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FRICTIONAL PRESSURE-DROP PREDICTION IN BUBBLY FLOW IN AN ANNULAR-DUCT USING DRIFT-FLUX MODEL

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Abstract. *Gas-liquid flows inside annular geometries are conditions often found in the petroleum industry, for example during the production process. In the last years, the interest in this kind of flow has increased substantially because of the fact that understanding the flow behavior in the wells is fundamental for the well logging and better downhole pressure control. It has been shown that frictional pressure-drop predictions using the 1D drift-flux model are relatively accurate for circular geometry. In this study, we use available experimental data of gas-liquid flow collected in an annular duct of real radial scale, for vertical and 45º inclination, to evaluate the drift-flux distribution parameter. We compare the pressure-drop predictions using the modified closure correlation with experimental data from literature for air-water and air-kerosene mixtures. We expect the proposed correlations to better capture the average spatial distribution of the gas over the cross section of the annulus and, therefore, an improvement of the directional well project, especially on horizontal stretch, where frictional component of the pressure drop is dominant.*

Keywords: *two-phase flow, gas-liquid flow, bubbly flow, pressure drop, drift-flux, annular duct*

1. INTRODUCTION

The oil and gas industry constantly relies on theoretical models and empirically adjusted correlations to predict working conditions during extraction and transport of product. Prediction of two-phase frictional pressure drop is a challenge that has generated extensive research and experimental studies during the last 70 years.

The modeling of flow in pipes has been adapted to other geometries through the concept of the hydraulic diameter. Caetano (1986) has shown that the error involved in predicting friction factor values in annulus configurations by applying the hydraulic diameter concept can vary between -40% to 50%, depending on the annulus pipe diameter ratio and the degree of eccentricity.

The drift-flux model has been experimentally evaluated with success to predict the pressure gradient (CARVALHO, 2013). However, the total frictional component does not seem well evaluated, and the good performance of the results is masked by the gravitational component of the pressure gradient, which is predominant.

Adjustments in the correlations of Hibiki and Ishii (2003a) and Petalas and Aziz (1998) for the distribution parameter are made with basis on data obtained by Caetano (1986) and Carvalho (2013), using vertical and inclined (45º) annular ducts with water-air and kerosene-air as working fluids. This adjustment allows the authors to obtain a better agreement between theoretical and experimental values. Their performances are then compared with other correlations.

2. ONE-DIMENSIONAL DRIFT FLUX MODEL

The drift-flux model is one of the most practical and accurate models for two phase flow. The model takes into account the relative motion between phases by a constitutive relation. It has been utilized to solve many engineering problems involving two-phase flow dynamics. The rational approach to obtain a one-dimensional model is to integrate the three-dimensional model over a cross-sectional area and then to introduce proper mean values (HIBIKI;ISHII, 2002).

The drift velocity of a gas phase, v_{gj}, is defined as the velocity of the gas phase, v_g, with respect to the volume center of the mixture, j:

$$v_{gj} = v_g - j = (1-\alpha)(v_g - v_f) = (1-\alpha)v_r \quad (1)$$

where α, v_f and v_r are the void fraction, the liquid velocity, and the relative velocity between phases, respectively. The void-fraction-weighted mean drift velocity is given by

$$\left\langle \frac{\alpha v_{gj}}{\alpha} \right\rangle = \left\langle \frac{\alpha v_g}{\alpha} \right\rangle - \left\langle \frac{\alpha j}{\alpha} \right\rangle = \left\langle \frac{j_g}{\alpha} \right\rangle - \left\langle \frac{\alpha j}{\alpha} \right\rangle \tag{2}$$

where a simple area average of a quantity, F, over the cross-sectional area, A, is defined by

$$\langle F \rangle = \frac{1}{A} \int_A F dA \tag{3}$$

The one-dimensional drift-flux model can be derived by recasting Eq. (2) as

$$\left\langle \frac{j_g}{\alpha} \right\rangle = \frac{\langle \alpha j \rangle}{\langle \alpha \rangle \langle j \rangle} \langle j \rangle + \frac{\langle \alpha v_{gj} \rangle}{\langle \alpha \rangle} = C_0 \langle j \rangle + V_{gj} \tag{4}$$

where C_0 and V_{gj} are the distribution parameter defined by Eq. (5) and the void-fraction-weighted mean drift velocity defined by Eq. (6), respectively.

$$C_0 = \frac{\langle \alpha j \rangle}{\langle \alpha \rangle \langle j \rangle} \tag{5}$$

$$V_{gj} = \frac{\langle \alpha v_{gj} \rangle}{\langle \alpha \rangle} \tag{6}$$

The void-fraction-weighted mean gas velocity and the cross sectional mean mixture volumetric flux are easily obtainable parameters in experiments. Therefore, Eq. (4) suggests a plot of $\langle j_g \rangle / \langle \alpha \rangle$ versus $\langle j \rangle$. An important characteristic of such plot is that, for two-phase flow regimes with fully developed void and velocity profiles, the data points cluster around a straight line. The value of the distribution parameter, C_0, has been obtained indirectly from the slope of the line, whereas the intercept of this line with the void-fraction-weighted mean gas velocity axis can be interpreted as the void-fraction-weighted mean local drift velocity, V_{gj}.

2.1 PRESSURE GRADIENT USING 1D DRIFT FLUX MODEL

The pressure gradient for steady-state flow can be obtained solving the following differential equation (CARVALHO, 2013),

$$\frac{d}{dz}\left[P + G_g v_g + G_l v_l \right] = -\rho_m g \sin\theta - T_w \tag{7}$$

where P, θ, g and T_w represent pressure, pipe inclination angle, gravitational acceleration and frictional force on the wall per unit volume, respectively. v and G are respectively the *in-situ* velocity and mass flux. The subscripts g and l indicate the gas and liquid phases. Equation (7) is derived from the instantaneous one-dimensional two-phase flow equations for the conservation of mass and momentum. Figure 1 summarizes the process for obtaining these equations (RODRIGUEZ, 2008a). The mathematical tools of Leibniz's rule and the Gauss Divergence Theorem are used to this purpose. The spatial average process is weighed by volumetric fraction.

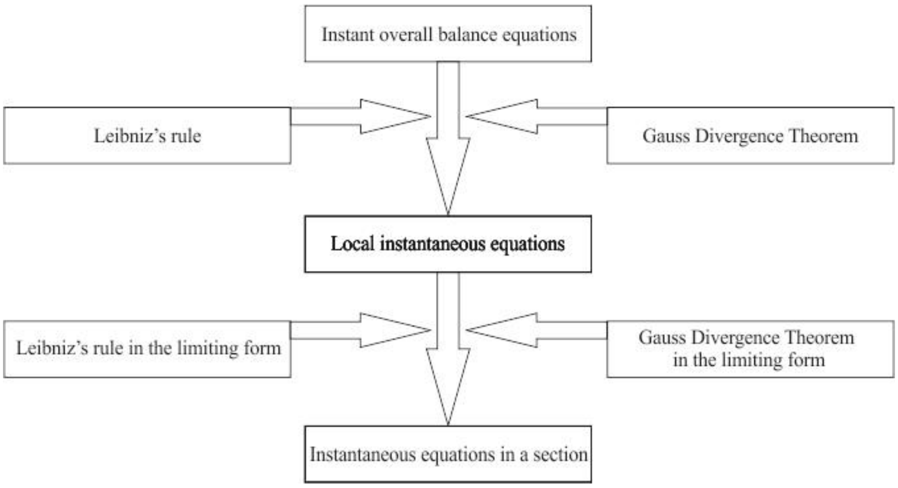

Figure 1. Summary of the instantaneous one-dimensional two-phase flow equations (RODRIGUEZ, 2008a)

For Drift-Flux model, T_w is calculated using the components of the mixture,

$$T_w = \frac{\tau_w S}{A_p} \tag{8}$$

where τ_w and A_p represents the shear stress and the lateral surface area of the annular duct, respectively. The cross-sectional area A and wet perimeter S of the annular duct are calculated by,

$$A = \frac{1}{4}\left(D_e^2 - D_i^2\right) \tag{9}$$

$$S = 2\pi\left(D_e + D_i\right) \tag{10}$$

where D_e is the internal diameter of the external pipe and D_i is the external diameter of the internal pipe. The shear stress is calculated from Darcy's expression as a function of the phase friction factor ($C_{f,n}$), phase density (ρ) and mixture superficial velocity (J) (RODRIGUEZ, 2008a)

$$\tau_w = \frac{1}{8}\left(C_{f,n}\rho_n j^2\right) \tag{11}$$

$$j = j_g + j_l \tag{12}$$

$$j_g = \alpha v_g \tag{13}$$

$$j_l = \left(1-\alpha\right) v_l \tag{14}$$

Equation (7) must be solved for each point along the z axis. Thus, explaining the components that vary with z, we obtain the following ordinary differential equation (LIMA; ROSA, 2009).

$$\frac{d}{dz}\left[\psi\left(z\right)\right] = -\rho_m\left(z\right) g \sin\theta - T_w\left(z\right) \tag{15}$$

where $\psi\left(z\right)$ is given by:

$$\psi\left(z\right) = P\left(z\right) + G_g V_g\left(z\right) + G_l V_l\left(z\right) \tag{16}$$

Equation (15) is an ordinary differential equation and can be solved using the fourth order Runge-Kutta method integrated along the z axis. The variable $P(z)$ is obtained from the solution of the non-linear implicit Eq. (11), obtained with an integration step of Δz:

$$f\left(P\right)=-\psi\left(z\right)+P\left(z\right)+G_gV_g\left(z\right)+G_lV_l\left(z\right) \tag{17}$$

To solve Eq. (11), it is necessary to know the *in-situ* gas and liquid velocities. Assuming steady-state and isothermal two-phase flow and ideal gas behavior for gas phase, they can be represented by

$$v_g\left(z\right)=\frac{\left[\dfrac{v_gP\left(L\right)}{P\left(z\right)}\right]}{\alpha\left(z\right)} \tag{18}$$

$$v_l\left(z\right)=\frac{j_l}{1-\alpha\left(z\right)} \tag{19}$$

It can be noted that both relations are void fraction dependent. Zuber and Findlay (1965) propose that the actual gas velocity can be represent by:

$$v_g=C_oj+v_{gj} \tag{20}$$

Combining Eq. (20) and Eq. (13), an implicit non-linear equation is obtained,

$$f\left(\alpha_l\right)=1-\alpha_l-\frac{j_g}{C_oj+v_{gj}}=0 \tag{21}$$

The void fraction is found from the solution of Eq. (21) by using a numerical method. As can be observed, Eq. (21) depends on the drift-flux distribution parameter C_0, which is related to the distribution of bubbles over the duct's cross section. Therefore, an accurate correlation of C_0 for two-phase flow in annular duct is needed to improve the 1D drift-flux model prediction capabilities.

2.2 DISTRIBUTION PARAMETER

The distribution parameter (C_0) was introduced to consider the effect of non-uniform distribution of the phases in a slug flow. Arguing that the bubble is in the region of highest velocity in the flow rate and that the maximum for the average speed turbulent flow is, approximately, 1.2, initially this value was attributed to the distribution parameter. [Nicklin et al. (1962) e Neal (1963) *apud* Zuber e Findlay (1965), Ishii (1977)].

Ishii (1997) developed a single correlation for the distribution parameter in upward two-phase flow. In this study, he first considered a fully developed bubbly flow and assumed that the distribution parameter would depend on the density ratio, ρ_g/ρ_l (Hibiki and Ishii, 2003a). Based on various experimental data in fully developed flows, the distribution parameter for a fully developed turbulent flow in a round tube was given by

$$C_0=1.2-0.2\sqrt{\frac{\rho_g}{\rho_l}} \tag{22}$$

Hibiki and Ishii (2003a) developed a correlation for two phase flow in a large diameter pipe at low mixture volumetric flux:

$$C_0=\sqrt{\frac{\rho_g}{\rho_l}}+\left(1-\sqrt{\frac{\rho_g}{\rho_l}}\right)e^{0.475\left(\frac{j_g}{j_l}\right)^{1.69}} \tag{23}$$

This drift-flux correlation gave reasonably good predictions for the low-flow-data taken under various experimental conditions such as flow channel diameters (0.102 – 0.480m) and fluid systems (air-water, nitrogen-water and steam-water) (Hibiki, Ishii, 2003a).

Petalas and Aziz (1998) had developed empirically for any geometry the correlation presented in the Eq. (24).

$$C_0 = (1.64 + 0.12\sin\theta)\left(\frac{j\rho_l D_H}{\mu_l}\right)^{-0.031} \tag{24}$$

When applied to predict the pressure gradient in upward two-phase flow in annular ducts, both correlations shown in Eq. (23) and Eq. (24) revealed to be good closure correlations. (CARVALHO, 2013)

3. SINGLE-PHASE FRICTION FACTOR COMPARISON

The hydraulic diameter concept, D_H, is a commonly used term when handling flow in non-circular tubes and channels. Using this term, it is possible to adapt round-tube equations for an annular-duct, for example. It is defined as:

$$D_H = \frac{4A}{S_W} \tag{25}$$

where A is the cross-sectional area and S_W is the wetted perimeter of the cross-section.

For an annular-duct, its value is equal to the difference between the internal diameter of the external pipe and the external diameter of the internal pipe.

$$D_H = \frac{4A}{S_W} = \frac{4\frac{\pi}{4}(D_e^2 - D_i^2)}{\pi(D_e + D_i)} = D_e - D_i \tag{26}$$

It is a common practice to predict friction pressure drop in noncircular conduits by applying the hydraulic diameter concept. However, this procedure should be limited to high Reynolds numbers, since unacceptable errors may occur for lower degrees of turbulence (CAETANO, 1986).

3.1. LAMINAR FLOW

For a Newtonian, fully developed, laminar, steady-state, axial upward vertical flow in a concentric annular-duct, the axial velocity distribution, $V_z(r)$, is:

$$V_z(r) = \frac{(\pi_O - \pi_L)}{4\mu L} R^2 \left[1 - \left(\frac{r}{R}\right)^2 + \frac{1-k^2}{\ln(1/k)}\ln\left(\frac{r}{R}\right)\right] \tag{27}$$

$$k = \frac{D_i}{D_e} \tag{28}$$

where π_O and π_L represents the combined effects of static pressure and gravitational force, μ is the viscosity and k is the annular pipe diameter ratio, defined by Eq. (28).

Applying the hydraulic diameter concept to the concentric annulus, the outer pipe inside radius, R, is given by:

$$R = \frac{D_H}{2(1-k)} \tag{29}$$

The average axial velocity, $\langle V_z \rangle$, over the cross-sectional area in terms of the hydraulic diameter of the annular-duct is given by:

$$\langle V_z \rangle = \frac{(\pi_O - \pi_L)}{32\mu L(1-k^2)} D_H^2 \left[\frac{1-k^4}{1-k^2} - \frac{1-k^2}{\ln(1/k)} \right] \tag{30}$$

The Darcy equation can be written by:

$$\pi_O - \pi_L = \frac{1}{2} C_f \frac{L}{D_H} \rho \langle V_z \rangle^2 \tag{31}$$

The Darcy friction factor in an annular-duct, C_f, is given by:

$$C_f = \frac{\overline{C_f}}{\text{Re}} \tag{32}$$

$$\text{Re} = \frac{\rho \langle V_z \rangle^2 D_H}{\mu} \tag{33}$$

where $\overline{C_f}$ is a friction geometry parameter and Re is the Reynolds number.

For a pipe flow, $\overline{C_f}$, its value is equal to 64. However, when applying the hydraulic diameter to a concentric geometry (*CA*), it becomes a function of the annular pipe diameter ratio, *k*.

$$\overline{C_{f,CA}}(k) = \frac{64(1-k)^2}{\left[\frac{1-k^4}{1-k^2} - \frac{1-k^2}{\ln(1/k)} \right]} \tag{34}$$

An analytical solution for laminar flow in an eccentric annulus by using bipolar coordinates was presented by Snyder and Goldstein (1965). According to this solution, the friction geometry parameter for an eccentric annular-duct (*EA*) is a function of both eccentricity, *e*, defined by Eq. (35), and the annular pipe diameter ratio, *k*.

$$e = \frac{2D_{BC}}{D_e - D_i} \tag{35}$$

$$\overline{C_{f,EA}}(k,e) = \frac{16(1-k)^2(1-k^2)}{\phi \sinh^4 \eta_O} \tag{36}$$

where D_{BC} is the distance between the centers of the tubes, ϕ is a function of the geometric characteristics and η_O is a coordinate of the transformed domain.

Figure 2 shows a comparison of the friction factor behavior in annular-duct terms of the pipe diameter ratio, *k*, assuming that all configurations have the same hydraulic diameter and Reynolds number.

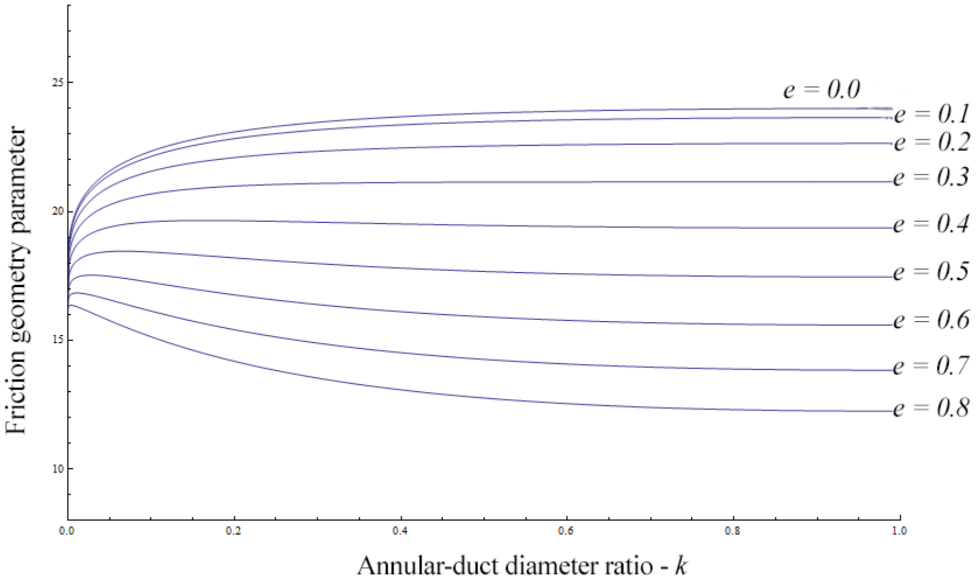

Figure 2 - Friction geometry parameter comparison

It is possible to observe that when $k \to 0$, for any eccentricity, the friction factor is 16, the value for a circular pipe. Besides, for the concentric annular-duct, when $k \to 1$ the friction geometry parameter approaches 24, which is the correct value for flow between infinite parallel plates.

For a fixed diameter ratio, the friction geometry parameter decreases with an increase in the degree of eccentricity. For a high degree of eccentricity, the friction geometry parameter and, consequently, the friction factor, is always smaller than for a circular pipe.

3.2. TURBULENT FLOW

Gunn and Darling (1963) described an approach to predict turbulent flow friction factor values in annular-ducts. The authors proposed a semi-empirical procedure that combined turbulent flow data for many non-circular configurations with the known behavior of friction factor as a function of Reynolds number and the ratio between friction geometry parameter for the circular and non-circular configurations. The friction factor for an annular-duct may be predicted from:

$$\frac{1}{\left\{C_{f,EA}\left(\frac{\overline{C_{f,P}}}{\overline{C_{f,An}}}\right)^{0.45e^{-\frac{\mathrm{Re}-3000}{10^6}}}\right\}^{0.5}} = 4Log\left\{\mathrm{Re}\left\{C_{f,EA}\left(\frac{\overline{C_{f,P}}}{\overline{C_{f,An}}}\right)^{0.45e^{-\frac{\mathrm{Re}-3000}{10^6}}}\right\}^{0.5}\right\} - 0.40 \qquad (37)$$

where $C_{f,x}$ is the Darcy's friction factor for the x configuration, $\overline{C_{f,x}}$ is the friction geometry parameter for the x configuration, defined by Eq. (34) and Eq. (36).

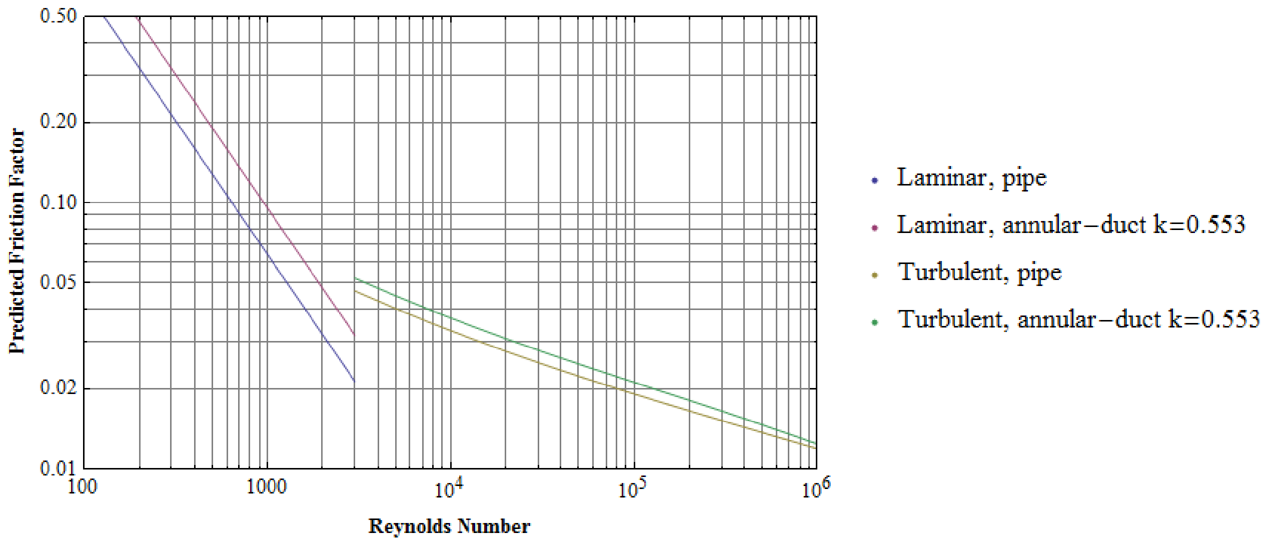

Figure 3 - Darcy friction factor

Figure 3 shows that, for the same hydraulic diameter, there is a considerable difference between friction factor values in an annular-duct as compared to circular pipe. Usually the friction factor for a concentric annular-duct is higher than for pipe flow. This difference is a constant value for laminar flow, and depends on the annular-duct pipe diameter ratio, *k*, and the degree of eccentricity, *e*. The annular-duct friction factor approaches the equivalent value for a circular pipe at high Reynolds numbers (CAETANO, 1992).

4. EXPERIMENTAL DATABASE

The experimental data of bubbly gas-liquid flow in annular-duct collected by Carvalho (2013) and Caetano (1985) is used in this study. The data of Carvalho (2013) was collected at the Thermal-Fluids Engineering Laboratory (LETeF) of the University of Sao Paulo (USP), Sao Carlos campus, in an experimental apparatus of 9 m of length composed by two concentric borosilicate-glass tubes forming an annular duct of external and internal diameters equal to 105 mm and 75 mm, respectively. The test line is attached to an inclinable structure and can be used at any angle between 0-90º. Pressure-drop data was acquired with a differential pressure transmitter, with maximum error equal to ±0.0055 KPa. Quick closing valves and a bypass system were also used to measure phase holdup. Caetano (1985) performed experiments at The University of Tulsa in a 13.7 m long vertical and concentric annular duct made of an outer acrylic tube and an inner PVC pipe of 76.2 mm and 42.2 mm, respectively. Caetano (1985) also used a bypass system with actuated ball valves to measure phase holdup, by closing the test section inlets and outlets and visually measuring the liquid level. That author used a differential pressure transmitter with maximum error equal to ±0.007 MPa. Table 1 shows the geometrical characteristics and the experimental conditions.

Table 1 - Experimental studies performed by Caetano (1986) and Carvalho (2013).

Source	Number of Points	Fluids	Outer-inner diameters (mm)	Inclination
Caetano (1986)	21	Air-water	76.2-42.2	Vertical
	19	Air-kerosene		
Carvalho (2013)	52	Air-water	105-75	Vertical
	41			45º

The full database is composed of 133 experimental points divided in air-water and air-kerosene mixtures and vertical and 45º inclination conditions. Figure 4a shows the bubbly flow data of Carvalho (2013) and Caetano (1985). It is observed that the Blanco *et al*. (2008)'s methodology for predicting the bubbly-slug flow-pattern transition has good agreement with experimental observations. The pressure drop data is shown in Figure 4b.

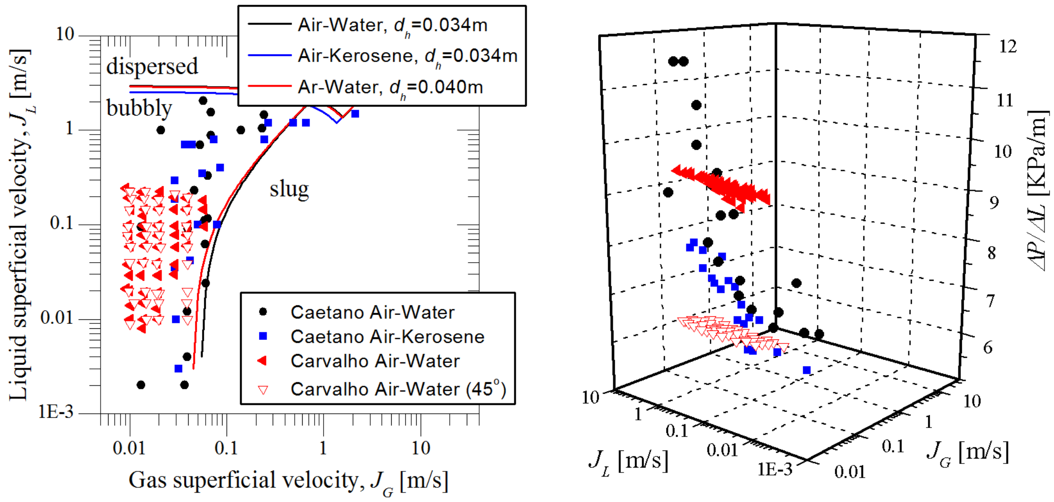

Figure 4. (a) Flow-pattern and (b) total pressure gradient of the experimental database. The symbols correspond to the test conditions presented in Table 1. The methodology of Blanco et al. (2008) was used to model the bubbly-slug flow-pattern transition.

5. METHODOLOGY

It has been shown that the 1D Drift-flux model predicts accurately the pressure gradient in upward gas-liquid pipe flow (LIMA; ROSA, 2009). The experimental data of Carvalho (2013) and Caetano (1986) is now used to develop a new drift-flux distribution parameter (C_o) for bubbly annular-duct flow by using the approach of Zuber and Findlay (1965). The ordinary differential equation is solved through a fourth order Runge-Kutta method (LIMA; ROSA, 2009; RODRIGUES; ROSA; MAZZA, 2008).

$$C_0 = \sqrt{\frac{\rho_g}{\rho_l}} + \left(1 - \sqrt{\frac{\rho_g}{\rho_l}}\right) e^{a\left(\frac{j_g}{j_l}\right)^b} \tag{38}$$

$$C_0 = (c + 0.12\sin\theta)\left(\frac{j\rho_l D_H}{\mu_l}\right)^{-d} \tag{39}$$

To adjust the model prediction to the experimental data, two coefficients of each distribution parameters were varied. As seen on equations (23) and (24), the original coefficients are $a = 0.475$, $b = 1.69$, $c = 1.64$ and $d = 0.031$. For each pair of $\{a,b\}$ or $\{c,d\}$, a graph was plotted, comparing the measured and predicted pressure drop. A wide range of values was tested and the generated graphs were visually analyzed to choose the best fitted one. The code was implemented in Wolfram Mathematica® platform.

6. RESULTS

The total pressure gradient for steady-state flow is composed of three components:

$$\left(\frac{dp}{dz}\right)_{Total} = \left(\frac{dp}{dz}\right)_{Gravity} + \left(\frac{dp}{dz}\right)_{Acceleration} + \left(\frac{dp}{dz}\right)_{Friction} \tag{40}$$

The acceleration pressure gradient in bubby flow is normally negligible (CAETANO, 1992). Applying the original distribution parameters equations of Hibiki and Ishii (2003a) – Eq. (23) – and Petalas and Aziz (1998) – Eq. (24) – to predict the theoretical pressure gradient, it is used for comparison with its respective experimental pressure gradient data. The results for Eq. (24), Petalas and Aziz (1998) are shown in Fig. 6. The results for Eq. (23), Hibiki and Ishii (2003a) are shown in Fig. 5.

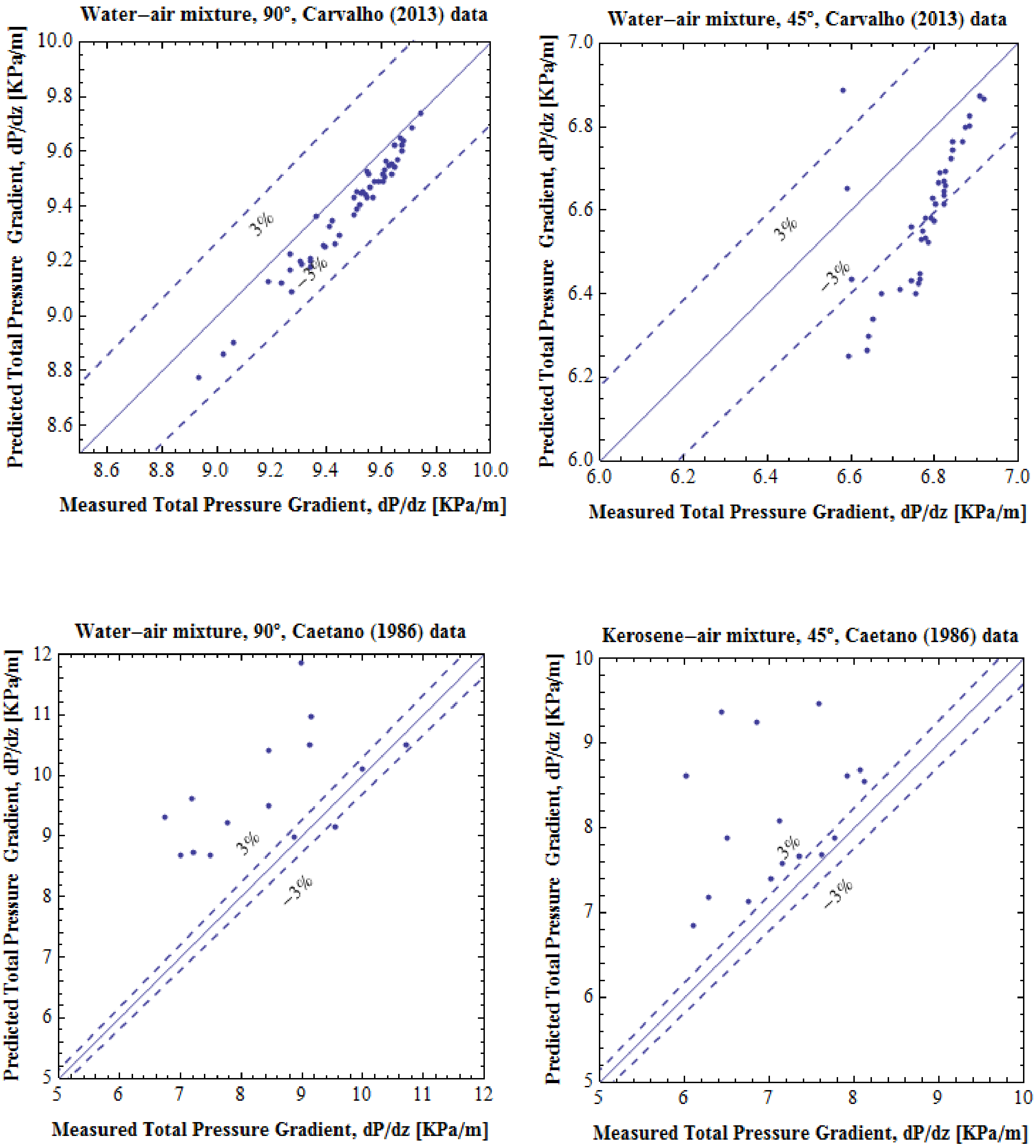

Figure 5 – Original Hibiki and Ishii (2003a) C_0 performance

In general, the Drift-flux model tends to overvalue the total pressure gradient for Carvalho (2013) and underestimate for Caetano (1986) data. The good agreement between predicted and measured results, especially for Carvalho (2013) data, show the good performance of the model.

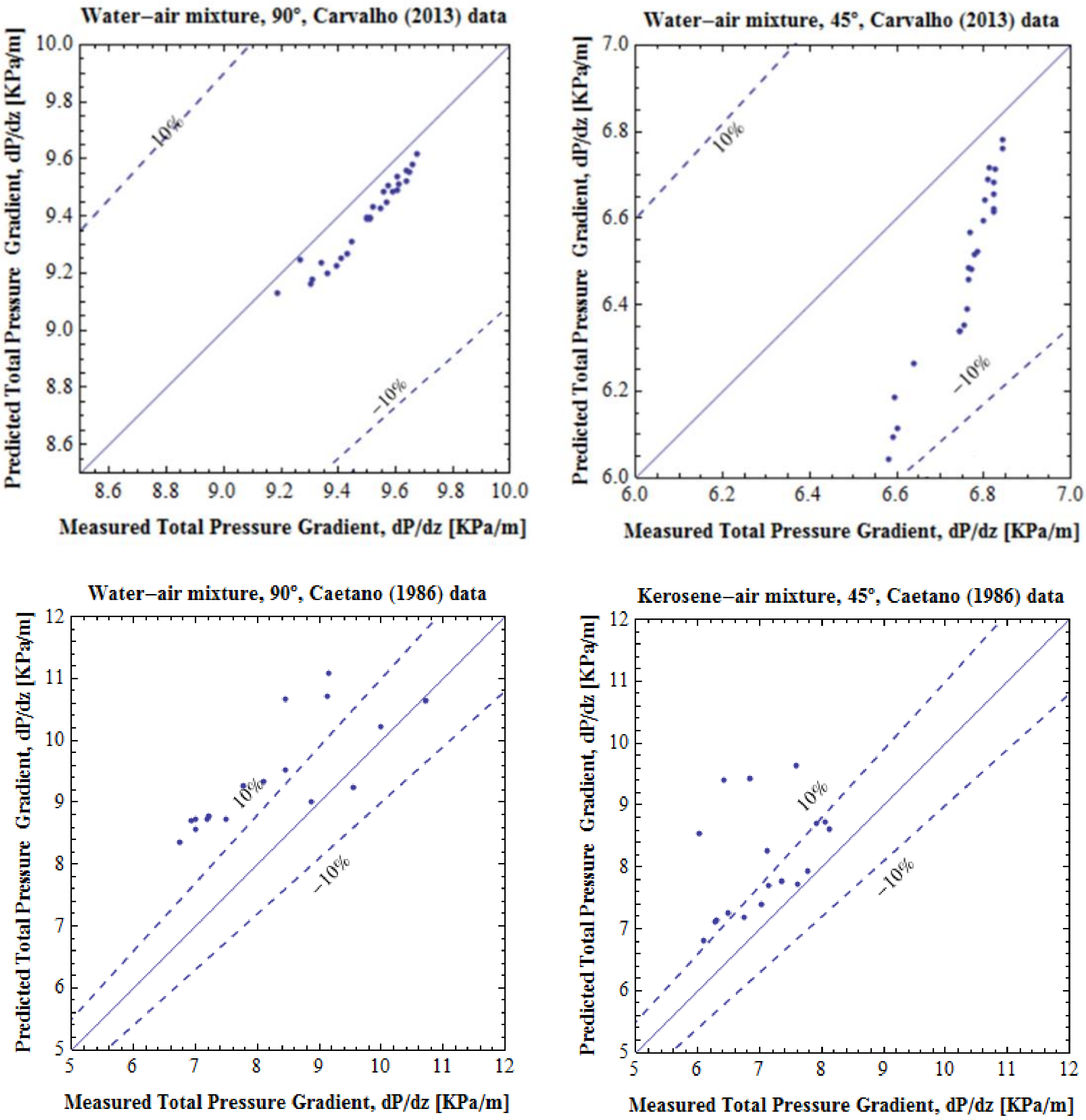

Figure 6 – Original Petalas and Aziz (1998) C_0 performance

In general, the Drift-flux model tends to overvalue the total pressure gradient for Carvalho (2013) and underestimate for Caetano (1986) data. The good agreement between predicted and measured results, especially for Carvalho (2013) data, show the good performance of the model. However, the gravity losses are predominant over the frictional ones. That way, the frictional component seems not well evaluated due the use of the hydraulic diameter.

For the distribution equation proposed by Hibiki and Ishii (2003a) was found a pair *{a,b} = {0.5,-0.26}* that well-fitted Carvalho (2013)'s data. However, for Caetano (1986), the same agreement was not found.

$$C_0 = \sqrt{\frac{\rho_g}{\rho_l}} + \left(1 - \sqrt{\frac{\rho_g}{\rho_l}}\right) e^{0.500\left(\frac{j_g}{j_l}\right)^{-0.26}} \tag{41}$$

The performance of Eq. (41) when applied to the Drift-Flux model can be seen on Fig. 7.

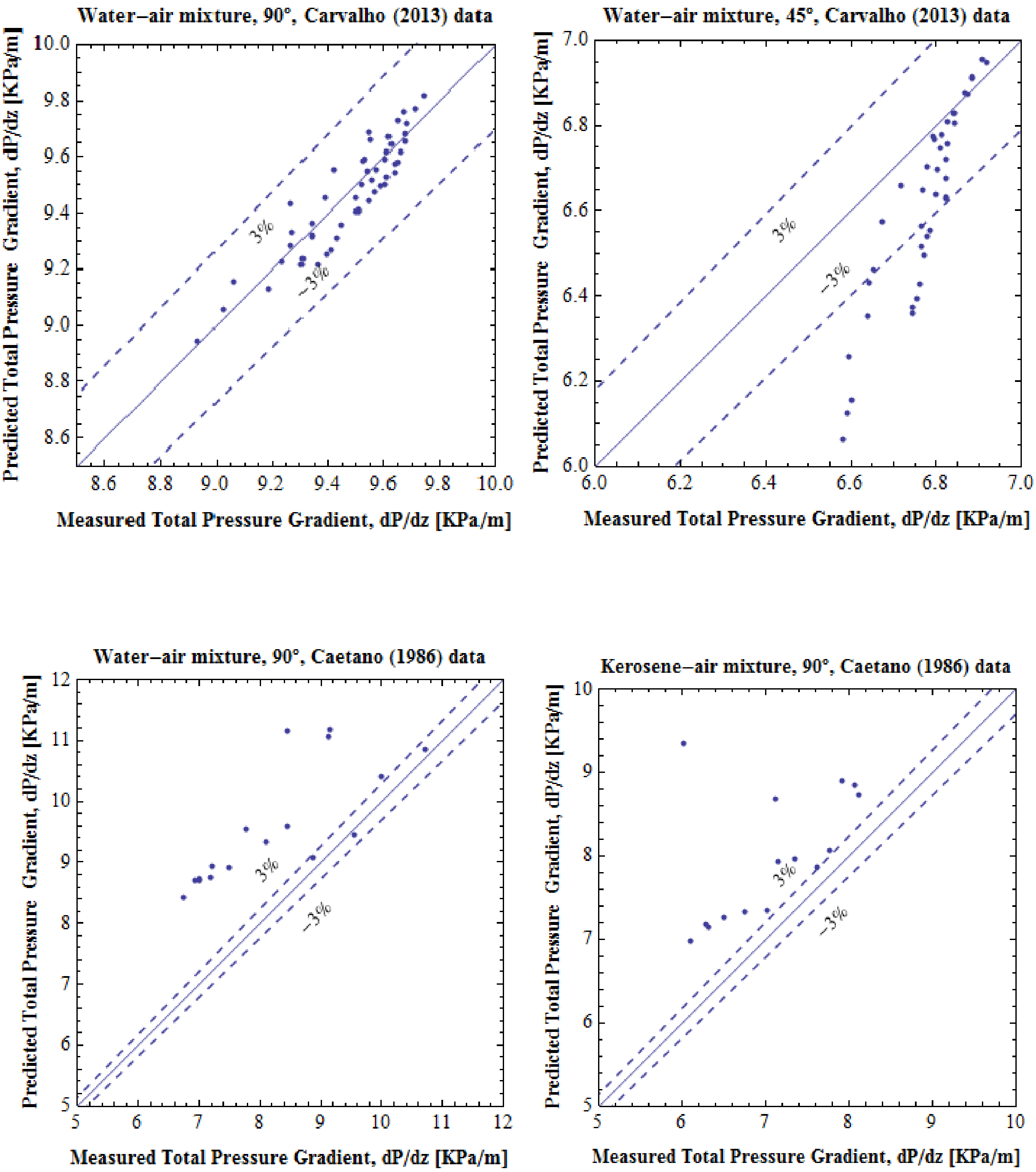

Figure 7 – Corrections proposed for the C_0 proposed by Hibiki and Ishii (2003a)

For the distribution parameter equation inspired by Petalaz and Aziz (1998), the pair *{c,d}* that best fit the data is *{0.97,0.012}*. This way, the final proposed correction in Eq. (24) for the annular geometry is shown on Eq. (41). This performance can be seen on Fig. 8.

$$C_0 = (0.97 + 0.12\sin\theta)\left(\frac{j\rho_l D_H}{\mu_l}\right)^{-0.012} \quad (42)$$

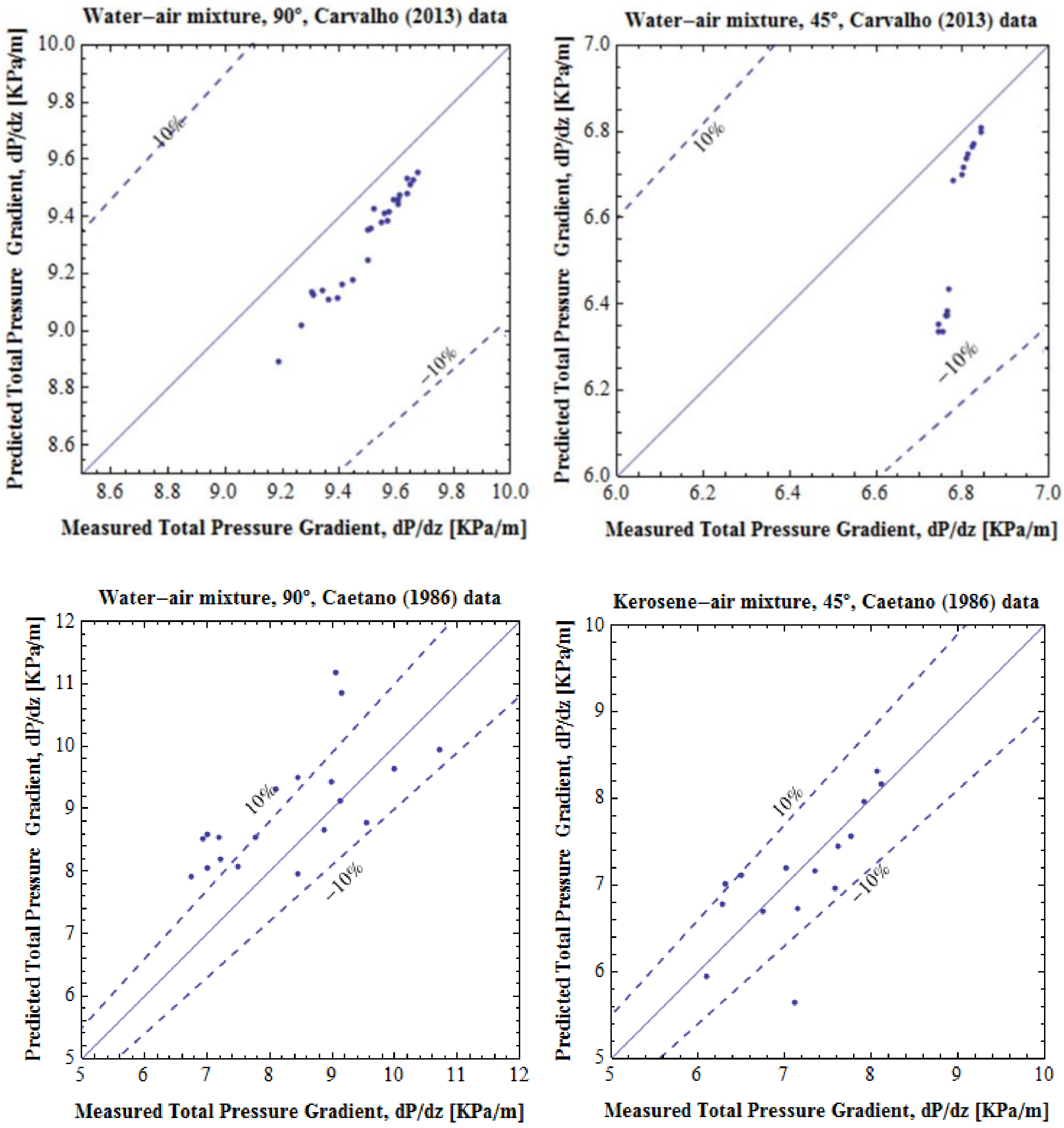

Figure 8 – Corrections proposed for the C_0 proposed by Petalas and Aziz (1998)

7. ACKNOWLEDGEMENTS

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