fluid, heating and cooling water flowed into the evaporator and the condenser jackets, respectively.



1. Test Tube 2. Heating Water Chamber 3. Cooling Water Chamber 4. Vacuum Valve 5. Vacuum Valve 6. Vacuum Valve 7.Vacuum Gauge 8.Vacuum Pump 9. Measuring Device for Liquid Level 10, Vacuum Rubber Hose 11. Insulation 12. Coolant Flow Meter 13. Coolant Pump 14. Coolant Tank 15. Coolant Control Valve 16. Heating Water Flow Meter 17. Heating Water Control Valve 18. Heating Water Control Pump 19. Heating Water Boiler 20. Liquid Level Gauge 21. Electric Heater 22. Thermo-Controller

Fig. 3 Schematic diagram of experimental appa-ratus

A small amount of non-condensable gas was collected at the end of the condenser after a few minutes of operation. This was the gas which had been solved in the working fluid at atmospheric pressure and temperature. This gas was removed through the plug by the vacuum pump mentioned previously.

The experimental conditions were set as follows. The heating and cooling water temperature was varied. The heat transfer rate was obtained as varying inclination angles, input temperatures and the mass flow rates of cooling water through the condenser jacket. The temperature distributions on the evaporator and the condenser walls were recorded.

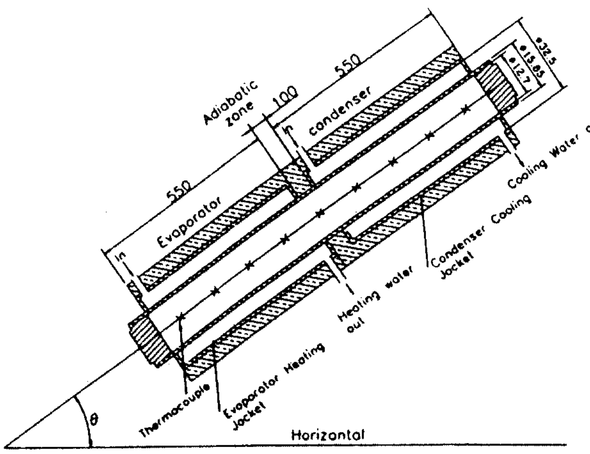


Fig. 4 Cross-sectional view of experimental incli-ned two-phase thermosyphon

4. RESULTS & DISCUSSION

4.1 Temperature Distribution in the Test Thermosyph-on

Temperature distributions on heated wall and cooled wall in the test tube are shown in Fig. 5. The working fluid used in experiment is the distilled water. Fig. 5 is the results for the effects of the heat flux with the other variables being almost constant. The temperature distributions on the heated wall scatter a little bit to some degree.

When the heat flux is small, the rate of vapor generation due to evaporation and boiling in the heated section was small and condensate film flowing down onto heated wall then was so thin that it was apt to break down easily. On the contrary, an increase of the heat flux raised the flow rate of the condensate film and the film became rather stable.

4.2 Heat Transfer in the Condenser

When the fill quantity is small, the film condensation occurs in the cooled section, but as the fill quantity increases, the surface of the

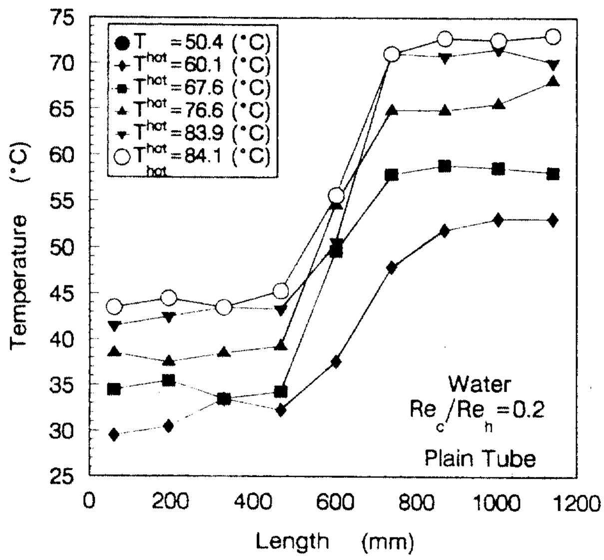


Fig. 5 Temperature distributions along the length of thermosyphon (Plain Tube)

two-phase mixture is elevated into the cooled section and two-phase mixture convection takes place there. Eq. (1) is Nusselt equation for the film condensation of vertical thermosyphon.

$$h_c=0.943\left[\frac{\rho^2 lgh_{fg}k^3 l}{L_c\mu l(T_{ai}-T_{ci})}\right] \tag{1}$$

Fig. 6 shows a comparison of the experimental results with Eq. (1). The heat transfer coefficients with water having a high latent heat of vaporization and high thermal conductivity scatter around the solid line and data scattering is not so small, but scattering becomes smaller with increasing the heat transfer rate. This may be due to a relatively small temperature difference between the vapor temperature and the inside wall temperature in a comparison of the accuracy of measurement.

Eq. (2) shows Yiwei's[11] semi-empirical equation for the film condensation of inclined thermosyphon:

$$\frac{h_c}{h_{Nu}}=P_s^{0.37}\left(\frac{L}{R}\right)\frac{\cos\theta}{4}[0.41-0.72\phi+(-62.7\phi^2+14.5\phi+7.1)\theta/100] \tag{2}$$

All the relevant experimental data are included

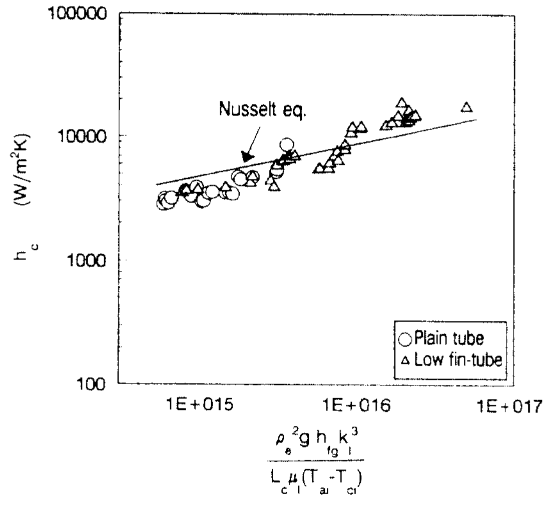


Fig. 6 Comparison of the experimental data with Nusselt's correlation

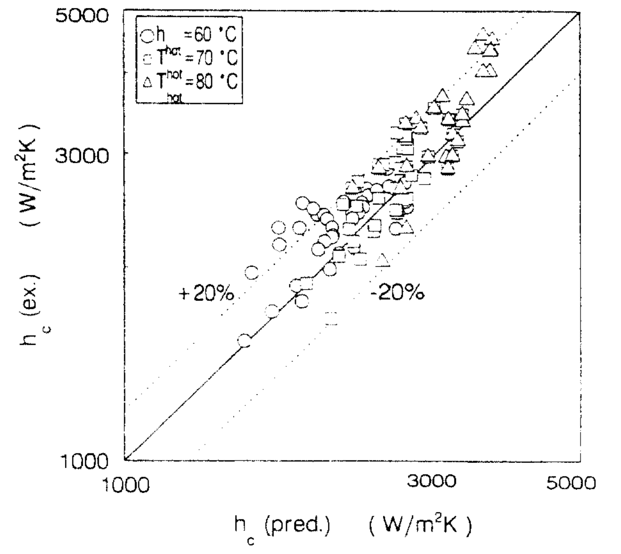


Fig. 8 Comparisons between Eq. (2) and the experimental data

in Fig. 7. It can be seen that the condensation coefficient increases as heating fluid temperature of the evaporator section becomes higher and the ratio of Reynolds number of cooling to heating water becomes greater. In our experiment, both theoretical analysis and experimental studies have indicated that heat flux or temperature difference has no effects on ratio h_c/h_{Nu} and the optimum inclination angle is about 20° ~ 50° . As the optimum inclination angle from the horizontal position, newly condensed surface was exposed to the vapor after surface was wiped by the dashing liquid.

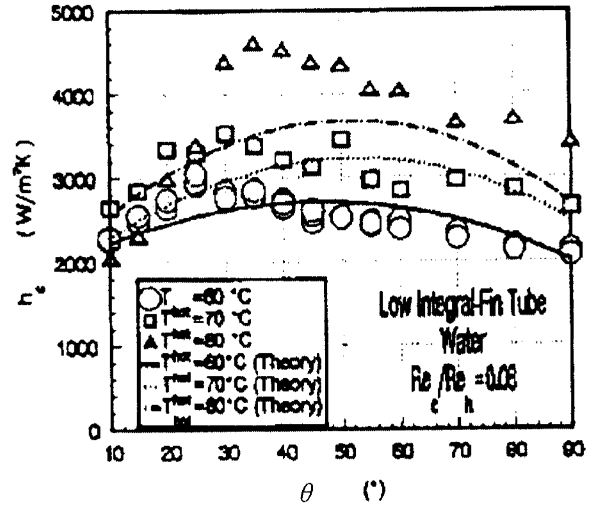


Fig. 7 Comparisons between Eq. (2) and the experimental data

Due to these factors, the heat transfer performance may have the highest value at this angle. Fig. 8 shows a comparison of the experiment results with Eq. (2). The maximum deviation was about 20% for all experimental data.

4.3 Effect of Thermosyphon Inclination

The thermosyphon can be mainly operated under the assistance of the gravity. Therefore, the heat transport capability of thermosyphon is highly affected by the direction of the gravity. This means that the inclination angle has a large effect on the operating characteristics of a thermosyphon. For this reason, it is necessary to study the effect of the inclination angle in order to search for the new application field and the clarification of the phenomena of the thermosyphon. The effect of the orientation of the test thermosyphon with respect to the horizontal position is shown in Fig. 9 through Fig. 13.

Distilled water and CFC-30 have been used as the working fluids. It is observed that for heating temperature of 60℃, 70℃, 80℃ respectively at a given condition. Thermosyphon exhibits the highest heat transfer performance at an operation angle between 30℃ and 50℃ from horizontal position. Hahne and Gross[10] explained

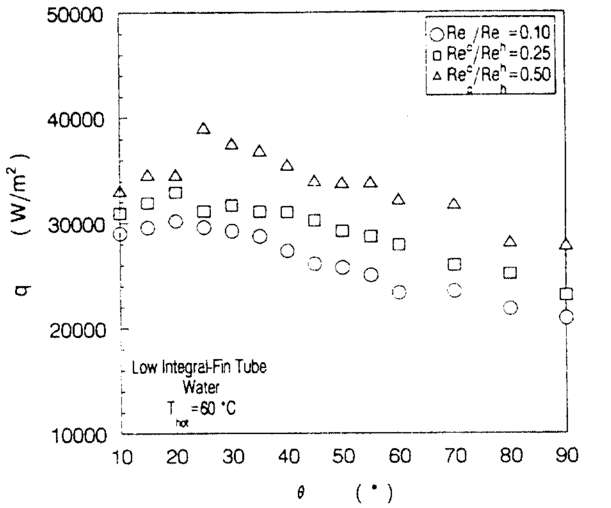


Fig. 9 Plot of heat flux against inclination angle (T_{hot} = 60℃)

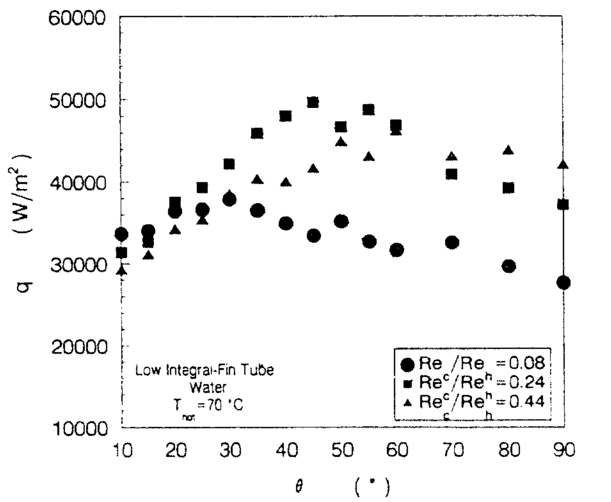


Fig. 10 Plot of heat flux against inclination angle (T_{hot} = 70℃)

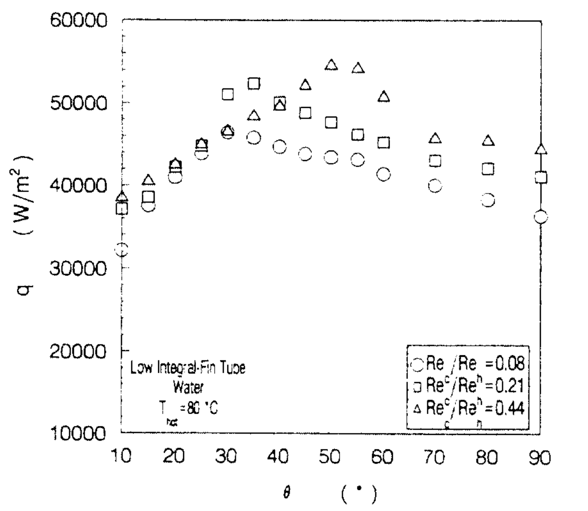


Fig. 11 Plot of heat flux against inclination angle (T_{hot} = 80℃)

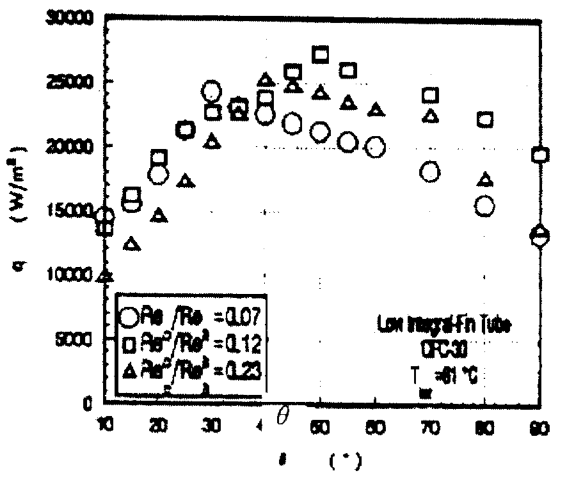


Fig. 12 Plot of heat flux against inclination angle (CFC-30)

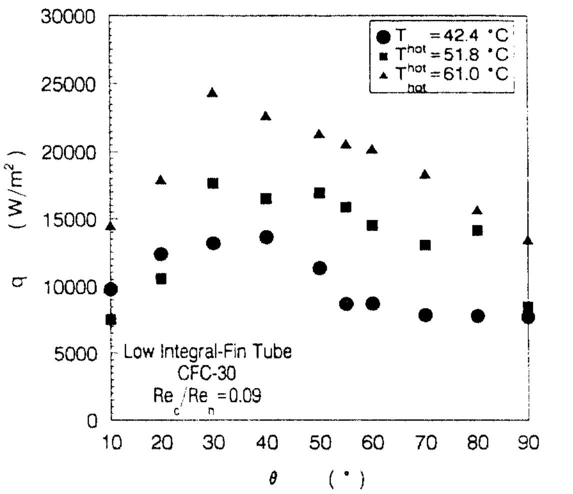


Fig. 13 Plot of heat flux against inclination angle inclination angle (CFC-30)

the higher heat transfer coefficients as a result of coupled effect of gravitational force and frictional force. As the angle of inclination is increased from the vertical to horizontal, both forces decrease. However, heat transfer coefficient is proportional to the gravity force and reversely proportional to the frictional force. Optimal angle could be anywhere between 0° and 90° . In our experiment, 20° ~50° is the range of the angle where the coupled effect of gravitational force and frictional force on heat transfer coefficient is at the maximum. Negishi and Sawada[12] explained that the heat transfer mechanism is considered to be affected by the condensation of vapor and forced convection by the dashing working fluid in the thermosyphon. As the inclination angle increased to this range from the vertical position, the dashing tip of the working fluid went up along the condenser and finally reached the end of condenser. Also newly condensed surface was exposed to the vapor after the surface was wiped by dashing liquid. Due to these factors, the transfer performance might have the highest value at this angle.

4.4 Overall Heat Transfer coefficients

The heat transfer mechanism of two- phase closed thermosyphon is an intricate complex consisting of evaporation with or without boiling, or unsteady condensate rivulet formation and the turbulent dashing motion of liquid as mentioned previously. Therefore it is almost impossible to represent the heat transfer coefficients by a simple formula. In the present study, the overall heat transfer coefficients have been obtained. The fill ratio of the working fluid was chosen to be 25% for water and methylene chloride. The overall heat transfer performance of a thermosyphon is generally characterized by the following equation introduced by Nguyen-Chi:

$$Q = UA_e(T_{ei} - T_{ci}) \tag{3}$$

Where U is an overall heat transfer coefficients, A_e is an inside surface area of the evaporator or the condenser, T_{ei} and T_{ci} are the temperatures on the inside surfaces of the evaporator and the condenser, respectively. The operating temperature is defined as an average temperature of the evaporator and condenser, hence it can be considered as the temperature of the working fluid. Typical results of the effect of operating temperature on the overall heat transfer coefficients for the plain and low integral fin-thermosyphon are illustrated in Fig. 14. It can be seen that the overall heat transfer coefficients for thermosyphon with low fins are higher than those of plain tubes. With water and CFC-30 as the working fluid, the overall heat transfer coefficient increases with the operating temperature. This is due to the fact that the operating temperature increases with increasing the activity of the dashing motion of liquid.

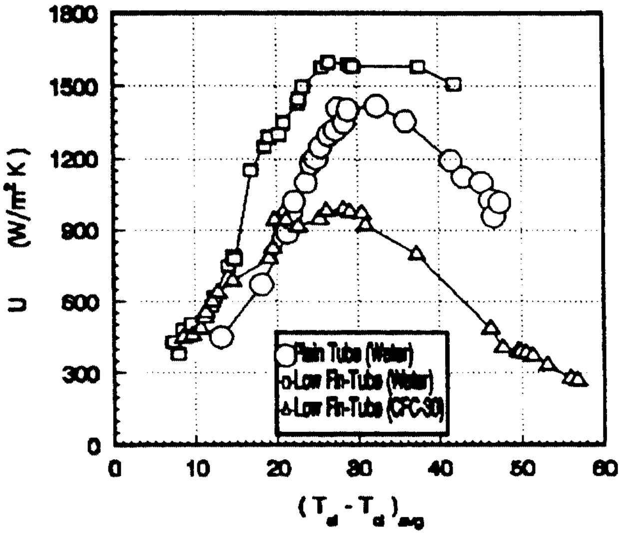


Fig. 14 Plot of overall heat transfer coefficient against operating temperatures

5. CONCLUSIONS

The conclusions of the present study can be summarized as follows;

1) The heat transfer coefficients of the vertical and inclined thermosyphon were estimated from Nusselt's and Yiwei's equation respectively, in which the heat transfer coefficients(h_c) agreed with the experimental values.
2) The heat flux or temperature difference has little effect on the ratio h_c/h_{Nu}.
3) The inclination angle of a thermosyphon has a notable influence on the condensation coefficient and the optimum inclination angle is between 20° and 50°.
4) The maximum overall heat transfer coefficient enhancement of thermosyphon (i.e, the ratio of overall heat transfer coefficient of finned to plain tubes) is about 1.3.

REFERENCE

1. H. Cohen and F. J. Baylay, "Heat Transfer Problems of Liquid Cooled Gas Turbine Blades", Proc. Inst. Mech. Eng., No. 169, pp. 1063~1080
2. B. S. Larkin, Trans. Canada. Soc. Mech. Engrg., 14-B6, pp. 1~8, 1971
3. Y. Lee and U. Mital, "A Two-Phase Closed Thermosyphon", Int. J. Heat and Mass Transfer, Vol. 15, pp. 1695~1707, 1984
4. A. I. Stret'tsov, "Heat Transfer - Soviet Research 7-1", pp. 23~27
5. G. A. Savochenkov and Z. R. Gorbis, "Heat transfer-Soviet Research 8-4", pp. 52~56
6. F. E. Andros and L. W. Florschuetz, "Two Phase Flow Heat Transfer Symp. Workshop", Vol. 4, pp. 1231~1267.
7. C. Tu et al, The two Phase Closed "Thermosyphon: an Experimental Study with Flow Pattern and Optimum Fill Charged", China-U.S. Seminar on Two-flows and Heat Transfer, Sian, pp. 395~405
8. H. Nguyen-Chi and M. Groll, "Entrainment of Foolding Limit in a Closed Two-Phase Thermosyphon", 4th IHPC, Tsukuba, 1-6, 1983
9. W. Nusselt, "Die Ooberflachen - kondensation des Waserdampfes", Zeitschr. V.D.I., Vol. 60, pp. 541~569, 1916
10. E. Hahne, and U. Gross, "The Influence of the Inclination Angle on the Performance of a Closed Two-Phase Thermosyphon", 4th IHPC, London, pp. 125~135, 1971
11. M. Yiwei, J. C. Y. Wang, L. Jifu and F. Yi, :Theoretical and Experimental Studies on Condensation Heat Transfer inside Vertical and Inclined Thermosyphons", National Heat Transfer Conference HTD-Vol. 108, pp. 11 1~116, 1987
12. K. Negishi and T. Sawada, "Heat Transfer Performance of an Inclined Two-Phase Closed Thermosyphon", Int. J. Heat Mass Transfer, 26, No. 8, pp. 1207~1213, 1983