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Comparative analysis of wall shear stress models to the drift-flux model applied to slug flow regime

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ABSTRACT

The one-dimensional drift-flux model efficiently predicts gas–liquid flows dominated by gravity force. The advantages of the drift-flux model applied to pipe flows are the absence of interfacial terms, well posedness and the reduced number of transport equations, but its weakness lays on the constitutive laws to predict the wall shear force of a gas–liquid mixture. Its success on upward vertical slug flows is, in part, due to the fact that for gravity dominated flows the friction contribution to the pressure gradient is usually small. In these applications the accuracy of the wall shear force model is not dominant. A challenging aspect is the application of the drift-flux model to the horizontal slug flows where the pressure gradient is due to friction force. The objective of this work is to develop a comparative analysis among wall shear stress models applied to the one-dimensional, steady state drift-flux approach applied to gas–liquid mixture flowing in the slug regime. Effective viscosity models based on the homogeneous and also on empirical propositions are employed. Additionally it is also introduced a mechanistic wall shear stress model. The effect of the use of distinct wall shear models into the drift-flux model is assessed by comparing the estimated pressure gradients against experimental data.

1. Introduction

The flow a gas and oil mixture challenges petroleum engineers to effectively monitor, control and optimize the oil production. The accomplishment of these tasks assures a successful and economical flow of hydrocarbon from the reservoir to the oil rigs. The flow assurance and the economical aspects are assessed mostly by the use of flow simulators which estimate the flow pattern, the phases' velocities, volumetric concentration, the pressure drop, heat transfer rate, mixture temperature among other flow properties. The accuracy of these estimates depends on the accuracy on the physical models built-in on the flow simulators.

This work addresses to the accuracy of the drift-flux model, one of the gas–liquid models frequently employed to steady and transient flow simulators. The drift-flux model is based on sound physical and mathematical principles (Drew and Passman, 1998) but, as many other models, it has more unknowns than equations demanding constitutive equations for closure. One weak point of this model is related to the constitutive equations for the viscous and turbulent stresses, which for one-dimensional models reduces to the wall friction forces. This is a complex issue since the stress tensor is represented as the contribution of the individual phases present in the mixture plus one extra term due to the interfacial forces. Manninen et al. (1996) discuss different approaches to this issue but there is no generalized theory. Usually the concept of a mixture viscosity or effective viscosity is adopted. This approach is largely accepted for dispersed flow regime where the phases are strongly coupled. But for separated flow regime, such as the stratified or the annular flows, the phases are weakly coupled and it is expected, beforehand, a poor performance of the drift-flux model.

The objective of this work is to assess the accuracy of the drift-flux model on predicting the pressure gradient for slug flow regime employing distinct wall shear stress models. The choice by the slug flow pattern is due to the application aspects and to the fundamental principles brought by this flow pattern. The application regards to the frequent occurrence of this pattern in crude oil pipelines. The fundamental principles consist of finding a suited wall friction model to capture the passage of liquid slugs trailed by elongated bubbles represented by alternating separated and dispersed phases flow pattern. This feature defines the wall shear stress models representation for having two flow patterns with a phase coupling shifting between strong and weak. Despite the fact that there were many attempts at developing various wall shear stress models for all kinds of gas–liquid flow patterns, it is not of the authors' knowledge that a comparative study of wall shear stress models performance for slug flow employing specifically the drift-flux model.

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The comparative wall shear stress analysis is developed using one dimensional, isothermal and steady state drift-flux model to estimate the pressure gradient on the upward vertical and horizontal slug flows. This analysis focus on the accuracy of the pressure gradient estimated by drift-flux model and guides the flow designer to a wall shear stress model selection according to his working scenario.

The work is structured as follows. The drift-flux equations and the wall friction models are defined in Sections 2 and 3. Sections 4 and 5 show the numerical algorithm and the experimental procedure. The data comparisons and conclusions are shown in Sections 6 and 7. The Appendix A describes the procedures to the closure equations to one of the wall friction models.

2. The gas–liquid drift-flux model

This section introduces the one-dimensional, isothermal and steady state drift-flux model. The model applies to a pipe with constant circular cross section with length, diameter and inclination with the horizontal represented by L, D and θ respectively. The definitions of the one-dimensional variables are introduced before the model for clarity and conciseness reasons.

2.1. Definitions of the flow variables and their kinematic relationships

The gas or the liquid phases are identified by index k which can be G or L. All variables express an average value at the pipe cross section here represented by A.

Gas phase fraction α is the area ratio between the pipe's section taken by gas phase, A_G, and the pipe cross section, A:

$$\alpha = A_G/A, \tag{1}$$

while the liquid holdup is simply $(1-\alpha)$.

Mixture density ρ is defined as the sum of the phase densities weighted by the void fraction:

$$\rho = \alpha\rho_G + (1-\alpha)\rho_L. \tag{2}$$

Phase velocity $(U_k)_\alpha$ is defined by the ratio between the volumetric flow rates of phase Q_k divided by the pipe cross section area taken by phase A_k:

$$(U_k)_\alpha = Q_k/A_k \equiv Q_k/(\alpha_k A), \quad k = \text{G or L}. \tag{3}$$

Phase superficial velocity J_k is the velocity of phase k as if it would take alone the whole pipe cross section:

$$J_k = Q_k/A \equiv \alpha_k (U_k)_\alpha, \quad k = \text{G or L}. \tag{4}$$

The mixture superficial velocity J is defined as the sum of the phase superficial velocities:

$$J = J_G + J_L \equiv \alpha(U_G)_\alpha + (1-\alpha)(U_L)_\alpha. \tag{5}$$

Drift kinematic law proposed by Zuber and Findlay (1965) establishes a linear relationship between the phase velocity, the mixture superficial velocity J and the local drift velocity $(V_{G,J})_\alpha$:

$$J_G/\alpha \equiv (U_G)_\alpha = C_0 J + (V_{G,J})_\alpha. \tag{6}$$

The distribution parameter C_0 is related to the cross section profiles of local values of J and α. The local drift velocity $(V_{G,J})_\alpha$ is a parameter dependent on the flow pattern, fluid transport properties and pipe size and inclination. The values of C_0 and $(V_{G,J})_\alpha$ are presented in Eq. (A.7) in the Appendix A.

2.2. The mass and momentum equations

The model applies to an isothermal gas–liquid mixture where the gas behaves as a real gas and the liquid phase is incompressible. There is no phase change and negligible surface tension effects, i.e., both phases share the same pressure. The phase mass conservation equations are

$$\frac{d}{dz}[\alpha\rho_G(U_G)_\alpha] = 0, \tag{7}$$

and

$$\frac{d}{dz}[(1-\alpha)\rho_L(U_L)_\alpha] = 0. \tag{8}$$

The mixture momentum equation arises by adding the phases' momentum (Pauchon et al., 1993; Pauchon and Dhulesia, 1994):

$$\frac{d}{dz}[\alpha\rho_G(U_G)^2_\alpha + (1-\alpha)\rho_L(U_L)^2_\alpha] = -\frac{dP}{dz} - \mathcal{T}_W - \rho g \sin\theta, \tag{9}$$

where P is the pressure, $\mathcal{T}_W$ is the wall friction force per unit volume, g is the gravitational acceleration and θ is the pipe inclination angle with the horizontal. The coordinate z is parallel to the pipe axis direction. The domain inlet and outlet are positioned at $z=0$ and $z=L$.

2.3. The model reduction

An inspection on Eqs. (7) (through 9) reveals a system of differential equations with five unknowns: α, $(U_G)_\alpha$, $(U_L)_\alpha$, P, and $\mathcal{T}_W$. The phases' transport properties ρ_L, μ_L and μ_G are known and considered as constants. The gas phase density is determined using the compressibility factor, Z, as

$$\rho_G = P/(Z R_G T_0), \tag{10}$$

where R_G is the specific gas constant and T_0 is the flow temperature. The pipe's length, diameter and inclination with the horizontal, L, D and θ respectively are also known. The boundary condition P_o and the input variables $J_{G,o}$ and $J_{L,o}$ are given at the pipe outlet, which represent, respectively, the pressure and the liquid and gas superficial velocities at the pipe's outlet. Since the flow is isothermal it is assumed that T_0 prevails over the whole flow domain.

Using the kinematic law, Eq. (6), the void fraction α is expressed as a function of the pressure only and of the values $J_{G,o}$, $J_{L,o}$ and P_o as

$$\alpha = \frac{J_{G,o}(P_o/P)}{C_0[J_{G,o}(P_o/P) + J_{L,o}] + (V_{G,J})_\alpha}, \tag{11}$$

where C_0 and $(V_{G,J})_\alpha$ are given in Eq. (A.7).

From Eqs. (7) and (8) it is possible to express the phases' velocities in terms of the void fraction and of the values $J_{G,o}$, $J_{L,o}$ and P_o as

$$(U_G)_\alpha = J_{G,o}(\rho_{G,o}/\rho_G)/\alpha \equiv J_{G,o}(P_o/P)/\alpha, \tag{12}$$

$$(U_L)_\alpha = J_{L,o}/(1-\alpha). \tag{13}$$

Furthermore, substituting Eq. (11) into Eqs. (12) and (13), turns the phases' velocities dependent of the pressure only. Inserting Eqs. (12) and (13) into Eq. (9) and isolating the pressure dependent terms turns out

$$\frac{d}{dz}\phi[P(z)] = -\mathcal{T}_W - \rho g \sin\theta, \tag{14}$$

where $\phi[P(z)]$ is the pressure dependent terms associated with the axial direction derivative:

$$\phi[P(z)] = P + \alpha\rho_G(U_G)^2_\alpha + (1-\alpha)\rho_L(U_L)^2_\alpha. \tag{15}$$

This procedure reduced the set of Eqs. (7) to (9) into Eq. (14) which is a single ordinary differential equation having the pressure as the dependent variable and the pipe axial distance, z, as the independent variable. While the second term of the RHS of Eq. (14) is readily evaluated once α is known the first term is the