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PRESSURE DROP IN VERTICAL CORE ANNULAR FLOW

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Abstract. *An experimental apparatus for the study of core annular flows of heavy oil and water at room temperature has been set up and tested at laboratory scale. The test section consists of a 2.75 cm ID galvanized steel pipe. Tap water and a heavy oil (17.6 Pa.s; 963 kg/m^3) were used. Pressure drop in a vertical upward test section was accurately measured for oil flow rates in the range 0.297 - 1.045 l/s and water flow rates ranging from 0.063 to 0.315 l/s. The oil-water input ratio was in the range 1-14. The measured pressure drop comprises gravitational and frictional parts. The gravitational pressure drop was expressed in terms of the volumetric fraction of the core, which was determined from a correlation developed by Bannwart (1998b). The existence of an optimum water-oil input ratio for each oil flow rate was observed in the range 0.07 – 0.5. The frictional pressure drop was modeled to account for both hydrodynamic and net buoyancy effects on the core. The model was adjusted to fit our data and shows excellent agreement with data from another source (Bai, 1995).*

Keywords: *Liquid liquid flow, Core annular flow, Modeling, Pressure drop, Heavy oil*

1. INTRODUCTION

In two-phase pipe flow of immiscible liquids, the annular flow pattern with the thicker fluid surrounded by the thinner one, is commonly observed when the conditions are such that both fluids form continuous phases. This flow configuration, known as *core annular flow* or *core flow*, has the very interesting feature that the frictional pressure drop is comparable to the single phase flow of the thinner fluid in the same pipe at mixture flow rate (see for example, Bannwart 1998a), because this fluid keeps in contact with the wall. The energetic advantage of core annular flow has been used for pipeline transportation of viscous oils, using water as lubricant.

The advantages of the core flow technology have been fully appreciated since the series of studies carried out by Russel & Charles (1959), Russell, Hodgson & Govier (1959), Charles (1960), and notably Charles, Govier & Hodgson (1961). Since then, many theoretical and experimental studies have been developed, concerning its stability and modeling aspects. Most of these studies have been made for horizontal lines, in order to apply the technology to heavy oil transportation (Oliemans *et al.*, 1987; Arney *et al.*, 1993; Ribeiro *et al.*, 1996; Bannwart, 1998a). Except for the experiments done by Bai (1995) in a 0.9525 cm ID glass tube, no experimental study has been found on vertical core annular flow.

In contrast with the horizontal case, where the net buoyancy force (which is proportional to the density difference) causes the oil core to be eccentric, in vertical flow this force favors the acceleration of the (lighter) oil and thus the stability of the flow itself.

The aim of this paper is to develop a model to calculate the frictional losses during vertical upward core annular flow, from experimental measurements of pressure difference and based on a simple theoretical approach. The resulting correlation is fitted to our measurements and compared with data by Bai (1995). Besides the fluid properties and flow rates, the correlation requires the volumetric fraction of the oil, which is determined from the model proposed by Bannwart (1998b). Through empiric parameters, this correlation considers the effects of interface irregularities, turbulence in the annulus flow as well as the effects of buoyancy force on the frictional pressure gradient.

2. EXPERIMENTAL APPARATUS

The setup used for studies on core annular flow was installed at the Department of Energy of the State University of Campinas – UNICAMP, Brazil, and comprises vertical and horizontal pipe test sections as shown in Fig. 1.

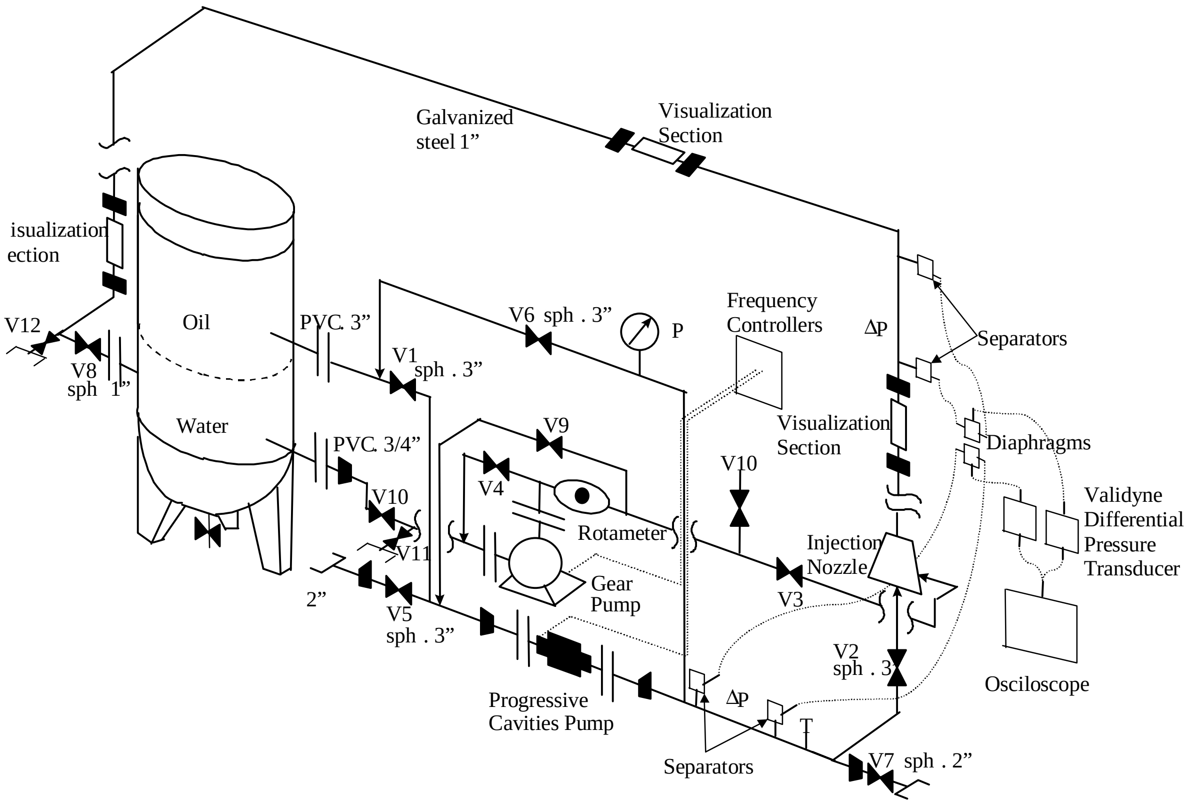

Figure 1- Experimental apparatus

Heavy oil was pumped from the separator tank to the pipe inlet by a progressive cavity pump through a 7.46 cm ID PVC pipe and its flow rate, after calibration (with a weighing tank and chronometer), was provided by the pump rotation. The oil used was a 17.6 Pa.s, 963 kg/m^3 fuel oil at room temperatures. From the bottom of the separator tank, water was driven through a 1.9 cm ID PVC pipe by a gear pump, measured by means of a rotameter and laterally injected into the pipe inlet. Each pump motor was controlled by its own frequency inverter. Both flow rates could be independently varied using either the by-pass valve of each pump or the frequency inverter.

A special injection nozzle with an entry section (30 diameters long) and a visualization section were provided in order to make sure that a stable core annular flow occurred in the pipe. The injection nozzle was designed to help stabilization of core flow while reducing the oil pipe diameter from 7.46 to about 2.5 cm. This is enough to penetrate the test section vertical pipe, forming an oil core surrounded by an annular gap of water in axial flow.

The oil-water mixture then flowed into the 2.75 cm ID test section pipe made in galvanized steel, through vertical and horizontal segments, returning to the separator tank. Pressure drop in a 84 cm segment of the vertical upward test section was measured by means of a Validyne differential pressure transducer (accuracy 3% of full scale) with appropriate diaphragm (88 mm of water). Before setting each pair of flow rates, the system was run with pure water until the pressure drop in the test section became low enough so as it could be assumed to be clean from any fouling action by the oil.

3. FRICTIONAL PRESSURE DROP MEASUREMENTS

The frictional pressure gradient in core flow (G_f) can be defined as the total pressure gradient minus the gravity term of the mixture (Arney *et al.*, 1993) and is determined from the measurements of pressure difference in the vertical test section, in the following way:

$$G_f = \frac{\Delta P_{friction}}{H} = \frac{\Delta P_{dpt}}{H} - (\rho_1 - \rho_2)g\alpha \tag{1}$$

where ΔP_{dpt} is the pressure difference read at the differential pressure transducer, α is the oil volumetric fraction, H is the length between pressure taps, ρ_1 is the density of the fluid at the core (oil), ρ_2 is the density of the fluid in the annulus (which is also the manometric fluid, i.e., water) and g is the gravity acceleration. Note that when only water is flowing in the pipe, the transducer gives the frictional pressure drop, because the transducer legs are filled with water; this corresponds to make $\rho_1 = \rho_2$ in Eq. (1). Each value of ΔP_{dpt} is read in Volts and converted to pressure units by previous calibration. The oil fraction (α) is determined from solution of the following equation for vertical core annular flow (Bannwart, 1998b):

$$j_1(1-\alpha) - s_o j_2 \alpha - V_{ref} F(\alpha) = 0 \tag{2}$$

with

$$F(\alpha) = k\alpha^2(1-\alpha)^n, \tag{3}$$

$$V_{ref} = \frac{(\rho_2 - \rho_1)gD^2}{16\mu_2} \tag{4}$$

and using s_o = 1, k = 0.0194, n = 1.75 (dimensionless parameters). The variables j_1 and j_2 are the oil and water superficial velocities, i.e. $(j_1, j_2) = (Q_1, Q_2)/A$, where (Q_1, Q_2) are the flow rates and $A=\pi D^2/4$; D (=2.76 cm) is the pipe diameter and μ_2 is the water viscosity. This correlation is based on the kinematic wave theory of interfacial waves, whose velocity was accurately measured by Bai (1995), and is in very good agreement with direct holdup measurements by the same author.

Pressure drop was measured for nine oil flow rates in the range 0.297 - 1.045 l/s, with different water flow rates ranging from 0.063 to 0.315 l/s. The total number of runs was 65. The measured values of the frictional pressure gradient are plotted in Fig. 2 as a function of the water-oil ratio (j_w/j_o), for each fixed oil superficial velocity (j_o).

The existence of a minimum pressure gradient for a certain input ratio, at a given oil flow rate, can be clearly observed. This happens because water addition helps the oil flow, but at the same time increases the total flow rate. This result has been reported for horizontal flow and is also confirmed in upward flow (Bai, 1995). The optimum input ratio (j_w/j_o), however, depends on the superficial velocity of the oil, and is observed to be in the range 0.07 - 0.5.

When the superficial oil velocity increases, the minimum pressure gradient point moves toward lower values of input ratio. In other words, that the largest oil flow rates need, proportionally, lower amounts of water to reach the minimum frictional pressure gradient. This is indeed a very attractive feature of this flow pattern.

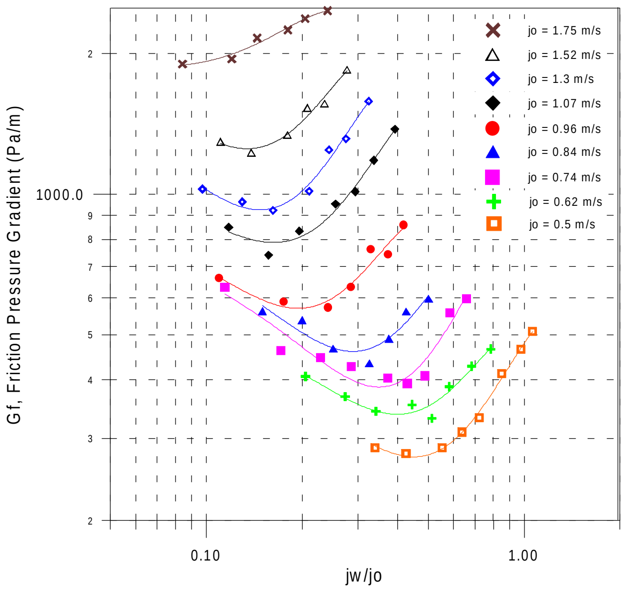

Figure 2 - Frictional pressure gradient in core flow as a function of input ratio, with the oil superficial velocity as parameter

4. SIMPLIFIED SOLUTION FOR "PERFECT CORE ANNULAR FLOW"

In the perfect core annular flow (PCAF) model the two Newtonian immiscible fluids flow inside a vertical pipe of inner radius R_2 (or inner diameter D), in a concentric configuration with a smooth circular interface placed at $r = R_1$, as shown in Fig. 3. According to this model, the frictional pressure gradient can be expressed as

$$G_f = \frac{128\mu_2 Q}{\pi D^4\left(1-\alpha^2(1-m)\right)} - \frac{(\rho_2-\rho_1)g(1-\alpha)\alpha[1-\alpha(1-m)]}{[1-\alpha^2(1-m)]} \tag{5}$$

where Q is the mixture flow rate, $m = \dfrac{\mu_2}{\mu_1}$ is the viscosity ratio and $\alpha = \left(\dfrac{R_1}{R_2}\right)^2$ is the oil volumetric fraction, which can be calculated by solving Eq. (2) using

$F(\alpha) = \alpha^2[2(\alpha - 1) - (\alpha + 1)ln\alpha]$ and s_o = 2. For the experiments reported here, m is negligible ($\approx 10^{-5}$) and Eq. (5) simplifies to

$$G_f = \frac{128\mu_2 Q}{\pi D^4(1-\alpha^2)} - \frac{(\rho_2 - \rho_1)g(1-\alpha)\alpha}{(1+\alpha)}. \qquad (6)$$

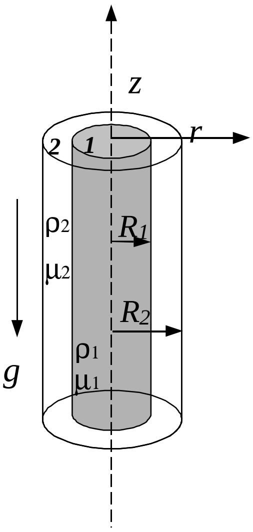

Figure 3 - Perfect core annular flow (PCAF)

Following Bannwart (1998a), the first term on the right side of the above equation can be interpreted as the frictional pressure drop of a laminar water flow at an equivalent flow rate Q_{PCAF} defined by

$$Q_{PCAF} = \frac{Q}{1-\alpha^2}, \qquad (7)$$

whereas the second term is the net buoyancy effect:

$$G_b = \frac{(\rho_2 - \rho_1)g(1-\alpha)\alpha}{(1+\alpha)}. \qquad (8)$$

Figure 4 shows a plot of the sum $G_{f,exp} + G_b$ as a function of the Q_{PCAF}, as suggested by Eq. (6), using the experimental values of G_f ; α. and G_b were calculated for the perfect case as described above. It can be clearly concluded that the PCAF model is not effective to describe our experiments. This fact can be attributed principally to two reasons: a) the presence of waves on the interface as observed in the experiments, and b) in all tests the water flow was turbulent, as seen in the Fig. 6. Both facts contradict essential assumptions of the PCAF theory. The Reynolds number for the water annulus flow is defined by

$$Re_2 = \frac{\rho_2 V_2 D_{H,2}}{\mu_2} = \frac{\rho_2 j_2 D}{\mu_2} \qquad (9)$$

where V_2 is the average velocity of the annulus flow and $D_{H,2}$ its hydraulic diameter.

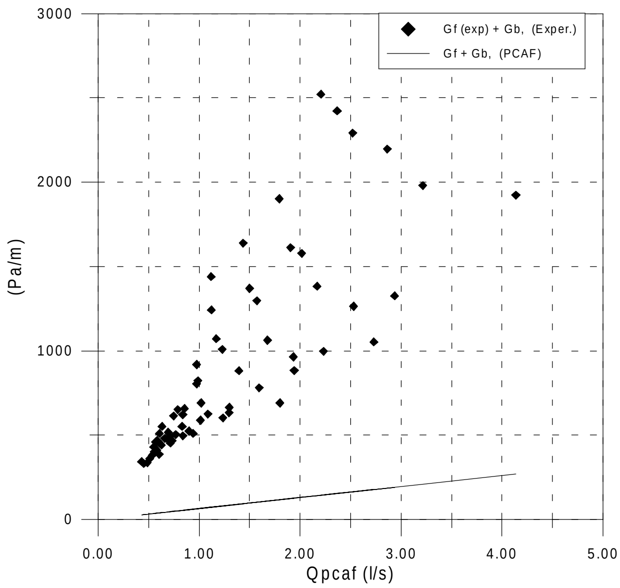

Figure 4 – The sum $G_{f,exp} + G_b$ as a function of the equivalent flow rate Q_{PCAF}

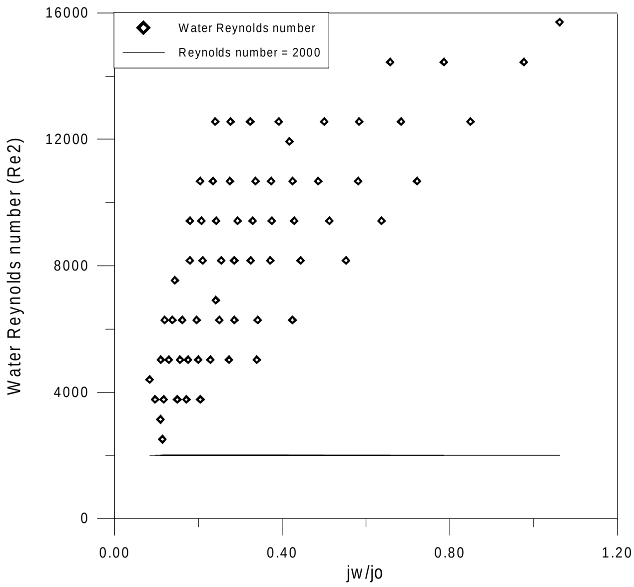

Figure 5 - Water Reynolds number versus oil-water input ratio

5. PROPOSED MODEL

In order to embody the wavy character and annulus turbulence effects together with the bouyancy effect in the pressure drop model, Eq. (5) can be rewritten in a more general form:

$$G_f = G_{f,h} - G_b \tag{10}$$

where $G_{f,h}$ is the hydrodynamic (irreversible) component and G_b is the net bouyancy effect. The later can be expressed as

$$G_b = (\rho_2 - \rho_1) g\ f(\alpha, m) \tag{11}$$

where $f(\alpha,m)$ is a function to be determined. The hydrodynamic term ($G_{f,h}$) can be written, as usual, as

$$G_{f,h} = a\left(\frac{\rho_m JD}{\mu_m}\right)^{-n} \frac{\rho_m J^2}{2D}, \tag{12}$$

where J is the superficial velocity of the mixture, ρ_m is the mixture density

$$\rho_m = \alpha\rho_1 + (1-\alpha)\rho_2 \tag{13}$$

and μ_m is the mixture viscosity. The coefficients a and n are parameters to be determined from experiments, and usually depend on the pipe wall properties. From Eq. (5) it can be concluded that for the PCAF model, a = 64, n = 1, and

$$\frac{1}{\mu_m} = \frac{\alpha^2}{\mu_1} + \frac{1-\alpha^2}{\mu_2} \cong \frac{1-\alpha^2}{\mu_2}, \tag{14}$$

$$f(\alpha, m) = \frac{(\rho_2 - \rho_1) g (1-\alpha)\alpha[1-\alpha(1-m)]}{[1-\alpha^2(1-m)]} \cong \frac{(\rho_2 - \rho_1) g (1-\alpha)\alpha}{(1+\alpha)}, \tag{15}$$

where the approximations hold for $m \to 0$. For turbulent-wavy annulus flow we suggest

$$\frac{1}{\mu_m} = \frac{\alpha}{\mu_1} + \frac{1-\alpha}{\mu_2} \cong \frac{1-\alpha}{\mu_2}, \tag{16}$$

$$f(\alpha) = k(1-\alpha)\alpha, \tag{17}$$

where k is an empirical parameter. Eq. (16) is compatible with a constant shear stress layer, and Eq. (17) satisfies the limiting values of Eq. (15) for $\alpha \to 0$ and $\alpha \to 1$. With the help of equations (11), (12), (16) and (17), Eq. (10) becomes

$$G_f = a\left(\frac{\rho_m JD}{\mu_2}\right)^{-n} \frac{1}{2D}\rho_m \frac{J^2}{(1-\alpha)^n} - k(\rho_2 - \rho_1) g (1-\alpha)\alpha \tag{18}$$

where a, n and k are parameters to be adjusted from experiments. The parameter n was set to 0.25 (turbulent flow in smooth walled pipe), then a and k were obtained from minimization of the total relative variance

$$S(a,k) = \sum_{runs} \left(\frac{G_f - G_{f,exp}}{G_{f,exp}} \right)^2 \tag{19}$$

where G_f is given by Eq. (18) and $G_{f,exp}$ is the measured value for each run, as described in the section 3. The following values were found

$$\begin{matrix} a = 0.257 \\ k = 0.159 \end{matrix} \qquad (n = 0.25). \tag{20}$$

Equation (18) with the set of constants of Eq. (20) and α determined by solving Eq. (2) is the final model proposed for the frictional pressure gradient in a vertical core annular flow, for turbulent-wavy annulus flow and accounting for buoyancy effects. Figure 6 compares the experimental hydrodynamic pressure gradient $G_{f,h}$ with its calculated value given by the first term of the right-hand side of Eq. (18), as a function of the equivalent flow rate

$$Q^* = \frac{Q}{(1-\alpha)^{\frac{n}{2}}} = \frac{Q}{(1-\alpha)^{0.125}}. \tag{21}$$

This plot is, in fact, similar to Figure 4 and shows the great improvement obtained through the use the turbulent-wavy annulus flow picture over PCAF model. A comparison of the calculated and measured friction pressure gradients is shown in Fig. 8, where the agreement between both is approximately ±25 %.

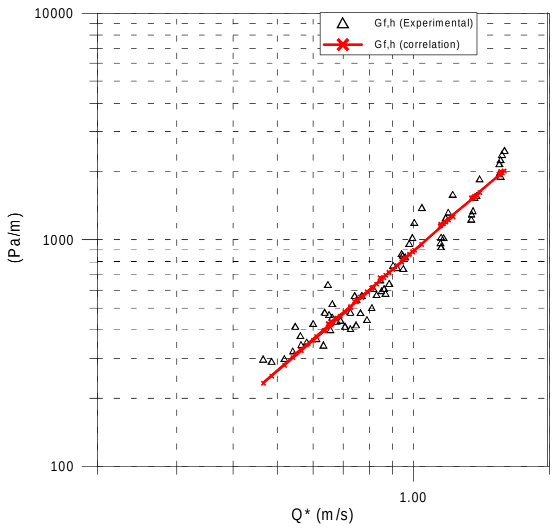

Figure 6 - Calculated and experimental hydrodynamic pressure gradient versus equivalent flow rate Q^*

The present model was also compared with friction pressure gradient data by Bai (1995), who studied the vertical core-annular flow inside a 0.9525 cm ID glass tube using an oil-water system with a much higher density difference than the present study (ρ_1 = 905 kg/m^3 , μ_1 = 0.601 Pa.s at 22 ºC). This comparison, shown in Fig. 8, shows an excellent agreement between calculated and measured frictional pressure gradients. In fact, this agreement is even better than our pressure drop data, because the correlation used to determine α , i.e. Eq.(2), was previously validated with Bai's wavespeed data and is also in very good agreement with direct holdup measurements in the same system (Bannwart, 1998b).

Finally, using Eqs. (13) and (16), Eq. (18) can be cast in a more general form as

$$G_f = G_{f,2}(Q)\left(\frac{\rho_m}{\rho_2}\right)^{1-n}\left(\frac{\mu_m}{\mu_2}\right)^n - k(\rho_2-\rho_1)g(1-\alpha)\alpha \qquad (22)$$

where $G_{f,2}(Q)$ is the friction pressure gradient for single phase flow of fluid 2 at mixture flow rate. Use of n = 0.25 and k = 0.159 is recommended.

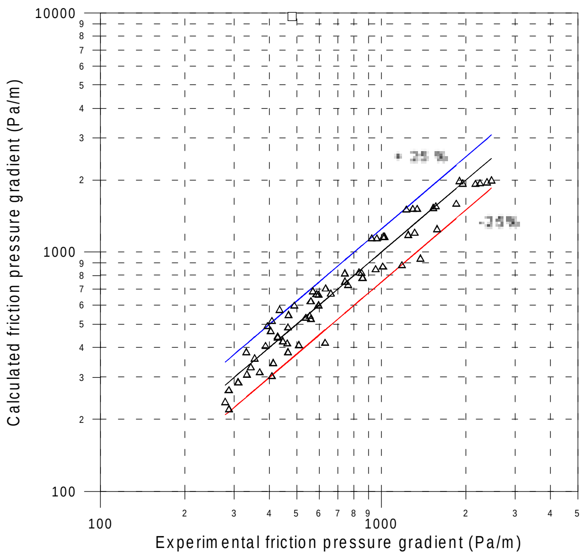

Figure 7 - Friction pressure gradient calculated by Eq. (18) *versus* experimental values (our data)

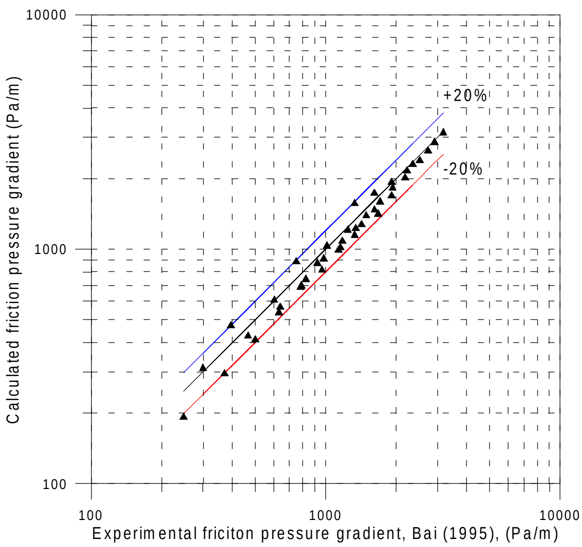

Figure 8 - Friction pressure gradient calculated by Eq. (18) *versus* experimental data by other source (Bai, 1995)

6. CONCLUDING REMARKS

Using a laboratory scale apparatus, the core annular flow technology was tested for lifting a heavy oil (μ_o = 17.6 Pa.s and ρ_o = 963.6 kg/m^3) with successful results. The vertical upward flow favors the stabilization of the core annular pattern.

First, it is shown that the PCAF solution is not appropriate to describe our friction pressure drop data, since the presence of a wavy interface and water turbulence contradict essential assumptions of that theory. To properly represent the friction pressure drop data, it is necessary to model the effects of the wavy core, annulus flow turbulence and buoyancy on friction. The resulting model can be adjusted to fit the data. The results obtained indicate that the buoyancy term which favors the flow of a lighter oil core, is affected by the wavy interface and water flow regime.

Comparisons of the present model with friction pressure drop data in a case where the difference of fluids densities is significant (Bai, 1995) as well as our data, provided very satisfactory agreement.

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