Studies on Condensation Heat Transfer in Two-phase Closed Thermosyphons

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Summary: In order to study the effects of shear stress acting on liquid-vapor interface at the condenser of two-phase closed thermosyphons, a simple analysis based on Nusselt's theory of filmwise condensation and heat transfer experiments were carried out. Analysis showed that interfacial shear stress caused by rising vapor makes condensate film thicker and condensation heat transfer coefficient and heat transfer rate decreased. However, according to heat transfer experiments, heat transfer rates increased and became larger than analytical results. To explain these discrepancies qualitatively, flow situations inside the thermosyphon were observed using a glass thermosyphon.

Nomenclature

d_i = inner diameter of thermosyphon (m)
g = acceleration due to gravity (m/s^2)
L = latent heat of evaporation (J/kg)
L_c = condenser length (m)
M = circulating mass flow rate in thermosyphon (kg/s)
Q = heat transfer rate (W)
q = heat flux without interfacial shear stress (W/m^2)
q' = heat flux with interfacial shear stress (W/m^2)
Re_{cri} = critical Reynolds number of condensate film for transition from laminar to turbulent flow
Re_L = liquid Reynolds number $4Q/(\pi d_i L \mu_L)$
Re_v = vapor Reynolds number $4Q/(\pi d_i L \mu_v)$
T = temperature (°C)
V_m = mean vapor velocity (m/s)
V^+ = dimensionless volume of working fluid in the thermosyphon (volume of working fluid)/(volume of evaporator)
X = downward vertical axis of thermosyphon from the top of condenser section (m)
α_c = mean condensation heat transfer coefficient with interfacial shear stress (W/m^2 °C)
α_m = mean condensation heat transfer coefficient without interfacial shear stress (W/m^2 °C)
α_x = local condensation heat transfer coefficient with interfacial shear stress (W/m^2 °C)
δ = liquid film thickness (m)

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λ = thermal conductivity (W/m °C)
μ = viscosity (kg/(ms))
ρ = density (kg/m^3)
τ = interfacial shear stress (N/m^2)

Subscripts
c = condenser
L = liquid
v = vapor

1. Introduction

Heat pipes are extremely excellent heat transfer devices and have been widely applied to various fields.

Two-phase closed thermosyphons investigated in this paper are basically identical with gravity-assisted wickless heat pipes which have the evaporator at their bottom. Vapor flow generated at the evaporator goes up along the axial direction of the tube, and condenses at the condenser wall. Condensate flows down along the wall surface, and returns to the evaporator by the gravitational force. As condensate falls against vapor flow, flow pattern in the thermosyphon is counter-current two-phase flow. Such condensation as counter-current two-phase flow is called "REFLUX CONDENSATION". In counter-current two-phase flow, liquid flow rate is restricted to a certain limit, because condensate is prevented from flowing down by the shear stress acting on liquid-vapor interface caused by rising vapor flow.

In general, flow patterns of counter-current two-phase flow are classified into the following four patterns /1/: laminar liquid-laminar vapor flow, laminar liquid-turbulent vapor flow, turbulent liquid-laminar vapor flow and turbulent liquid-turbulent vapor flow. In open systems, each flow pattern can be realized by changing liquid and vapor flow rates independently. But in closed systems such as two-phase closed thermosyphons, condensate and vapor flow rates depend on heat transfer rate, so the above four flow patterns cannot be realized freely. Consequently, flow patterns and interfacial shear stresses are determined by heat transfer rates.

In this paper, effects of interfacial shear stress on condensation heat transfer at the condenser of two-phase closed thermosyphons were investigated experimentally and analytically. And flow situations inside the thermosyphon were observed using a glass thermosyphon.

2. Analysis

Nusselt analyzed the filmwise condensation on a vertical wall at a constant temperature in the stagnant vapor, and obtained the following equation as the mean condensation heat transfer coefficient/2/.

$$\alpha_m = 0.943\left[\frac{\lambda_L^3 \rho_L(\rho_L - \rho_v) g L}{\mu_L(T_v - T_c) L_c}\right]^{1/4} \qquad (1)$$

In two-phase closed thermosyphons, shear stress acts on liquid-vapor interface due to rising vapor and falling liquid. So, taking into account of interfacial shear stress for Nusselt's analysis, liquid film thickness δ

$$\left[\frac{\rho_L(\rho_L-\rho_v)gL}{4\mu_L}\right]\delta^4-\left[\frac{\rho_L L\tau_i}{3\mu_L}\right]\delta^3=\lambda_L(T_v-T_c)x \tag{2}$$

Solving the above biquadratic equation, local condensation heat transfer coefficient α_x is determined as follows.

$$\alpha_x=\frac{\lambda_L}{\delta} \tag{3}$$

Mean condensation heat transfer coefficient α_c is given by integrating eq. (3) along the condenser section and being divided by the condenser length L_c.

$$\alpha_c=\frac{1}{L_c}\int_0^{L_c}\alpha_x dx \tag{4}$$

Interfacial shear stress τ_i must be known in calculating liquid film thickness δ. In this paper, shear stresses are given by the following two equations. (Eqs. (5) and (6))

〈1〉 In case of laminar vapor flow, ($Re_v \leqq 2300$)
From the theory of Hagen-Poiseuille flow/3/, interfacial shear stress is

$$\tau_i=\frac{8\mu_v V_m}{d_i} \tag{5}$$

〈2〉 In case of turbulent vapor flow, ($Re_v > 2300$)
From the experimental equation of Blasius/3/, interfacial shear stress is

$$\tau_i=0.03955\rho_v V_m^2\left[\frac{\nu_v}{V_m d_i}\right]^{1/4} \tag{6}$$

Mean vapor velocity V_m in eqs. (5) and (6) is derived from the following circulating mass flow rate M.

$$M=\frac{Q}{L}, \qquad V_m=\frac{4M}{\rho_v \pi d_i^2} \tag{7}$$

Calculation procedures are as follows:

(1) Temperature difference at the condenser T_v-T_c is assumed.

(2) Mean condensation heat transfer coefficient α_m and heat flux q without interfacial shear stress are calculated. (Eq. (1))

(3) Condenser section is divided into 200 parts at the same interval. For each part, liquid film thickness δ and local heat transfer coefficient α_x are calculated. (Eqs. (2) and (3))

(4) Mean condensation heat transfer coefficient α_c with interfacial shear stress stress is calculated from eq. (4) using trapezoidal approximation. Then heat flux q' is obtained.

(5) If the difference $|q-q'|$ is larger than the desired error tolerance, temperature difference at the condenser T_v-T_c is corrected and calculations of (3)–(5) are repeated.

Calculations were carried out for three brass thermosyphon models: inner diameter 54 mm, 34 mm, 14 mm, conderser length 200 mm uniformly, ethyl alcohol as a working fluid. Thermal properties of ethyl alcohol at 50°C were used in the above calculations. Fig. 1 is the analytical results with respect to heat flux vs temperature difference at the condenser. When heat fluxes are lower than 1×10^4 w/m^2, effects of interfacial shear stress are negligible. For larger heat fluxes than 1×10^4 w/m^2, effects of interfacial shear stress appear, and the smaller pipe diameter, the larger vapor velocity, and the larger effects of shear stress are found.

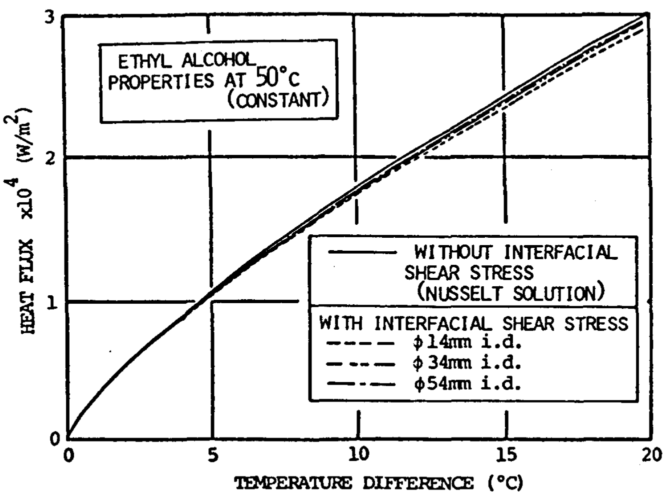

Fig. 1. Analytical Result.

3. Experimental Apparatuses and Procedures

3.1. Brass Thermosyphons

A schematic diagram of experimental apparatus is shown in Fig. 2. The experimental apparatus was constructed by thermosyphon, charging line of the working fluid, evacuating line, power supply and cooling system. Dimensions of three brass thermosyphons and the number of thermocouples are summarized in Table-1. Wall temperatures were measured with fourteen or fifteen copper-constantan thermocoples soldered at the position of 2 mm from the outer wall. To measure the vapor temperatures in the thermosyphon, a copper pipe with outer diameter of 2 mm and wall thickness of 0.3 mm was soldered at the center of the thermosyphon, and c-c sheath thermocopule of 1 mm o. d. was traversed in it. Heat was supplied by the electrical heater of Nichrome belt wrapped around the evaporator with glass wool insulation.

Before operation, thermosyphons were cleaned and vacuum tested. After the

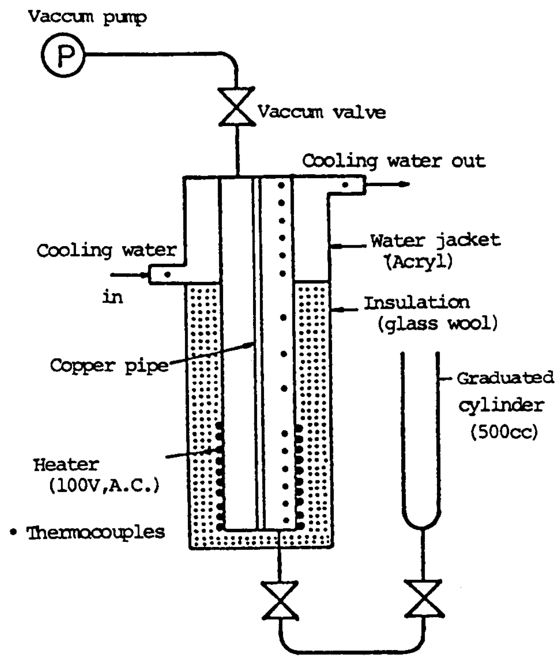

Fig. 2. Experimental Apparatus of Brass Thermosyphon

Table 1. Thermosyphon dimensions and the number of thermocouples

i.d. (mm)	o.d. (mm)	Evaporator		Adiabatic		Condenser	
		Length (mm)	T.C.	Length (mm)	T.C.	Length (mm)	T.C.
54	60	200	6	140	3	200	6
34	40	210	6	100	2	210	6
14	20	300	6	400	3	300	6

Table 2. Experimental conditions

i.d. (mm)	Quantity of working fluid V^+	Heat input Q (W)	Working fluid
54	1/8, 1/4, 1/2, 3/4, 1	100—900	Ethyl alcohol
34	1/8, 1/4, 1/2, 3/4, 1	100—600	Ethyl alcohol
14	1/8, 1//, 1/2, 3/1, 1	50—350	Ethyl alcohol

thermosyphons were evacuated, a known quantity of ethyl alcohol was charged. Then thermosyphons were operated at fixed heat input levels. Wall and vapor temperature distributions, inlet and outlet colling water temperatures and flow rates of cooling water were recorded at each heat input level for steady state conditions. Heat transfer rates, heat fluxes, condensation heat transfer coefficients, vapor and liquid Reynolds numbers were calculated. Experimental conditions of this paper are summarized in Table-2.

3.2. The Glass Thermosyphon

Fig. 3 is a schematic view of the glass thermosyphon which was made for the

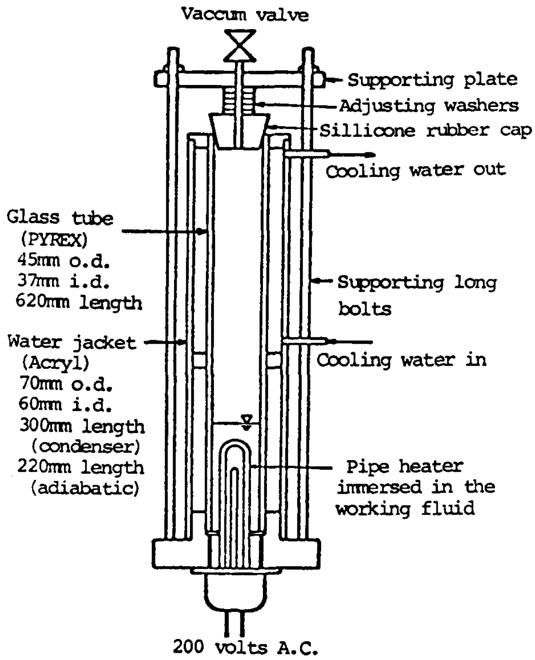

Fig. 3. Experimental Apparatus of the Grass Thermosyphon

observations of inside flow situations. The main body of thermosyphon was a heat-pressure-proof PYREX glass pipe with the diameter of 45 mm outer diameter, 37 mm inner diameter and 620 mm length. Acryl pipe with the diameter of 70 mm outer diameter, 60 mm inner diameter and 520 mm length was attached to the glass pipe coaxially. Cooling water was supplied to the upper 300 mm region between glass and acryl pipe where became the condenser section. Pipe heater with the capacity of 1000 W was immersed in the liquid reservoir at the bottom. For each heat input level from 100 W to 1000 W, flow situations of condensate and vapor inside the thermosyphon were observed.

4. Experimental Results and Discussions

4.1. Heat Transfer Experiments of Brass Thermosyphons

A typical wall temperature distribution is shown in Fig. 4. This figure is a uniform temperature distribution peculiar to heat pipes. Temperatures at the evaporator were high in the direction of the bottom for lower heat transfer rates, but constant for higher ones. This is considered as the effects of pressure distribution in the liquid reservoir.

Comparisons of analytical results with experimental ones are shown in Figs. 5, 6 and 7. In each figure, analysis agrees with experiment for lower heat fluxes, but experimental values are larger than analytical results for high heat fluxes. These disagreements between analysis and experiment will be explained qualitatively by the observations of inside flow situations at the condenser using a glass thermosyphon.

4.2. Observations of the Glass Thermosyphon

At lower heat input levels up to $Q=300$W, filmwise condensation was observed for

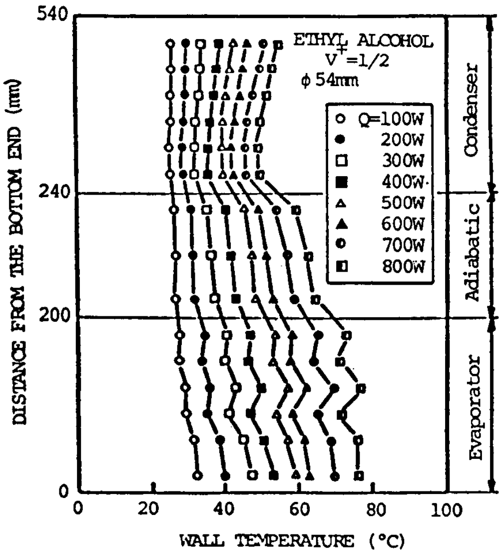

Fig. 4. A Typical Wall Temperature Distribution.

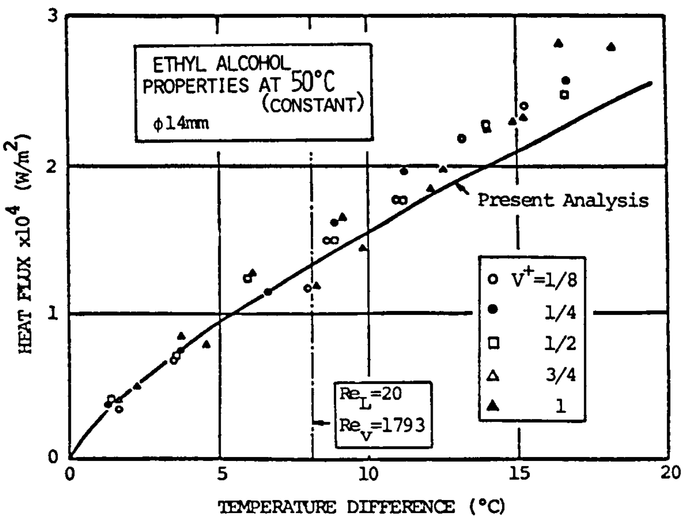

Fig. 5. Comparison of Analytical with Experimental Results.

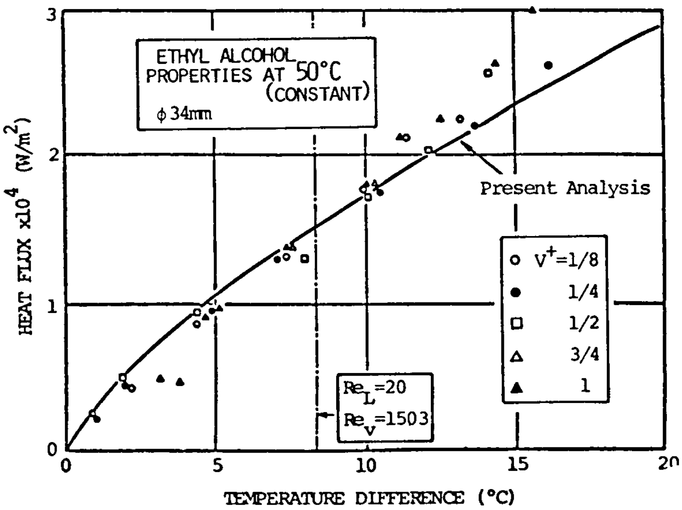

Fig. 6. Comparison of Analytical with Experimental Results.

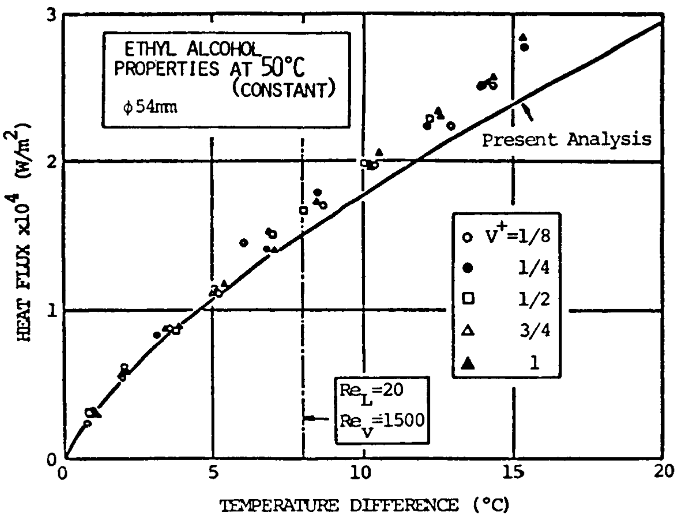

Fig. 7. Comparison of Analytical with Experimental Results.

the most part of condenser, but dropwise condensation partly.

Heat inputs were gradually increased. Being heat input 370 W, ripple condensation was observed when liquid and vapor Reynolds numbers were $Re_L=2\ 1$ and $Re_v=1535$ respectively. Photos 1, 2 and 3 show ripples at the condenser of the glass thermosyphon. From these photos, generation point of ripple moves upward in the condenser as heat input increases. As heat inputs were increased, vapor flow run against the upper end cap, rebound and generate vortexes at the upper portion of the condenser section.

Observations of the glass thermosyphon showed that ripples generate at $Re_L=20$ and $Re_v=1500$. The value of $Re_L=20$ agrees with that of reference/4/ which is the study of filmwise condensation in the stagnant vapor.

Dot-dash-lines in Figs. 5, 6 and 7 indicate the borderlines of being $Re_L=20$ and $Re_v=1500$ which were obtained from the observations of the glass thermosyphon. These lines form the boundaries where analysis doesn't agree with experiment. The exact critical Reynolds number of condensate film for the transition from laminar to turbulent flow is still unknown, but it is said that the transition occurs approximately at $Re_{cri}=400$–2000/5/. As $Re_L=20$ of ripple condensation is much smaller than the above critical Reynolds number, condensate film is considered as laminar flow, but heat transfer rates of experiment are larger than those of analysis. The followings are guessed from the above considerations. When liquid and vapor Reynolds numbers exceed 20 and 1500 respectively, ripples take place in the condenser and liquid film is disturbed. Increasing liquid and vapor Reynolds number further more, and being turbulent vapor flow formed, fluctuations of turbulent vapor flow makes liquid film disturbed and then heat transfer rates increase and become larger than analytical values.

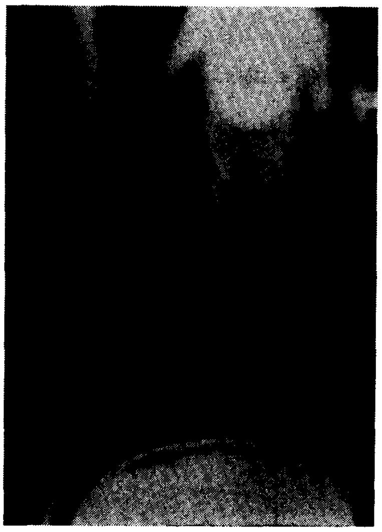

Photo 1
Ripples at the condenser
$Q=600$ W, $Re_L=33.6$,
$Re_v=2490$.

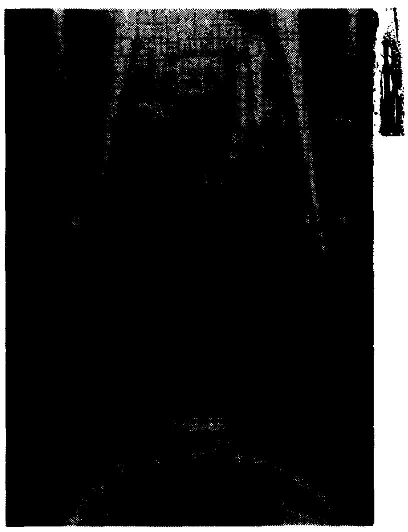

Photo 2
Ripples at the condenser
$Q=800$ W, $Re_L=44.7$,
$Re_v=3320$.

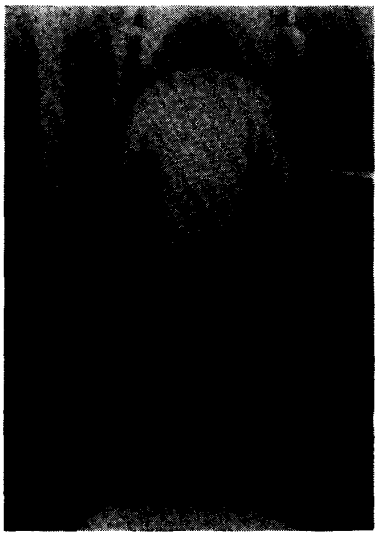

Photo 3
Ripples at the condenser
$Q=900$ W, $Re_L=50.3$,
$Re_v=3735$.

5. Summary and Conclusions

The effects of interfacial shear stress on condensation heat transfer at the condenser of two-phase closed thermosyphons were investigated experimentally and analytically, and flow situations of considensate film and vapor flow inside the thermosyphon were observed using a glass thermosyphon. From this paper, the following conclusions can be drawn.

(1) Ripples generate at $Re_L=20$ and $Re_v=1500$ for ethyl alcohol.

(2) Fluctuations of turbulent vapor flow makes liquid film disturbed and heat transfer rates increase.

6. Acknowledgement

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