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TECHNICAL NOTES

Prediction of circulation rates in vertical tube thermosiphon reboiler

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INTRODUCTION

Vertical tube thermosiphon reboilers have an application in chemical, petrochemical and many other allied industries as energy efficient equipment. The prediction of rates of heat transfer and thermally induced flow (circulation rate) is the primary requirement for the design of such equipment. The experimental measurement of these rates as carried out earlier by Piret and Isbin [1] on an electrically heated pipe. Johnson [2] measured the circulation rate in a steam-heated vertical thermosiphon reboiler of standard design. The effects of heat flux and physical properties were studied experimentally by many workers [3–7]. Recently some studies have been carried out by Smith [8], Shah [9], Agarwal [10], Johnson and Yukawa [11], Johnson [12] and Ali and Alam [3]. Some workers [13–18] tried to develop models, design methods and/or computer programs for vertical tube thermosiphon reboilers.

The aim of the present study is to show the exclusive effects of heat flux and submergence on circulation rates in a single vertical tube thermosiphon reboiler, and to develop an empirical correlation.

EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental set up used for the study has been described in detail elsewhere [3, 19, 20]. The main unit consisted of a circulation loop made of two long vertical stainless steel tubes. One of the tubes which served as a test section had a 25.56 mm i.d. and a length of 1900 mm, and was electrically heated. The upper end of the test section was connected to a vapor–liquid separator. The vapors from the separator were led to a condenser vessel, where total condensation was ensured. The condensate and the liquid from the separator entered the top end of the other vertical tube, through which the total liquid moved downwards and ultimately re-entered the test section. The stabilized power input to the test section was measured by a calibrated voltmeter and ammeter. The temperature of the heated surface was monitored by 21 copper–constantan thermocouples. The temperatures at the inlet and exit of the test section, separator and condenser were measured by thermocouple probes. The cooling water flow rate was maintained and measured by means of a rotameter. The entire set-up was thoroughly lagged with asbestos rope and glass wool, and finally covered with a thin aluminum sheet to reduce the heat losses, which were less than $\pm 2.5\%$.

The heat transfer surface was stabilized by boiling and the aging process. The test liquid was boiled off to remove traces of dissolved gases while starting a series of runs. Readings for cooling water flow rate, inlet and exit temperatures, energy input and liquid level in the downflow pipe were recorded after the steady state was established. The ranges of the parameters covered during the experiment are given in Table 1.

The effective lengths of the boiling and non-boiling zones, the liquid temperature distributions and the circulation rates were computed from the heat balance as discussed earlier [3, 19, 20]. The hydrostatic head effect on the liquid saturation temperature at various liquid submergences has been studied.

The wall temperature distribution along the heated length of the test section was obtained from experimentally measured values of the surface temperatures at 21 locations on it. A typical representative plot of this distribution with heat flux as a parameter is shown in Fig. 1. This distribution was indicative of the effective lengths of the boiling and non-boiling zones in the reboiler tube, and was helpful in verifying the Z_{NB} obtained by the equilibrium model, as shown by Kamil [19] and Kamil *et al.* [20].

RESULTS AND DISCUSSION

The circulation of fluid through a closed-loop thermosiphon system is established due to the differential head existing between the cold and hot legs. The hydrostatic head in the cold leg (downflow pipe) of a thermosiphon reboiler depends upon the liquid submergence, the maximum value of which in the present study equals the liquid level corresponding to the top end of the test section. This has been termed 100% submergence ($S = 100$). Other values of the submergence were 75, 50 and 30%. The hydrostatic head in the hot leg (reboiler tube) is provided by a column of fluid consisting of a two-phase mixture whose quality changes with boiling and vapor generation as the fluid flows upwards. The rate of circulation, therefore, depends upon the liquid submergence, heat flux, inlet liquid subcooling, vapor fraction and all the parameters involved in the frictional resistance of the circulation loop.

The fluid circulation rates through the reboiler tube were computed from the heat balance as discussed earlier. The mass flow rates have been expressed in dimensionless form as the Reynolds number. The variation of *Re* with heat flux, q, in dimensionless form with the Peclet number for boiling, Pe_B, is represented in Fig. 2 for methanol and toluene. The submergence and two-phase quality have been used as parameters. The values of S were held constant while the variation in X_{tt} could not be avoided for a set of data. In Fig.

NOMENCLATURE

a	cross sectional area of heated tube [m^2]
d	inside diameter of heated tube [m]
k	thermal conductivity [W m^{-1} °C^{-1}]
m	circulation rate [kg s^{-1}]
q	heat flux [W m^{-2}]
S	submergence [%]
T_s	saturation temperature of liquid [°C]
T_w	wall temperature [°C (K)]
ΔT_{sub}	degree of subcooling, $(T_s - T_L)$ [°C]
x^o	vapor fraction
Z	distance along the test section [m].

Greek symbols

λ	latent heat of vaporization [J kg^{-1}]
μ	dynamic viscosity [N s m^{-2}]
ρ	density [kg m^{-3}]
σ	surface tension [N m^{-1}].

Subscripts

L	liquid
s	saturation
v	vapor.

Dimensionless groups

K_{sub}	subcooling number, $1+\rho_L\Delta T_{sub}/\rho_v T_s$
Pe_B	Peclet number for boiling, $q\rho_L c_L/\lambda\rho_v k_L[\sigma/g(\rho_L-\rho_v)]^{0.5}$
Re	Reynold's number, $dm/a\mu$
X_{tt}	Lockhart–Martinelli parameter, $(1-x^o/x^o)^{0.9}(\rho_v/\rho_L)^{0.5}(\mu_L/\mu_v)^{0.1}$.

Table 1. Range of experimental parameters

System	$q \times 10^{-4}$ [W m^{-2}]	S [%]	ΔT_{sub} [°C]	x^o
Distilled water	0.57–4.3	30, 50, 75, 100	0.5–4.6	0.0028–0.493
Methanol	0.41–2.1	30, 50, 75, 100	1.0–3.7	0.0024–0.123
Benzene	0.41–2.9	30, 50, 75, 100	0.7–3.6	0.0026–0.571
Toluene	0.41–3.2	30, 50, 75, 100	1.9–8.7	0.0524–0.963
Ethylene glycol	1.6–3.0	30, 50, 75, 100	5.8–11.6	0.0063–0.485

2, it is observed that Re increases linearly with Pe_B in a logarithmic plot. The straight lines are shifted to lower values of Re as the submergence is reduced from 100 to 50%. The slight change in the slopes and scatter of data may be due to the variation in X_{tt} and inlet liquid subcooling in the above plot. At a given submergence, the liquid head in the cold leg remains unchanged while an increase in heat flux shifts the point of boiling incipience towards tube inlet. Thus, the

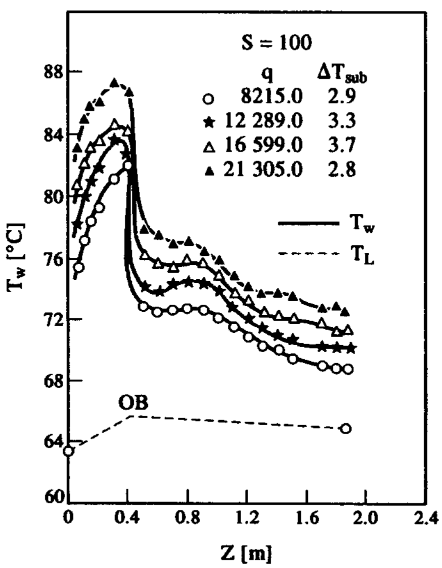

Fig. 1. Variation of wall temperature along the tube length with heat flux as a parameter for methanol.

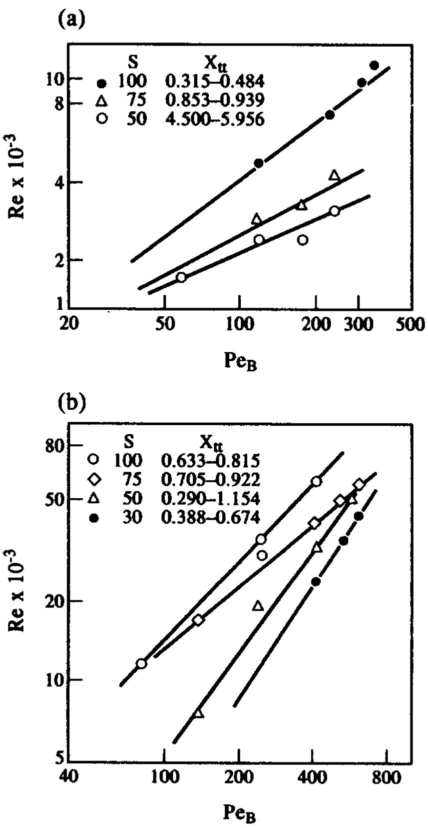

Fig. 2. Variation of circulation rate with heat flux for: (a) methanol, and (b) toluene.

saturated boiling occupies a long length of the reboiler tube, resulting in a higher vapor fraction. The average density, and hence the head of the two-phase mixture, is reduced. The differential head responsible for fluid circulation thus gets enhanced with heat flux, explaining the increase in Re with Pe_B. As the submergence is lowered, the liquid head is decreased while the vapor fraction increases due to the enhanced effect of saturated boiling in the tube. However, the differential head causing circulation becomes smaller than that at higher value of S. This explains the shifting of lines to lower values of Re with S in Fig. 2. The effect of S on Re is also shown in Fig. 3, where an increase in Re with S is observed. The data points are found to lie almost on straight lines. As a result of the increase in the circulation rate with submergence, the change in liquid temperature along the tube length is diminished, and it requires a longer length of tube to attain the saturation value. This seems to be the reason why the points of fully developed saturated boiling are shifted to larger Z at higher values of S. In the experimental runs the inlet liquid subcooling could not be regulated to the desired value and it progressively increased on decreasing the heat flux as submergence was varied from 100 to 30%. The values of X_{tt} also changed in various runs. This seems to be responsible for the scatter of data and changes in the slopes of the lines.

The linear variation of Re with Pe_B and S in Figs. 2 and 3 (log–log plots) suggest that a functional relationship of the following form may be used to correlate the data:

$$Re = C_2(Pe_B)^{m_1}(S)^{m_2}(x_{tt})^{m_3}(K_{sub})^{m_4}. \quad (1)$$

The inlet liquid subcooling and exit fluid quality have been included as K_{sub} and X_{tt}, respectively, in equation (1).

The values of the indices m_1, m_2, m_3 and m_4, and the constant C_2, were determined by a regression analysis using the data for all the test liquids, and the following equation resulted:

$$Re = 2.7494(Pe_B)^{0.9605}(S)^{0.7235}(X_{tt})^{0.6922}(K_{sub})^{-0.3089}. \quad (2)$$

The above correlation is applicable for Re ranging from 1042 to 10.1×10^4, Pe_B from 64.5 to 629.6, S from 30 to 100, X_{tt} from 0.05 to 47.0, and K_{sub} from 3.6 to 74.0.

A comparison of calculated Re with those predicted by equation (2) is shown in Fig. 4. Almost all the data points of present study and those of similar investigations [10, 21] lie within a maximum error of $\pm 20\%$.

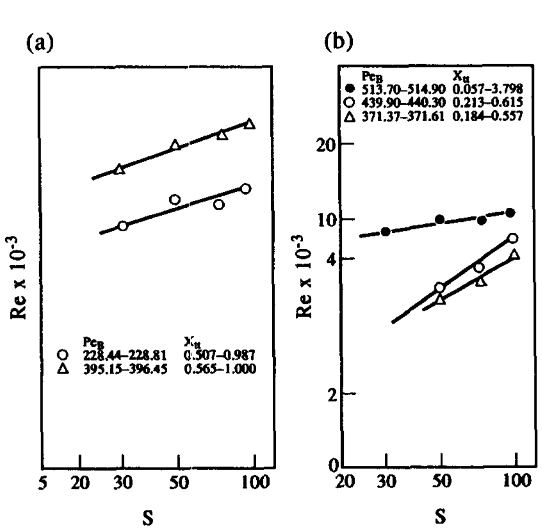

Fig. 3. Variation of circulation rate with submergence for: (a) benzene, and (b) ethylene glycol.

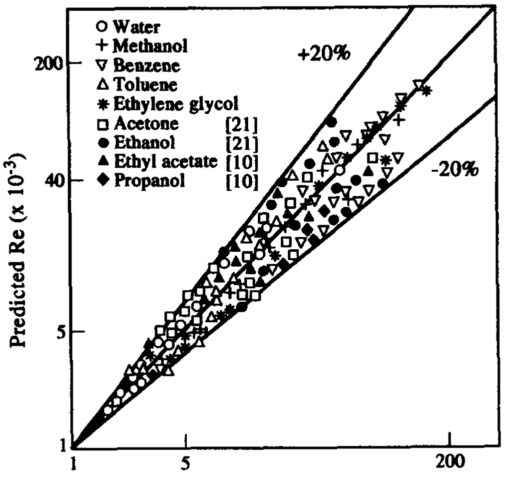

Fig. 4. Comparison of calculated Re ($\times 10^{-3}$) and those predicted by equation (2).

The properties used in the above dimensionless groups are for the saturation temperature of the test liquids.

CONCLUSION

An effort has been made to develop an empirical correlation to predict the circulation rates. The form of the equation is fairly general; however, its applicability to other systems may have to be substantiated with more data. The results may be used in uniform wall temperature problems if the heat flux profile is known.

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Rigorous solution of unsteady forced convection heat transfer

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1. INTRODUCTION

The fundamental problem of unsteady-state heat transfer between solid surfaces and steady forced flows has long been the subject of many investigations [1–6]. However, all the previous series expansion solutions (refs. [1–3] and the literature cited therein), local similarity and nonsimilarity solutions [4, 5], and even the finite-difference solution [6] are approximate ones. Only the solutions in the initial and final stages of transient have been verified by comparison with the unsteady conduction and steady convection solutions. The accuracy of these approximate solutions in the transition stage remains uncertain. There is still a need for a simple and very effective solution method that will give precise solutions over the entire transient history of unsteady convection.

In the present study, we introduce a new method for analyzing unsteady forced convection heat transfer. The method is based on the concept that the whole transient history consists of the initial stage of unsteady conduction, the final stage of steady convection, and the transition stage between these two limiting cases. Our approach is to model the thermal boundary-layer thickness of the unsteady convection as an appropriate combination of those of unsteady conduction and steady convection. In addition, a proper dimensionless time is proposed as the ratio of the thermal boundary-layer thickness of unsteady convection to that of steady convection. As a result, the transformed energy equation describes accurately the entire transient history and can be reduced readily to the conventional similarity equations of unsteady conduction and steady forced convection. Therefore, very precise finite-difference solutions and a simple correlation equation can be obtained for $0.001 \leqslant Pr \leqslant \infty$.

We demonstrate the proposed solution method for unsteady forced convection heat transfer with the case of a rotating disk.

2. ANALYSIS

The fluid of the steady laminar flow induced by a rotating disk is assumed to be incompressible and with constant properties. Initially the fluid and the solid surface are at the same temperature T_∞. At a certain instant the surface temperature is changed from T_∞ to T_0 and maintained thereafter. This situation is the case of a step change in surface temperature. Another case considered is a step change in heat flux from 0 to q_0.

The energy equation of unsteady heat transfer from the suddenly heated surface to the steady laminar flow can be written as

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial r}+w\frac{\partial T}{\partial z}=\alpha\frac{\partial^2 T}{\partial z^2} \tag{1}$$

where u and w are the velocity components in the radial and axial coordinates, respectively. This equation is subject to the initial and boundary conditions

$$T(r,z,0)=T_\infty \tag{2}$$

$$T(r,\infty,t)=T_\infty. \tag{3}$$