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Heat transfer characteristics of a two-phase closed thermosyphon

S.H. Noie *

Engineering Faculty, Chemical Engineering Department, Ferdowsi University of Mashhad, Mashhad, P.O. Box 9177948944-1111, Iran

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Abstract

The applications of closed two-phase thermosyphons are increasing in heat recovery systems in many industrial practices because of their high effectiveness. Various parameters affect the heat transfer performance of thermosyphons. In this paper, the effect of three parameters: input heat transfer rates ($100 < \dot{Q} < 900\,\mathrm{W}$), the working fluid filling ratios ($30\% \leqslant \mathrm{FR} \leqslant 90\%$), and the evaporator lengths (aspect ratios) were investigated experimentally. The aspect ratios for these experiments were 7.45, 9.8, and 11.8. A series of experiments were carried out to find the influence of the above parameters on steady-state heat transfer characteristics of a vertical two-phase closed thermosyphon. A smooth copper tube of total length of 980 mm with inside and outside diameters of 25 and 32 mm was employed with distilled water as working fluid. The temperature distribution along the thermosyphon was monitored; input heat to evaporator section and output heat from condenser were measured, as well. The experimental boiling heat transfer coefficients were compared with existing correlations. Conclusions have been drawn for the optimum-filling ratio at which the thermosyphon operates at its best for a certain aspect ratio.

Keywords: Heat pipes; Filling ratio; Aspect ratio; Themosyphon; Boiling heat transfer; Thermal characteristics

* Corresponding author. Tel.: +98 915 1105515; fax: +98 511 8612295.
E-mail address: noie@ferdowsi.um.ac.ir

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Nomenclature

AR	aspect ratio (ratio of evaporator section length to inside diameter, Le/ID)
C_{sf}	constant, determined from experimental data
c_p	specific heat of water (j/kg °C)
g	gravitational acceleration (m/s^2)
FR	filling ratio (ratio of volume of working fluid to volume of evaporator section)
k_l	thermal conductivity of liquid (W/m °C)
h_e	boiling heat transfer coefficient (W/m^2 °C)
h_{fg}	latent heat of vaporization (J/kg)
I	current (A)
ID	inside diameter of tube (m)
L_a	length of adiabatic section (m)
L_c	length of condenser section (m)
L_e	length of evaporator section (m)
$\dot{m}$	mass flow rate of water in condenser (kg/s)
OD	outside diameter of tube (m)
Pr	Prandtl number
P_{atm}	Standard atmospheric pressure (Pa)
P_{sat}	vapor saturation pressure (Pa)
q	heat flux (W/m^2)
$\dot{Q}_{av}$	heat transfer rate (W)
$\dot{Q}_{in}$	inlet heat by evaporation (W)
$\dot{Q}_{out}$	outlet heat by condensation (W)
$\dot{Q}_{av} = \frac{\dot{Q}_{in}+\dot{Q}_{out}}{2}$	average of heat transfer rate (W)
T_{av}	average temperature of evaporator section (°C)
T_i	inlet water temperature of condenser (°C)
T_o	outlet water temperature of condenser (°C)
T_v	vapor temperature (°C)
V	voltage (V)
R	resistance (Ω)
ρ_l	density of liquid (kg/m^3)
ρ_v	density of vapor (kg/m^3)
μ_l	viscosity of liquid (kg/m s)
σ	surface tension (N/m)

1. Introduction

The thermosyphon has been proved as a promising heat transfer device with very high thermal conductance. In practice, the effective thermal conductivity of thermosyphon exceeds that of copper 200–500 times [1]. A two-phase closed thermosyphon is a high performance heat transfer device which is used to transfer a large amount of heat at a high rate with a small temperature

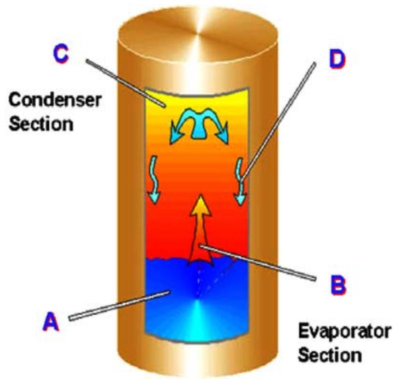

Fig. 1. Schematic of a two-phased closed thermosyphon.

difference. This is achieved by evaporation of the working fluid (A) in the evaporator section (B) and condensing in the condenser section (C) and return of the condensate (D), as shown in Fig. 1.

The two-phase closed thermosyphon is widely used because of their simple structure when compared with other types of heat pipes. Therefore, thermosyphons are being used in many applications such as: heat exchangers (air pre-heaters or systems that use economizers for waste heat recovery), cooling of electronic components, solar energy conversion systems, spacecraft thermal control, cooling of gas turbine rotor blades, etc. [1].

A considerable experimental and theoretical works have been done on the application and design modification for improving thermosyphon performance. Nguyen-Chi et al. [2] investigated experimentally the performance of vertical two-phase close thermosyphon and used water as a working fluid. They investigated the influence of operating parameters on the maximum performance either by dry out or burn out limits. Li et al. [3] investigated experimentally the steady-state heat transfer characteristics of a vertical two-phase closed thermosyphon at low temperature differences with R11, R22, and water as working fluids. Park and Lee [4] made an experimental study on the performance of stationary two-phase closed thermosyphon with three working fluid mixtures (water–glycerin, water ethanol, and water–ethylene glycol). Shiraishi et al. [5] investigated experimentally a critical heat transfer rate in thermosyphons by taking into account the aspect ratio, filling ratio, working fluid property, and operating pressure. Terdtoon et al. [6] studied experimentally the effect of aspect ratio and Bond number on heat transfer characteristics of two-phase closed thermosyphon at normal operating condition. They used R-22, ethanol, and water as working fluids. Also, Terdtoon et al. [7,8] studied experimentally internal flow patterns of thermosyphon and the effect of aspect ratio and Bond number on it at various angles. In addition they changed the filling ratio from 80% to 150% with R123 as a working fluid. Nitipong et al. [9] introduced an analytical model for the performance limit of an inclined two-phase closed thermosyphon. In that model the prediction of the performance determining the average liquid film thickness and heat conduction in the condenser section when dry out limit occurs. Shalaby et al. [10] studied experimentally the heat transfer performance of a two-phase closed thermosyphon with R22 as working fluid. Booddachan [11] studied experimentally the effect of inclination angles, aspect ratios, and Bond numbers on the thermal behavior of a closed two-phase HDPE with R113 and R11 as working fluids. Hong et al. [12] studied the heat transfer characteristics of a FC-72 thermosyphon experimentally. Han et al. [13] investigated the boiling heat transfer characteristics of a two-phase closed thermosyphon with grooves and developed a simple mathematical model to predict the performance of the thermosyphon.

Although numerous works have been carried out in the field of high temperature applications, there is a lack of sufficient data on the actual thermal performance of thermosyphons for low temperature especially the geometric effects. This paper considers the effect of sizing, which are aspect ratio and filling ratio on thermal performance of the thermosyphon in the range of heat transfer rate $100 < \dot{Q} < 900\,\mathrm{W}$.

2. Experimental apparatus and test procedure

Fig. 2 shows the experimental setup for studying the thermal characteristics of a thermosyphon for medium temperatures. The test rig consists of a heater, a liquid reservoir for charging, a thermosyphon, a cooling section, and also measuring instruments. The upper part of the thermosyphon was equipped with a vacuum seal valve for connection to a mechanical vacuum pump and to the working fluid charging line. A mechanical vacuum pump capable of up to −86 kPa and pumping capacity of 142 l/min was used for partial elimination of the non-condensable gases (NCG) from the thermosyphon. Complete extraction of NCG was achieved by purging.

The details of thermosyphon with aspect ratios (AR) and defined as Le/ID of 9.45, 9.8, and 11.8, an electric heater for evaporator section, and a water jacket for condenser section are shown in Fig. 3.

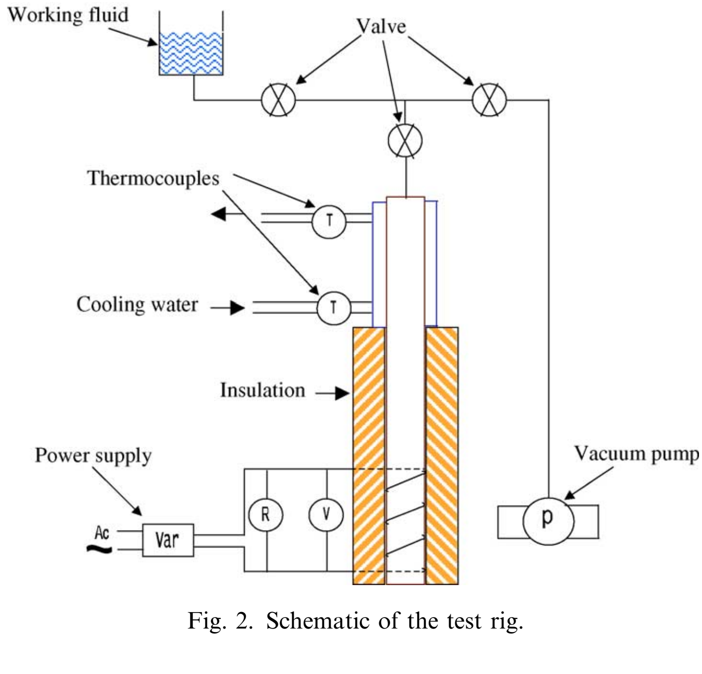

Fig. 2. Schematic of the test rig.

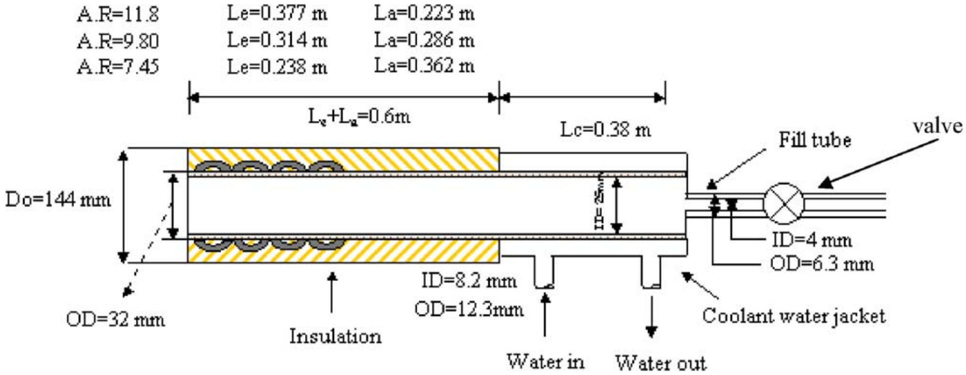

Fig. 3. Details of thermosyphon with AR = 11.8, 9.8 and 7.45.

The thermosyphon consisted of a 980 mm long copper tube having an inside diameter of 25 mm and outside diameter of 32 mm. The tube was sealed at one end and was provided with a vacuum valve at the other. The evaporator and adiabatic sections had a total length of 0.6 m. The length of the evaporator section was varied by varying the length of the electrical resistance. The condenser section of the pipe consisted of a 0.38 m long (51 mm OD) concentric tube acting as cooling water jacket surrounding the pipe. An electrical resistance of a nominal power 1000 W, which was wrapped around the evaporator section, heated the evaporator section. To prevent the heat loss, the electric elements were insulated by Rock Wool having a thickness of 55 mm. Heat was removed from the condenser section by the water jacket as described.

The power supplied to the evaporator section was determined by monitoring the applied voltage and resistance with accuracy of ±2%.

Flow rate of the cooling water was determined by measuring the amount of the water over an interval of time, while water inlet and outlet temperatures were measured using digital mini-thermometer with an accuracy ±1 °C. The accuracy of flow rate measurement was estimated to be around ±3%. The distilled water was charged into the tube under −86 kPa vacuum pressure. A variable voltage controlled the rate of heat transfer to the evaporator. Temperature distribution along the surface of the thermosyphon was measured by Ni–Cr thermocouples. The thermocouples were mechanically attached to the surface of the pipe. The locations of the thermocouples are shown in Fig. 4. The vapor temperature (T_v) was measured by thermocouple T_{a1} attached to the surface of the adiabatic section as shown in the figure. The upper surface of the thermocouple was fully insulated.

The rate of heat transfer to the evaporator section was calculated from the following relation:

$$\dot{Q}_{in} = V^2/R \tag{1}$$

The rate of heat removal from the condenser section was obtained from the following relation:

$$\dot{Q}_{out} = \dot{m}c_p(T_o - T_i) \tag{2}$$

The heat losses from evaporator and condenser sections were assumed to be negligible.

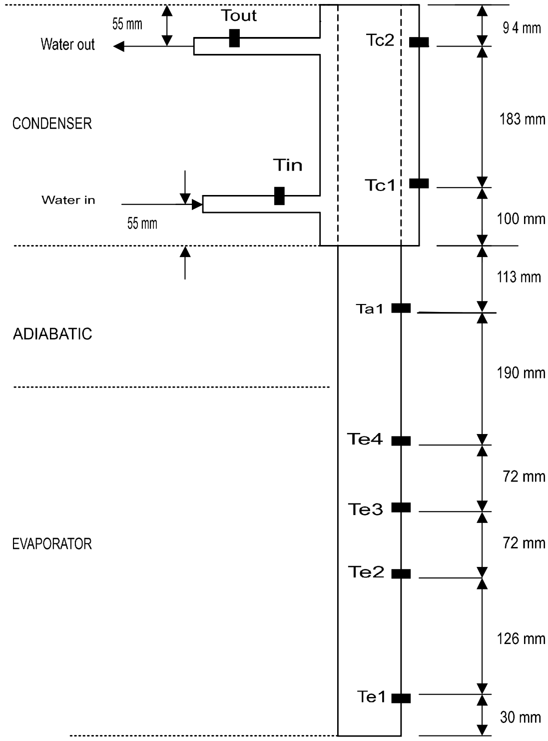

Fig. 4. Location of thermocouples.

In order to find out the effects of input heat transfer rates, aspect ratio, and filling ratio on the thermal performance of the thermosyphon, a series of tests were carried out for the following conditions:

Range of input heat transfer rate :	$100 < \dot{Q} < 900\text{W}$
Working fluid :	Distilled water
Filling ratio :	$30\% \leqslant \text{FR} \leqslant 90\%$
Aspect ratio :	$7.45, 9.8,$ and 11.8

Test procedure began by charging distilled water as working fluid. In the first series of experiments, the effects of filling ratio on the thermal performance of the thermosyphon with constant aspect ratio were investigated. For aspect ratios 7.45, 9.8, and 11.8 the thermosyphon was filled with different filling ratio equal to 30%, 60%, and 90%. Aspect ratio was varied by varying the length of the heated section of the evaporator. Experiments were carried out with increasing input heat in the range of $100 < \dot{Q} < 900\,\text{W}$ to the evaporator section. As a result, the average temperature of the surface of the evaporator section increased from 60 to 120 °C.

Data were recorded when the system reached steady-state condition.

3. Experimental results and discussion

3.1. Temperature distribution along the outside wall of the thermosyphon

The temperatures at four points on the evaporator section, one point on the adiabatic section, and two points on the condenser section, are simultaneously monitored to observe the temperature distribution over the entire length of the thermosyphon as shown in Fig. 4.

Figs. 5 and 6 present the variation of the temperature along the thermosyphon for an aspect ratio of 7.45 and three different filling ratios (30%, 60%, and 90%) for heating rate of 215 and 830 W respectively. Figs. 7 and 8 show the same information for heating rate of 670 and 830 W and for an aspect ratio of 11.8 respectively.

As shown in the figures the temperature distribution along the wall of the thermosyphon in the evaporator section is almost isothermal, especially at the lower $\dot{Q}_{in}$. Points T_{e2} exhibit slightly a higher temperature with respect to T_{e1} because the end of the pipe was not insulated. Due to the coolant water flow inside the cooling jacket, the temperature distribution in the condenser section was lower as expected.

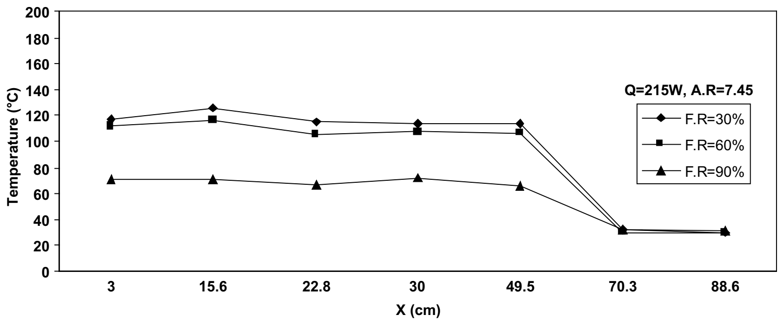

Fig. 5. Variation of the temperature along the thermosyphon for aspect ratio of 7.45 and three different filling ratio for heating rate of 215 W.

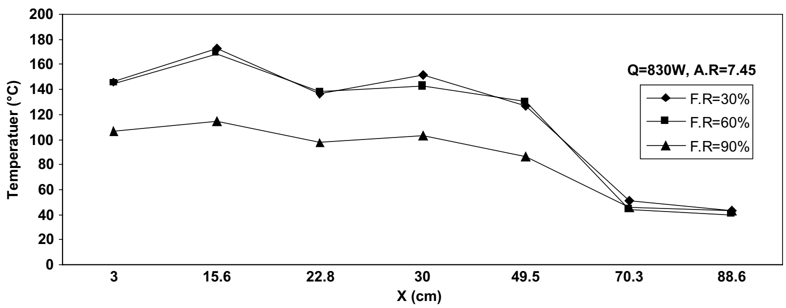

Fig. 6. Variation of the temperature along the thermosyphon for aspect ratio of 7.45 and three different filling ratio for heating rate of 830 W.

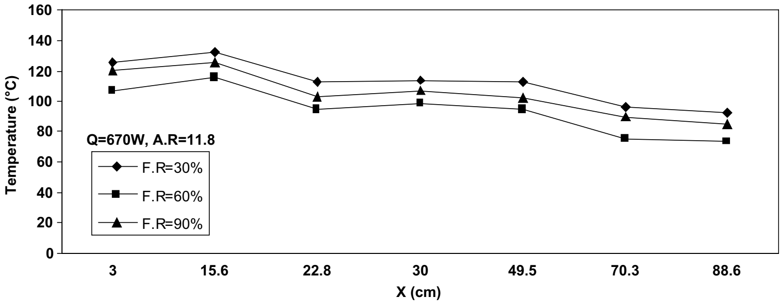

Fig. 7. Variation of the temperature along the thermosyphon for aspect ratio of 11.8 and three different filling ratio for heating rate of 670 W.

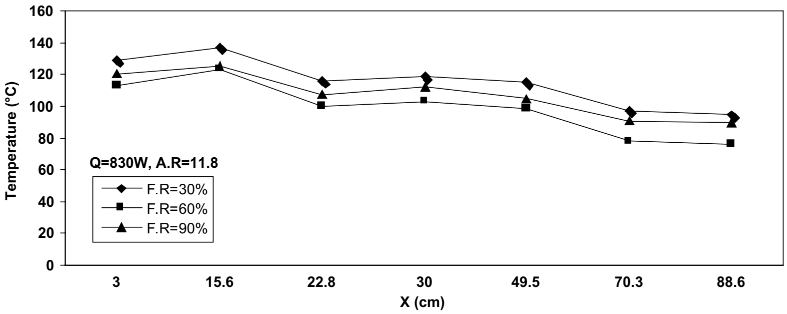

Fig. 8. Variation of the temperature along the thermosyphon for aspect ratio of 11.8 and three different filling ratio for heating rate of 830 W.

It can also be seen in Figs. 5 and 6 that the outside temperatures of evaporator section are lower when filling ratio is 90% for aspect ratio of 7.45, while in Figs. 7 and 8 the temperature for aspect ratio of 11.8 is lower when filling ratio is 60%.

3.2. Maximum heat transfer rate

In Figs. 9–11 the results have also been plotted as heat transfer rate with respect to the average temperature of the evaporator surface for the aspect ratios 7.45, 9.8, and 11.8 and the filling ratios of 30%, 60%, and 90%. The heat transfer rate of the thermosyphon was assumed to be equal to the average of the heat generation rate supplied by electric heater and heat absorption by cooling water under steady-state conditions.

As shown in the figures the maximum heat transfer rates for each aspect ratio take place at different filling ratios. For aspect ratio of 11.8 the maximum heat transfer rate occurs when filling ratio is 60%. For aspect ratios of 7.45 and 9.8 the corresponding filling ratios for maximum rate of heat transfer were 90% and 30% respectively. These results for aspect ratios of 7.45 and 11.8 confirm the results presented previously in Figs. 5–8.

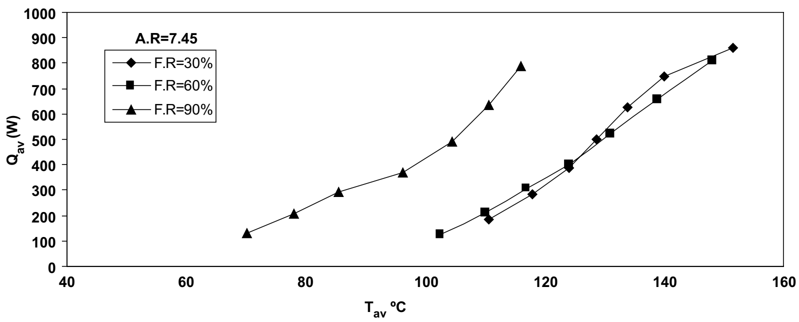

Fig. 9. Heat transfer rate vs average temperature of evaporator surface for AR = 7.45.

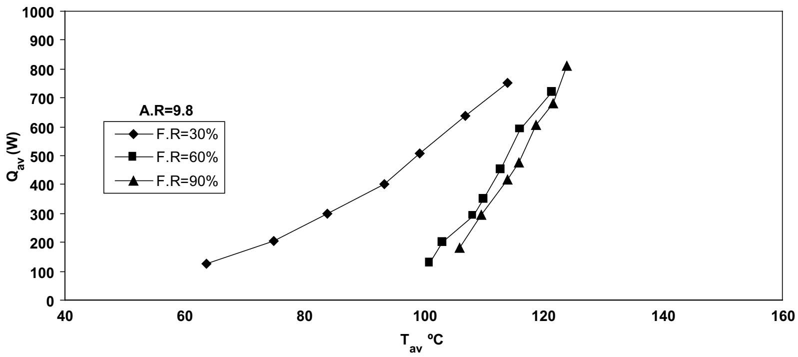

Fig. 10. Heat transfer rate vs average temperature of evaporator surface for AR = 9.8.

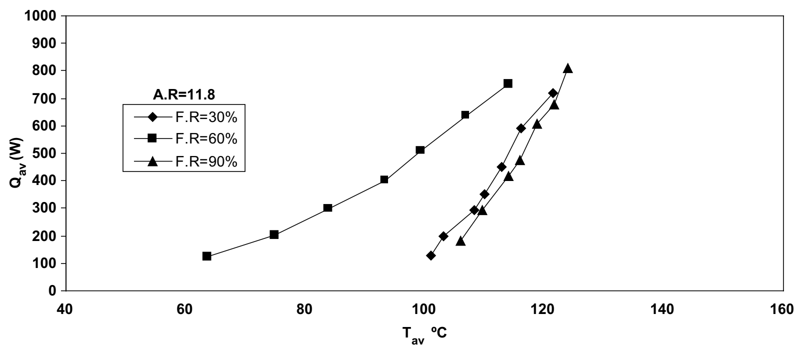

Fig. 11. Heat transfer rate vs average temperature of evaporator surface for AR = 11.8.

3.3. Comparison of experimental results with empirical correlations

From the measured data on wall temperature distribution, vapor temperature and value of thermal load, the heat transfer coefficient in the evaporator can be evaluated using the following equation:

$$h_e = \frac{\dot{Q}_{av}}{\pi IDL_e(T_{av} - T_v)} \tag{3}$$

In a previous study [12], it was found out that nucleate boiling is the dominant mechanism in the evaporator, when the filling ratio is higher than 30%. Therefore, the following two correlations based on nucleate boiling were chosen to compare with experimental data. These are Rohsenow's correlation [14]:

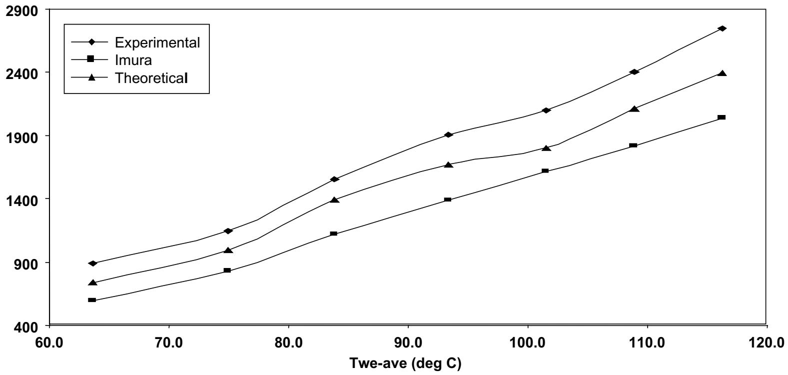

Fig. 12. Heat transfer coefficient vs average temperature of evaporator surface for AR = 9.8 and FR = 30%.

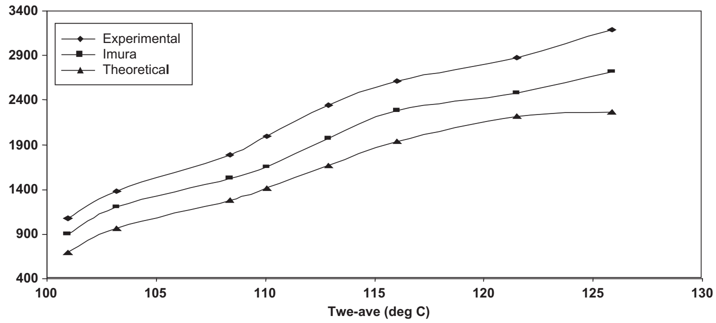

Fig. 13. Heat transfer coefficient vs average temperature of evaporator surface for AR = 9.8 and FR = 60%.

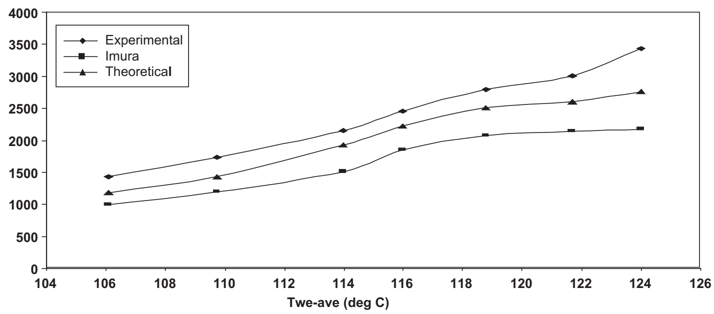

Fig. 14. Heat transfer coefficient vs average temperature of evaporator surface for AR = 9.8 and FR = 90%.

$$h_e = \frac{q^{2/3}}{\frac{C_{sf} h_{fg}}{C_{p,l}} \left\{ \frac{1}{h_{fg}\mu_l} \left(\frac{\sigma}{g[\rho_l - \rho_v t]} \right)^{1/2} \right\}^{0.33} Pr^{1.7}} \tag{4}$$

and Imura's correlation [15]:

$$h_e = 0.32 Z \left(\frac{P_{\rm sat}}{P_{\rm atm}} \right)^{0.3} \tag{5}$$

where

$$Z = \frac{\rho_l^{0.65} k_l^{0.3} c_{pl}^{0.7} g^{0.2} q_e^{0.4}}{\rho_v^{0.25} h_{fg}^{0.4} \mu_l^{0.1}} \tag{6}$$

Comparisons of heat transfer coefficient in the evaporator as calculated by Eq. (3) are compared with predictions by Rohsenow and Imura in Figs. 12–14 for aspect ratio of 9.8 and filling ratios 30%, 60% and 90%. As shown in the figures, the experimental results were found to be in reasonable agreement with correlations of Rohsenow and Imura.

4. Conclusions

The effects of the aspect ratio and filling ratio on the heat transfer characteristics of a closed two-phase thermosyphon under normal operating conditions was investigated in this paper in the range of input heat $100 < \dot{Q} < 900\,\mathrm{W}$. The experimental results were compared with the correlations of Rohsenow and Imura. It has been concluded that:

1. The temperature distribution along the thermosyphon wall in the evaporator section was almost isothermal. The measured temperature along the condenser showed lower values. This drop of temperature is expected because of the internal resistances due to boiling and condensation.
2. The outside temperature of evaporator section are lower when filling ratio is 90% for aspect ratio of 7.45, while the temperature for aspect ratio of 11.8 is lower when filling ratio is 60%.
3. Maximum heat transfer rates for each aspect ratio take place at different filling ratios. For aspect ratio of 11.8 maximum heat transfer rate occurs when filling ratio is 60%, while for aspect ratios of 7.45 and 9.8 the corresponding filling ratios for maximum rate of heat transfer are 90% and 30% respectively.
4. The boiling heat transfer coefficients for aspect ratio of 9.8 and filling ratios 30%, 60% and 90% were found to be in reasonable agreement with empirical correlations.

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