Influence of the Inclination Angle and Liquid Charge Ratio on the Condensation in Closed Two-Phase Thermosyphons with Axial Internal Low-Fins

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This study concerns the performance of the heat transfer of the thermosyphons having 60, 70, 80, 90 axial internal low-fins in which boiling and condensation occurr. Water, HCFC-141b and CFC-11 have been used as the working fluids. The operating temperature, the liquid charge ratio and the inclination angle of thermosyphons have been used as the experimental parameters. The heat flux and heat transfer coefficient at the condenser are estimated from experimental results. The experimental results have been assessed and compared with existing theories. As a result of the experimental investigation, it was found that the maximum heat flow rate in the thermosyphons is dependent upon the liquid charge ratio and inclination angle. A relatively high rate of heat transfer has been achieved by the thermosyphon with axial internal low-fins. The inclination of a thermosyphon has a notable influence on the condensation. In addition, the overall heat transfer coefficients and the characteristics at the operating temperature are obtained for the practical applications.

Key Words: Condensation, Inclination Angle, Liquid Charge Ratio, Heat Transfer Coefficient, Thermosyphon

Nomenclature

A : Heat transfer surface area (m^2)
C_p : Specific heat (J/kgK)
D : Tube diameter (m)
H : 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 : Length (m)
$\dot{m}$: Mass flow rate (kg/s)
P : Pressure (Pa)
Q : Rate of heat flow (W)
q : Heat flux (W/m^2)
R : Radius (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 (°)
ϕ : Liquid charge ratio

Subscript

avg : Average
c : Condensation
$cool$: Cooling fluid
hot : Heating fluid
i : Inner

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Nu : Nusselt number
l : Liquid
w : Wall
o : Outer

1. Introduction

New fields of application and growing concern of a more intensive use of energy have stimulated and revived many suggestions for economic heat transfer devices. One of such devices is the thermosyphon and growing interest in it is documented by a great number of investigations. Two-phase closed thermosyphon can operate 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 flowrates through two-phase closed thermosyphon may be obtained. In some cases, the two-phase closed thermosyphon needs to be inclined from the direction of gravity. Research on the heat transfer in the two-phase closed thermosyphon was first performed by Cohen and Bayley (1955) and followed by Lee and Mital (1972), Rhi et al. (1997), Kim et al. (1998), Lee (2001), Noie at al. (2002) so that considerable information on the heat transfer and flow phenomena have been obtained so far.

Many investigators agree that there exists a range of inclination angle at which the heat transfer coefficient of the thermosyphon is higher than that at the vertical position. Because the thermosyphon works under the assistance of gravity, its transport capability is highly dependent on the direction of gravity. Tu et al. (1984) have reported the effect of inclination angle on the thermosyphon using a carbon steel tube with water as the working fluid. They suggested an operation angle between 50° and 55° from horizontal position. Negishi and Sawada (1983) carried out an investigation with a copper thermosyphon of 330 mm in length and 13 mm in diameter. High heat transfer coefficient was observed at an inclination angle of 20～40° for water and 40～50° for freon 113 in a glass tube. This study aims at getting better insight into the transport processes of a thermosyphon by observing local phenomena in different parts of the device. In this paper, experimental studies on vertical and inclined thermosyphons with axial internal fins have been carried out. Especially, the effects of inclination, liquid filling of thermosyphon and the heat flow rate on the heat transfer coefficient within the heating zone and the cooling zone are investigated.

Table 1 Geometric specifications of thermosyphons with axial internal low-fins

No.	Fins	D_o (mm)	D_i (mm)	H (mm)	L (mm)
1	60	15.9	14.3	0.3	1500
2	70	15.9	14.3	0.3	1500
3	80	15.9	14.3	0.3	1500
4	90	15.9	14.3	0.3	1500

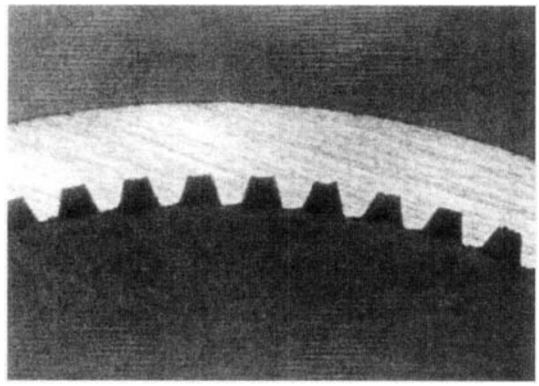


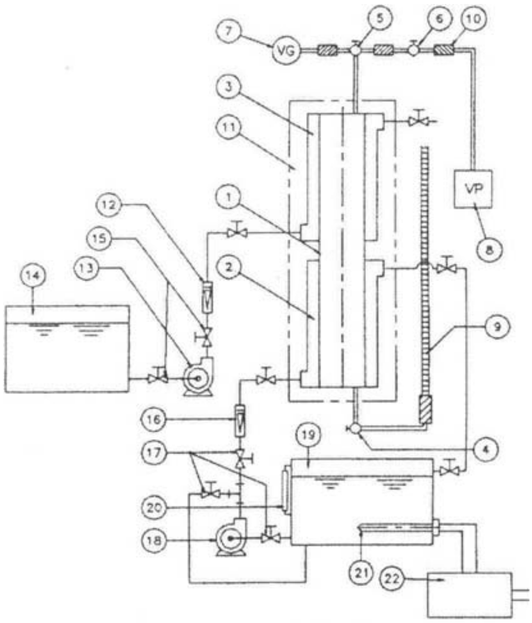
Fig. 1 Cross section of thermosyphon with axial internal fins

2. Description of Experimental Thermosyphons

The main characteristics of the test thermosyphon are shown in following Table 1. The cross section of the finned thermosyphon is shown in Fig. 1.

3. Experimental Apparatus and Procedure

The experimental apparatus is illustrated in Fig 2. The system consists of the test thermosyphon, a hot-water boiler, a cooling and heating



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 pump
19. Heating water boiler
20. Liquid level gauge
21. Electric heater
22. Thermo-controller

Fig. 2 Schamatic digram of the experimental apparatus

jacket, cooling and heating systems and appropriate instrumentation. The test thermosyphon is made of a 15.9mm O. D. standard cooper tube. The tube is 1500 mm long and had a 14.3 mm I. D., with a wall thickness of 0.8 mm. Two 600 mm long water jackets were set on the tube. One was used as a heating jacket for an evaporator and the other as a cooling jacket for a condenser. Small tubes to guide heating and cooling water to each jacket were directed at a tangent to the inside surface of the jacket so as to prevent the thermosyphon from direct exposure to the water flow. The thermosyphon can be positioned with any inclination, from 0° to 90° with respect to the horizontal position. The inner 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. Thirteen thermocouples were soldered on the outer surface of the tube along its length and 9 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 outlets of two water jackets. The outputs of these thermocouples were recorded on a digital thermometer. A mechanical vacuum pump (Edwards EM80) and a diffusion pump with a rating of 10^{-6} Torr were used to remove air and other non-condensable gases. In the present study, distilled water, CFC-11 and HCFC-141b were chosen as the working fluids, since they are compatible with copper and safe 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. A small amount of non-condensable gas was collected at the end of the condenser after a few minutes of operation. This was 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 temperatures were varied. The heat transfer rate was obtained for different experimental conditions varying the inclination angle, fill quantity, temperature rise and the mass flow rate of cooling water through the condenser jacket. The temperature distributions on the evaporator and the condenser walls were recorded.

4. Results and Discussion

4.1 Heat transfer in the condenser

The mass flow rate of condensate $\dot{m}_c$ is determined

$$Q_c = \dot{m}_c h_{fg}''. \qquad (1)$$

The heat flow rate of condenser zone are given by

$$Q_{cool} = \dot{m}_{cool} C_{p,cool} (T_o - T_i)_{avg}. \qquad (2)$$

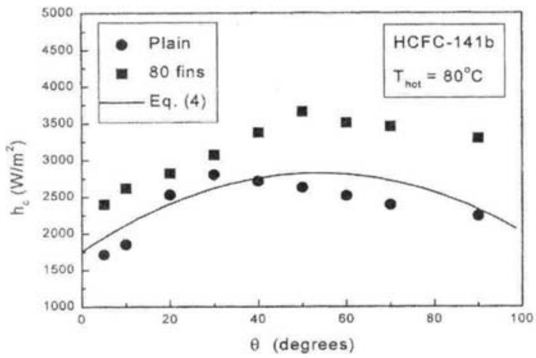


Fig. 3 Comparison of the condensation heat transfer coefficient with a correlation

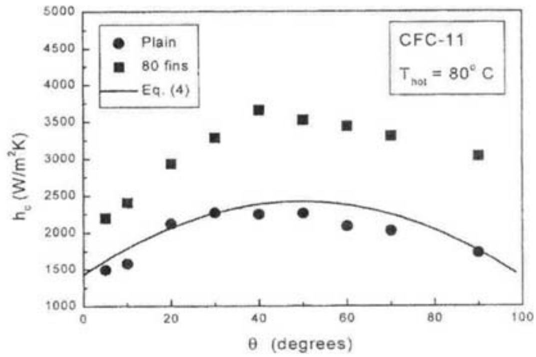


Fig. 4 Comparison of the condensation heat transfer coefficient with a correlation

and the condensation heat transfer coefficient is obtained from experimental data by Eq. (3).

$$h_c = \frac{Q_{cool}}{A_i(T_{wc} - T_{sc})_{avg}}. \quad (3)$$

When the fill quantity is small, film condensation occurs in the cooling section, but as the fill quantity increases, the surface of the two-phase mixture is elevated into the cooled section and two-phase mixture convection takes place there. In this work, the results are correlated with the following semi-empirical equation for the film condensation in inclined thermosyphons proposed by Yiwei (1989):

$$\frac{h_c}{h_{Nu}} = P_s^{0.37}\left(\frac{L}{R}\right)^{\frac{\cos\theta}{4}}[0.41 - 0.72\phi + (62.7\phi^2 + 14.5\phi + 7.1)\,\theta/1000] \quad (4)$$

where h_c is the condensation heat transfer coefficient, h_{Nu} Nusselt's condensation heat transfer coefficient (1916), P_s vapor pressure, θ angle from the horizontal, ϕ liquid filling defined as the ratio of working-fluid volume to total volume of a thermosyphon. All the relevant experimental data are included in Figs. 3 and 4. In our experimental results, consistent with the theoretical prediction, indicated that heat flux or temperature difference has no effects on ratio h_c/h_{Nu} and that the optimum inclination angle is about 40°~50°. The thermosyphon with 80 fins gives a 30~50% increase in h_c. This corresponds to an enhancement ratio of 1.3~15, compared with the plain test thermpsyphon. At the optimum inclination angle from the horizontal position, newly condensed surface was exposed to the vapor after the surface was wiped by the dashing liquid. Therefor, it is likely that the heat transfer performance has the highest value at the angle.

4.2 Effect of inclination angle

The thermosyphon should operate under the assistance of the gravity. Therefore, the heat transport capability of the thermosyphons is highly affected by the direction of the gravity. This means the inclination angle has a large effect on the operating characteristics of thermosyphons. 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 in the thermosyphon. The effect of the orientation of the test thermosyphons is shown in Figs. 5~8.

Water, CFC-11 and HCFC-141b have been used as the working fluids. It can be observed that for heating-fluid temperatures 60℃ and 80℃, the thermosyphons exhibits the highest heat transfer performance at an operation angle between 40° and 50° from the horizontal. Hahne and Gross (1981) explained the einhanced 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 frictional force. Optimal angle could be anywhere between 0° and 90°. In our experiment, 35°~50° is the range of the angle where the coupled effect of

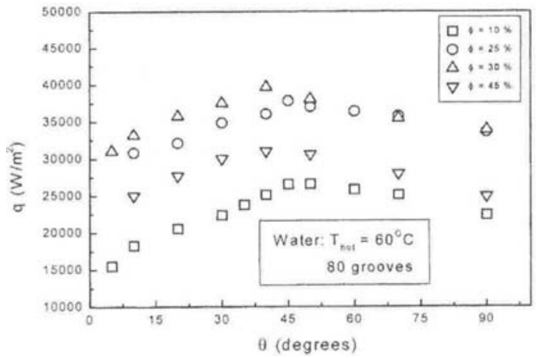


Fig. 5 Plot of condensation heat flux against inclination angle. (water, $T_{hot}=60℃$)

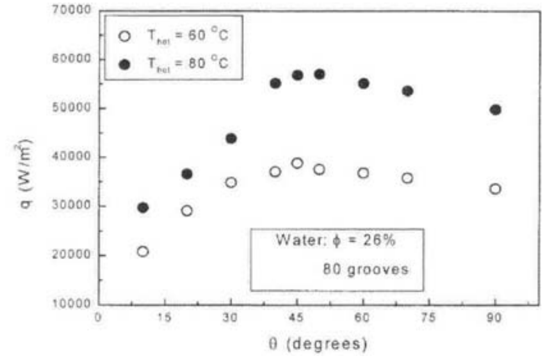


Fig. 6 Plot of condensation heat flux against inclination angle. (water, $\phi=26\%$)

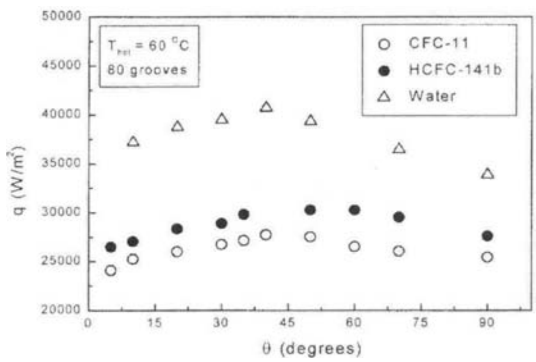


Fig. 7 Plot of condensation heat flux against inclination angle. ($T_{hot}=60℃$, 80 Grooves)

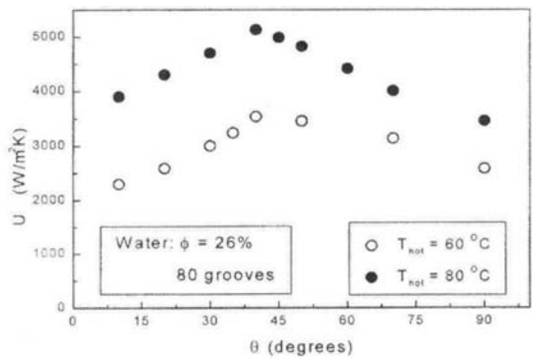


Fig. 8 Plot of overall heat transfer coefficients at the condensing section against inclination angle. ($T_{hot}=60℃$, 80℃, 80 Grooves)

gravitational force and frictional force on heat transfer coefficient is at the maximum. Negishi and Sawada (1983) explained that the heat transfer mechanism is 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 goes up along the condenser and finally reaches 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 may have a maximum value at this angle.

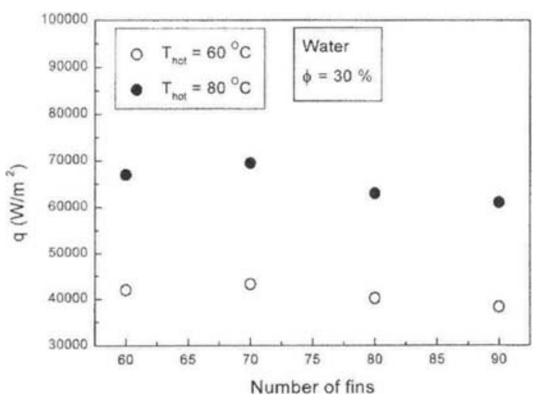


Fig. 9 Plot of condensation heat flux against grooves

4.3 Effect of fins

The relation between heat flux and fin density is shown in Fig. 9 with $\phi=30\%$. Experiments were carried out for thermosyphons with 4 axial internal fins. The fill ratio of working fluid was chosen to be 30% for water. The geometric speci-

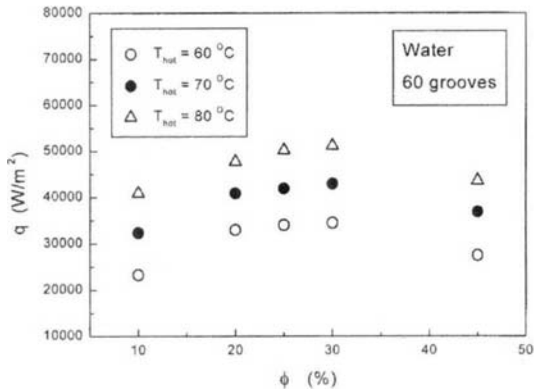


Fig. 10 Plot of condensation heat flux against charge ratio

fications of thermosyphons are given in Table 1. Heat flux has a maximum at 70 fins. It may be noted that this performance is to be attributed to both the fins and the fin depth. The best performance is obtained for 70 fins. It yields a maximum value of 69502 W/m^2K at T_{hot}=80℃.

4.4 Effect of fill quantity

The relation between heat flux and charge ratio of finned thermosyphons is shown in Fig. 10. Heat flux has a maximum value around ϕ=25～30%. The fill quantity of 0.1 is not suitable because a low heat flux resulted from the liquid film breakdown. On the other hand, for the fill quantity over 0.3, heat flux decreases because of elavation of the two-phase mixture level. For the above reasons, the optimum fill quantity of the liquid in finned thermosyphons is in the range of ϕ=25～30%.

5. Conclusions

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

(1) The experimental results indicate that heat flux or temperature difference has no effects on the ratio h_c/h_{Nu}.

(2) The best performance is obtained for 70 fins. It yields a maximum value of 69502 W/m^2K at T_{hot}=80℃.

(3) The optimum charge ratio (ϕ) of water for thermosyphons with axial internal low-fins is in the range of 25～30%.

(4) The inclination angle of a thermosyphon has a substantial influence on the condensation heat transfer coefficient and the optimum inclination angle shows between 40° and 50°.

(5) The maximum enhancement of the condensation heat transfer coefficients of thermosyphon (i.e. the ratio of overall heat transfer coefficient of finned to plain thermosyphon) is about 1.3～1.5.

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