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A void fraction model for annular two-phase flow

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Abstract—An analytical model has been developed for predicting void fraction in two-phase annular flow. In the analysis, the Lockhart–Martinelli method has been used to calculate two-phase frictional pressure drop and von Karman's universal velocity profile is used to represent the velocity distribution in the annular liquid film. Void fractions predicted by the proposed model are generally in good agreement with available experimental data. This model appears to be as good as Smith's correlation and better than the Wallis and Zivi correlations for computing void fraction.

1. INTRODUCTION

VOID fraction is one of the fundamental quantities necessary to describe the flow characteristics of a two-phase mixture. The value of void fraction is required for the computation of acceleration and hydrostatic pressure drops, mean fluid density, etc. Therefore, prediction of the void fraction as a function of the design and operating parameters of the heat exchanger (geometry, working pressure, fluid flow rate, flow pattern, thermodynamic and transport properties of the two-phase, etc.) is of considerable importance.

Several attempts have been made in the past to develop methods for the determination of void fraction in two-phase flow. Semiempirical and empirical correlations based on experimental data have been suggested by a number of investigators [1–10].

From the design point of view annular flow is particularly important. For a wide range of pressure and flow conditions, annular flow is observed over a large vapour quality range during condensation and boiling inside tubes [11].

2. PREVIOUS WORK

Many investigations have correlated void fraction with quality for annular flow [12–15]. Among these, the correlation of Zivi [1] is most widely used. His equation, which is based on the principle of minimum entropy production, is given below:

$$\alpha = 1\Big/\left[1+\left(\frac{1-x}{x}\right)(\rho_V/\rho_L)^{2/3}\right]. \qquad (1)$$

However, Zivi's analysis is based on two hypothetical conditions: namely (i) the principle of minimum entropy production is applicable to a turbulent two-phase flow and (ii) the wall shear stress is negligible (which is far from reality in actual two-phase flows). It may be noted that the wall shear is an extremely important parameter characterizing two-phase flow. Moreover, expression for entropy production rates are not available for turbulent flows [1].

Wallis [2] proposed a simple theory for annular two-phase flow in terms of equations for interfacial and wall shear stress which was based on two simplified conditions: namely (i) there is no liquid entrainment and (ii) the liquid film velocity is low compared with the velocity of the gas core. He proposed separate correlations for upward cocurrent and horizontal annular flow. For vertical annular flow the following correlations were proposed:

$$\Delta P^* = 10^{-2} j_V^* \left[\frac{1+75(1-\alpha)}{\alpha^{5/2}}\right] \qquad (2a)$$

turbulent flow:

$$\Delta P^* = (1-\alpha)+10^{-2}\frac{j_L^*|j_L^*|}{(1-\alpha)^2} \qquad (2b)$$

laminar flow:

$$\Delta P^* = \frac{j_L^*}{(1-\alpha)^2}+0.684(1-\alpha). \qquad (2c)$$

For turbulent flow equation (2a) and (2b) have to be solved simultaneously to evaluate α and ΔP^*; similarly, for laminar flow, equation (2a) and (2c) have to be solved simultaneously.

The laminar film correlation was found to be close to the data up to a value of the liquid Reynolds number of about 3000, and thereafter, the turbulent film line followed the data.

For horizontal flow (either laminar or turbulent) the following equation was proposed:

$$X^2 = \frac{(1-\alpha)^2[1+75(1-\alpha)]}{\alpha^{5/2}}. \qquad (3)$$

Subsequently, Wallis [3] modified his simple theory to include the effects of liquid entrainment, the relative velocity between liquid film and gas core and several other effects. However, the effects of these modifications were found to be secondary.

Smith developed a model based on equal velocity heads of the homogeneous mixture core and the annulus liquid phase. The void fraction is given by

NOMENCLATURE

A	tube inside flow area [m^2]
A_V	cross-sectional area of vapour core [m^2]
$\bar{d}$	mean deviation
D	tube inside diameter [m]
$F(X_{tt})$	function of Lockhart–Martinelli parameter defined by equation (20)
g	acceleration due to gravity [$m\ s^{-2}$]
g_c	gravitational constant [$N\ m^2\ kg^{-2}$]
G	total mass flux [$kg\ s^{-1}\ m^{-2}$]
j	volumetric flux [$m\ s^{-1}$]
j_V^*	dimensionless turbulent vapour flux, $(j_V\rho_V^{1/2})/\{[gD(\rho_L-\rho_V)]^{1/2}\}$
j_L^*	dimensionless turbulent liquid flux, $(j_L\rho_L^{1/2})/\{[gD(\rho_L-\rho_V)]^{1/2}\}$
$j_L'^*$	dimensionless laminar liquid flux, $(32j_L\mu_L)/[D^2g(\rho_L-\rho_V)]$
ΔP^*	dimensionless pressure drop, defined by equation (2)
(dP/dZ)	pressure gradient [$N\ m^{-2}\ m^{-1}$]
r_0	inner radius of condenser tube [m]
r_0^+	dimensionless radius defined by equation [9]
Re_L	liquid Reynolds number
u	axial velocity [$m\ s^{-1}$]
u^+	dimensionless axial velocity, u_L/V^*
V^*	friction velocity defined by equation (10) [$m\ s^{-1}$]
W_V	vapour refrigerant flow rate [$kg\ s^{-1}$]
x	vapour quality
X	Lockhart–Martinelli parameter, $[(dP/dZ)_L/(dP/dZ)_V]^{1/2}$
y	distance normal to tube wall [m]
y^+	dimensionless radial distance, yV^*/ν_L
z	axial distance [m].

Greek symbols

α	void fraction
ρ	density [$kg\ m^{-3}$]
μ	dynamic viscosity [$kg\ s^{-1}\ m^{-1}$]
ν	kinematic viscosity [$m^2\ s^{-1}$]
δ	liquid film thickness [m]
δ^+	dimensionless liquid film thickness defined by equation (8)
ϕ_V	parameter defined by equation (15)
τ_w	shear stress at the tube well [$N\ m^{-2}$]
σ	standard deviation.

Subscripts

L	liquid phase
V	vapour phase
TPF	two-phase friction
tt	turbulent liquid turbulent vapour.

equation (4)

$$\alpha = \left(1+(\rho_V/\rho_L)\left(\frac{1-x}{x}\right) \times \left\{K+(1-K)\sqrt{\frac{(\rho_L/\rho_V)+K\left(\frac{1-x}{x}\right)}{1+K\left(\frac{1-x}{x}\right)}}\right\}\right)^{-1}. \tag{4}$$

Smith found equation (4) to be in excellent agreement with experimental data with $K = 0.4$ [4].

The present paper describes the development of a new but simple model for void fraction prediction during annular two-phase flow. In the light of the above discussion, the comparison of the proposed model is restricted to the much cited Zivi's [1] correlation, the general correlation of Smith [4], and the Wallis [2] analysis based on the simple theory.

3. FLOW MODEL

The proposed analysis is based on the physical model shown in Fig. 1. The annular flow is considered to be characterized by an axisymmetric liquid annulus and a vapour core with no liquid entrainment. The flow is assumed to be steady and one-dimensional and that no radial pressure gradient exists. In addition, both liquid and vapour flows are considered to be turbulent and that at any section both the phases have constant properties corresponding to the saturated state.

In annular flow at very high vapour flow rates, entrainment of liquid droplets in the vapour core may occur. The computation of void fraction with liquid entrainment requires the knowledge of the fraction of liquid entrained in the gas core. Some correlations have been suggested in literature for estimation of liquid entrainment. But, as pointed out by Hewitt and Hall-Taylor [16], their range of applicability is severely

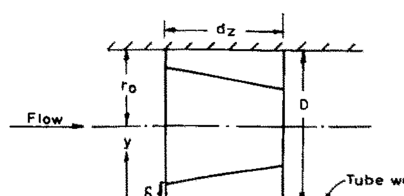

FIG. 1. Coordinate system of two-phase annular model.

limited. Moreover, the fraction of liquid entrained is likely to be very small in many cases [16–22]. It may be noted that the assumptions of no liquid entrainment and smooth liquid–vapour interface have been used in several condensation analysis [12–15] as well as in other investigations [8, 23]. Wallis [2] observed that "for many engineering problems added complexity is not worth while and the simple theory is quite adequate". It is, nevertheless, recognized that allowance for the above-mentioned assumptions made in the model, if incorporated, will slightly reduce the discrepancy between the experimental data and the analysis. However, as discussed in a later section, the proposed simplified analysis has been found to give an agreement within $\pm 15\%$ with the experimental values. This is quite acceptable for many engineering design calculations.

The void fraction α is defined as

$$\alpha = A_V/A. \tag{5}$$

Referring to Fig. 1

$$A_V = \pi(r_0-\delta)^2 \quad \text{and} \quad A = \pi r_0^2$$

$$\text{thus } \alpha = [1-(\delta/r_0)]^2. \tag{6}$$

Writing equation (6) in terms of dimensionless parameters

$$\alpha = [1-(\delta^+/r_0^+)]^2 \tag{7}$$

where

$$\delta^+ = \delta V^*/\nu_L \tag{8}$$

$$r_0^+ = r_0 V^*/\nu_L \tag{9}$$

and

$$V^* = [(g_c\tau_w)/\rho_L]^{1/2}. \tag{10}$$

The determination of dimensionless liquid layer thickness δ^+ in terms of liquid Reynolds number Re_L follows the procedure of Bae *et al.* [13] and is briefly outlined below.

The von Karman universal velocity distribution for the liquid layer in non-dimensional form can be expressed as

$$u^+ = y^+ \qquad 0 < y^+ < 5 \tag{11a}$$

$$u^+ = -3.05+5 \ln y^+ \qquad 5 < y^+ < 30 \tag{11b}$$

$$u^+ = 5.5+2.5 \ln y^+ \qquad y^+ > 30. \tag{11c}$$

Using von Karman's velocity profile and continuity equation for the liquid mass flow rate, the inter-relationship between dimensionless liquid layer thickness and liquid Reynolds number can be expressed as [13]

$$Re_L = 2\delta^{+2} \qquad 0 < \delta^+ < 5 \tag{12a}$$

$$Re_L = 50-32.2\delta^+ +20\delta^+ \ln \delta^+ \qquad 5 < \delta^+ < 30 \tag{12b}$$

$$Re_L = -256+12\delta^+ +10\delta^+ \ln \delta^+ \qquad \delta^+ > 30. \tag{12c}$$

Traviss *et al.* [15] approximated equation (12a)–(12c) by the following equations with an error of less than 4%.

$$\delta^+ = 0.707\, Re_L^{0.5} \qquad 0 < Re_L < 50 \tag{13a}$$

$$\delta^+ = 0.482\, Re_L^{0.585} \qquad 50 < Re_L < 1125 \tag{13b}$$

$$\delta^+ = 0.095\, Re_L^{0.812} \qquad Re_L > 1125. \tag{13c}$$

The wall shear stress for annular flow can be expressed in terms of two-phase frictional pressure drop as

$$\tau_w = -\frac{D}{4}(\mathrm{d}P/\mathrm{d}Z)_{TPF}. \tag{14}$$

Dukler *et al.* [24] critically compared the predictions of Lockhart–Martinelli [25] correlation for two-phase frictional pressure drop with several other correlations in literature and concluded that it shows the best agreement with experimental data on pressure drop. This correlation, though based on adiabatic mixture data, has been extensively used for evaluating two-phase frictional pressure drop for phase change processes also [12, 13, 15, 26]. The Lockhart–Martinelli [25] method is adopted for evaluation of two-phase frictional pressure drop.

Soliman *et al.* [12] approximated the data of Lockhart–Martinelli [25] over the range of $0 < X_{tt} \leqslant 1$ within $\pm 5\%$ by the following relationship:

$$\phi_V = 1+2.85\, X_{tt}^{0.523} \tag{15}$$

where

$$X_{tt} = (\mu_L/\mu_V)^{0.1}\left(\frac{1-x}{x}\right)^{0.9}(\rho_V/\rho_L)^{0.5}. \tag{16}$$

Following the Lockhart–Martinelli method the two-phase frictional pressure gradient is related to the pressure gradient for vapour only by

$$(\mathrm{d}P/\mathrm{d}Z)_{TPF} = \phi_V^2\,(\mathrm{d}P/\mathrm{d}Z)_V. \tag{17}$$

The pressure drop in the turbulent vapour as given in reference [25] may be expressed as

$$(\mathrm{d}P/\mathrm{d}Z)_V = -0.143(\mu_V^{0.2} W_V^{1.8}/g_c\rho_V D^{4.8}). \tag{18}$$

Using equations (10) and (14)–(18) the friction velocity V^* can be expressed as [26]

$$V^* = (\nu_L/D)Re_L^{0.9}[F(X_{tt})] \tag{19}$$

where

$$F(X_{tt}) = 0.15[X_{tt}^{-1}+2.85X_{tt}^{-0.476}]. \tag{20}$$

Combining equations (7), (9) and (19) the expression for void fraction for annular flow can be obtained as

$$\alpha = \{1-4\delta^+ Re_L^{-0.9}[F(X_{tt})]^{-1} + 4\delta^{+2} Re_L^{-1.8}[F(X_{tt})]^{-2}\}. \tag{21}$$

In the above analysis it was assumed that both liquid and vapour flows are turbulent. This is a reasonable assumption. Carpenter and Colburn [27] have reported that at high vapour velocity, the transition from laminar to turbulent flow may occur at a liquid Reynolds number as low as 240. Substituting equation (13b) and (13c) into equation (21) the expression for α in annular flow is given by the following equations:

$$\alpha = \{1 - 1.928\, Re_L^{-0.315}[F(X_{tt})]^{-1} + 0.9293\, Re_L^{-0.63}[F(X_{tt})]^{-2}\} \quad 50 < Re_L < 1125 \tag{22a}$$

$$\alpha = \{1 - 0.38\, Re_L^{-0.088}[F(X_{tt})]^{-1} + 0.0361\, Re_L^{-0.176}[F(X_{tt})]^{-2}\} \quad Re_L > 1125. \tag{22b}$$

4. VERIFICATION OF PROPOSED MODEL

In order to demonstrate the validity of the present model, it has been compared with the accessible experimental data of Isbin *et al.* [28] and Rouhani and Becker [29].

Figure 2 shows the comparison of the proposed analysis with the data of Isbin *et al.* [28], along with the predictions of Zivi [1], Wallis [2] and Smith [4]. The data of Isbin *et al.* were obtained by using the gamma ray absorption technique for steam–water during annular flow in a 22 mm i.d. vertical tube. As is apparent from Fig. 2 the data of Isbin *et al.* show only a very small effect of velocity on void fraction values. However, equations (22a) and (22b) predicted a still smaller effect. Prediction of void fraction from proposed analysis for high and medium flow rates, i.e. at mass flow rates of 0.3184 and 0.2386 kg s^{-1} could be correlated by a single curve. The predicted void fraction for low mass flow rate has been shown by a separate curve. The agreement of the proposed model with the experimental data is satisfactory except at very low vapour quality ($x < 0.008$), even though Smith's correlation continues to give a slightly better agreement. However, the agreement of Zivi's [1] correlation is rather unsatisfactory with the bulk of the data ($x < 0.02$). Wallis' [2] simple theory shows still larger deviation from the data for all three mass flow rates.

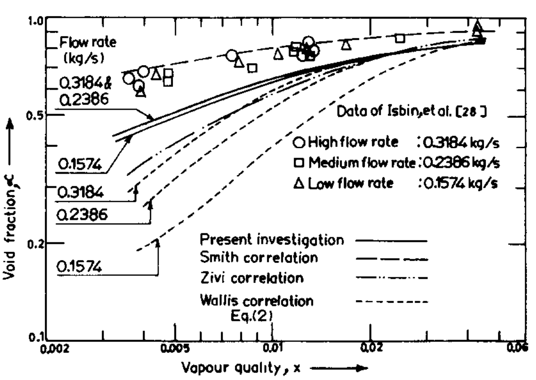

FIG. 2. Comparison of void fraction correlations with experimental data of reference [28].

The data of Rouhani and Becker [29] which were selected to evaluate the proposed model, pertained to the flow of boiling heavy water in a vertical round duct of 6.10 mm i.d. and a heated length of 2500 mm. The data covered the following ranges of variables:

Pressure	7×10^5–60×10^5 Pa
Vapour quality	0.0–0.41
Mass velocity	650–2050 kg s^{-1} m^{-2}
Void fraction	0.24–0.92
Surface heat flux	380–1200 kW m^{-2}.

It may be noted that the table of the data of Rouhani and Becker [29] does not identify the flow regime for each run. Isbin *et al.* [28] have reported data for

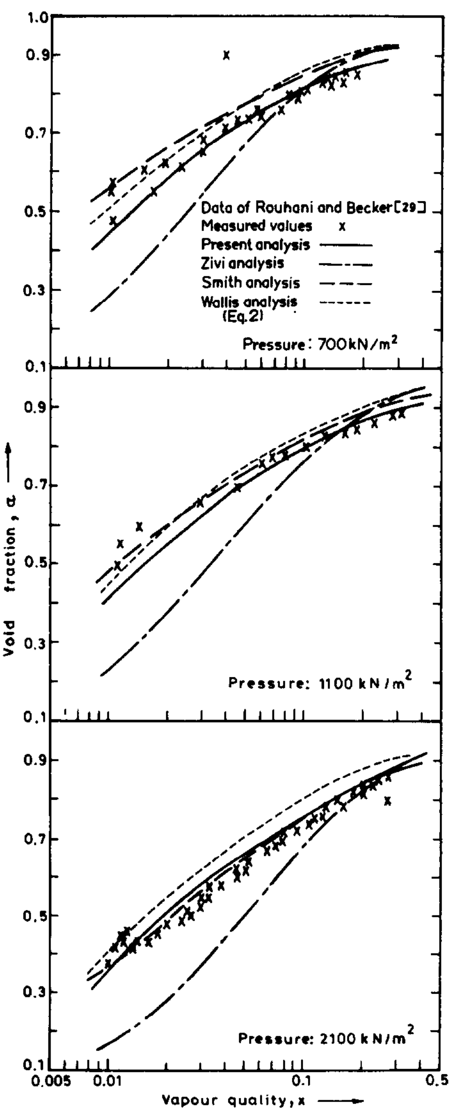

FIG. 3(a).

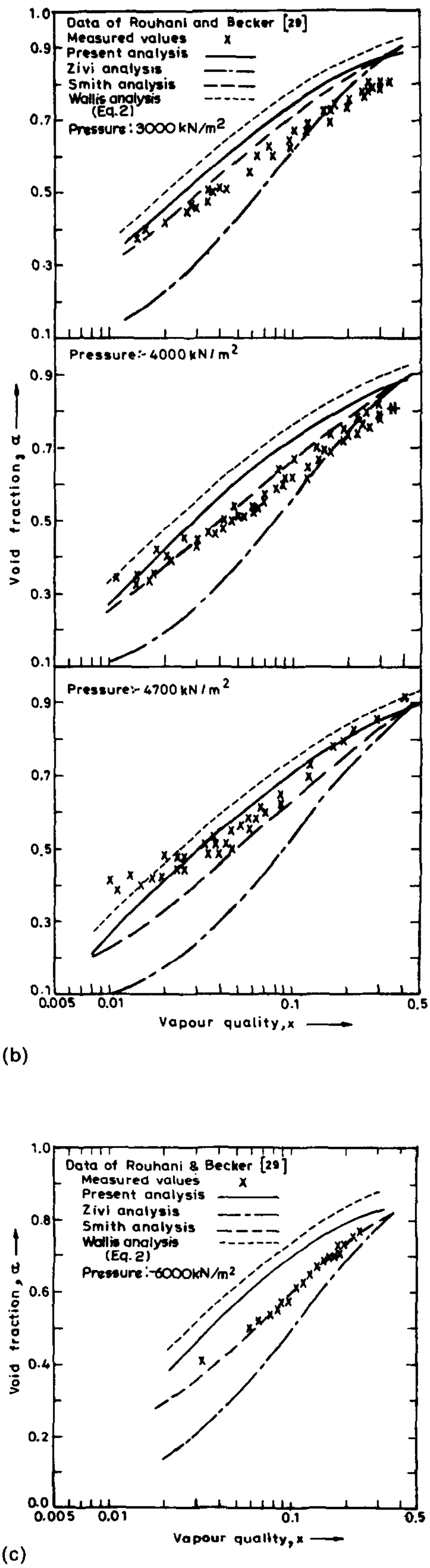

FIG. 3. (a)–(c) Comparison of void fraction correlations with experimental data of reference [29].

annular flow patterns in the case of steam at vapour quality as low as 0.005. It is, therefore, believed that a good number of void fraction data (for vapour quality above 0.01) must be belonging to the annular flow regime.

The comparison of the present model with those of Zivi [1], Smith [4], and Wallis [2] is shown in Fig. 3(a)–(c) for progressively increasing pressures for vapour quality above 0.01. The Wallis analysis, equation (2), and the proposed model, equation (22), express the void fraction as a function of test fluid mass velocity. However, the effect of mass velocity is found to be very small in the data range of Rouhani and Becker using the proposed model and that of Wallis. This is in accordance with the observation of Smith [4]. Therefore, the proposed and Wallis analyses are shown by a single curve for each pressure. Figure 3(a)–(c) reveal that, excepting for very high pressure (6000 kN m^{-2}), the agreement between the proposed model and the experimental data is, in general, very good. Smith's correlation is also in very good agreement, in fact slightly better than the proposed model, with experimental results. However, Wallis correlation shows relatively larger deviation from experimental data, as compared to the present analysis. Zivi's correlation was found to indicate still larger deviation from the data. It is apparent from Fig. 3(a)–(c) that the proposed analysis shows a definite improvement over the predictions of Zivi's [1] correlation.

Figure 4 shows an overall agreement of the present model with the data of Rouhani and Becker [29] above a vapour quality of 0.01. The predictions of the present analysis are, in general, within $\pm 15\%$ of the experimental values. Wallis' simple theory agreement with the data was found unsatisfactory, whereas Zivi's correlation showed marked deviation for void fractions below 0.70. However, Smith's correlation showed an agreement within $\pm 10\%$.

It is observed from Fig. 3(a)–(c) that pressure influences the degree of prediction of void fractions by the proposed model and Wallis theory. Figure 5(a) and (b) depict the experimental and calculated void

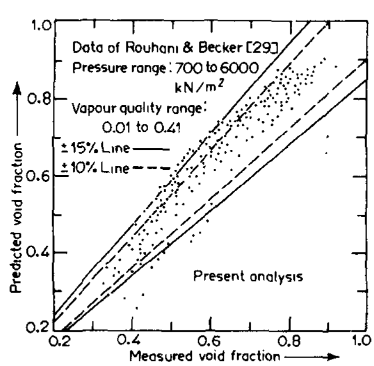

FIG. 4. Comparison of measured and predicted void fractions for pressure range 700–6000 kN m^{-2}.

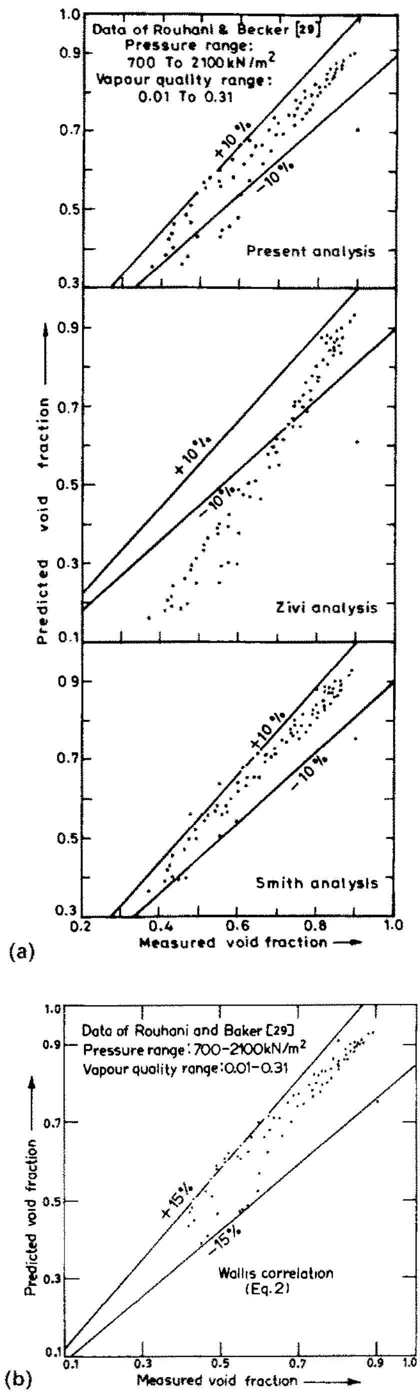

FIG. 5. (a), (b) Comparison of measured and predicted void fractions for pressure range 700–2100 kN m^{-2}.

fractions given by the four models for pressures up to 2100 kN m^{-2}. Up to this pressure, the present analysis shows an agreement within $\pm 10\%$ as in the case with Smith's correlation. Wallis' [2] theory compares within $\pm 15\%$ of the data. Even in this range, agreement of Zivi's correlation is unsatisfactory for void fractions below 0.70. The mean and standard deviation between the calculated and measured values of α for different pressure ranges are given in Table 1. It is apparent from this table that for all three pressure ranges, the mean deviation of the present model from the data is less than the Wallis theory which is in confirmity with Fig. 3(a)–(c). Smith's correlation gives the lowest value of mean deviation. It seems that the wide use of Zivi's [1] correlation, which gives the largest mean and standard deviations from experimental data, is perhaps due to its simplicity.

From the foregoing discussion some of the features of the proposed analysis may be summarized in Table 1.

The proposed model, successfully correlates the void fraction experimental data. The model represents a significant improvement over widely used Zivi's [1] model in that the effect of wall friction is accounted for. The proposed simplified analysis has the advantage (for annular flow regime) over the Smith's model in that it has been developed on detailed description of flow behaviour and momentum exchange processes during annular two-phase flow; whereas, as stated by Smith, his model does not describe the physical processes that actually occur in two-phase flow [4]. Further, the proposed model takes into account the effect of mass velocity, though very small, which has been noted by several investigators and not accounted by the correlations of Zivi and Smith. The prediction of proposed model appears to be better than the prediction of Wallis' [2] theory.

5. CONCLUSIONS

A simplified model has been developed for predicting void fraction in two-phase annular flow. This analysis is confirmed qualitatively and, in general, satisfactory quantitative agreement is also obtained. The following inferences may be drawn:

(1) The agreement of both the proposed model and Smith's correlation [4] with the data of Isbin *et al.* [28] is, in general, satisfactory and nearly similar. Comparison of these two models with the data of Rouhani and Becker [29] is very good.

Table 1. Comparison of accuracy of prediction between the proposed model and others

Pressure range (kN m^{-2})	Proposed model		Wallis' theory		Zivi's model		Smith's model		Number of data points
	$\bar{d}$	σ	$\bar{d}$	σ	$\bar{d}$	σ	$\bar{d}$	σ	
700–2100	0.16	7.86	−6.14	7.96	17.90	19.60	−3.04	5.23	79
3000–6000	−8.61	9.18	−15.81	8.23	21.49	21.39	1.13	10.61	150
700–6000	−5.58	9.68	−12.47	9.24	20.25	20.82	0.31	9.32	229

(2) The prediction of proposed model appears to be better than the predictions of Zivi's [1] correlation and Wallis [2] simple theory.

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UN MODELE DE FRACTION DE VIDE POUR UN ECOULEMENT ANNULAIRE DIPHASIQUE

Résumé—Un modèle analytique est développé pour prédire la fraction de vide dans un écoulement annulaire diphasique. Dans cette analyse, la méthode de Lockhart–Martinelli est utilisée pour calculer la chute de pression par frottement et le profil universel de Von Karman est utilisé pour représenter la distribution de vitesse dans le film liquide annulaire. Les fractions de vide calculées par ce modèle sont généralement en bon accord avec les données expérimentales disponibles. Ce modèle est aussi bon que la corrélation de Smith et meilleur que les corrélations de Wallis et Zivi pour calculer la fraction de vide.

EIN MODELL FÜR DEN DAMPFVOLUMENANTEIL EINER ZWEIPHASEN-RINGSTRÖMUNG

Zusammenfassung—Es wurde ein analytisches Modell zur Berechnung des Dampfvolumenanteils einer Zweiphasen-Ringströmung entwickelt. Die Lockhart–Martinelli-Methode wurde verwendet, um den Reibungsdruckverlust der Zweiphasenströmung zu berechnen. Mit dem universellen Geschwindigkeitsprofil von von-Karman wurde die Geschwindigkeitsverteilung im ringförmigen Flüssigkeitsfilm dargestellt. Dampfvolumenanteile, die mit dem vorgeschlagenen Modell berechnet werden, sind im allgemeinen in guter Übereinstimmung mit vorhandenen experimentellen Daten. Dieses Modell scheint so gut zu sein wie die Korrelation von Smith und besser als die Korrelationen von Wallis und Zivi für die Berechnung des Dampfvolumenanteils.

МЕТОД РАСЧЕТА ИСТИННОГО ОБЪЕМНОГО ПАРОСОДЕРЖАНИЯ КОЛЬЦЕВОГО ДВУХФАЗНОГО ПОТОКА

Аннотация—Разработана аналитическая модель расчета истинного объемного паросодержания двухфазного кольцевого потока. При анализе используется метод Локкарта–Мартинелли для расчета перепада давления в двухфазном потоке, а также универсальный профиль Кармана для описания распределения скорости в кольцевой пленке жидкости. Значения истинного объемного паросодержания, рассчитанные по предложенной модели, хорошо согласуются с имеющимися экспериментальными данными. Предлагаемая модель позволяет получать такие же по точности значения истинного объемного паросодержания, что и зависимость Смита, но более точные, чем те, которые рассчитываются по соотношениям Уоллиса и Циви.