HEAT TRANSFER PERFORMANCE OF A TWO-PHASE CLOSED THERMOSYPHON

أداء انتقال الحرارة لسيفون حرارى مغلق ذى طورين

M. A. Shalaby, F. F. Araid, G. I. Sultan, and M. M. Awad

Mech. Power Eng. Department, Faculty of Engineering
Mansoura University, Egypt, P. O. 35516
E-mail: gisultan@mum.mans.eun.eg

خلاصـــــة:

يقدم هذا البحث دراسة معملية لأداء سيفون حرارى مغلق ذى طورين يعمل بوسيط تبريد R22 . السيفون الحرارى مصنوع من أنبوبة نحاسية ملساء بقطر داخلى ٢١ مم وطول ١٥٠٠ مم، وطول كل من المبخر والمكثف ٦٠٠ مم بينما طول الجزء الادياباتيكى ٣٠٠ مم. أثناء التجارب تم تغيير معدل انتقال الحرارة من ١٠٠ إلى ٣٠٠ وات ونسبة مئوية لامتلاء المبخر من ٣٠% إلى ١٠٠% وزاوية الميل من ٩٠° إلى ٢٢,٥° مقاسة من الوضع الإفقى. وقد استنتج من البحث أن أحسن أداء للسيفون الحرارى عندما تكون نسبة امتلاء المبخر ٥٠% وزاوية ميل ٣٠° مقاسه من الوضع الأفقي. وقد تم استنتاج صيغة لا بعدية لحساب النسبة بين معامل انتقال الحرارة الكلى عند أى زاوية ومعامل انتقال الحرارة الكلى عند الوضع الراسى كدالة فى رقم كوتاتلادزى ونسبة الامتلاء وزاوية الميل ونسبة انخفاض الضغط.

ABSTRACT

An experimental investigation has been made in order to study the heat transfer performance of a two-phase closed thermosyphon tube. A smooth copper tube of total length 1500 mm and 21 mm inside diameter is used as a container of the thermosyphon. Each of the evaporator section and the condenser section has a length of 600 mm, while the remaining part of the thermosyphon tube is the adiabatic section. The working fluid used is R22. The effects of heat transfer rate (100 W $\leq Q \leq$ 300 W), filling percentage (30%$\leq V^{+} \leq$100%) and the inclination angle of the thermosyphon ($22.5^{\circ} \leq \theta \leq 90^{\circ}$) are studied. It is found that, the optimum filling percentage and the best inclination angle values are 50% and 30°, respectively. The overall heat transfer coefficient ratio (U/U_{90}) has been correlated in a dimensionless form as a function of Kutateladze number, filling ratio, inclination angle, and reduced pressure ratio.

INTRODUCTION

A closed thermosyphon is a passive device, which is used to transfer a large amount of heat rate, with small temperature difference, by evaporation of the working fluid through the evaporator section and condensing it in the condenser section. In practice, the effective thermal conductivity of thermosyphon (k_{eff}), exceeds that of copper 500 times [9]. The thermosyphon is widely used in many energy and industrial applications. Examples include heat exchangers, solar collectors, cooling of electronic equipment, systems for the prevention of ice formation on roads, and cooling of gas turbine rotor blades, preservation of permafrost in arctic regions, and extraction of geothermal energy.

So, a considerable experimental and theoretical work had been done on the application and design modifications for improving thermosyphons performance. The heat transfer

Accepted June 20, 2000

performance of a two-phase closed thermosyphon together with a simple theoretical analysis for its maximum heat transfer capacity were studied by Lee and Mital [1]. They used water and R11 as working fluids. The heat transfer in a two-phase closed thermosyphon with water and ethanol as working fluids were studied by Imura et al. [2]. They found that the optimum quantity of working fluid was 10-20 percent of the tube inside volume and established an empirical formula for the heat transfer coefficient in the evaporator section for a thermosyphon installed in a vertical position. Nguyen-Chi et al. [3] investigated experimentally the performance of vertical closed two-phase thermosyphons and used water as a working fluid. They investigated the influence of operating parameters on the maximum performance either by dry-out or burnout limits. Nguyen-Chi and Gross [4] investigated experimentally the influence of the surface roughness on the maximum thermosyphon performance and used water as a working fluid. The optimum tilt angle was found to be of the order of 40° to 60°, and the optimum-filling percentage was of the order of 40 % to 70 %, depending on operating conditions. Shiraishi et al.[5] studied experimentally the heat transfer characteristics of a two-phase closed thermosyphon and developed a simple mathematical model to predict the performance of such thermosyphons. They used water, ethanol and R113 as the working fluids. Hahne and Gross [6] performed an experimental investigation in order to observe the effect of the inclination angle on the transport behavior of a closed two-phase thermosyphon. The working fluid used was R115. They concluded that the highest heat transport rate exhibited at an inclination angle around 40°. Negishi and Sawada [7] made an experimental study on the heat transfer performance of an inclined two-phase closed thermosyphon. They used water and ethanol as working fluids. They found that for high heat transfer rates, it was necessary to fill between 25 to 60% of the evaporator volume with water as a working fluid, between 40 to 75% for ethanol and the inclination angle must be between 20° and 40° for water, and more than 5° for ethanol. Gross and Hahne [8] studied experimentally the effects of pressure, heat flow rate and inclination angle on the internal heat transport near critical state. They used R115 as a working fluid and found that the optimum inclination angle was at about 40°. In experimental work [9], the effects of pressure and inclination angle on the internal heat transport over a wide range were studied. They used R13B1 as a working fluid and found that the evaporation heat transfer coefficient was strongly affected by pressure and inclination angle. The condensation heat transfer coefficient depended both on pressure and inclination. Terdtoon et al. [10] described the effect of the fluid's property on the heat transfer characteristics of the inclined thermosyphon by taking into account the filling ratio. With the experiments taken very near to the boiling point, the ratio of Q_{cr} at any inclination angle to the Q_{cr} at 90° was determined as a function of the inclination angle. They found that the ratio of $Q_{cr}/Q_{cr,90}$ at any inclination angle was higher than that for the vertical position. Negishi et al. [11] studied the influences of the filling ratio and inclination angle on the heat transfer performance of a corrugated tube thermosyphon. They found that the optimum-filling percentage was found to be 40 %. The maximum performance was obtained at an inclination angle of 30°. Park and Lee [12] made an experimental study on the performance of stationary two-phase closed thermosyphons with three working fluid mixtures (water-glycerin, water-ethanol, and water- ethylene glycol). The amount of the working fluid of any mixture percentage was about 20% of the total test tube by volume. They found that for all values of filling ratio, the tested thermosyphon had a highest performance at an angle of inclination of about 60° from the direction of the gravitational force. Gunnerson and Zuo [13] studied theoretically the heat transfer characteristics and constraints of an inclined two-phase thermosyphon. The effects of the inclination angle, induced secondary circumferential liquid flow, minimum working fluid, flooding limit maximum heat transfer rate and mean heat transfer coefficient of the thermosyphon were studied. They found that the highest flooding limit was at the inclination angle ranging from 30° to 45°. Shiraishi et al.[14] conducted a flow visualization study of the inside flow phenomena of an inclined two-phase closed thermosyphon at several angles. They used R 113 as a working fluid. They found that the basic flow patterns inside an inclined thermosyphon were classified as an annular flow and a stratified flow. An annular flow could be observed at the vertical position, while a stratified flow could be seen at other angles. Terdtoon et al. [15] investigated the effect of aspect ratio (ratio of evaporator section length to diameter) and Bond number on the heat transfer characteristics of an inclined two-phase closed thermosyphon. They used R22, ethanol, and water as working fluids, and a filling percentage of 80%. They found that the aspect ratio and Bond number did not affect the angle at which the highest heat transfer rate occurred. The heat transfer in a vertical annular two-phase closed thermosyphon has been studied experimentally with distilled water as a working fluid by Abdel-Aziz [16]. The effects of heat flux, liquid fill charge, and evaporator to condenser length ratio on the overall heat transfer coefficient, was investigated. He showed that the maximum overall heat transfer coefficient occurred at an evaporator to condenser length ratio ranged between 0.33 to 1.0, and the liquid fill charge was about 16% which based on the total thermosyphon inside volume. A correlation between the above three parameters was suggested to calculate the overall heat transfer coefficient. The effect of thermosyphon inclination angle is not taken into consideration. Hussein et al.[17] studied theoretically and experimentally the thermosyphon flat-plate solar collector under transient conditions. The governing equations of the different parameters of the collector are presented and generalized in dimensionless forms. The finite difference technique is used to solve the set of dimensionless governing equations by means of simulation program. For verifying the simulation program, a thermosyphon flat –plate collector is designed, constructed, and tested at transient conditions, different mass flow rates of cooling water, and different inlet cooling water temperatures. The experimental results are compared with their corresponding simulated ones. The comparison showed a considerable agreement between the experimental and simulated results. In addition, the simulation model proved to be an effective tool for the design of a thermosyphon flat-plate solar collector and the prediction of its transient thermal behavior.

From the available literature review, one may observe that a limited number of working fluids have been used and some of these working fluids are rarely used, such as R22. In addition to that, the parameters suggested to have some effects on the performance of the two-phase closed thermosyphon still need more work in order to satisfy a quite acceptable agreement of the results.

In this work, the performance of the two-phase closed thermosyphon is studied with R22 as a working fluid, and the heat transfer rates are 100 W, 200 W, and 300 W. The filling percentage used are 100%, 80%, 60%, 50%, 40%, and 30%. The inclination angle existed are 90° (vertical position), 60°, 45°, 37°, 30°, and 22.5° measured from the horizontal level.

EXPERIMENTAL APPARATUS

The schematic diagrams of the experimental test rig and test section are shown in Fig. 1 and Fig. 2, respectively. The thermosyphon (1) is made of a smooth copper tube, of inside diameter 21 mm, 2 mm thickness, and a total length of 1500 mm. The length of the evaporator section is 600 mm, adiabatic section is 300 mm, and condenser section is 600 mm. R22 is used as a working fluid (22). A prespix glass tube "water jacket" (5) surrounds the he condenser section, and the cooling water is flowing through the annular passage. The cooling water enters from the bottom section in tangential direction to the inside surface of the jacket so as to prevent the thermosyphon from direct exposure to the water flow. A constant head tank (9) is placed 2 m above the head of the thermosyphon tube and connected to the cooling section through a plastic tube (10). The make-up water provides the constant head tank by a float valve (8) to ensure that the flow of water is nearly steady. Also, an over flow tube (6) allows the exceeding water to flow to the drain. The flow rate of cooling water is controlled by a valve (7) and is measured by a rotameter (11).

The main heater (24) of evaporator section is made of a nickel-chrome wire. The main heater is wound around the evaporator section at equal pitches. Before winding the main heater, the bare smooth copper tube is covered by an electrical insulating tape (23). After winding the main heater, it is covered by an electrical insulating tape again followed by a semi cylindrical glass wool insulation of 4.0 cm thick (25). The guard heater (26) is used to eliminate the heat loss from the main heater to the environment, so the power input to the main heater is converted to heat and transferred completely to the evaporator section. The adiabatic section (21) is covered by a semi-cylindrical glass wool insulation.

For measuring the outside wall temperatures of the thermosyphon, 10 copper-constantan thermocouples (T-Type) of 1-mm diameter are used, 4 for the evaporator section, 3 for the adiabatic section, and 3 for the condenser section. The thermocouples are embedded in slots and glued by a high conductive epoxy. All thermocouples are connected to a digital temperature recorder (16), which has an inaccuracy of $\pm 0.1^{\circ}C$. Two autotransformers (14) are used to control the power input to both main heater and guard heater individually. The power input to both main and guard heaters are calculated from the values measured by two analogue ammeters (12), and two analogue voltmeters (13) with an inaccuracy of ± 0.01 A, and ± 0.1 V respectively. A stabilizer (15) is connected with the two autotransformers to ensure that the input power fluctuations during the experiment are negligible. The thermosyphon is fitted to a vertical holder (4) by a special clamp and can be turned to have different angles of inclination in a vertical plane. A pointer (2) and a protractor (3) are used to indicate the inclination angle.

CALCULATION PROCEDURE

In this research, the overall heat transfer coefficients of the thermosyphon (U), the convective heat transfer coefficient of both evaporator (h_e), and condenser sections (h_c), respectively [12], are defined as the following:

$$U = q/(T_{e,ave} - T_{c,ave}) = 1/(1/h_e + L^+/h_c) \quad (1)$$
$$h_e = q/(T_{e,ave} - T_{a,ave}) \quad (2)$$
$$h_c = q\, L^+/(T_{a,ave} - T_{c,ave}) \quad (3)$$

The heat transfer coefficients h_e and h_c are calculated using $T_{a,ave}$ instead of T_{op} as used in equations (2) and (3) because the value of T_{op} is more or less, equal $T_{a,ave}$ as in [18] when $L^+=1$.

RESULTS AND DISCUSSION

Experimental runs are performed to study the effect of heat transfer rate Q, filling percentage V^+(volume of the working fluid to the evaporator volume), and the inclination angle θ, on the performance of low temperature two-phase closed thermosyphon.

Temperature Distribution along the Thermosyphon

Figure 3 shows the temperature distribution along the thermosyphon at a filling percentage of $V^+ = 50\%$ in a vertical position at different heat transfer rates. It is clear that the local temperature in the condenser increases with position while in the adiabatic section decreases with position. The local temperature in the evaporator is nearly uniform. Also, the local temperature increases with increasing the heat transfer rate for all positions.

Figure 4 shows the temperature distribution along the thermosyphon at a heat transfer rate of Q = 200W in a vertical position at different filling ratios. It is clear that the local temperature in the condenser increases with position while in the adiabatic section decreases with position. The local temperature in the evaporator is nearly uniform (±3% deviation at mean value).

Figure 5 shows the variation of local temperature distribution along the thermosyphon at a heat transfer rate of Q = 200 W, and a filling percentage of 50% and at different inclination angles. It is clear that the local temperature al ong the evaporator section increases with length to a certain value and then decreases. The adiabatic section shows that, the local temperature decreases with the length, meanwhile in the condenser section, the local temperature increases with length. It is also seen that the local temperature along the evaporator section is decreased with the decrease of the inclination angle. On the other side, in the condenser section, the local temperature distribution increases with the decrease of the inclination angle, but at the highest point on the condenser, the local temperature decreases with the decrease of inclination angle. While in the adiabatic section, the local temperature is more or less, equal constant with decreasing the inclination angle.

Effect of Heat Transfer Rate on the Temperature

Figure 6 shows the effect of heat transfer rate on the average temperature of both evaporator and condenser at a filling percentage of 50% in a vertical position. It is clear that with increasing the heat transfer rate, the average temperature of both evaporator and condenser increases. Also, the evaporator-condenser average temperature difference increases with increasing the heat transfer rate.

Effect of the Operating Temperature

Figure 7 shows the relation between the overall heat transfer coefficient (U) and operating temperature (T_{op}) (which is the saturation temperature corresponding to the operating pressure) at a filling percentage of 50%, and the inclination angle $\theta = 90^\circ$. The overall heat transfer coefficient varies from 854 to 998.5 $W/m^2.^\circ C$ for the operating temperature between 30.5°C and 36.9°C. The overall heat transfer coefficient increases with increasing the operating temperature. This is because the boiling heat transfer coefficient increases with pressure.

Effect of the Inclination Angle

Figure 8 shows the dependence of the overall heat transfer coefficient (U) on the inclination angle (θ) at a filling percentage 50% and different heat transfer rates. It is clear that the overall heat transfer coefficient increases with increasing the inclination angle having its maximum value of 1104 W/m^{2}C, which corresponds to an inclination angle of 30° "measured from horizontal position" and Q = 300 W. At inclination angles "greater than 30°", the overall heat transfer coefficient decreases. This is because the difference in the potential energy between the condenser and the evaporator is largest and the flying-up distance of the fluid along the tube axis caused by the expansion of the boiling bubbles of the same energy level becomes smaller. Also, in a vertical tube, the film flows downward in axial direction with an approximately constant film thickness around the circumference but in an inclined tube, the condensate flows along a curved path to the lower part of the cross-section, where it accumulates and flows back to the evaporator. For small inclinations ($\theta < 30°$), the reduction of axial gravity component is minimum (due to the behavior of the sine function) but the secondary circumferential flow can provide a large relief for the returning liquid film flow. Even when the topside liquid film is retarded, the liquid can still migrate to the underside via the secondary circumferential flow, which makes the underside film thickness (correspondingly the gravitational force) large enough to overcome the shear forces. Also as the thermosyphon is tilted, the flooding limit at first increased due to the effect of secondary circumferential liquid flow. However, for large inclinations, the reduction of axial gravity component is significant and the flooding limit steeply drops. Also, the overall heat transfer coefficient increases with increasing the heat transfer rate. This is because with increasing heat flux, more bubbles generate and flying-up to the surface where it is converted to vapor. The growth and movement of these bubbles tends to increase the overall heat transfer coefficient.

Effect of the Filling Ratio

Figure 9 shows the variation of the overall heat transfer coefficient (U) with the filling percentage (V^+) at an inclination angle $\theta = 30°$ "optimum inclination angle", and different heat transfer rates. It is obvious that the overall heat transfer coefficient increases with increasing the filling ratio and reaches a maximum value of 1104 W/m^2.°C which corresponds to Q = 300 W at a filling percentage of 50% after which it begins to decrease again. Also, the overall heat transfer coefficient increases with increasing the heat transfer rate.

Correlating the Experimental Data

The present experimental data (108 points) are correlated in a dimensionless form, to represent the overall heat transfer coefficient ratio (U/U_{90}) as a function of Kutateladze number (Ku) which represents ratio of the heat flux to the critical heat flux, filling ratio (V^+), inclination angle (θ), and reduced pressure (P/P_{cr}). By the mathematical statistical analysis method (least square method), the present experimental data are correlated as:

$$\frac{U}{U_{90}} = \begin{cases} [1.012(Ku)^{-0.015}(V^+)^{0.009}(\sin\theta)^{-0.099}(P/P_{cr})^{0.071}], & \text{for } V^+ > V^+_{opt} \\ [1.004(Ku)^{-0.033}(V^+)^{-0.021}(\sin\theta)^{-0.101}(P/P_{cr})^{0.167}], & \text{for } V^+ \leq V^+_{opt} \end{cases} \quad (4)$$

Where $2526.3 \leq q \leq 7578.8$, $0.3 \leq V^+ \leq 1.0$, $22.5° \leq \theta \leq 90°$, and $0.23 \leq P/P_{cr} \leq 0.3$

Figures 10 and 11 show the deviation between present experimental data and the present correlation when $V^{+}>V^{+}_{opt}$ and $V^{+}\leq V^{+}_{opt}$, respectively. The maximum deviation between the present experimental data and the present correlation is ± 4.5% and ± 5% respectively.

CONCLUSION

Study of the heat transfer performance of a two-phase closed thermosyphon tube with R22 as a working fluid is presented. This study is concerned to investigate the effect of heat transfer rate, filling percentage, and angle of inclination. The main results can be summarized as follows:

- Increasing heat transfer rate, the thermosyphon wall temperatures, the operating temperature, and the evaporator-condenser temperature difference increase.
- Increasing operating temperature, the overall heat transfer coefficient increases.
- The optimum-filling percentage is found to be 50%.
- The maximum performance is obtained at an inclination angle of 30°.
- The obtained correlation is a helpful tool for studying closed type of thermosyphons.

NOMENCLATURE

d	diameter, m
g	gravitational acceleration, m/s^2
h	convective heat transfer coeff., W/m^2C
h_{fg}	latent heat of evaporation, J/kg
k	thermal conductivity, W/mC
L	length, m
L^+	length ratio = L_e/L_c
P	pressure, bar
q	heat flux, W/m^2
Q	heat transfer rate, W
T	temperature, °C
U	overall heat transfer coeff., $W/m^2.°C$
V^+	filling percentage

Greek Symbols

θ	inclination angle, degree
ρ	density, kg/m^3
σ	surface tension, N/m

Subscripts

a	adiabatic
ave	average
c	condenser
cr	critical
e	evaporator
eff	effective
i	inside
l	liquid
op	operating
opt	optimum
v	vapor
90	vertical position

Dimensionless Groups

Ku Kutateladze number

$$=(Q/\pi d_i L_e)/\{h_{fg}[\sigma g \rho_v^2(\rho_l-\rho_v)]^{1/4}\}$$

REFERENCES

1. Lee Y., and Mital U. (1972), A Two-Phase Closed Thermosyphon, Int. J. Heat Mass Transfer, Vol.15, pp. 1695-1707.
2. Imura H., Kusuda H., Ogata J., Miyazaki T., and Sakamoto N. (1979), Heat Transfer in a Two-Phase Closed Thermosyphon, Heat Transfer Japanese Research, Vol.8, No.2, pp.41-57.
3. Nguyen-Chi H., Groll M., and Dang-Van Th. (1979), Experimental Investigation of

Closed Two-Phase Thermosyphons, AIAA 14th Thermophysics Conference June 4 - 6, Orlando, Florida, pp.239-246.

4. Nguyen-Chi H., and Groll M. (1980), The Influence of Wall Roughness of Closed Two-Phase Thermosyphon, AIAA 15th Thermophysics Conference July 14-16, Snowmass, Colorado, pp.259-267.
5. Shiraishi M., Kikuchi K., and Yamanishi T. (1981), Investigation of Heat Transfer Characteristics of a Two-Phase Closed Thermosyphon, Proceedings of the 4th International Heat Pipe Conference, London, UK, pp. 95-104.
6. Hahne E., and Gross U. (1981) The Influence of the Inclination Angle on the Performance of a Closed Two-Phase Thermosyphon, Proceedings of the 4th International Heat Pipe Conference, London, UK, pp.125-136.
7. Negishi K., and Sawada T. (1983), Heat Transfer Performance on an Inclined Two-Phase Closed Thermosyphon, Int. J. Heat Mass Transfer, Vol.26, No.8, pp.1207-1213.
8. Gross U., and Hahne E. (1985), Heat Transfer in a Two-Phase Thermosyphon Operating with a Fluid in the Near Critical State, Int. J. Heat Mass Transfer, Vol.28, pp.589-601.
9. Gross U., and Hahne E. (1986), Heat Transfer in a Closed Thermosyphon Over a Wide Range Pressures and Inclinations, Ger. Chem. Eng. 9, pp.292-299.
10. Terdtoon P., Shiraishi M., and Murakami M. (1990), Investigation of the Effect of Inclination Angle on Heat Transfer Characteristics of Closed Two-Phase Thermosyphon, Proceedings of 7th International Heat Pipe Conference, Minsk, USSR, pp.517-524.
11. Negishi K., Kaneko K., Matsuoka T., Hirashima M., Nishikawa Y., and Taguchi M. (1991), Heat Transfer Performance of a Corrugated-Tube Thermosyphon. Part 1. Evaporator Performance, Heat Transfer Japanese Research, Vol.20, No.2, pp.144-157.
12. Park R. J., and Lee, Y. (1992), Two-Phase Closed Thermosyphons With Two Fluid Mixture, Proceedings of the 8th International Heat Pipe Conference, Beijing, China, Vol.2, pp.220-225.
13. Gunnerson F. S., and Zuo Z. J. (1995), Modeling of an Inclined Two-Phase Closed Thermosyphon, ASME / JSME Thermal Engineering Conference, Vol.2, pp. 27-34.
14. Shiraishi M., Terdtoon P., and Murakami M. (1995), Visual Study on Flow Behavior in an Inclined Two-Phase Closed Thermosyphon, Heat Transfer Eng., Vol.16, No.1, pp.53-59.
15. Terdtoon P., Ritthidej S., and Shiraishi M. (1996), Effect of Aspect Ratio and Bond Number on Heat Transfer Characteristics of an Inclined Two-Phase Closed Thermosyphon at Normal Operating Condition, Proceedings of the 5th International Heat Pipe Symposium, Melbourn, Australia, pp.261-266.
16. Abdel-Aziz M. (1996), Experimental Analysis of the Optimum Performance of Closed Vertical Thermosyphon Based on Geometry and Liquid Fill Rate Considerations, The 9th Int. Conf. For Mech.Power Eng.,(ICMPE.9), Shebin El-Kom, Egypt, December, 21-24.
17. Hussein H.M.S., Mohamed M.A., and El-Asfouri A.S.(1999), Transient Investigation of a Thermosyphon Flat-Plate Solar Collector, Applied Thermal Eng., 19, pp.789-800.
18. Lee Y., and Bedression A.(1978), The Characteristics of Heat Exchangers Using Heat Pipes or Thermosyphons, Int. J. Heat and Mass Transfer, Vol.21, pp.221-229.

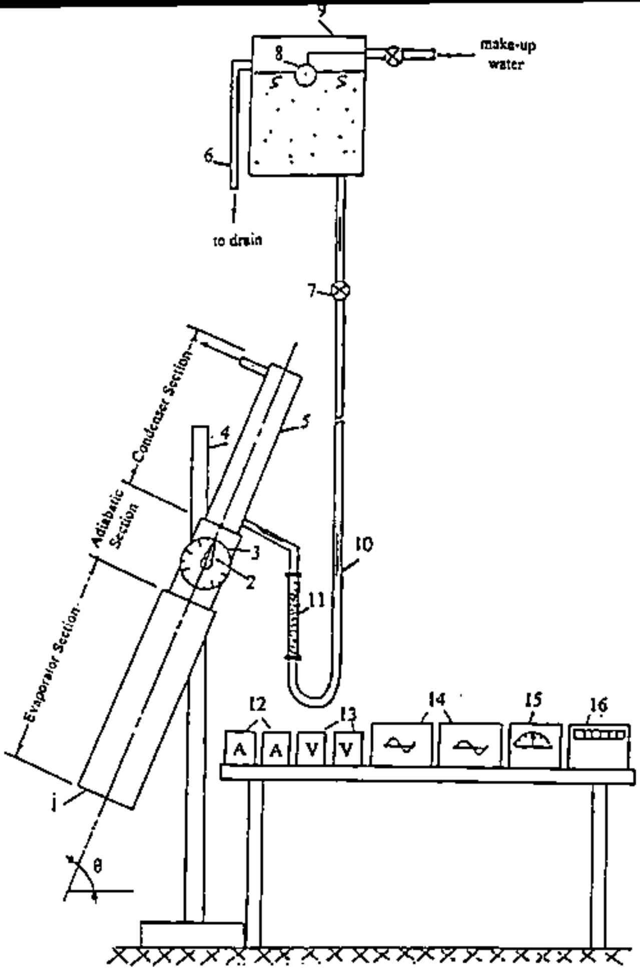

Fig. 1 The experimental test rig.

1. thermosyphon, 2. pointer, 3. protractor, 4. vertical holder, 5. water jacket, 6. over flow tube, 7. flow control valve, 8. float valve, 9. constant head tank, 10. plastic tube, 11. rotameter, 12. ammeters, 13. voltmeters, 14. autotransformers, 15. stabilizer. and 16. temperature recorder.

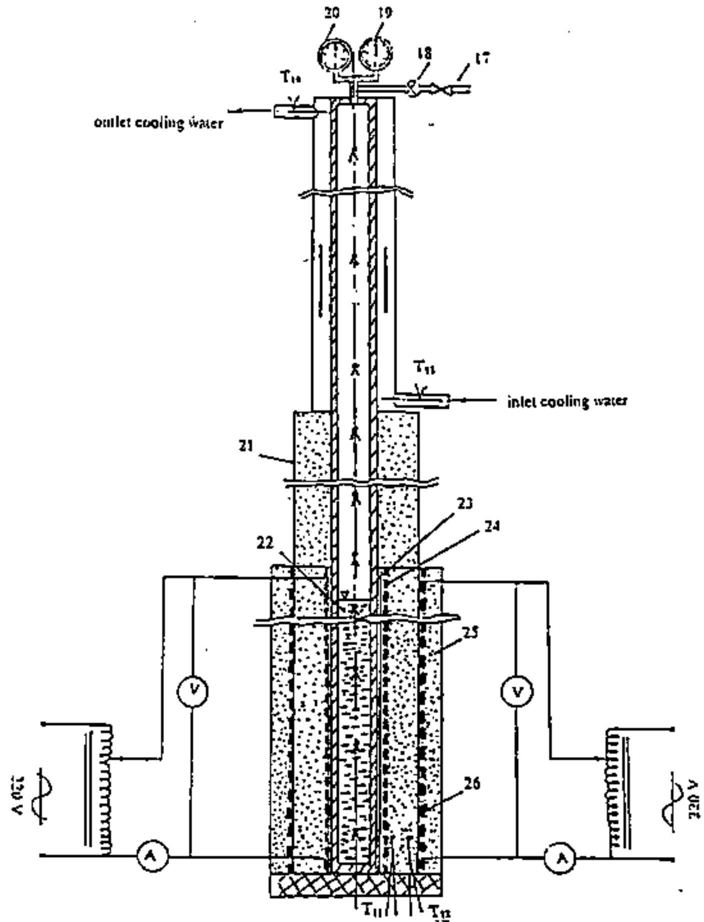

Fig. 2 The experimental test section.

17. valve, 18. shut-off valve, 19. low pressure gauge, 20. high pressure gauge, 21. adiabatic section, 22. working fluid, 23. electrical insulating tape, 24. main heater, 25. insulation, and 26. guard heater.

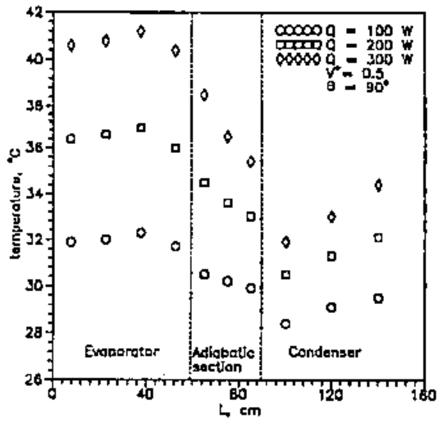

Fig. 3 The temperature distribution along the thermosyphon at constant both filling ratio and inclination angle.

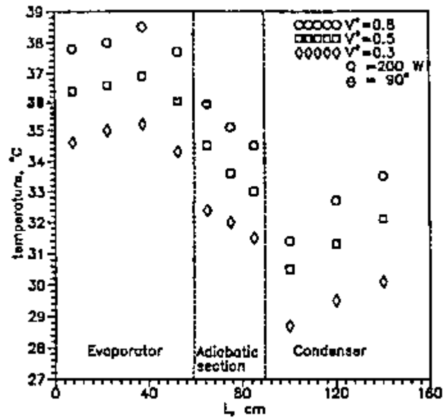

Fig. 4 The temperature distribution along the thermosyphon at constant both heat flow rate and inclination angle.

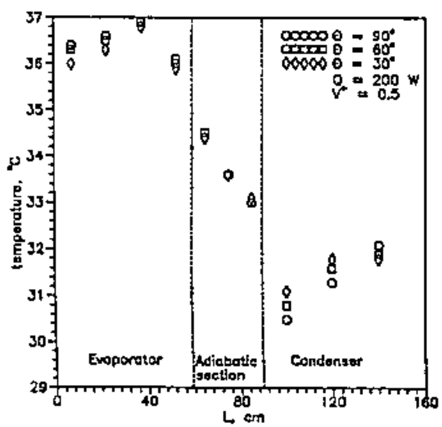

Fig. 5 The temperature distribution along the thermosyphon at constant both heat flow rate and filling ratio.

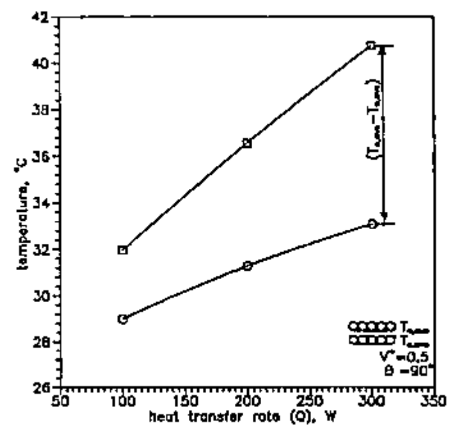

Fig. 6 Effect of heat transfer rate on the average temperature of both evaporator and condenser.

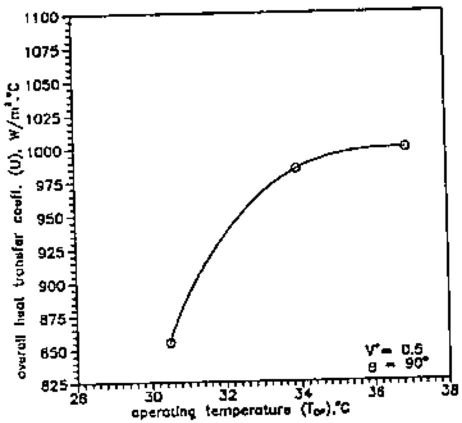

Fig. 7 Effect of thermosyphon operating temperature on the overall heat transfer coefficient.

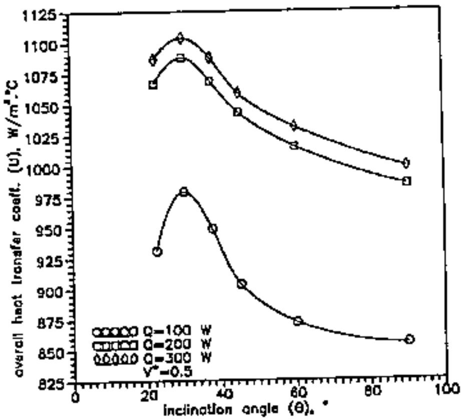

Fig. 8 Effect of thermosyphon inclination angle on the overall heat transfer coefficient.

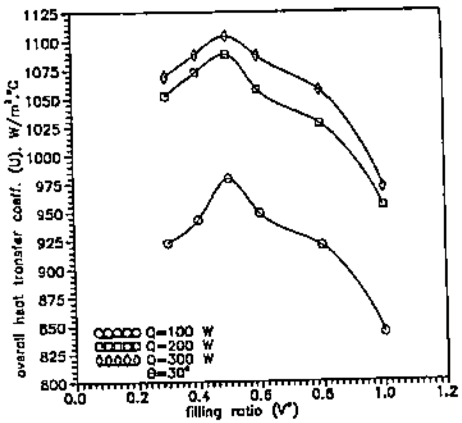

Fig. 9 Effect of thermosyphon filling ratio on the overall heat transfer coefficient.

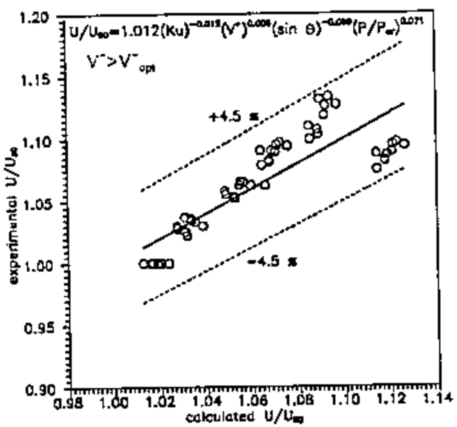

Fig. 10 Deviation between the experimental data and the present correlation.

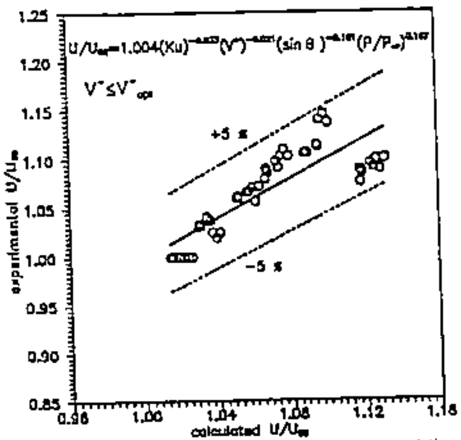

Fig. 11 Deviation between the experimental data and the present correlation.