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EVALUATION OF THE LIQUID CHARGE FOR A CLOSED TWO-PHASE THERMOSYPHON

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We present experimental results and a simplified theoretical model on the volume filled by the liquid-vapor mixture in the evaporator of a closed two-phase thermosyphon as a function of the power throughput. The volume filled by the mixture depends on its average density. Thus, we first correlate the mixture density to the vapor mass flow rate; then, we evaluate the density average value over the mixture volume. Finally, we derive an analytical correlation between the mixture volume, the liquid charge and the power throughput. Experimental results on the mixture volume agree well with the values calculated with the proposed correlation.

NOMENCLATURE

A	device cross section area ($= \pi D^2/4$)
c_0	distribution parameter of void fraction ($= \langle \alpha j \rangle / \langle \alpha \rangle \langle j \rangle$)
c	dimensionless constant depending on the flow regime
D	device diameter
h_b	evaporator height
h_{fg}	vaporization enthalpy
j	total volumetric flux ($= j_f + j_g$)
j_g	vapor volumetric flux
j_g^*	vapor dimensionless volumetric flux ($= j_{g_{max}}/k\upsilon$)
$j_{g_{max}}$	maximum vapor volumetric flux for $x = h_b$ ($= \dot{Q}/\rho_g h_{fg} A$)
k	dimensionless constant depending on the flow regime
M	mass of the liquid charge
p_r	reduced pressure ($=p/p_{cr}$)
$\dot{Q}$	power throughput
v_{gj}	drift velocity
V_b	evaporator volume
V_m	volume filled by the mixture during operation
υ	characteristic velocity depending on the flow regime
α	void fraction
α_l	void fraction for $x = h_b$
$\bar{\alpha}$	void fraction averaged over the mixture volume V_m
$\bar{\alpha}_{0l}$	void fraction averaged over the evaporator volume V_b
ε	apparent filling degree during operation ($=V_m/V_b$)
ε_0	filling degree for zero power throughput
ρ_f	liquid density
ρ_g	vapor density
$\bar{\rho}_m$	mixture density averaged over the mixture volume V_m

INTRODUCTION

Closed two-phase thermosyphons can operate, depending on the liquid charge or filling (volume filled by the liquid at zero-power throughput), in two quite different ways, i.e., falling-film evaporation for small fillings (liquid volume smaller than 10% of the evaporator volume), and pool boiling for medium and large fillings (greater than 30% of the evaporator volume). In the former mode, the liquid wets the evaporator walls with a continuous film, and evaporation occurs at the free surface. In most cases however, due to wettability problems, the liquid does not form a continuous film and flows in rivulets down the evaporator walls. Consequently, the power throughput is greatly reduced. For medium and large filling, instead, the thermosyphon evaporator is flooded and boiling occurs. As boiling may be sustained over a range of operating conditions far wider than falling-film evaporation, pool boiling is the operation mode of practical interest. There are, however, many boiling regimes which characterize differently the thermosyphon performances, as discussed by Niro and Beretta (1990), but it is the fully-developed boiling regime which yields optimal operation.

The fully-developed boiling regime is characterized by high nucleation frequency and thick jets of bubbles. Depending on power throughput, operating pressure and device diameter, bubbles may have a small size and rise quite uniformly (bubble flow), or they may tend to agglomerate increasing in number for unit volume, and making the mixture strongly turbulent (churn flow); or, finally, bubbles may have dimensions of the same order of the device diameter (slug flow). Due to bubble flow through the liquid in the evaporator, the volume of the liquid-vapor mixture may be up to two or three times the liquid volume at zero power throughput. However, for best performance of a closed two-phase thermosyphon, preliminary experimental data reported by Petroff and Beretta (1988), suggest that the liquid-vapor mixture during operation should fill a volume slightly greater than the evaporator volume. Thus, the designer needs to correlate the liquid charge to the mixture volume during operation and the power throughput. However, no such a correlation is available in literature even though a first attempt is reported by Petroff and Beretta (1988).

The volume filled by the liquid-vapor mixture during operation depends on the average density of the mixture. Thus, in this paper we first correlate the mixture density to the vapor mass flow rate, and evaluate the density average value over all the mixture volume. Then, we derive an analytical correlation between the volume filled by mixture during operation, the zero-power liquid charge and the power throughput. Finally, we discuss the experimental results.

AVERAGE DENSITY OF THE MIXTURE

Assuming a one-dimensional flow for the liquid-vapor mixture in the thermosyphon evaporator, the mixture density ρ_m on a section at a distance x from the evaporator bottom is

$$\rho_m = \alpha \rho_g + (1 - \alpha)\rho_f \tag{1}$$

that may be also written as

$$\rho_m = \rho_f - (\rho_f - \rho_g)\alpha \tag{1'}$$

where ρ_f and ρ_g are the densities of the liquid and vapor phases, respectively, and $\alpha = A_g / A$ the void fraction, i.e., the ratio of the vapor flow area to the total cross-section area. To estimate the void fraction we use the drift-flux Wallis' model that approximates well the experimental data for bubble, churn and slug flows, i.e, all the flow regimes which may occur in fully-developed boiling. Therefore, the void fraction is

$$\alpha = \frac{j_g}{j + v_{gj}} \tag{2}$$

where $j_g = Q_g/A$ is the vapor volumetric flux, i.e., the vapor volumetric flow rate relative to the total flow area, $j = j_f + j_g$ the total volumetric flux, and v_{gj} the drift velocity of the bubbles, defined as the difference between the bubble velocity and the mixture homogeneous velocity (the velocity the mixture would have if homogeneous) that corresponds to the total volumetric flux.

If the thermosyphon is operated in a steady-state, at any section the mass flow rates of liquid and vapor must balance, i.e., $\rho_f j_f A = \rho_g j_g A$. Therefore for $\rho_g << \rho_f$, we have $j_f << j_g$ (i.e., $j \approx j_g$) and Equation 2 becomes

$$\alpha \approx \frac{j_g}{j_g + v_{gj}} \tag{2'}$$

For churn flow, the drift velocity is well approximated by the correlation proposed by Zuber and Findlay (1965)

$$v_{gj} = (c_0 - 1)j + 1.53\,[\sigma g(\rho_f - \rho_g)/\rho_f^2\,]^{0.25} \quad \text{(churn)} \tag{3'}$$

where σ is the surface tension of the liquid phase, g the gravitational acceleration, and c_0 is the distribution parameter as defined by Zuber and Findlay (1965), which takes into account the actual void distribution across the flow cross-section; Zuber and Findlay (1965) have proposed to assume $c_0 = 1.2$ (all variables in SI units) for a fully-developed flow, i.e., in a region sufficiently far from the point of bubble injection. For the slug and bubble flow, instead, the drift velocity may be evaluated by the Wallis formulas

$$v_{gj} = k_1\,[g D(\rho_f - \rho_g)/\rho_f\,]^{0.5} \quad \text{(slug)} \tag{3''}$$

$$v_{gj} = 1.18\,(1 - \alpha)\,[\sigma g(\rho_f - \rho_g)/\rho_f^2\,]^{0.25} \quad \text{(bubble)} \tag{3'''}$$

where $k_1 = 0.345\,[1 - \exp(0.01\,N_f/0.345)]$ as defined by Wallis (1969), $N_f = [D^3 g(\rho_f - \rho_g)\rho_f]^{0.5}/\mu_f$ is the inverse dimensionless viscosity relative to the liquid phase, D the device diameter, and σ is the surface tension of the liquid phase. For $N_f \geq 300$, the dimensionless function k_1 becomes a constant equal to 0.345. For bubble flow, the drift velocity depends on the void fraction.

Substituting Equations 3 into Equation 2', we obtain an estimate of the void fraction for churn, slug and bubble flow, respectively,

$$\alpha = \frac{j_g}{1.2\,j_g + 1.53\,[\sigma\, g(\rho_f - \rho_g)/\rho_f^2\,]^{0.25}} \qquad \text{(churn)} \quad (4')$$

$$\alpha = \frac{j_g}{j_g + k_1\,[gD(\rho_f - \rho_g)/\rho_f\,]^{0.5}} \qquad \text{(slug)} \quad (4'')$$

$$\alpha = \frac{j_g}{1.18\,[\sigma\, g(\rho_f - \rho_g)/\rho_f^2\,]^{0.25}} \qquad \text{(bubble)} \quad (4''')$$

The Equations 4 may be written in a more general form as shown

$$\alpha = \frac{j_g}{c\,j_g + k\upsilon} \qquad (5)$$

where υ is a characteristic velocity and c and k are dimensionless constants depending on the flow regime (churn, bubble or slug flow) which in turn depends on the device diameter.

Since the vapor is produced along all of the heated zone, the volumetric flux of the vapor phase and, therefore, the void fraction and the density of the liquid-vapor mixture are functions of the distance x from the evaporator bottom. Specifically, as x increases, j_g and α increase, whereas ρ_m decreases. For simplicity, we assume j_g increases linearly with the distance x because, as experimentally observed, nucleation occurs almost uniformly along the evaporator. Therefore

$$j_g = \frac{\dot{Q}}{\rho_g A h_{fg}} \frac{x}{h_b} = j_{g_{max}} x^* \qquad (6)$$

where $\dot{Q}$ is the overall power supplied to the evaporator, h_{fg} the vaporization enthalpy, $j_{g_{max}} = \dot{Q}/(\rho_g A h_{fg})$ the maximum vapor volumetric flux obtained for $x = h_b$, h_b the evaporator length, and $x^* = x/h_b$ is the dimensionless distance. Obviously, Equation 6 holds for $x \le h_b$ ($x^* \le 1$), whereas for $x > h_b$ ($x^* > 1$) the vapor volumetric flux becomes a constant equal to $j_{g_{max}}$.

The average density of the liquid-vapor mixture over the volume V_m, filled by the mixture during operation, is

$$\bar{\rho}_m = \frac{1}{h_m}\int_0^{h_m} \rho_m(x)\,dx \qquad (7)$$

where $h_m = V_m/A$ is the height of the zone wetted by the mixture. Substituting Equation 1 into Equation 7, this becomes

$$\bar{\rho}_m = \rho_f - (\rho_f - \rho_g)\,\bar{\alpha} \qquad (8)$$

where $\bar{\alpha}$ is defined according to

$$\bar{\alpha} = \frac{1}{h_m}\int_0^{h_m} \alpha(x)\,dx \qquad (9)$$

and represents the average void fraction over V_m. For $\rho_g << \rho_f$, Equation 8 yields

$$\bar{\rho}_m \approx \rho_f\,(1 - \bar{\alpha}) \qquad (8')$$

Finally, substituting Equations 3, 5 and 6 into Equation 9, and integrating, we obtain

$$\bar{\alpha} = \frac{1}{c}\left[1 - \frac{\ln(1 + c j^*_{g_{max}}\varepsilon)}{c j^*_{g_{max}}\varepsilon}\right] \qquad \text{for } \varepsilon < 1 \quad (10')$$

$$\bar{\alpha} = \frac{1}{\varepsilon c}\left[1 - \frac{\ln(1 + c j^*_{g_{max}})}{c j^*_{g_{max}}}\right] + \frac{\varepsilon - 1}{\varepsilon c}\,\frac{c j^*_{g_{max}}}{1 + c j^*_{g_{max}}} =$$

$$= \frac{\bar{\alpha}_{01}}{\varepsilon} + \frac{(\varepsilon - 1)\,\alpha_1}{\varepsilon} \qquad \text{for } \varepsilon \ge 1 \quad (10'')$$

where ε is the apparent filling degree during operation, defined as the ratio of the mixture volume during operation to the evaporator volume, i.e., $\varepsilon = V_m/V_b$, $j^*_{g_{max}} = j_{g_{max}}/k\upsilon$ is the vapor dimensionless volumetric flux for $x = h_b$, $\bar{\alpha}_{01}$ the void fraction averaged over the evaporator volume V_b, and α_1 the void fraction for $x = h_b$. For bubble flow, the constant c is null and Equations 10 reduce to

$$\bar{\alpha} = j^*_{g_{max}}\,\varepsilon/2 \qquad \text{for } \varepsilon < 1 \quad (11')$$

$$\bar{\alpha} = j^*_{g_{max}}\,(2\varepsilon - 1)/2\varepsilon \qquad \text{for } \varepsilon \ge 1 \quad (11'')$$

EVALUATION OF THE LIQUID CHARGE

If the mass of vapor in the device-core and the mass of liquid in the film at the wall are negligible compared to the mass of the liquid-vapor mixture in the evaporator, this latter is almost equal to the mass of liquid charge, i.e., $M_m \approx M$. Thus, the volume filled by the liquid-vapor mixture during operation is

$$V_m \approx M/\bar{\rho}_m = (\rho_f/\bar{\rho}_m)\,V_0 \qquad (12)$$

where $V_0 = M/\rho_f$ is the volume filled by the liquid charge for zero-power throughput. Substituting Equation 8', which holds for $\rho_g << \rho_f$, into Equation 12, we obtain

$$V_m = V_0/(1-\bar{\alpha}) \qquad (13)$$

or equivalently

$$\varepsilon = \varepsilon_0/(1-\bar{\alpha}) \qquad (14)$$

where $\varepsilon_0 = V_0/V_b$ the static filling degree, i.e., the filling degree for zero power throughput. Substituting Equations 10 into Equation 14 yields

$$\varepsilon = \left[c - 1 + \frac{\ln(1+\varepsilon c j^*_{g\,max})}{\varepsilon c j^*_{g\,max}}\right]^{-1} c\varepsilon_0 \qquad \text{for } \varepsilon < 1 \quad (15')$$

$$\varepsilon = \frac{\varepsilon_0 - (\alpha_1 - \bar{\alpha}_{01})}{1-\alpha_1} \qquad \text{for } \varepsilon \geq 1 \quad (15'')$$

where $j^*_{g\,max} = \dot{Q}/(\rho_g A h_{fg} k \upsilon)$. Equation 15 correlate, even though in implicit form, the apparent filling degree ε during operation to $j^*_{g\,max}$ and ε_0, and therefore, to the power throughput $\dot{Q}$ and the liquid charge M. Finally, solving Equation 15" with respect to ε_0, we obtain

$$M = \frac{\rho_f V_b}{c}\left[\varepsilon c - 1 + \frac{\ln(1+cj^*_{g\,max})}{cj^*_{g\,max}} - \frac{(\varepsilon-1)cj^*_{g\,max}}{1+cj^*_{g\,max}}\right] \qquad (16)$$

Equation 16 gives the mass of the liquid charge as a function of the power throughput and the apparent filling degree during operation. Assuming the value ε_{opt} of the apparent filling degree that optimizes the device operation is known, then equation 16 can be used to determine the optimal liquid charge M_{opt} for a given design power throughput $\dot{Q}$.

EXPERIMENTAL APPARATUS AND PROCEDURES

Experiments were performed to collect data on the height and, therefore on the volume, of the zone filled by the liquid-vapor mixture during operation as a function of power throughput, operating pressure, liquid charge and device-diameter. To this end, we specifically designed a facility equipped with transparent pipes made of boronsilicate glass, 1000 mm long, 16 and 36 mm o.d., 1.8 and 2.8 mm wall thickness, 11 and 7 bar maximum pressure, respectively. The evaporator, the adiabatic section and the condenser were respectively 300, 300 and 400 mm long. The evaporator was heated by an electric cartridge supplying a maximum power of 900 W , having a 300 mm long with a 6.35 mm o.d.. The heater was inserted directly inside the device through the bottom end cap, so that the supplied power was transfered entirely to the liquid-vapor mixture. The maximum heat flux at the heater surface was of 15 W/cm^2. The condenser was cooled by a stream of water flowing in the cooling jacket, i.e., a tube made of boronsilicate glass assembled externally and coaxially to the thermosyphon.

Measurements of the vapor temperature and pressure, and of the input power were performed. The vapor temperature was measured by means of a micro-thermocouple sheathed in a stainless tube of 0.25 mm o.d. . The probe was sealed in a glass tube of 3 mm o.d. with the tip exposed directly to the vapor, and placed at 330 mm from the condenser top. The accuracy on the temperature measure was of 0.1 °C. The pressure was measured by a strain-gage transducer of 2 bar full scale and 0.3 % f.s. accuracy, connected to the test pipe through the upper end cap. The input power was measured by an electrodynamic wattmeter to an accuracy better than 1%. Standardized procedures in test-section preparation and filling operations were adopted. In addition, at regular intervals the calibration of the pressure transducers was performed.

For each test, after the steady-state operation was reached, the measurements are sampled for a statistically significant time, and a video-record of the liquid-vapor mixture in the evaporator was recorded. Next, video images were analyzed frame by frame and the height of the zone wetted by the mixture was measured for 300 consecutive frames, corresponding to a time interval of 6 s (the video-camera frequency was 50 Hz). Finally the data collected in this way was digitized and stored on a personal computer.

RESULTS AND DISCUSSION

The experimental results concern tests run with two closed two-phase thermosyphons of 12.4 and 30.4 mm i.d., oriented vertically, with two static filling degrees, ε_0, corresponding to 50% and 100% of the evaporator volume. Water, acetone and refrigerant R113 were used as operating fluids. The operating pressure, expressed as reduced pressure, ranges between 0.001 and 0.05. The tests were performed in sequences carried out by changing the input power and refrigerant temperature in such a way that the operating pressure was kept constant during the sequence.

As the thermosyphon operation is steady only "on average" as discussed Niro and Beretta (1990), for given operating conditions, the height h of the zone wetted by the mixture is not a time-constant, but fluctuates around a constant average value. The

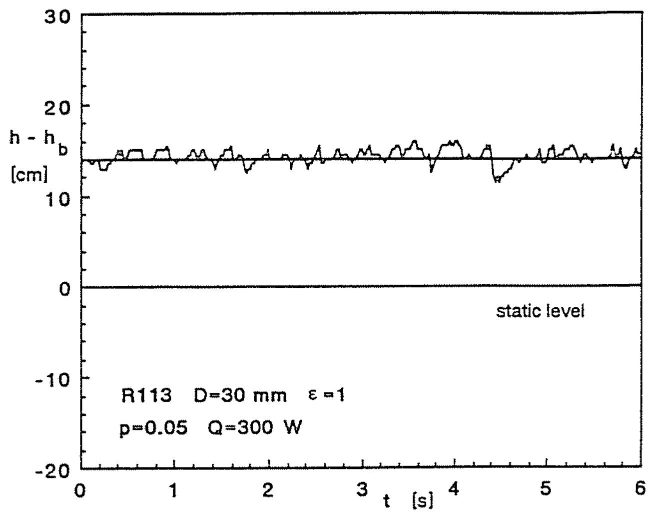

FIG. 1. Mixture level in boiling with churn flow for $j_{g_{max}} = 0.26$ m/s and $p_r = 0.05$ ($h_b = 30$ cm).

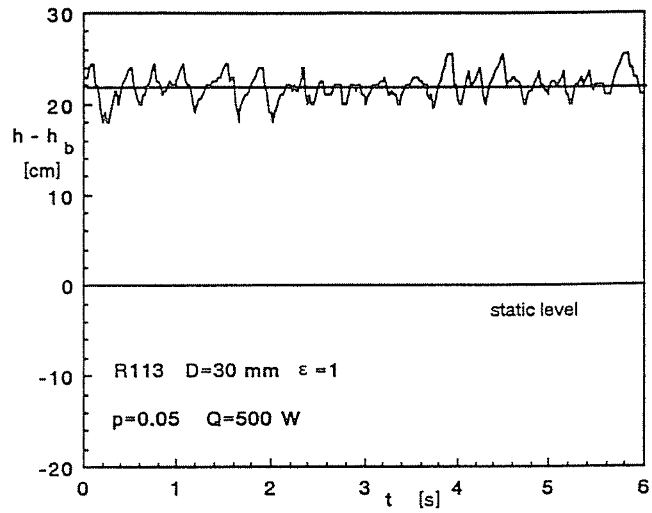

FIG. 2. Mixture level in boiling with churn flow for $j_{g_{max}} = 0.43$ m/s and $p_r = 0.05$ ($h_b = 30$ cm).

fluctuation amplitudes depend on the power throughput and the flow regime (bubble, churn or slug) and, therefore, on the operating pressure and the device diameter.

Fig. 1 shows a characteristic trend of the height h (more precisely, of $h - h_b$) as a function of time for a low value of the maximum vapor volumetric flux, i.e., $j_{g_{max}} = 0.26$ m/s. In the figure, the liquid level for zero-power throughput (static level) is also shown by a solid line drawn for $h - h_b = 0$. Obviously, for a unit static filling degree, ($\varepsilon_0 = 1$), the height h_0 coincide with h_b. As experimentally observed, the boiling is fully-developed but the bubbles are small and the mixture is weakly turbulent, because of high operating reduced pressure ($p_r = 0.05$) and low vapor volumetric flux; consequently, the fluctuation amplitude is small. However, these fluctuations increase as the power throughput increases. Figure 2 shows in fact the time trend of $h - h_b$ for a value of $j_{g_{max}}$ nearly double that of the former case ($j_{g_{max}} = 0.43$ m/s), with all other conditions equal. The bubbles are still small, but the flow is more turbulent so that the level of fluctuations are larger. Figure 3 shows the time trend of $h - h_b$, instead, for $j_{g_{max}} = 0.84$ m/s and $p_r = 0.003$. Boiling occurs with churn flow, i.e., characterized by quite large bubbles that make the mixture strongly turbulent and cause very large fluctuations of the mixture level. In Figures 1, 2 and 3, the average value of the height wetted by the liquid-vapor mixture is also shown by a solid line.

In the most cases observed experimentally, boiling occurs with churn flow even though it may be characterized by bubble dimensions quite different depending on the operating pressure (the smaller p_r, the larger bubbles become). Obviously, for a given device diameter, bubbles with larger dimension cause a stronger turbulent flow. Boiling with slug flow, instead, has been observed only for the 12.4 mm i.d. thermosyphon operating at $p_r = 0.001$ and $p_r = 0.003$. However, no experimental results are available for this latter case because the slugs rose much further than the measure zone ($h - h_b > 30$ cm). Therefore, all the experimental results concern measurements for boiling with churn flow, except for a few cases where boiling with bubble flow was observed.

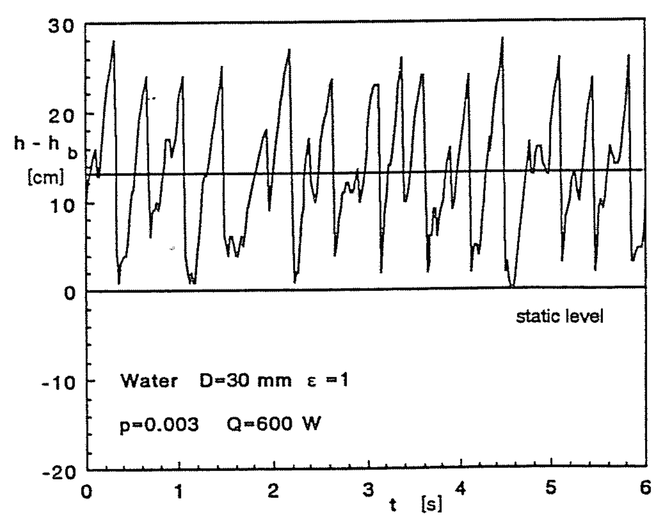

FIG. 3. Mixture level in boiling with churn flow for $j_{g_{max}} = 0.84$ m/s and $p_r = 0.003$ ($h_b = 30$ cm).

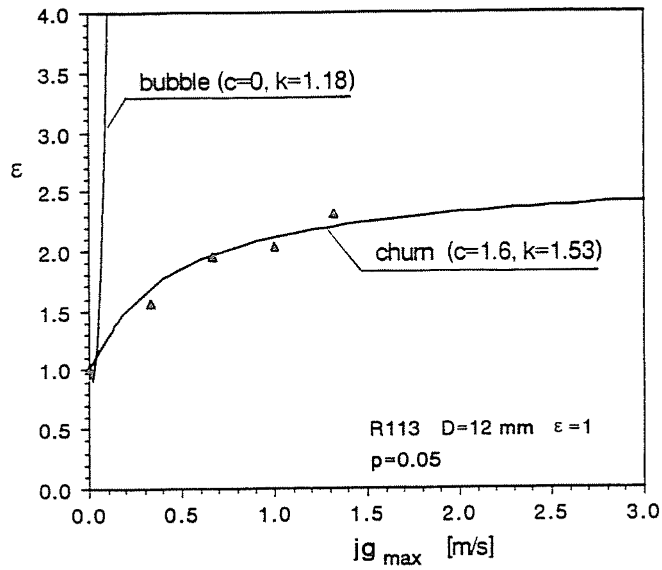

FIG. 4. Comparison of calculated trend with experimental values of filling degree during operation ε vs $j_{g_{max}}$ for boiling with churn flow.

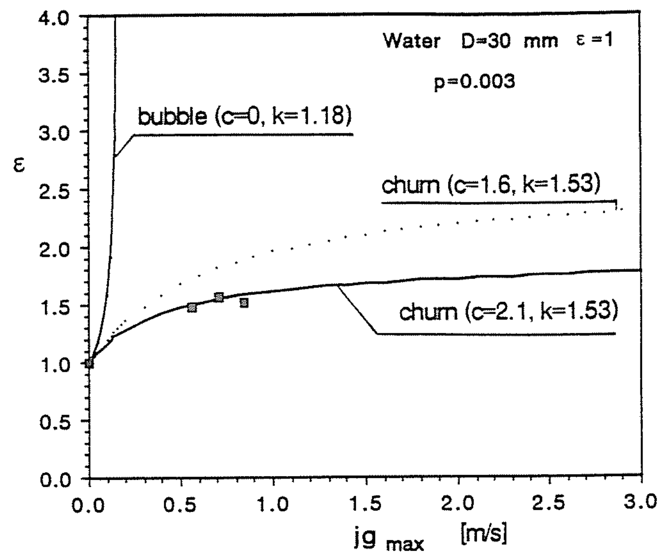

FIG. 6. Comparison of calculated trend with experimental values of filling degree during operation ε vs $j_{g_{max}}$ for boiling with churn flow.

Figures 4, 5 and 6 show the measured values of ε as a function of $j_{g_{max}}$ (therefore, mixture volume versus power throughput) for different operating conditions. For comparison, the trends of ε calculated for churn flow (Equation 15) and for bubble flow are also plotted in the figures. For churn flow, however, the measured values of ε are overestimated by Equation 15 if the values of 1.2 (Zuber and Findley, 1965) and 1.53 (Wallis, 1969) are used respectively for the constants c and k. Thus, for every series of data, we have evaluated the values of c and k which minimize the error between the measured and calculated values of ε. For the constant c, we have found values ranging between 1.4 and 2.1, whereas for the constant k we have kept the value of 1.53 because the error is weakly influenced by variation of k. These calculated values of c higher than the value proposed by Zuber and Findlay (1965) indicate a higher concentration of bubbles in the central region of our thermosyphons. Probably, this difference between the values of c is due to the different experimental conditions. Indeed, in the thermosyphon churn flow occurs with bubble nucleation distributed along the entire evaporator, whereas the value of 1.2 for c has been proposed by Zuber and Findlay (1965) for a fully-developed churn flow, i.e., in a zone far from the point of bubble injection. For boiling with bubble flow, assuming for k the value of 1.18 proposed by Wallis (1969), we obtain a trend (not shown) of ε that seems to agree well with the limited experimental data.

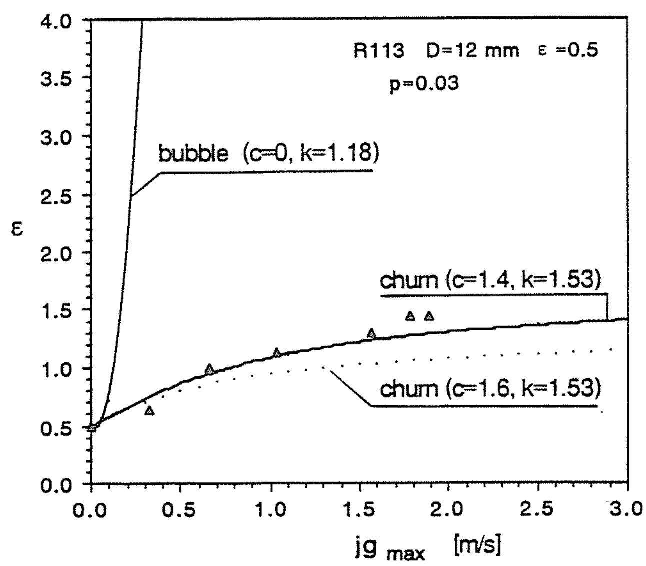

FIG. 5. Comparison of calculated trend with experimental values of filling degree during operation ε vs $j_{g_{max}}$ for boiling with churn flow.

Figure 7 shows an interesting transition from bubble flow to churn flow. Experimentally we observed that for values of $j_{g_{max}}$ less than 0.2 m/s, the liquid is fully emulsified, whereas for

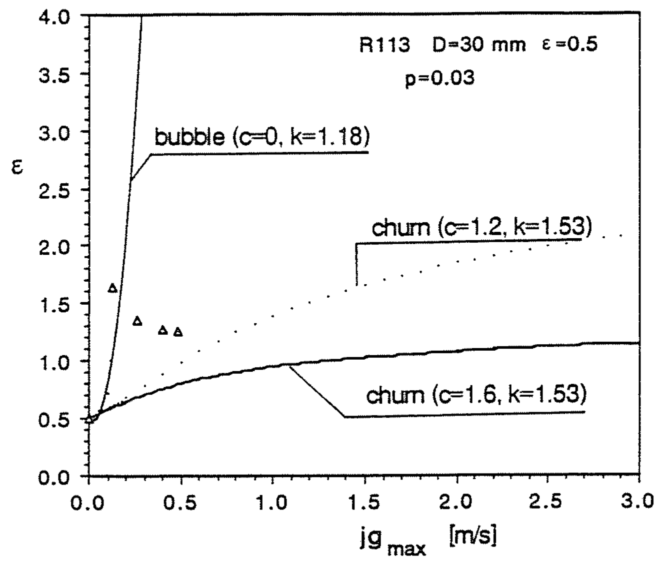

FIG. 7. Comparison of calculated trend with experimental values of filling degree during operation ε vs $j_{g_{max}}$ for boiling in transition between bubble flow and churn flow.

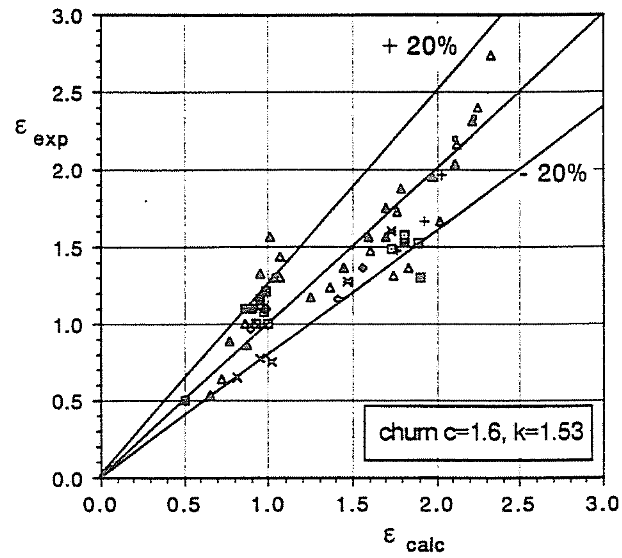

FIG. 8. Comparison of experimental results with calculated values of the mixture volume for all the collected data.

$j_{g_{max}} > 0.2$ m/s the bubbles begin to coalesce and, therefore, the mixture level decreases as power throughput increases.

In Figure 8 the experimental data for ε are plotted versus the values evaluated using Equation 15 for the corresponding values of $j_{g_{max}}$. The experimental data refer to all the test runs and therefore represent a variety of working fluids, vapor volumetric fluxes, liquid charges and device diameters. As can be seen from the figure, irrespective of liquid charge and working fluid, Equation 15 correlates the experimental data well enough with a correlation coefficient of 0.90 and an average scatter smaller than 20%, provided that we assume c = 1.6. The agreement between experimental and calculated values is slightly improved if the data are subdivided according to the device diameter and, consequently, two different values of the constant c are used. In this case, the average scatter is lower than 15% using both c = 1.5 and c = 2.1, respectively, for the 12.4 mm and 30.4 mm i.d. thermosyphons.

SUMMARY AND CONCLUSIONS

For closed two-phase thermosyphons operating for medium and large fillings under fully-developed boiling, the volume of the liquid-vapor mixture during operation may be up to two or three times the liquid volume at zero-power throughput because of the bubble flow through the liquid in the evaporator. In this paper, we have derived an analytical correlation which gives the volume filled by the liquid-vapor mixture during operation as a function of the liquid charge and the power throughput.

An experimental analysis performed on two transparent devices 12.4 and 30.4 mm i.d., using water, refrigerant R113 and acetone as working fluids, has yielded data on the volume filled by the liquid-vapor mixture during operation for many values of power throughput, operating pressure and static filling degrees. The data are in reasonable agreement with the proposed correlation (Equation 15) except for the constant c which has been evaluated experimentally.

For given power throughput, operating temperature and working fluid, we can evaluate the amount of the liquid charge by means of Equation 16, provided we know the optimal value ε_{opt} of the apparent filling degree during operation. Based on preliminary experimental data given by Petroff and Beretta (1988), we may tentatively assume a value of ε_{opt} in the range between 1 and 1.2, provided that mixture does not flood the condenser. Thus, further experimental work is needed to establish the value of ε_{opt} for optimal thermal performance of thermosyphons.

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