A Comparison of the Heat Transfer Performance of Thermosyphon Using a Straight Groove and a Helical Groove

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This study is focused on the comparison of heat transfer performance of two thermosyphons having 60 straight and helical internal grooves. Distilled water has been used as working fluid. Liquid fill charge ratio defined by the ratio of working fluid volume to total internal volume of thermosyphon, the inclination angle and operating temperature were used as experimental parameters. The heat flux and heat transfer coefficient are estimated from experimental results. The conclusions of this study may be summarized as follows; Liquid fill charge ratio, inclination angle and geometric shape of grooves were very important factors for the operation of thermosyphon. The optimum liquid fill charge ratio for the best heat flux were 30%. The heat transfer performance of helically grooved tube was higher than that of straight grooved tube in low inclination angle (less than 30°), but the results were opposite in high inclination angle (more than 30°). As far as optimum inclination angle concerns, range of 25°~30° for a helically grooved tube and about 40° for a straight grooved tube are suggested angles for the best results.

Key Words : Thermosyphon, Internal Groove, Liquid Fill Charge Ratio, Heat Flux, Operating Temperature, Condensation, Evaporation

Nomenclature

A_c : Surface area on condensation section [m^2]
C_{pw} : Specific heat of cooling water [kJ/kgK]
D_i : Internal diameter [m]
h_c : Condensation heat transfer coefficient [W/m^{2}K]
h_{Nu} : Nusselt's condensation heat transfer coefficient [W/m^{2}K]
L_c : Length of condensation section [m]
$\dot{m}_c$: Mass flow rate of cooling water [kg/s]
P_s : Internal saturation pressure [Pa]
P_{atm} : Atmospheric pressure [Pa]
Q_{cool} : Heat gain of cooling water [W]
Q_{hot} : Heat loss of heating water [W]
q : Heat flux [W/m^2]
T_{we} : Surface temperature of evaporation section [K]
T_{wc} : Surface temperature of condensation section [K]
T_{sc} : Vapor temperature of condensation section [K]
T_{in} : Inlet temperature of cooling water [K]
T_{out} : Outlet temperature of cooling water [K]
$\emptyset$: Liquid fill charge ratio
θ : Iinclination angle

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1. Introduction

One of the effective management and application of energy is the usage of heat transfer by latent heat exchange caused by phase change. The typical closed heat transfer mechanisms using this principle are heat pipe and thermosyphon named as a wickless heat pipe. The heat pipe transfers large amount of heat even with a small temperature difference under boiling and evaporation of working fluid.

Gaugler (1951) first designed the heat pipe, and many other researchers conducted this kind of researches in many fields. Also, Schmidt (1994) developed the thermosyphon for the cooling of turbine blade, and its effectiveness is highly evaluated. Especially, the research of heat transfer performance of thermosyphon is conducted continuously by Cohen and Bayley (1995), Larkin (1971), Lee and Mital (1970), Stet'tsov (1975). They all insisted that the heat transfer performance of the thermosyphon is effected by the parameters such as the type of working fluid, quantity of working fluid, inclination angle, internal diameter of tubes, heat flux, working fluid pressure, geometric shape of internal surface. And better heat transfer coefficient of thermosyphon exists in the inclined position compared to vertical position because the thermosyphon is effected by a gravitational force as well as a frictional force caused by inclination angle.

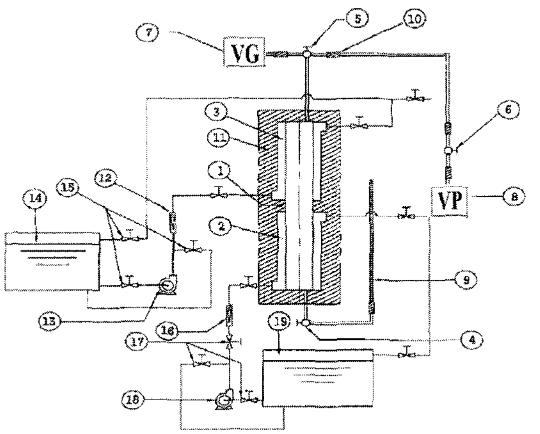
Tu et al.(1984) did an experiment using a thermosyphon with 1000 mm in length and 25 mm in diameter and water as a working fluid. His result showed that the maximum heat transfer coefficient is obtained by the inclination angle between 50° and 55°. Hahne and Gross (1981) carried out an experiment using a steel thermosyphon with 2000 mm in length, 40 mm in diameter and CFC-115 as a working fluid and came up with a result showing that the maximum heat transfer coefficient is obtained by the inclination angle between 45° and 50°. Cho and Kwon (1998) had an experiment with a thermosyphon that has an outer surface of low fin using distilled water and CFC-30 as a working fluid with the tube length of 1200 mm, diameter of 15 mm. His result said that the most heat transfer rate is obtained by the inclination angle between 20° and 45°.

Recently, Park (2001) made a thermosyphon with the tube length of 1200 mm, 15.8 mm in diameter and a plain copper tube and came up with a result that showed the maximum heat transfer rate is obtained by 30° of inclination angle. The experiments conclude that the inclination angle having maximum heat transfer coefficient varies by different conditions given to the experiment and importantly the selection of the working fluid along with the liquid fill charge ratio affects the efficiency of the thermosyphon. But these researches were mostly done by using a plain tube and only several researches were done by using grooved tubes. The usage of the triangular grooved tube was performed in Peterson and Ma's heat transfer research (1996), and Hong (1998) also performed a thermosyphon experiment using a heat pipe with an inner groove. Recently, researches for the heat transfer characteristics done by differences in number of grooves were reported by Han (2001) and Yee (2001) but more research is still necessary.

In this study, straight and helically grooved copper tube were chosen to make the thermosyphon with other different parameters that effect heat transfer performance. And the experiment is done by using the distilled water working under low temperature region. Its heat transfer performance is compared by the parameter variation of liquid fill charge ratio and inclination angle.

2. Experimental Apparatus and Method

Figure 1 shows the schematic diagram of experimental apparatus. It consists of five parts such as test section, cooling water circulation line, system, temperature measurement and recording system. The total length of test section is 1200 mm and it consists of boiling and evaporation section with 550 mm in length respectively and 100 mm of adiabatic section. The test tube has a 14.35 mm of internal diameter and a 15.85 mm of a outer diameter and material is copper. The outer surface of the tube is smooth (plain) and internal surface has grooves. Two types of internal grooved tube are used. One is straight in axial direction and the other is 18 degree of helically grooved tube from axial direction. Geometric specifications of two grooved tubes and internal surface configura-



1. Test Tube 2. Heating Water Jacket 3. Cooling Water Jacket 4. Vacuum Valve 5. Vacuum Valve 6. Vacuum Rubber Hose 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 Constant Temperature Bath 15. Coolant Control Valve 16. Heating Water Flow Meter 17. Heating Water Control Valve 18. Heating Water Pump 19. Heat Water Constant Temperature Bath

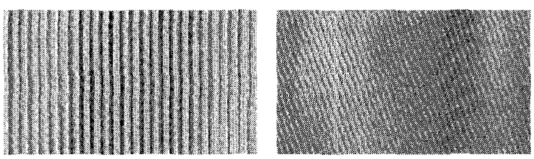
Fig. 1 Schematic diagram of the experimental apparatus

tions are shown in Table 1 and Fig. 2, respectively.

Two 550 mm long annular shaped water jackets (38 mm of internal diameter and 42 mm of outer diameter) are set on both ends of the test tube.

Table 1 Geometric specifications of two grooved thermosyphons

Classification	Tube		Groove		
	Do (mm)	Di (mm)	Skew Angle (°)	Depth (mm)	Angle (°)
Straight Groove	15.85	14.35	0	0.29	42
Helical Groove	15.85	14.35	18	0.29	42



(a) Straight groove (b) Helical groove

Fig. 2 Internal surface configuration of two grooved tubes

One is used as a heating jacket for an evaporator and the other is used as a cooling jacket for a condenser. Outside of thermosyphon is perfectly insulated with 50 mm of glass wool insulator. The thermosyphon can be positioned with any inclination angle from 0° to 90° with respect to the horizontal position.

Nine thermocouples (T type Cu-CuNi) are soldered on the outside surface of the tube along its length to measure surface temperatures. Another nine thermocouples (K type NiCr-Ni) are inserted into the inside of thermosyphon to measure inside vapor temperatures. Four more Pt 100 temperature sensors are placed at the inlets and the outlets of two water jackets. The location of thermocouples is shown in Fig. 3. The temperature outputs are recorded on a data logger and it is connected to personal computer to analyze recorded data.

A rotary vacuum pump is used first and a diffusion pump with a rating of 10^{-6} Torr is used to remove air and other non-condensable gases. Distilled water is chosen as working fluid. The working fluid is injected into the tube after evacuating air. After injecting the working fluid, heating and cooling water flow into the evaporator and the condenser jackets, respectively. A small amount of non-condensable gas was col-

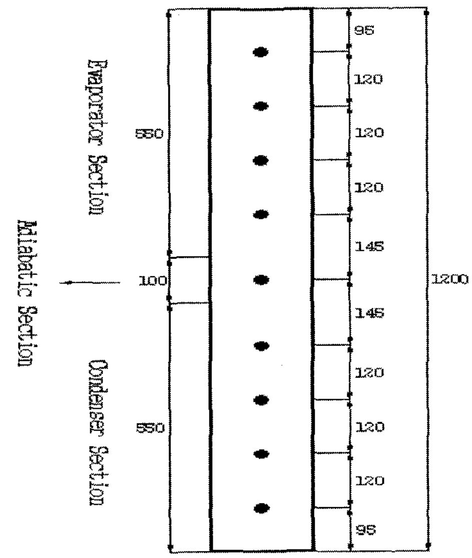


Fig. 3 Location of thermocouples

lected at the end of the condensation section after a few minutes of operation. This gas is removed again by vacuum pump for perfect operation of thermosyphon. All of the residual working fluid in the test tube was collected after each run of experiment and its volume was measured to compare the exact quantity of working fluid. Only a data with no volumetric difference of working fluid before and after each test was taken.

The experimental conditions are set as follows. The heating and cooling water temperature are varied. Two constant water temperature baths support hot or cool water continuously within ±0.2℃ difference of each setting temperature. The capacity of constant water temperature bath for heating water is 100 liter and that for cooling water is 130 liter and temperature control range is 0℃ to 100℃.

3. Resusts and Discusson

3.1 Equilibrium of energy for thermosyphon

Preliminary test has been done for the test of energy equilibrium on the evaporation and condensation zone of the test section. All of the data is obtained after steady state condition is reached. Energy balance for the heat loss of heating water and the heat gain of cooling water is well maintained within 10% of error as represented in Fig. 4. Heat loss and heat gain for both sections are calculated using eq. (1) and eq. (2), respectively.

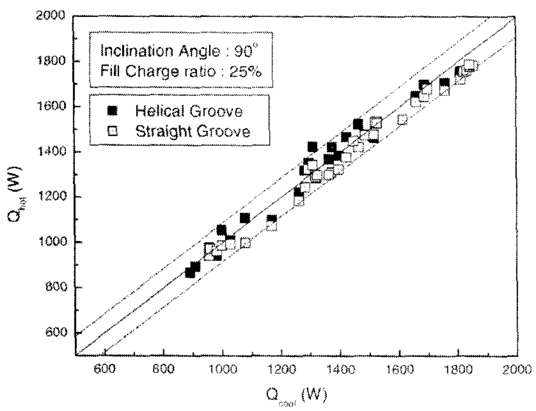


Fig. 4 Heat balance of two thermosyphons

$$Q_{cool} = \dot{m}_{cool} C_{p,cool} (T_{out} - T_{in})_{avg} \tag{1}$$

$$Q_{hot} = \dot{m}_{hot} C_{p,hot} (T_{in} - T_{out})_{avg} \tag{2}$$

Where, $\dot{m}_{cool}$, $\dot{m}_{hot}$ shows the mass flow rate of cooling water and heating water and T_{in} and T_{out} represent the inlet and outlet water temperature.

3.2 Temperature distribution of outside wall of thermosyphon

Figure 5 through Fig. 7 show the temperature distributions along the length of two thermosyphons using straight and helically grooved tube at the 80℃ of operating temperature. Fig. 5 represents the profile based on the data of 90° of inclination angle and 25% of fill charge ratio. Temperature distribution of straight grooved tube moves close to the adiabatic temperature compared to that of helically grooved tube. This means that more saturated working fluid flows to the evaporation section from condensation section because of the lower frictional force in the straight groove compared to that in the helical groove.

Figures 6 and 7 show the temperature distribution with 30% of fill charge ratio and 4 cases of inclination angle (10°, 30°, 40°, 70°) for two types of thermosyphon (straight and helical). Highest heat transfer performance is 30° of inclination angle for helically grooved tube and 40°

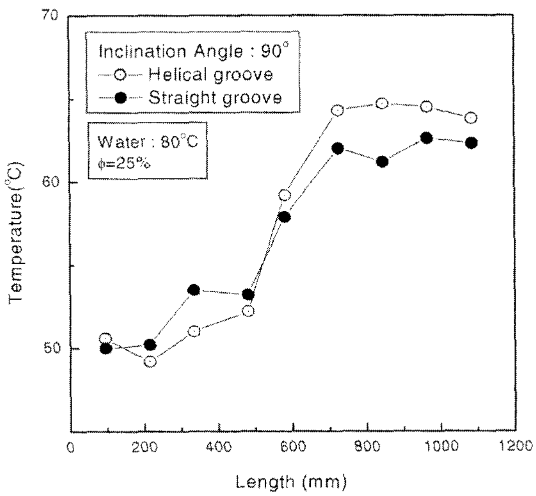


Fig. 5 Temperature distribution along the length of two grooved thermosyphons

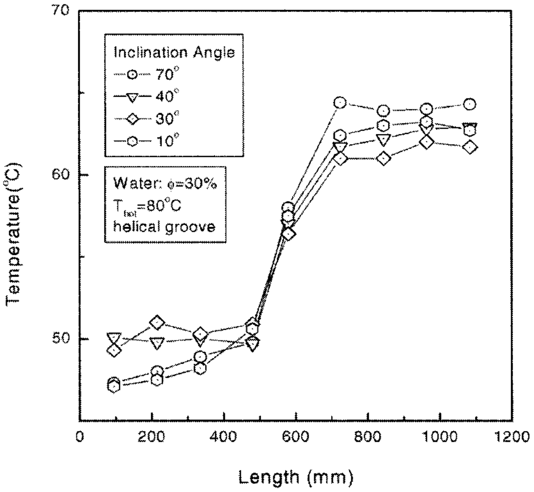


Fig. 6 Temperature distribution along the length of helical grooved thermosyphon

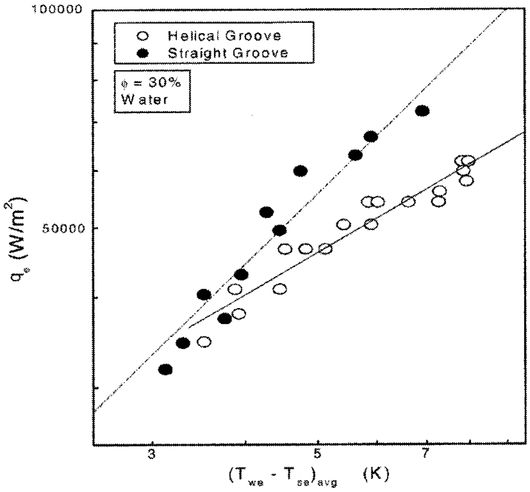


Fig. 8 Plot of boiling heat flux vs. excess temperature

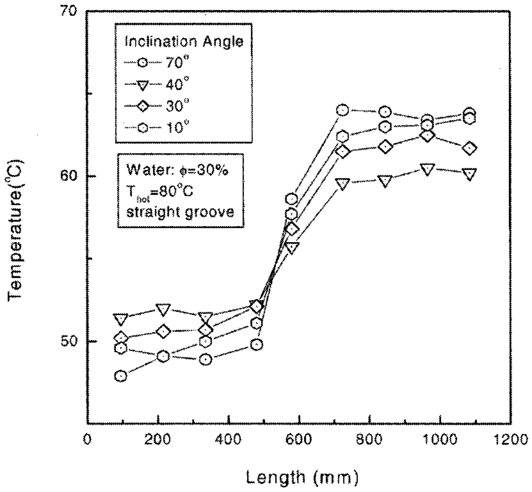


Fig. 7 Temperature distribution along the length of straight grooved thermosyphon

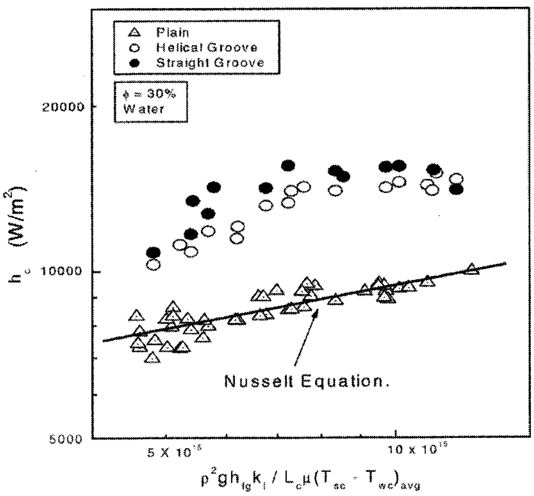


Fig. 9 Comparison of the experimental data with Nusselt's equation

for straight grooved tube. More temperature fluctuation in the condensation section compared to evaporation section is caused by the turbulence wave of the interface of condensing liquid flowing downward on the internal surface and evaporating vapor flowing upward through the core of the tube.

3.3 Heat flux and heat transfer coefficient

Figure 8 is showing the variation of the boiling heat flux with respect to the temperature difference of average internal wall temperature and average saturated vapor temperature on the evaporation section. The fill charge ratio is 30% and inclination angle is 90° in this case. The boiling heat flux increases as the excess temperature increases. Heat transfer performance for the straight grooved tube is higher than that for helically grooved tube in all of the range and it can be justified by the slope difference of two lines.

Figure 9 shows the condensation heat transfer coefficient for grooved tubes and plain tube at the 30% of fill charge ratio. Nusselt's equation matches well with the data of plain tube. The coefficient for the grooved tube is higher than that of plain tube in all of the experimental range. The condensation heat transfer coefficient is calculated by the eq. (3). The eq. (4) represents Nusselt's equation of condensation theory.

$$h_c = \frac{(Q_{cool} + Q_{hot})}{A_o (T_{wc} - T_{sc})_{ava}} \tag{3}$$

$$h_c = 0.943 \left[\frac{\rho_l^2 g h_{hg} k_l^3}{L_c \mu_l (T_{sc} - T_{wc})} \right]^{1/4} \tag{4}$$

3.4 Heat transfer rate as the function of inclination angle and fill charge ratio

Figures 10 and 11 show the variation of condensation heat flux against inclination angle for two types of grooved tubes. Data is obtained using two heating water temperature of 60° C and 80° C. As shown in two Figures, between 35° and 40° of inclination angle is the approximate region at which the trend of heat transfer rate is changing. This means that the heat transfer rate in the case of helically grooved tube has higher value at lower than these range of angle and the value of heat transfer rate is reversed after these range of angle. The highest condensation heat transfer rate is obtained at about 30° of inclination angle for the case of helically grooved tube and at about 40° in the case of straight grooved tube.

Figure 12 shows the condensation heat flux against the fill charge ratio. Five cases of fill charge ratio (10%, 20%, 25%, 30%, 40%) are used. Heat flux is low at 10% of fill charge ratio because of the dry out of working fluid. And heat flux is increasing until the fill charge ratio reaches to about 30%.

The heat flux is decreasing after 30% of ratio. This is considered that if the volume of working

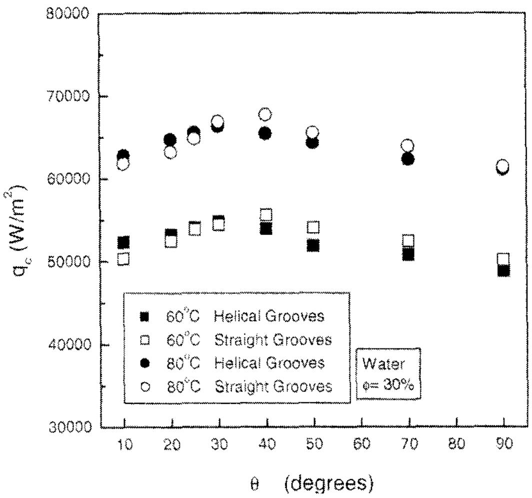


Fig. 11 Plot of Heat Flux against inclination angle (∅=30%)

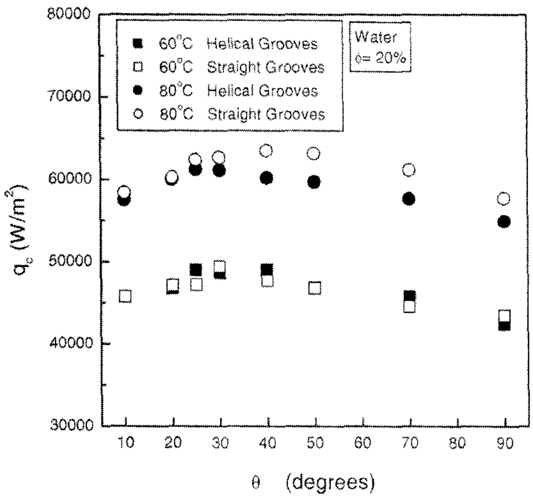


Fig. 10 Plot of Heat Flux against inclination angle (∅=20%)

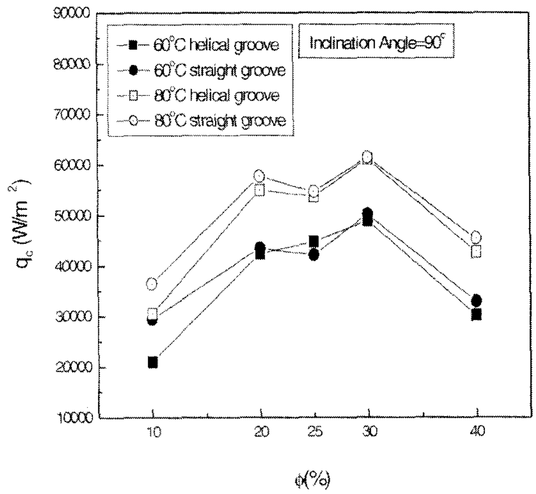


Fig. 12 Plot of heat flux against fill charge ratio

fluid increases to the certain limit, two phase flow mixed with liquid and vapor reaching to the condensation section results in decreasing the condensation heat transfer performance.

4. Conclusions

The experimental results are summarized as follows.

(1) The thermosyphon using grooved tube has the higher heat transfer rate compared to the thermosyphon using plain tube. This is well verified by using Nusselt's condensation theory.

(2) The heat transfer performance is tested by changing of inclination angle between 10° and 90° with respect to the horizontal position. The highest result is obtained at about 25°~30° in case of helically grooved tube and at about 40° in case of straight grooved tube.

(3) The highest heat flux is obtained at 30% of fill charge ratio regardless of the types of grooved tube.

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