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THE MIST–ANNULAR TRANSITION DURING CONDENSATION AND ITS INFLUENCE ON THE HEAT TRANSFER MECHANISM

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Abstract—Influence of the mist–annular transition on the heat transfer mechanism during condensation is studied. It is shown that the applicability of annular flow correlations cannot be extended with accuracy to the mist region and that a dimensionless parameter We can be used in defining the departure from annular flow conditions. A heat transfer correlation is suggested for the mist region and shown capable of good agreement with a data base of different fluids and tube orientations.

1. INTRODUCTION

Studies of the two-phase flow patterns and their transitions during condensation have gained increasing interest and importance from the well-accepted view that the pressure drop and heat transfer characteristics are strongly dependent on the flow pattern. In the development of almost all the two-phase heat transfer correlations available in the literature (analytical or empirical), a certain flow pattern was assumed either explicitly or implicitly. Using a certain heat transfer correlation under operating conditions inconsistent with the flow pattern for which it was developed could lead to serious errors in magnitude and trend. This point was illustrated quantitatively for the case of horizontal intube condensation of pure vapors by Bell *et al.* (1970). Consequently, it is very important to know which flow pattern is expected to exist at certain flow conditions inside a condenser and to develop heat transfer correlations consistent with the prevailing heat transfer mechanism in each flow pattern.

Several experimental investigations (e.g., Soliman & Azer 1971, 1974; Traviss & Rohsenow 1973; Fathi 1980) were undertaken having as major objectives the generation of information on the flow patterns of condensing flows and the operaing conditions under which each pattern is supposed to exist. Figure 1 shows schematically the flow pattern progression along the condenser as observed in these investigations for conditions of high and low mass fluxes. A major difference between the two parts of figure 1 is the existence of mist flow in the entry part of the condenser at high mass fluxes. This flow pattern was commonly characterized by a continuous vapor phase carrying a major portion of the liquid phase in the form of entrained droplets with no visible stable liquid film at the tube wall. Further along the condensation path, the mist flow pattern was consistently followed by the annular flow pattern in which a stable liquid film covers the whole circumference of the tube. The range of quality x over which mist flow was observed varied according to the total mass flow rate, tube diameter and fluid properties. However, this range was found to be as wide as $1 > x > 0.3$ under some operating conditions (Fathi 1980).

The differences in flow structure between mist and annular flows were found, as expected, to reflect significantly on the heat transfer and pressure drop characteristics of two-phase, gas–liquid flows. For example, Chien & Ibele (1964) reported that an observed flow pattern transition from annular to mist–annular during air–water flow in a vertical tube coincided with a change in the slope of measured frictional pressure gradient. Groothuis & Hendal (1959) obtained experimental results which indicate clearly that at any liquid flow rate the heat transfer coefficient increases as the gas flow rate increases up to a maximum beyond which the heat transfer coefficient decreases with further increase in the gas flow rate. This reversal of trend in the heat transfer data (which was confirmed in later experiments by Ravipudi & Godbold 1978) was attributed to the transition from annular to mist–annular flow. Other investigations resulted in the recommendation of specific models or specific correlating methods for predicting the pressure drop and heat transfer coefficient in the mist flow region (Husain & Weisman 1978; Vijay 1977).

While the mist flow pattern and the influence of the mist–annular transition on the heat transfer mechanism having received some considerable attention for cases of two-phase, gas–liquid flow (as mentioned above) and two-phase boiling particularly in the dryout and post-dryout regions (Groeneveld 1975), limited attention was directed to the case of two-phase condensing flow. Many heat transfer correlations were reported for condensation assuming annular flow with a stable liquid film containing all the liquid phase (no entrainment); most, if not all, of these correlations did not specify an upper limit for their application. As a result, wide discrepancies between measurements and predictions of the heat transfer coefficient were reported in several sources (e.g., Rohsenow & Hartnett 1973; Shah 1979) at the high end of the design flow range and also at the high end of the vapor quality range (conditions at which mist flow is expected). No alternative correlations were suggested for this flow pattern which is charaterized by high entrainment and breakdown of a continuous liquid film.

The objectives of this investigation are to define the boundary between mist and annular flows during condensation and to study the influence of this transition on the heat transfer mechanism with the ultimate goal of developing a correlation capable of predicting the heat transfer coefficient in the mist flow region with reasonable accuracy.

2. THE MIST–ANNULAR TRANSITION

Entrainment during annular flow is known to occur because of the removal of the tops of waves travelling on the surface of the liquid film. However, this mechanism has been difficult to model so far because of the complexity of the phenomenon itself as well as the shortage of reliable data on the entrainment rate (Bergles *et al.* 1981). In spite of the fact that the mechanism of entrainment still needs further study, it is apparent from the present state of knowledge that the inertia force of the vapor phase is the dominant destructive force acting on the liquid film while the surface tension and liquid viscous forces are the major stabilizing forces. For the purpose of defining the mist–annular transition, Soliman (1983) suggested that a simple, easy to use, and reasonably accurate correlation can be developed based on a balance between the aforementioned forces. The hypothesis adopted

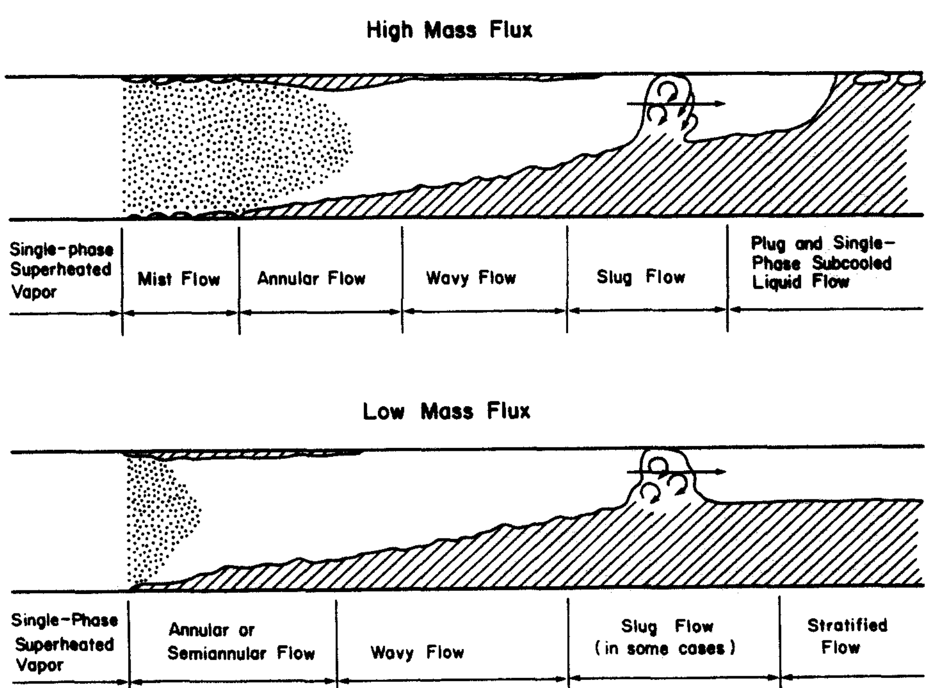

Figure 1. Flow-pattern development during horizontal intube condensation.

was that the balance between the destructive and stabilizing forces can be expressed in terms of a modified Weber number, as follows:

$$\mathrm{We}^2 = [(\rho_G V_G^2)/(\sigma/D)][(\sigma/D)/(\mu_L V_L/\delta)]^b \quad , \tag{1}$$

where We is the modified Weber number, ρ_G the density of vapor, V_G the average velocity of vapor, σ the surface tension, D the tube diameter, μ_L the liquid viscosity, V_L the average velocity of the liquid film, δ the average thickness of the liquid film and b a correlating constant. The quantities $(\rho_G V_G^2)$, $(\mu_L V_L/\delta)$ and (σ/D) are measures of vapor inertia, liquid viscosity and surface tension forces, respectively. It was shown by Soliman (1983) that hypothesis [1] is probably valid since good agreement with a wide data base of flow patterns was possible with characteristic values for the parameters b and We. From this analysis, the following formulation of the modified Weber number can be derived:

$$\mathrm{We} = 2.45\,\mathrm{Re}_{Gs}^{0.64}\left(\frac{\mu_G^2}{\rho_G \sigma D}\right)^{0.3} \Big/ \phi_G^{0.4}, \quad \mathrm{Re}_{Ls} < 1250 \tag{2a}$$

and

$$\mathrm{We} = 0.85\,\mathrm{Re}_{Gs}^{0.79}\left(\frac{\mu_G^2}{\rho_G \sigma D}\right)^{0.3}\left[\left(\frac{\mu_G}{\mu_L}\right)^2\left(\frac{\rho_L}{\rho_G}\right)\right]^{0.084}\left(X_{tt}/\phi_G^{2.55}\right)^{0.157}, \qquad \mathrm{Re}_{Ls} > 1250 \quad , \tag{2b}$$

where μ_G is the vapor viscosity and ρ_L the liquid density, while the liquid superficial Reynolds number Re_{Ls}, Lockhart–Martinelli parameter X_{tt} and the two-phase multiplier ϕ_G are given by

$$\mathrm{Re}_{Ls} = G(1-x)D/\mu_L \quad , \tag{3a}$$

$$\mathrm{Re}_{Gs} = xGD/\mu_G \quad , \tag{3b}$$

$$X_{tt} = [(1-x)/x]^{0.9}(\rho_G/\rho_L)^{0.5}(\mu_L/\mu_G)^{0.1} \quad , \tag{3c}$$

$$\phi_G = 1 + 1.09\,X_{tt}^{0.039} \quad , \tag{3d}$$

where G is the total mass flux.

Comparisons between correlation [2] and a large data base showed that, according to visual observation, the flow is always annular for We < 20 and always mist for We > 30. This method of defining the mist–annular transition during condensation will be used in the present study. At the high velocities associated with mist flow, it is expected that tube orientation would not have a significant influence on predictions based on [2] and hence this equation, while developed from horizontal flow data, would be applied later to situations involving vertical as well as horizontal flows.

3. INFLUENCE ON THE HEAT TRANSFER MECHANISM

While the existence of mist flow with appreciable liquid entrainment in the upstream parts of condensers and the fact that this flow pattern may persist for a considerable length at high mass fluxes are well recognized no comprehensive studies are available yet on the prevailing heat transfer characteristics. It is expected that at mist flow conditions the liquid film thickness is decreased by entrainment and consequently the analytical models based on annular flow (assuming no entrainment) would underpredict the experimental heat transfer data. However, this trend was accomodated in some earlier studies by modifying the annular flow correlations in order to achieve better agreement with the data in the mist region rather than abandoning the annular flow assumption and developing correlations consistent with the mist flow situation.

The objective here is to demonstrate the necessity of a special model or a special correlating method for predicting the condensing heat transfer coefficient during mist flow.

Table 1. Heat-transfer data base

Data Set no.	Inside Tube Diameter D mm	Tube Orien-tation	Condensing Fluid	Saturation Temperature T_s °C	Mass Flux G $kg/m^2.s$	Quality x	Modified Weber Number We	Liquid Prandtl Number Pr_l	Source
1	7.4	V	Steam	77 – 127	80-440	0.20-0.95	10-69	1.3-2.3	Goodykoontz & Dorsch (1967)
2	7.4	V	R-113	37 – 85	830-1470	0.20-0.95	29-90	5.7-7.7	Goodykoontz & Brown (1967)
3	12.7	H	R-12	37 – 50	190-440	0.50-0.95	12-40	3.1-3.2	Azer et al. (1971)
4	8.0	H	R-12	21 – 59	140-1530	0.20-0.95	8-120	3.0-3.3	Traviss et al. (1971)
5	8.0	H	Steam	164 – 310	380-1610	0.20-0.95	22-140	1.0-1.1	Ananiev et al. (1961)

To do so, a heat transfer data base was selected and compared with the predictions of some of the well known correlations.

3.1 *Heat-transfer data base*

The data base selected for this study is given in table 1. It consists of five sets encompassing both the horizontal and vertical orientations, and three test fluids with widely different fluid properties. The main distinguishing feature of these data sets is the high values of G and We which make their selection appropriate for the present study. According to the values of We listed in table 1, sets nos. 2, 4 and 5 are predominantly in the mist flow region, set no. 3 is mainly in the annular flow region, while set no. 1 evenly covers both regions. Because of these favorable features, it is expected that the present data base (while certainly not containing all the published data) would provide an appropriate test on the findings of the present study.

3.2 *Comparisons with annular flow correlations*

An exhaustive review of all available correlations would be a very worthwhile exercise, however, the required effort is immense and probably is beyond the objectives of this investigation. Instead, attention was focussed on two of the leading correlations, and based on the results of other comparisons reported in the literature, the conclusions drawn later are expected to be valid.

A well-known model was developed by Akers *et al.* (1958) who argued that during annular flow the heat transfer coefficient is determined by the flow character of the liquid film, which is mainly influenced by the interaction between the vapor core and the liquid film. By replacing the vapor core by a liquid flow which would produce the same shear stress they arrived at the following correlation:

$$\mathrm{Nu} = 0.0265\,\mathrm{Pr}_L^{1/3}\,\mathrm{Re}_{eq}^{0.8} \quad , \qquad \mathrm{Re}_{eq} > 5 \times 10^4 \qquad [4a]$$

and

$$\mathrm{Nu} = 5.03\mathrm{Pr}_L^{1/3}\,\mathrm{Re}_{eq}^{1/3} \quad , \qquad \mathrm{Re}_{eq} < 5 \times 10^4 \quad , \qquad [4b]$$

where Nu is Nusselt number, Pr_L the liquid Prandtl number and the equivalent Reynolds number Re_{eq} is given by

$$\mathrm{Re}_{eq} = \mathrm{Re}_{Ls} + \mathrm{Re}_{Gs}\left(\frac{\mu_L}{\mu_G}\right)\left(\frac{\rho_L}{\rho_G}\right)^{0.5} \quad . \qquad [5]$$

Equations [4a] and [4b] are analogous to single-phase heat transfer correlations, and it must be noted that the saturation to wall temperature difference ΔT is not a parameter. This correlation received mixed assessment from different investigators working with the same test fluids but different flow rates, however, its use is recommended by the *ASHRAE Fundamentals Handbook* (1981).

A comparison between [4a] and [4b] and the heat-transfer data base is shown in figure 2 using We as the main parameter in the comparison. Values of We were calculated from [2] and it is understood that the data base of table 1 has a strong bias towards mist flow. It appears from the results of figure 2 that the accuracy of prediction for all fluids is quite reasonable for We < 20 (annular flow according to visual observations), while for We > 30 (mist flow) the deviation is generally high and can reach 50%. The trends displayed in figure 2 do not seem to be related to tube orientation or fluid properties but rather to a departure from the flow conditions on which [4a] and [4b] were based.

A more rigorous analysis was reported by Traviss *et al.* (1971) who applied the momentum and heat transfer analogy to an annular flow model with all the liquid phase flowing in a turbulent film of uniform thickness. The von Karman universal velocity profile was used in the film as well as the Lockhart & Martinelli method for correlating the frictional pressure gradient. The resulting correlation was given by

$$\mathrm{Nu} = \mathrm{Pr}_L\mathrm{Re}_{Ls}^{0.9}F(X_{tt})/F_2 \quad , \qquad [6]$$

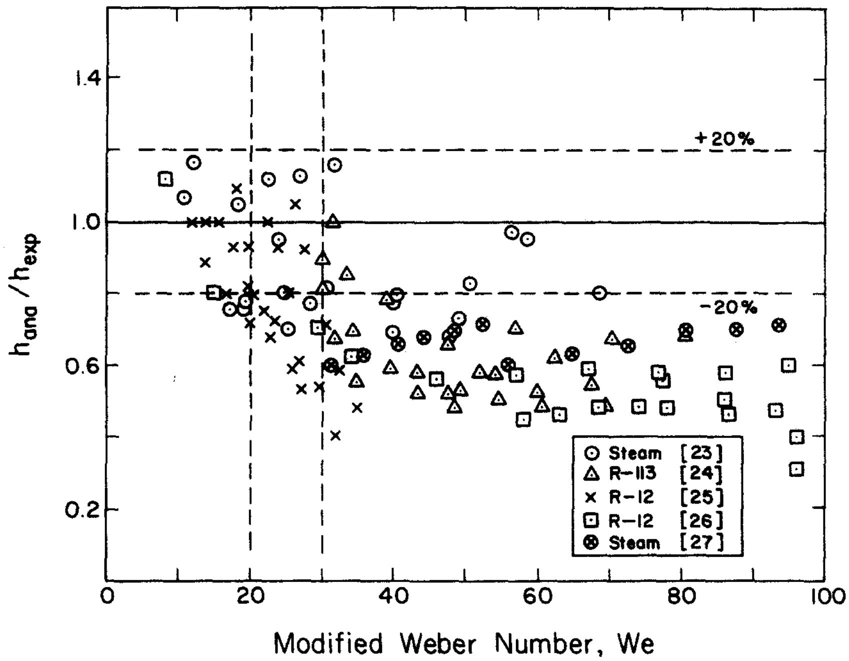

Figure 2. Comparison between the heat-transfer data base and the correlation of Akers *et al.* (1958).

where

$$F(X_{tt}) = 0.15[1/X_{tt} + 2.85/X_{tt}^{0.476}] \quad , \qquad [7a]$$

$$F_2 = 5\,\mathrm{Pr}_L + 5\ln(1 + 5\,\mathrm{Pr}_L) + 2.5\ln(0.00313\,\mathrm{Re}_{Ls}^{0.812}) \quad , \qquad \mathrm{Re}_{Ls} > 1125 \qquad [7b]$$

and

$$F_2 = 5\,\mathrm{Pr}_L + 5\ln[1 + \mathrm{Pr}_L(0.09636\,\mathrm{Re}_{Ls}^{0.585} - 1)] \quad , \qquad \mathrm{Re}_{Ls} < 1125 \quad . \qquad [7c]$$

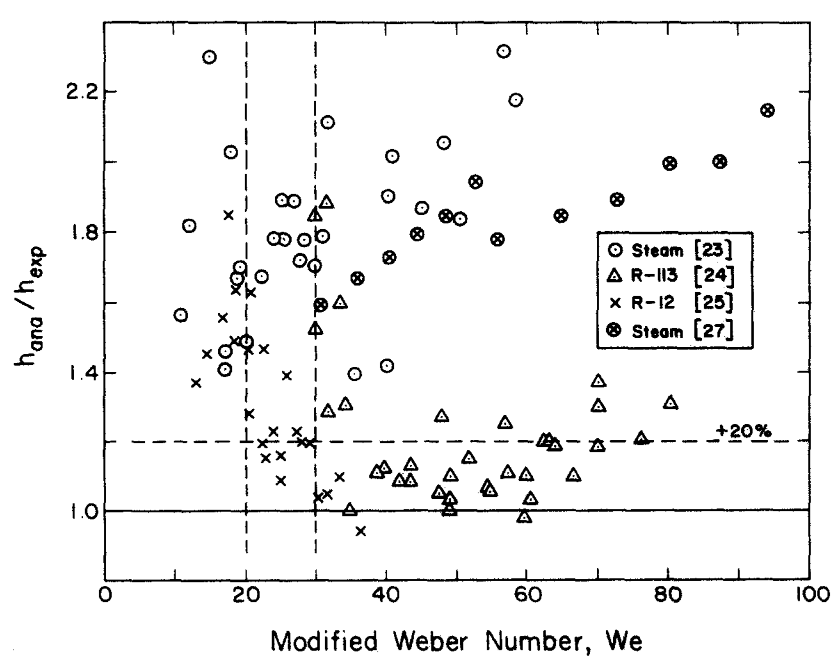

Figure 3. Comparison between the heat-transfer data base and the correlation of Traviss *et al.* (1971).

When comparing [6] with their own data (data set no. 4 of table 1) Traviss *et al.* (1971) found that the data was underpredicted by [6]. Consequently, [6] was limited to the range $F(X_{tt}) < 2$ and the following correlation was proposed for the range $F(X_{tt}) > 2$:

$$\mathrm{Nu} = \mathrm{Pr}_L \mathrm{Re}_{LS}^{0.9}\,[F(X_{tt})]^{1.15}/F_2 \quad . \qquad [8]$$

A comparison between the predictions of this model and the data base of table 1 is shown in figure 3. Data set no. 4 is not shown since good agreement is a foregone conclusion. Figure 3 shows good agreement with data sets no. 2 and 3 in the mist region, but clearly the agreement is poor with the sets no. 1 and 5. This may be attributed to property effects, while tube orientation does not seem to be a factor. It is interesting to note that the present comparison showing correlations [6] and [8], which were developed from R-12 data (set no. 4), to seriously overpredict the steam data seem to be in line with an earlier comparison by Traviss *et al.* (1971) in which it was found that a correlation by Boyko & Kruzhilin (1967) developed from steam data seriously underpredicts the R-12 data. Both correlations were developed from data which are believed (based on the values of We) to be predominantly in the mist region where the heat transfer mechanism may be different from that described by annular flow models assuming no entrainment.

4. PROPOSED CORRELATION

As a starting point, it is assumed that in the mist region the vapor phase and the bulk of the liquid phase are flowing as a homogeneous mixture in thermodynamic equilibrium with all heat transfer taking place at the wall. This assumption is supported by visual observations of the flow situation. Because of heat transfer, condensate forms at the wall and is mostly blown off by the high velocity vapor leaving behind very thin ridges of liquid with alternate dry spots. The pattern of this broken film and frequency of dry spots are determined by the wall shear stress induced by the flowing mixture. Further condensation occurs mostly in the dry spots at a rate which would be influenced by the temperature difference between the flowing mixture and the wall. This mechanism (based on the interaction between the mixture and the wall, and the saturation to wall temperature difference) would result in heat transfer rates considerably higher than those predicted on the basis of a stable thin film covering the tube wall.

Analogy between momentum and heat transfer in single-phase turbulent flow resulted in correlations of the form,

$$\mathrm{Nu} = C\,\mathrm{Re}^{0.875}\,\mathrm{Pr}^{1/3} \quad ,$$

where Re is the Reynolds number, Pr the Prandtl number and C a correlating coefficient. Treating the vapor–liquid mixture as a single-phase homogeneous stream, the influence of wall–mixture interaction on the overall heat transfer was explored by calculating $\mathrm{Nu}\cdot\mathrm{Re}_m^{-0.9}\cdot\mathrm{Pr}_G^{-1/3}$ for all the data points of the five data sets of table 1, where Nu is the experimental Nusselt number, Pr_G is Prandtl number of vapor, Re_m the misture Reynolds number defined as

$$\mathrm{Re}_m = GD/\mu_m \qquad [9]$$

and μ_m the homogeneous viscosity given by

$$1/\mu_m = x/\mu_G + (1-x)/\mu_L \quad . \qquad [10]$$

A representative sample of the behavior of $\mathrm{Nu}\cdot\mathrm{Re}_m^{-0.9}\cdot\mathrm{Pr}_G^{-1/3}$ against We for some runs is shown in figure 4. For all test runs irrespective of fluid properties or tube orientation the quantity $\mathrm{Nu}\cdot\mathrm{Re}_m^{-0.9}\cdot\mathrm{Pr}_G^{-1/3}$ assumed an almost constant value for We > 35 and deviated considerably at lower We. The deviation may go towards higher or lower values depending on fluid properties and quality at the mist–annular transition. For each data set, the runs

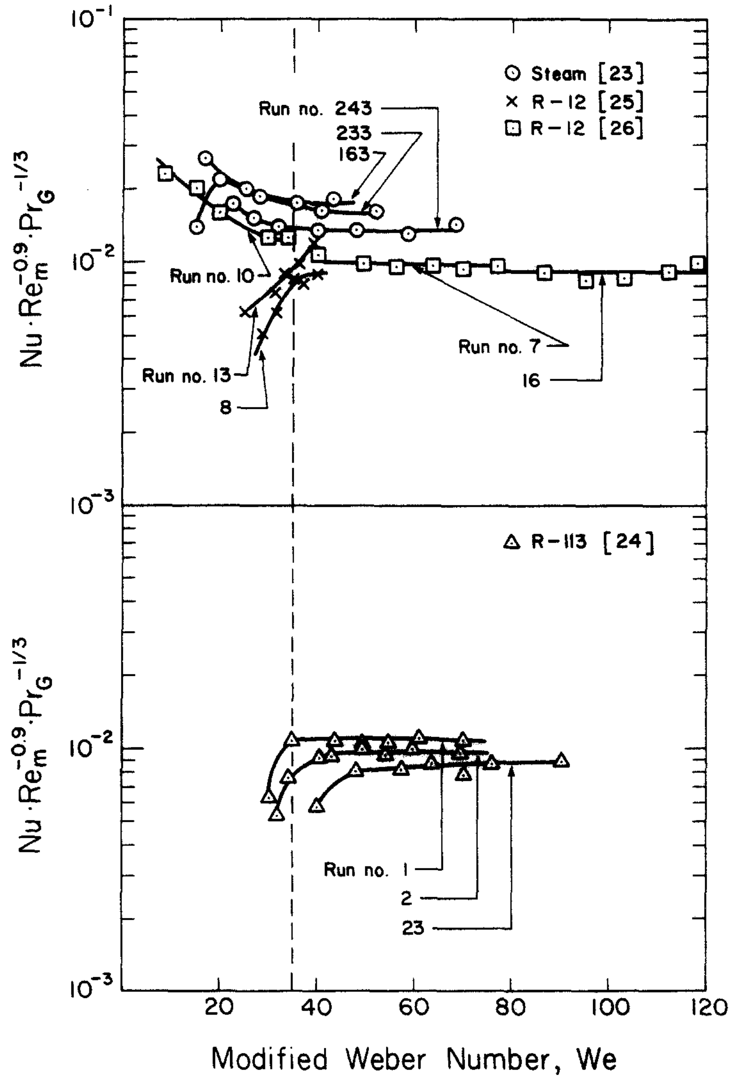

Figure 4. Behavior of the product Nu · $Re_m^{-0.9}$ · $Pr_G^{-1/3}$ along the condensation path.

shown in figure 4 correspond to the maximum, minimum and an intermediate flow rate, and it is clear that the results of each data set occupy a narrow band. Data set no. 5 is not shown because these results were reported graphically without providing values for the saturation to wall temperature difference ΔT, which are required later.

In order to isolate the influence of ΔT on the heat transfer coefficient, it would be desirable to compare results of the same fluid, same saturation temperature T_s and same total mass flux G, but different ΔT. This was not possible to do in a rigorous manner since G was the main parameter which was varied from one run to another within each set. Changing ΔT while keeping all other flow parameters unchanged requires the ability to control the heat flux which is difficult to do for long condensers. The only study in which ΔT was controlled involved a very short condenser (Akers & Rosson 1959), and it was found that Nu is proportional to $\Delta T^{-1/6}$ at conditions of low liquid loading. In another study involving high velocity refrigerant R-22 condensing inside a horizontal tube, Altman *et al.* (1959) found that the substantial deviation between the experimental and analytical heat transfer coefficients at high qualities can be adjusted by a correction factor involving $\Delta T^{-0.52}$, while the amount of inlet superheat was irrelevant.

The influence of ΔT on the heat transfer coefficient for the present data base is illustrated in the two parts of figure 5. In this figure the product Nu · $Re_m^{-0.9}$ · $Pr_G^{-1/3}$ is plotted against (C_{pG} · $\Delta T/h_{LG}$) for all the data points of sets no. 1 to 4 corresponding to We > 35, where C_{pG} is the specific heat of vapor and h_{LG} is the latent heat. Within each data set, where variations in C_{pG} and h_{LG} are small, it can be seen that Nu · $Re_m^{-0.9}$ · $Pr_G^{-1/3}$ has a tendency

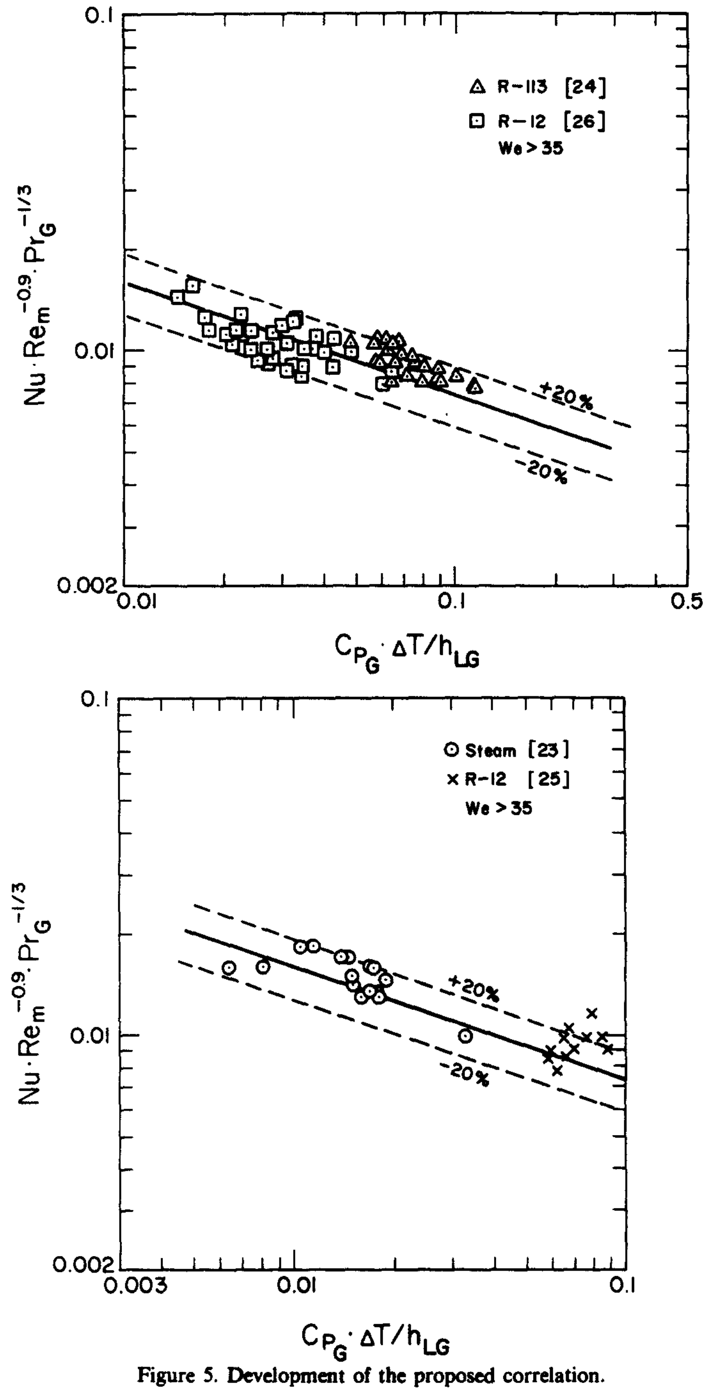

Figure 5. Development of the proposed correlation.

to increase as ΔT decreases. A slope of $-1/3$ provides a reasonable fit with the present data of different fluids and different tube orientations. Figure 5 indicates that the heat-transfer data base of table 1 agree within $\pm$ 20% with the following correlations:

$$\mathrm{Nu} = 0.00345\,\mathrm{Re}_m^{0.9}\mathrm{Pr}_G^{1/3}(C_{P_G} \cdot \Delta T/h_{LG})^{-1/3}\ , \qquad \mathrm{We} > 35\ , \qquad [11]$$

which can be reduced to

$$\mathrm{Nu} = 0.00345\,\mathrm{Re}_m^{0.9}\left(\frac{\mu_G h_{LG}}{k_G \Delta T}\right)^{1/3}, \qquad \mathrm{We} > 35\ . \qquad [12]$$

where k_G is the thermal conductivity of vapor.

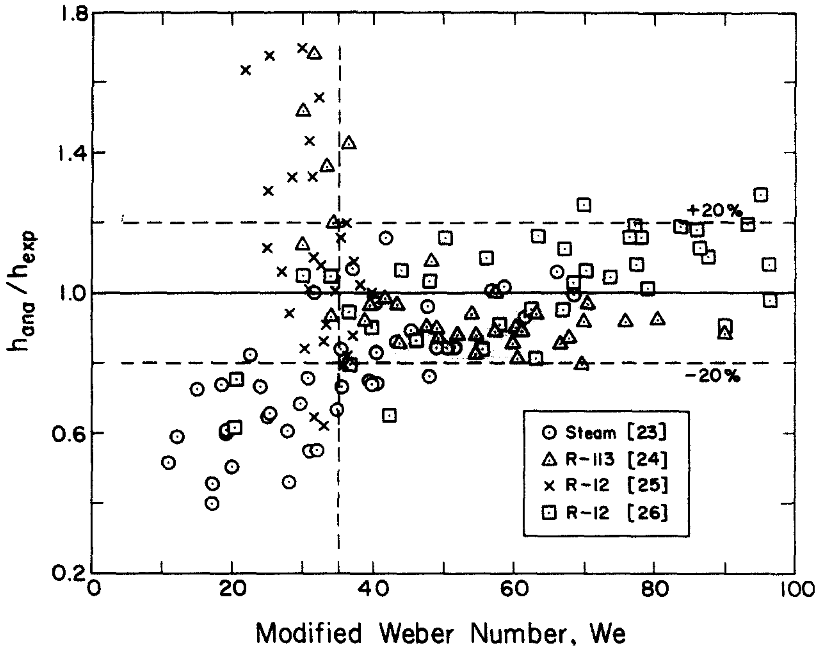

Figure 6. Comparison between the heat-transfer data base and the proposed correlation.

Figure 6 was prepared to illustrate that correlation [12] should not be used in the annular region (We < 30). Substantial deviations can be seen there which are attributed to the establishment of a stable liquid film and consequently a departure from the heat transfer mechanism on which correlation [12] is based.

5. CONCLUDING REMARKS

Correlations of heat transfer during annular flow condensation are normally given in the literature without specification of upper limits for their application. It was reported by several investigators, and also demonstrated here, that these correlations would substantially underpredict the experimental heat transfer coefficients at high qualities and/or high mas fluxes. This was correctly attributed to excessive entrainment and break down of the liquid film (mist flow conditions). It is shown here that a dimensionless parameter We, derived earlier from a theoretical consideration of the mist–annular transition (Soliman 1983), has relevance to heat transfer and that We = 30 may serve as an upper limit for models based on the assumption of annular flow.

A new correlation based on the mixture–wall interaction and saturation to wall temperature difference is developed. This correlation is limited to the mist region and would result in severe errors if applied outside this region. Also, in view of the fact that the exponents and coefficients of this correlation were obtained empirically not as a result of rigorous analysis, the accuracy of the present correlation is ensured only for conditions encompassed by the heat-transfer data base of table 1.

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NOMENCLATURE

C_{p_G} specific heat of the vapor phase
D tube diameter
G total mass flux
h heat transfer coefficient in figures 2, 3 and 6
h_{LG} latent heat

Nu Nusselt number
Pr_L Prandtl number of the liquid phase
Pr_G Prandtl number of the vapor phase
Re_{eq} equivalent Reynolds number
Re_{Ls} superficial liquid Reynolds number
Re_m Reynolds number of the mixture
Re_{GS} superficial vapor Reynolds number
ΔT saturation to wall temperature difference
V_L average velocity of the liquid phase
V_G average velocity of the vapor phase
We modified Weber number
X_{tt} Lockhart–Martinelli parameter
x quality
δ average thickness of the liquid film
μ_L liquid dynamic viscosity
μ_m mixture dynamic viscosity
μ_G vapor dymanic viscosity
ρ_L liquid density
ρ_G vapor density
σ surface tension
ϕ_G two-phase pressure drop multiplier

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