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On the drag reduction for the two-phase horizontal pipe flow of highly viscous non-Newtonian liquid/air mixtures: Case of lubricating grease

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Abstract

The prediction of the pressure drop gradient and the evaluation of the drag reduction phenomenon observed during the piping multiphase flow of a lubricating grease/air mixture have been investigated. With this aim, viscous flow tests in rotational rheometers and pressure drop measurements in pipelines have been carried out using different geometries with both smooth and rough surfaces. The Sisko model has been used to predict the pressure drop gradient. The drag ratio, as a function of air flow rate, for highly viscous pastes such as lubricating greases, significantly differs, qualitative and quantitatively, from that found in the literature for other non-Newtonian fluids with viscosities of around 200 times lower. The pressure drop gradient in the intermittent multiphase flow regime can be predicted by modifying the classical approach of Lockhart and Martinelli with an empirical correction factor. An empirical model, with a combination of power-law and sigmoidal-type equations, has been proposed to describe the experimental evolution of the drag ratio as a function of $Re'_{\mathrm{L}}/Re'_{\mathrm{TP}}$. The accuracy of the proposed model has been tested by estimating the classical Fanning friction factor for a non-Newtonian fluid, $f = 16/Re'$, once the pressure loss has been corrected with the drag ratio previously obtained.

Keywords: Laminar flow; Lubricating greases; Two-phase flow; Non-Newtonian fluid; Rheology; Wall slip

1. Introduction

The two-phase flow of gas and liquid along pipelines is of considerable industrial importance in the transport of liquids and related unit operations. The mechanism of flow of these systems should be fully understood before other transport processes are addressed, such as for instance heat transfer. In a two-phase flow the most important hydrodynamic features to be considered are the determination of the flow pattern, holdup and

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pressure drop, which imply some difficulties due to the complex flow mechanism and the number of parameters involved. During the last thirty years there have been many studies concerning the determination of these design parameters for co-current flow of gas and liquids in pipelines (see for instance Govier and Aziz, 1972; Chisholm, 1983), including the two-phase flow of non-Newtonian liquid and gas mixtures (Carleton et al., 1973; Farooqi and Richardson, 1982; Chhabra et al., 1983, 1984; Dziubinski and Chhabra, 1989; Dziubinski, 1995). However, most of them are referred to relatively low viscous materials, like kaolin and clay suspensions or CMC dispersions, and not to highly viscous paste materials, such as lubricating greases. In the case of highly viscous materials, the effect of inserting gas into the liquid could be extremely important, since slight discrepancies in the prediction of the pressure drop may imply important differences in the power consumption of the pumping system or even a malfunction in liquid distribution.

It is a well-known fact that, when a shear-thinning fluid is flowing through a pipeline at a fixed volumetric flow rate, pressure drop from frictional losses may be reduced by the injection of gas into the fluid. The earliest studies concerning this phenomenon were carried out by Ward and Dallavalle (1954), who injected air into clay suspensions flowing in the laminar regime. This effect, defined as drag reduction, may be explained in a simply and qualitative manner considering two opposite effects. As the rate of air injected into the fluid is increased, the linear velocity rises and, consequently, the pressure gradient along the slug length of fluid becomes greater. On the other hand, air injection reduces the wetted area of pipe surface also reducing shear stresses at the wall. However, the pressure gradient will increase less rapidly than fluid velocity. Therefore, as the pressure gradient along the gas bubble is usually neglected, there will be a net reduction in the overall pressure drop over the system. Nevertheless, at high air flow rates, this effect was found to be reduced, and some conditions have been found where the average pressure gradient with air injection exceeded that for liquid flowing alone at the same superficial suspension velocity.

Oliver and Young-Hoon (1968) studied the drag reduction phenomenon for the slug and elongated-bubble flow regimes in the case of aqueous polymeric solutions. In addition to this, Carleton et al. (1973) observed how air split the liquid into a series of plugs when conveying bentonite pastes, obtaining a series of alternate air and paste plugs, each filling the whole cross-section of the pipe (plug flow mechanism). Dziubinski (1986) and Dziubinski et al. (2004) presented the flow pattern maps for two-phase flow of gas and non-Newtonian fluid in horizontal and vertical pipes, respectively. In general, the distinction between elongated-bubble, plug and slug flows is not so clear and these two-phase flow patterns are usually described as intermittent flow, which is considered of special practical importance because, only in this case, pressure drop has been found to be reduced (Dziubinski, 1995).

Pressure drop gradient in the intermittent multiphase flow regime can be predicted by different correlations, which modify the classical approach of Lockhart and Martinelli (Farooqi and Richardson, 1982). Thus, for instance, it is possible to use the general correlation adding an empirical parameter, C_L, which is believed to depend on the Reynolds number and takes into account the efficiency of plug formation (Carleton et al., 1973). This treatment was initially applied for shear-thinning fluids which clearly exhibit a power-law behaviour, such as kaolin and anthracite suspensions, being tested later on other systems of similar viscosity giving satisfactory results for low values of the Reynolds number (Chhabra et al., 1983). Dziubinski (1995) developed a similar approach leading to a general expression of drag ratio for two-phase pressure drop during the intermittent flow of gas and non-Newtonian liquids based on the concept of loss coefficient, which is an alternative form of the friction factor.

If normally the flow of a non-Newtonian fluid/air mixture through a pipeline is a very complex situation, the case of a lubricating grease is even more dramatic. Lubricating greases are generally highly structured suspensions consisting of a thickener, usually a metal soap, dispersed in mineral or synthetic oil (Mas and Magnin, 1994). The thickener is added to prevent loss of lubricant under operating conditions but, evidently, this implies a considerably resistance to the flow of these materials. There are, at least, two problems which must be taken into account in the pipeline flow of greases: wall slip and the effect of air bubbles entrainment when the system is primed, even when no air injection was introduced deliberately. Some previous studies (Sacchettini et al., 1985; Froishtener et al., 1989; Mas and Magnin, 1994) have reported wall slip phenomena in disperse or colloidal systems, including greases. The most common explanation for these systems is the formation of a depleted layer at the boundaries as, for instance, the wall of the sensor system in a rheometer or the wall of pipes or tubes, with lower viscosity than the bulk, which induces a lubrication effect (Barnes, 1995).

It depends on the solid surface, the nature of the fluid and the system geometry. In some applications, the most important feature is not to eliminate wall slip but quantify the extension of it in order to make a correct design of industrial operations. On the other hand, there is a certain amount of air that is inserted in the pipe by the pump, and this is not avoidable, for instance in the case of screw pumps or when handling highly viscous materials. It has been verified that small flow rates of air cause large reductions in the pressure drop, and this effect seems to be more important as the viscosity of the fluid increases (Delgado et al., 2005).

Rein and McGahey (1965) introduced a simple method for the determination of the pressure drop due to friction losses for greases. This is a practical method from a practical point of view and is still used nowadays as an ASTM standard method (ASTM D 1092). It is based on a simple substitution of the Newtonian viscosity, μ, for an apparent non-Newtonian viscosity, η, at each flow rate, in the Hagen–Poiseuille equation for laminar flow (Metzner and Reed, 1955). The applicability of this short-cut method and the discussion of the above mentioned flow problems found in the pumpability of greases were discussed elsewhere (Delgado et al., 2005). Taking these considerations into account, the main aim of this work was to develop a systematic study of the influence of air injection on the pressure drop reduction during the flow of lubricating greases through a pipeline and establish a general method, based on empirical modifications of the Lockhart–Martinelli approach, to validate the experimental pressure drop data, especially at very low values of the Reynolds number, not easily found in the literature.

2. Theoretical background

The comparison between the pressure drop with and without air injection has been traditionally carried out in the form of the drag ratio coefficient (see Eq. (1)), which is applicable to any flow pattern:

$$\Phi_{\mathrm{L}}^2 = \frac{(-\Delta P_{\mathrm{TP}}/L)}{(-\Delta P_{\mathrm{L}}/L)} \tag{1}$$

where $(-\Delta P_{\mathrm{TP}}/L)$ is the two-phase pressure gradient and $(-\Delta P_{\mathrm{L}}/L)$ is the pressure gradient for a liquid flowing alone at the same superficial velocity as in the two-phase mixture.

At low air superficial velocities, less than about 1 m/s, the magnitude of the drag reduction may be adequately predicted by a simple plug flow model, initially proposed by Carleton et al. (1973) and developed later by Richardson and co-workers (Heywood and Richardson, 1978; Farooqi and Richardson, 1982; Chhabra et al., 1983). This model considers that air and liquid are formed into discrete flat-ended plugs filling the whole cross-section of the pipe, which implies no slip velocity between phases, pressure gradient along air plugs is negligible, pressure gradient along liquid plugs is equivalent as in conventional pipeline operating at the same velocity and the length of plugs is short compared with the total length of pipeline. The two-phase pressure drop may be then written as

$$-\frac{\Delta P_{\mathrm{TP}}}{L} = \frac{2 f_{\mathrm{TP}} u_{\mathrm{M}}^2 \rho_{\mathrm{L}} \lambda_{\mathrm{L}}}{D} \tag{2}$$

where f_{TP} is the two-phase Fanning friction factor. The mixture velocity is given by

$$u_{\mathrm{M}} = u_{\mathrm{SG}} + u_{\mathrm{SL}} \tag{3}$$

with u_{SG} and u_{SL} as the superficial velocities of gas and liquid, respectively, taken as volumetric flux divided by pipe cross-sectional area of diameter D. ρ_{L} is the liquid density, and the input volume fraction of liquid is

$$\lambda_{\mathrm{L}} = \frac{u_{\mathrm{SL}}}{u_{\mathrm{M}}} \tag{4}$$

For the estimation of f_{TP}, the rheological model that describes the flow behaviour of the liquid must be taken into account. Considering the power-law model,

$$\tau = k \cdot \dot{\gamma}^n \tag{5}$$

where τ is shear stress, $\dot{\gamma}$ shear rate, n the flow index, related to the slope of the shear-thinning region, and k the consistency coefficient, the usual expression (Heywood and Richardson, 1978; Chhabra et al., 1983) for the two-phase generalized Reynolds number is given as

$$Re'_{\mathrm{TP}} = \frac{8D^n \rho_{\mathrm{L}} u_{\mathrm{M}}^{2-n}}{k\left(\frac{3n+1}{4n}\right)^n 2^{3n}} \tag{6}$$

More elaborated expressions must be developed for other phenomenological models such as the Sisko model (Turian et al., 1998):

$$\tau = \eta_\infty \cdot \dot{\gamma} + m \cdot \dot{\gamma}^n \tag{7}$$

$$Re'_{\mathrm{TP}} = \frac{\rho_{\mathrm{L}} u_{\mathrm{M}} D}{\eta_\infty} \cdot \frac{G(n,X)}{(1+X)} \tag{8}$$

where η_∞ is the high shear rate limiting viscosity, m the consistency parameter for the Sisko model and n the flow index, where

$$G(n,X) = \frac{\left[1 + 4\left[\left(\frac{n+2}{n+3}\right)X + \left(\frac{2n+1}{2n+2}\right)X^2 + \left(\frac{n}{3n+1}\right)X^3\right]\right]}{(1+X)^3} \tag{9}$$

$$X = \frac{m}{\eta_\infty} \cdot \dot{\gamma}^{n-1} \tag{10}$$

For the liquid flowing alone at the same superficial velocity, the pressure drop is

$$\frac{-\Delta P_{\mathrm{L}}}{L} = \frac{2 f_{\mathrm{L}} u_{\mathrm{SL}}^2 \rho_{\mathrm{L}}}{D} \tag{11}$$

Therefore, combining Eqs. (1), (2) and (11), the following expression for the drag ratio can be obtained:

$$\Phi_{\mathrm{L}}^2 = \frac{1}{\lambda_{\mathrm{L}}} \cdot \frac{f_{\mathrm{TP}}}{f_{\mathrm{L}}} \tag{12}$$

In the laminar flow regime, the friction factor may be expressed as

$$f = \frac{16}{Re'} \tag{13}$$

As a result, a general expression for the drag ratio is as follows:

$$\Phi_{\mathrm{L}}^2 = \frac{1}{\lambda_{\mathrm{L}}} \cdot \frac{Re'_{\mathrm{L}}}{Re'_{\mathrm{TP}}} \tag{14}$$

For a power-law liquid/air two-phase flow in the laminar regime, using Eqs. (4), (6), (13) and the classical definition of Re'_{L}, the drag ratio is reduced to the well-known following expression:

$$\Phi_{\mathrm{L}}^2 = \lambda_{\mathrm{L}}^{1-n} \tag{15}$$

In order to calculate the deviations between the theoretical values of the drag ratio deduced from Eq. (14) or (15) and the values obtained from the experimental tests, it is necessary to introduce a coefficient C_{L}, related to the efficiency of the plug formation (Carleton et al., 1973), which is defined as the ratio between experimental and theoretical two-phase pressure drops:

$$C_{\mathrm{L}} = \frac{\left(-\frac{\Delta P_{\mathrm{TP}}}{L}\right)_{\mathrm{EXP}}}{\left(-\frac{\Delta P_{\mathrm{TP}}}{L}\right)_{\mathrm{TH}}} \tag{16}$$

Consequently, the general expression for the drag ratio applicable to any rheological model is

$$\Phi_{\mathrm{L}}^2 = C_{\mathrm{L}} \cdot \frac{1}{\lambda_{\mathrm{L}}} \cdot \frac{Re'_{\mathrm{L}}}{Re'_{\mathrm{TP}}} \tag{17}$$

In this paper, the experimental values of the drag ratio have been obtained as the ratio between the pressure drop measured in the experimental device and that calculated from the corresponding pressure drop expression, according to the Sisko model, for the single phase flow of the lubricating grease. This equation may be written as (Turian et al., 1998):

$$(-\Delta P) = \left[\frac{8\eta_\infty^3 \dot{\gamma}_W^4 L^3}{u_{SL} D^2} \cdot (1+X)^3 \cdot G(n,X)\right]^{1/3} \tag{18}$$

where η_∞, m and n are the parameters obtained from rotational rheometry, with $G(n, X)$ and X defined by Eqs. (9) and (10), and where $\dot{\gamma}_W$ is the wall shear rate.

3. Materials and methods

3.1. Materials

A commercial lithium complex soap grease kindly supplied by Verkol S.A. (Spain), classified into NLGI grade 2, was used as received. This lubricating grease presents a density, at 20 °C, of 914 kg/m^3 and shows a marked shear-thinning behaviour with apparent viscosities ranging from 5.6×10^4 Pa s at 0.01 s^{-1} to 3 Pa s at 1000 s^{-1}, at the same temperature. Other technological characteristics are shown in Table 1.

3.2. Experimental pumping system

Pumping setup consisted of a force feed screw pump (MV2.6IVA10, PCM Moineau, France), with maximum capacity of 0.26 m^3/h and discharge pressure of 10 bar, which continuously drove the grease stored in a hopper tank to the pipeline section. A controlled-speed electric motor of 0.37 kW and 1500 rpm, model Varmeca-10 (Leroy–Sommer, France) was incorporated in order to apply different flow rates, ranging from 0.04 to 0.26 m^3/h. Grease flow rates were measured by collecting the fluid in a weighting tank during a given time. Six horizontal stainless steel exchangeable pipes (Table 2) with different diameters and roughness were used. Stainless steel meshes were fixed to the internal wall of three of them to modify their roughness. For smooth pipes, a roughness under 3×10^{-3} mm was assumed negligible. Two bourdon-type manometers placed at enough distance from the entrance and the exit, longer than 40 D and 30 D, respectively, (see Table 2), were used to measure the local pressure avoiding any end effect. Entrance and exit lengths were selected according to previous work (Delgado et al., 2005). These manometers had a measurement range of 0–10 bars and 1% error. Pressure drop measurements were always taken over the same L/D ratio in the pipeline test section. Finally, a loop dealing with a 1 1/4″ transparent PVC tube, which returns the grease to the hopper tank, provides experimental evidences of the two-phase flow pattern involved in each measurement. This tube was also

Table 1
Technical specifications of the lubricating grease studied

Property	Value
NLGI grade	2
Penetration (after 60 strokes, dmm)	265–295
Operating temperature (°C)	−30 to 180
Oil separation (% w)	<5
Base oil	Mineral

Table 2
Geometrical characteristics of pipelines

Pipe	Inside diameter (mm)	Relative roughness	L/D	Entrance length (mm)	Test section length (m)	Exit length (mm)
1 1/4″ smooth	36.62	0.001	60	603	2.197	200
1″ smooth	27.86	0.001	60	1128	1.672	200
3/4″ smooth	22.45	0.001	60	1453	1.347	200
1 1/4″ rough	34.22	0.033	60	603	2.197	200
1″ rough	25.46	0.043	60	1128	1.672	200
3/4″ rough	20.85	0.037	60	1453	1.347	200

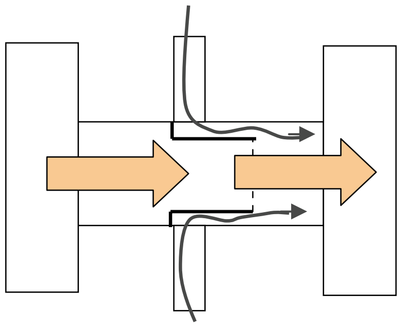

Fig. 1. Schematic picture of the air distributor in the pipeline.

placed along the test section to ensure that there was no change in the flow pattern all along the experimental system. No significant differences in pressure drop were observed in pipes of stainless steel and PVC for the same diameter, therefore it was assumed that pipe material does not influence the flow pattern.

Air was taken from a pressurized line and injected after the discharge point of the pump through a stainless steel piece (see Fig. 1), which was designed to axially inject air lengthways near the pipe wall while the grease is flowing across the axis. The injection of the entire amount of air perpendicularly in a given point makes the adequate distribution between phases impossible. Thus, although Carleton et al. (1973) concluded that the geometry of air distributor did not seem to be a critical factor, further research on this topic should be addressed especially for highly viscous pastes such as lubricating greases. Air flow rate was determined using two flowmeters for gases model 2100 (TECFLUID, Spain), which provide flow measurements in the range of 0.02–0.2 and 0.2–1.2 m^3/h, respectively. Pressure drop experiments were carried out at 22 ± 2 °C. Tests never lasted more than 15 min and a new test was always carried out at least 1 h later than the previous one, in order to let the structure of the grease recover from one test to another. Measurements were replicated at least three times.

3.3. Rheological measurements

The lubricating grease tested was rheologically characterized by using different rotational rheometers, which provide data in a wide range of shear rates. Thus, viscous flow measurements were performed with a Bohlin CS controlled-stress rheometer (Bohlin, Sweden), in a range of shear stress of 4–2500 Pa, using a plate–plate geometry (25 mm of diameter and 1 mm of gap). In addition, measurements with both a RV20-CV20N controlled-shear rate rheometer (Haake, Germany), in a range of shear rate of 10–2000 s^{-1}, using a coaxial cylinder geometry (15 mm external diameter, 12 mm length, 0.545 mm gap), and an ARES (Rheometric Scientific, USA) controlled-strain rheometer, in a range of 0.01–10 s^{-1}, using a plate–plate geometry of 25 mm with 1 mm gap, were also carried out. Plate–plate geometries with both smooth and rough surfaces (roughness: 0.4 mm) were used in order to quantify the extension of wall slip phenomena. All measurements were done at 22 ± 0.5 °C, following the same thermal protocol, i.e., 30 min resting time at the selected temperature, and replicated, using new unsheared samples, at lest three times.

4. Results and discussion

4.1. Viscous flow characterization

Traditionally, lubricating greases have been considered as a classical yielding material (Bondi, 1960), with an apparent yield stress as a characteristic parameter over an extended region and a subsequent shear-thinning flow region at higher shear rates. Fig. 2 shows the viscous flow curves (shear stress vs. shear rate and viscosity vs. shear rate) for the lubricating grease studied, obtained in rotational rheometers using smooth and rough plate–plate geometries, in a relatively wide range of shear rate. Two different flow regions are noticed. Thus, at low shear rates, a slight increase in shear stress with shear rate is observed, associated to the above mentioned

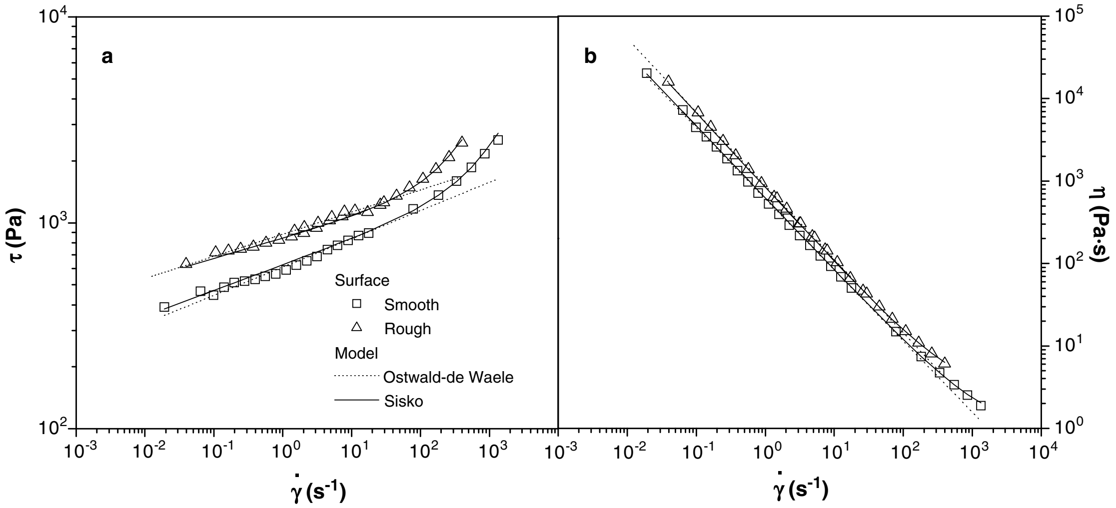

Fig. 2. Viscous flow curves, obtained with both smooth and roughened geometries, for the lubricating grease studied. (a) Shear stress vs. shear rate and (b) viscosity vs. shear rate.

Table 3
Fitting parameters of the Sisko and power-law models for measurements carried out in smooth and roughened geometries

Model	Geometry	k (Pa s^n)	n	m	η_∞ (Pa s)
Ostwald-de Waele	Smooth	610	0.14		
	Rough	881	0.11		
Sisko	Smooth		0.12	625	0.89
	Rough		0.098	842	2.54

yielding behaviour, which is followed, at shear rates higher than 10 s^{-1}, by a less pronounced shear-thinning region. However, this last region is limited to shear rates of around 10^3 s^{-1}, due to the appearance of shear fracture and the consequent ejection of the sample from the gap. This particular flow behaviour was associated to the non-monotonous evolution of the shear stress vs. shear rate observed, which corresponds to a dynamically non-stable region (Coussot et al., 1993; Bertola et al., 2003) and may be related to a non-homogeneous field of velocities during the viscometric flow. This fact makes the experimental results highly dependent on the geometry used (Balan and Franco, 2001). Thus, as can be clearly observed in Fig. 2, a significant influence of the roughness of the measuring tool exists in the shear rate range studied. This effect is widely accepted as an experimental evidence of wall slip phenomena. In a previous work (Delgado et al., 2005), the dependence of the surface, geometry and gap of the measuring tool used on the lubricating grease viscosity has been demonstrated. Thus, the use of rough surfaces overcame slip effects in both rotational rheometry and pipeline flow. Taking into account these considerations, the power-law and the Sisko models have been used to fit the experimental rheological response obtained with both smooth and rough measuring tools. The fitting parameters for both models are presented in Table 3. These values represent the viscous flow behaviour of typical greases classified into NLGI grade 2. As shown in Fig. 2, both models fit fairly well the experimental results obtained in a range of shear rates between 10^{-2} and 10 s^{-1}, with very low values of the flow index. However, at higher shear rates, due to the above mentioned flow behaviour, a considerable deviation between both models can be noticed, with the Sisko model closer to experimental flow behaviour in the whole shear rate range studied.

4.2. Experimental pressure drop

Selected experimental two-phase pressure drop raw data are shown in Fig. 3, as a function of air flow rate, for a lubricating grease/air mixture flowing along a smooth pipe (1 in.) at different liquid flow rates. This figure

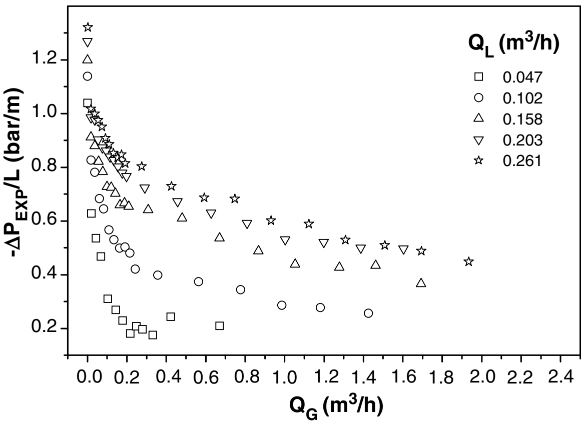

Fig. 3. Experimental pressure drop, as a function of the volumetric air rate, in a smooth pipeline (1 in.) for the lubricating grease studied.

demonstrates that the injection of air reduces pressure drop significantly. As can be observed, this drag reduction is very important upon injection of relatively low air flow rates and increases as liquid flow rate decreases. However, the pressure drop tends to reach constant values after further increases in air flow rate. The final values must be close to the characteristic minimum which determines the transition to the turbulent flow regime (Heywood and Richardson, 1978; Farooqi et al., 1980; Chhabra et al., 1984). This transition has not been experimentally found for the lubricating grease studied, due to its high viscosity.

These results are also presented in the form of the drag ratio vs. superficial gas velocity plots (Fig. 4), for a wide range of experimental conditions and pipes studied. In this way, the pressure drop reduction due to air injection, in relation to the grease flowing on its own at the same volumetric rate, can be easily evaluated. The pressure drop due to the grease flow ($-\Delta P_L/L$) has been estimated from the expression derived using the Sisko model (Eq. (18)), after a recurrent calculation of the necessary parameters (Delgado et al., 2005). Although the Sisko model fits the experimental viscous flow curves much better, ($-\Delta P_L/L$) has also been calculated by using the power-law model parameters. No significant differences have been found.

The rheological parameters determined from the fitting of the above mentioned rheological models to the experimental flow curves obtained with roughened tools should be used to calculate the drag ratio when a lubricating grease/air mixture flows along a pipe with internal rough surfaces (Fig. 4d). On the contrary, the use of rheological parameters calculated from flow curves obtained with smooth geometries (see Table 3) for describing the flow along rough pipes would yield higher discrepancies between experimental and predicted (Eq. (14)) values. This fact is due to the absence of wall slip phenomena in roughened pipelines but not in the smooth measuring tools of the rheometer.

As can be seen, Fig. 4 also shows the theoretical prediction of the drag reduction, deduced from Eq. (14). The most relevant aspect deduced from this plot is that this model overestimates the values of the drag ratio coefficient for low superficial air velocities and, on the contrary, underestimates this coefficient at higher values of the superficial air velocity. Moreover, when liquid velocity is increased, deviations from the model become progressively larger. The fact that the consideration of an idealised plug model is likely to underestimate the magnitude of the two-phase pressure drop was previously mentioned by Chhabra et al. (1984). In this sense, Dziubinski (1995) obtained very good accuracy in the description of a wide range of experimental data in the case of the intermittent flow of gas and non-Newtonian liquid mixtures by inserting a correction factor (see Eq. (17)) which depended on Re'_{TP} and Re'_L. This author concluded that, for $Re'_{TP} < 500$, no correction coefficient was needed. However, as it may be seen in Fig. 4, this is obviously not the case of highly viscous materials such as lubricating greases, where the superficial liquid velocities and the values of the Reynolds number are extremely low. In fact, so highly viscous systems have never been studied hitherto. As can be seen in Fig. 5, which compares the experimental drag ratio values obtained with lubricating greases with those found in the literature for non-Newtonian slurries for approximately the same pipe diameters (Heywood and Richardson, 1978; Farooqi et al., 1980; Farooqi and Richardson, 1982), the drag reduction shown by a lubricating grease/

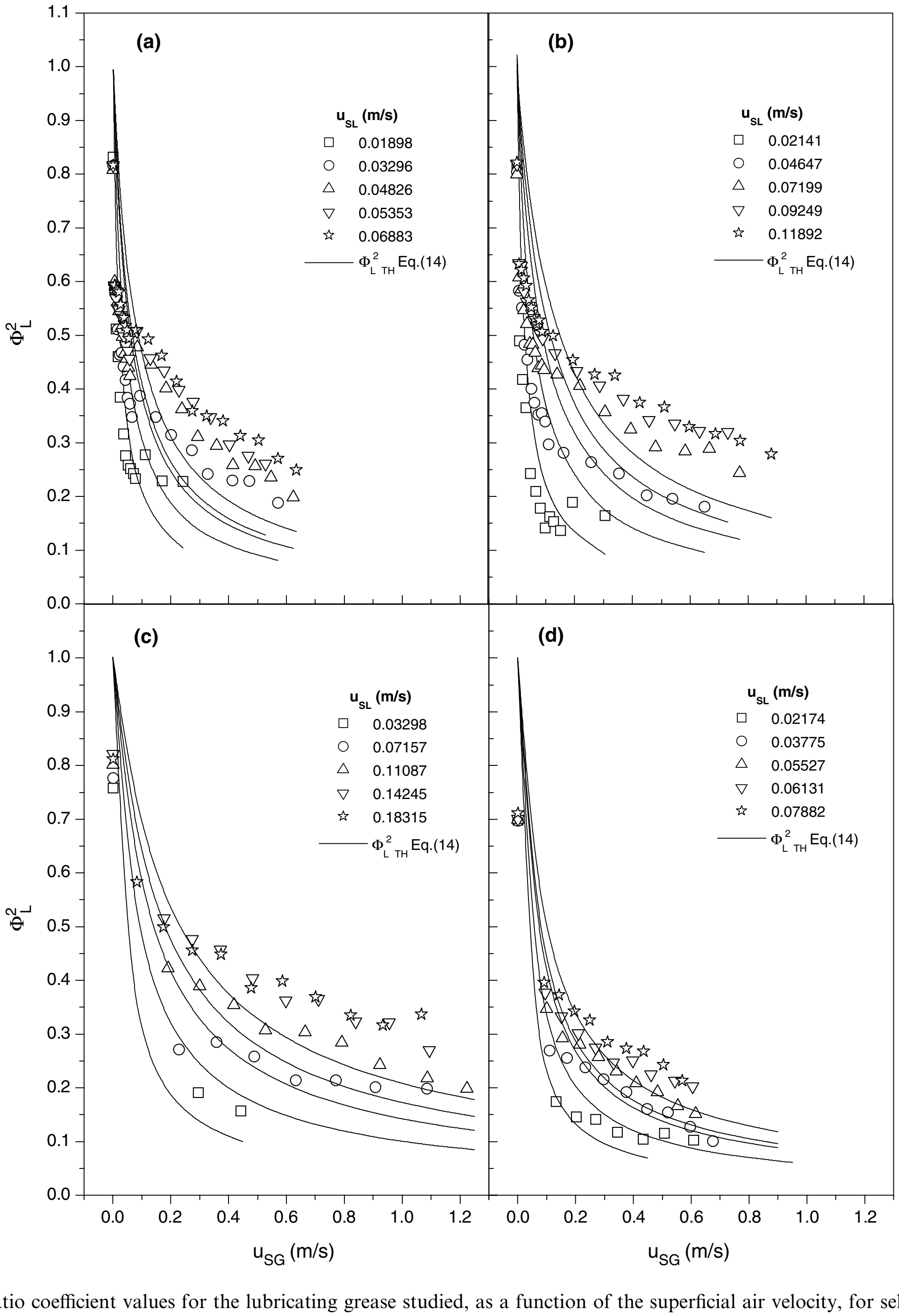

Fig. 4. Drag ratio coefficient values for the lubricating grease studied, as a function of the superficial air velocity, for selected pipelines: (a) 1 1/4 in. smooth, (b) 1 in. smooth, (c) 3/4 in. smooth and (d) 1 1/4 in. rough.

air mixture is significantly higher than that found with relatively low viscous non-Newtonian suspensions/air systems, at very low values of the superficial air velocity.

Dukler et al. (1964) discussed the effect of pipe diameter as early as 1964. It was noted that the correlations developed from data in small pipes cannot be expected to predict adequately the total pressure loss in larger diameters, unless an acceleration term is accounted for, because the effect of gas expansion takes more relevance

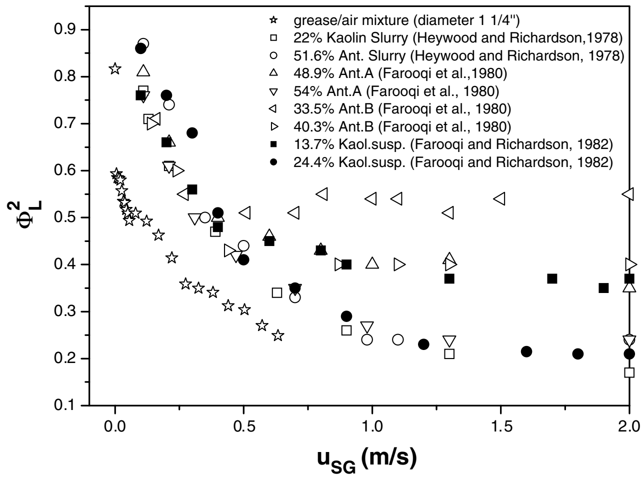

Fig. 5. Comparison between the drag ratio coefficient values for a lubricating grease/air mixture and those found in the literature for other disperse systems, as a function of the superficial gas velocity.

for small diameters. Nevertheless, most of the work reported up to date has been carried out in horizontal pipes of around 42 mm and, consequently, little information is available about the effect of pipe diameter. Fig. 6 depicts the effect of pipe diameter on the experimental drag ratio for the piping flow of lubricating grease/air mixtures at selected grease flow rates. As can be observed, the drag ratio decreases as pipe diameter increases, especially at high superficial air velocities. Therefore, upon air injection, the pressure drop reduction seems to be more important in large pipe diameters. These results are consistent with those reported by Heywood and Richardson (1978), who used pipelines of 42 and 158 mm diameters and pointed out that the drag reduction appears to be consistently more marked in large pipelines, concluding that the effect increases with the pipe diameter for constant superficial air and liquid velocities. Farooqi and Richardson (1982) found some similar evidences.

4.3. Two-phase flow patterns

Three two-phase flow patterns have been detected by visual observation through the PVC transparent tube. Initially, at low gas flow rates, air flows quite homogeneously, in the form of small elongated bubbles dispersed in the lubricating grease (see Fig. 7a), with larger sizes as air flow rate increases. This flow pattern rapidly evolves, approximately for $Q_G \cong 0.05\ m^3/h$, to a slug flow pattern in which bubbles coalesce into more elongated bubbles, taking up nearly the whole section of the pipe, which leads to a disturbed slug flow. In this regime the length of air bubbles, as well as its frequency, changes remarkably (Fig. 7b). Finally, for $Q_G > 0.15\ m^3/h$, plugs of air appear clear and completely developed, increasing their length, in relation to the grease plugs, with air flow rate (Fig. 7c and d). As has been mentioned before, and despite the initial bubbles, air tends to split the lubricating grease in portions, due to its viscosity, which fill the entire cross-section of the pipe.

It must be pointed out that a higher drag reduction than expected was observed at low air flow rates (Fig. 4) which corresponds to the flow pattern showed in Fig. 7a. On the contrary, at high air flow rates, especially those corresponding to a developed plug flow pattern, the experimental drag ratio was higher than predicted by the traditional model (Eq. (14)). As it was mentioned above, the drag reduction phenomenon is mainly due to two opposite physical effects. In this case, phase distribution is highly affected by the liquid viscosity. On one hand, at low air flow rates, the presence of relatively small dispersed bubbles significantly reduces the wetted area of pipe surface but, however, these bubbles are not able to increase the linear velocity of grease by decompression. On the other hand, the decompression effect is predominant above a critical value of air flow rate, as a consequence of a dramatic change in phase distribution. Instead of taking the form of bubbles, air tends to form flat-ended plugs following the grease ones whose net effect is to increase the linear velocity of grease, thus damping two-phase pressure drop decay, which is favoured by the high viscosity.

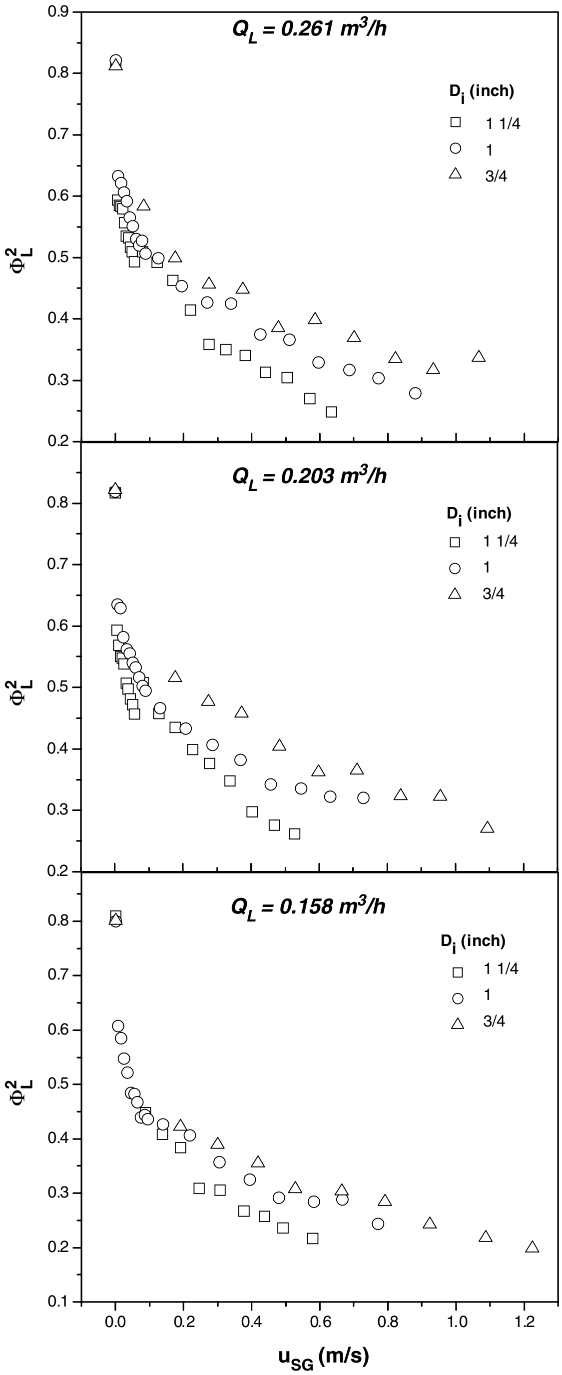

Fig. 6. Influence of pipe diameter on the drag ratio coefficient for the lubricating grease studied.

4.4. General correlation for the experimental drag ratio

Fig. 8 summarises all the experimental data obtained, in the form of $\Phi_L^2 \cdot \lambda_L$ vs. Re'_L/Re'_{TP}. In this figure, the dash line represents the prediction of the theoretical general expression for the drag ratio under idealised plug flow conditions (Eq. (14)). As may be seen in this figure, the experimental data clearly deviate from the theoretical prediction in two different manners. At low values of Re'_L/Re'_{TP}, Eq. (14) underestimates the

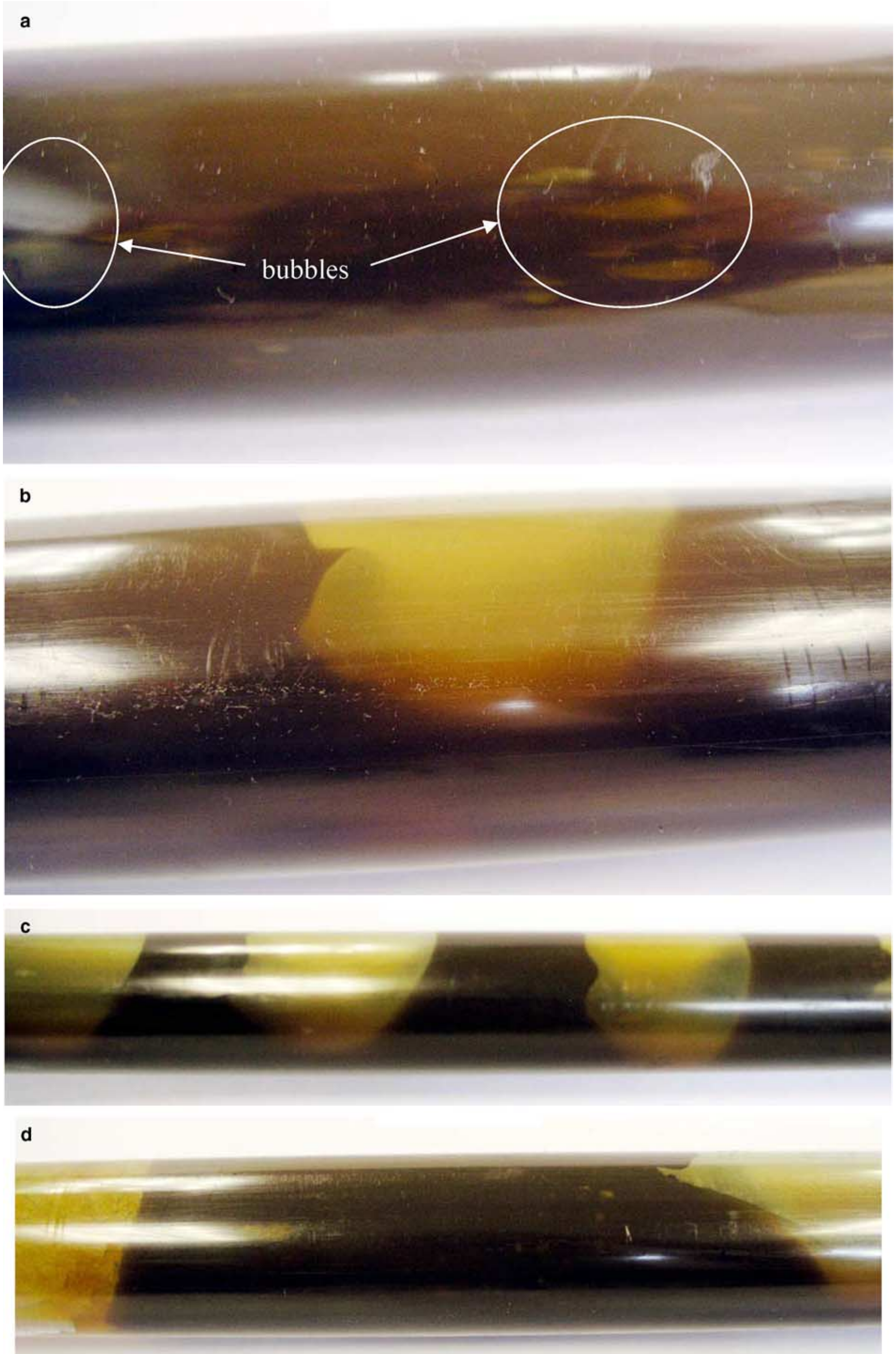

Fig. 7. Experimental flow patterns found during the lubricating grease/air two-phase flow: (a) bubble flow, (b) slug flow, (c) incipient plug flow and (d) fully developed plug flow.

experimental drag ratio. This deviation has been previously reported in the literature (see Fig. 8) and even empirically modelled by Dziubinski (1995). However, as has been mentioned before, this model predicts almost the same values that Eq. (14) for $Re'_{\mathrm{TP}} < 500$, and, therefore, it is not adequate to describe the experimental drag ratio found with highly viscous fluids like greases. On the other hand, for values of $Re'_{\mathrm{L}}/Re'_{\mathrm{TP}}$

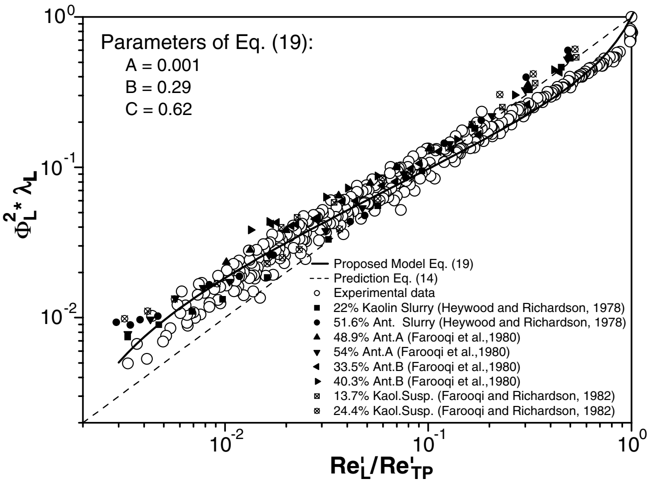

Fig. 8. Experimental and predicted values of $\Phi_L^2 \cdot \lambda_L$ vs. Re'_L/Re'_{TP}.

close to 1, i.e., at very low air flow rates, the opposite deviation is found, being the experimental data overestimated by Eq. (14), in accordance with a dramatic decrease in pressure loss (see Fig. 3) for extremely low air flow rates. This dramatic decrease in pressure loss was reported in a previous work (Delgado et al., 2005), where just the amount of air introduced by the pump was considered. Consequently, this complex behaviour cannot be predicted by a relatively simple power-law model. Thus, a combination of a power-law and a sigmoidal-type equation was proposed to describe the experimental evolution of the drag ratio.

$$\Phi_L^2 \cdot \lambda_L = \left(\frac{Re'_L}{Re'_{TP}}\right)^A + \left[\frac{B \cdot Re'_L/Re'_{TP}}{(B+1) - Re'_L/Re'_{TP}}\right]^C - 1 \tag{19}$$

where A, B and C are experimental fitting parameters. As can be observed, this model fits fairly well the experimental data obtained from pipes of different diameters and internal relative roughness, in all the experimental range of air and grease flow rates studied. In addition to this, Eq. (19) reasonably predicts, at low and intermediate values of Re'_L/Re'_{TP}, the drag ratio values found in the literature, obtained in the laminar regime with liquids of significantly lower viscosity, although fails for very low values of air flow rates, for which the drag ratio tends to follow Eq. (14). In this sense, the influence of the superficial velocity and the apparent viscosity of the liquid phase on the drag ratio, for similar flow indexes, has been previously pointed out (Heywood and Richardson, 1978; Dziubinski and Chhabra, 1989), i.e., the higher the consistency index or the lower the superficial liquid velocity, the higher the drag reduction is. This fact is in accordance with the results shown in this work, taking into account that the consistency index of the lubricating greases is 200 times higher, or even more, than those available in the literature. However, in view of the increasing importance of the highly viscous non-Newtonian fluids, the effects of viscosity and flow index require a more intensive research.

From Eqs. (17) and (19), a general expression for the correction coefficient C_L, which modifies the theoretical drag ratio, may be easily deduced as a function of the ratio Re'_L/Re'_{TP}:

$$C_L = \left(\frac{Re'_L}{Re'_{TP}}\right)^{A-1} + \frac{B^C \cdot \left(Re'_L/Re'_{TP}\right)^{C-1}}{\left[(B+1) - Re'_L/Re'_{TP}\right]^C} - \frac{Re'_{TP}}{Re'_L} \tag{20}$$

The accuracy of this correlation has been tested by plotting the experimental values of the drag ratio vs. the predicted ones, calculated from Eq. (19). As Fig. 9 shows, the use of Eq. (20) allows a good prediction of the drag reduction for the lubricating grease/air mixture studied. Thus, most of the experimental values are well inside the 30% deviation region, which is a reasonably good agreement between theoretical and experimental data. Similar deviations between experimental and predicted values have been found in other investigations with non-Newtonian/air mixtures like, for instance, by applying the correction proposed by Dziubinski (1995), which shows a deviation of ±15% for laminar flow and ±25% for turbulent flow, or the extension of the Hubbard–Dukler model with maximum errors higher than ±20% (Farooqi et al., 1980). However, for rough pipes, the use of rheological fitting parameters obtained with smooth surfaces (see Table 3), or vice versa, provides errors significantly higher than 30% (data not shown) because of the non-adequate consideration of wall slip phenomena as previously mentioned. Data collected from different references (Heywood and Richardson, 1978; Farooqi et al., 1980; Farooqi and Richardson, 1982) have also been included to test the proposed model with other non-Newtonian fluids showing relatively low viscosities. As has been previously pointed out, in those cases, the model clearly fails at low air flow rates, because a relatively good agreement with Eq. (14) is observed.

Finally, the proposed method has also been checked by estimating the Fanning friction factor from Eq. (11), once the pressure loss has been corrected with the drag ratio obtained from Eq. (19), as a function of the lubricating grease generalized Reynolds number. As can be observed in Fig. 10, the classical expression for the laminar regime is obeyed in all cases fairly well.

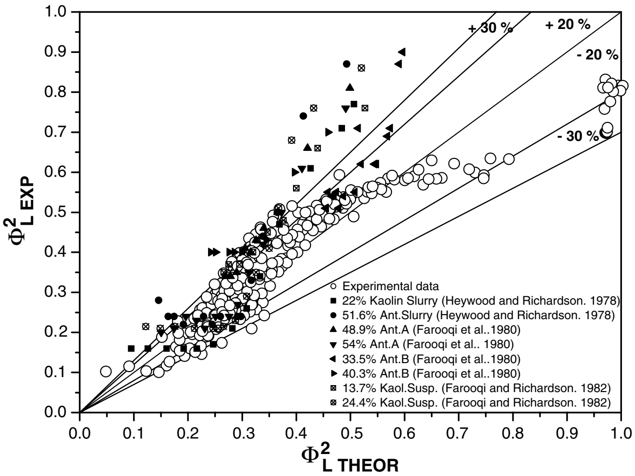

Fig. 9. Comparison between experimental and theoretical, obtained from Eq. (19), values of the drag ratio, for the lubricating grease/air two-phase flow studied in this work and for others systems reported in the literature.

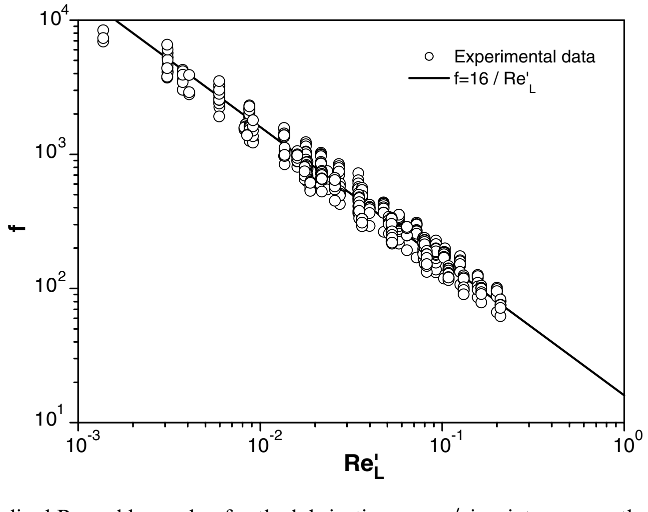

Fig. 10. Friction factor vs. generalised Reynolds number for the lubricating grease/air mixture, once the pressure loss has been corrected with the drag ratio obtained from Eq. (19).

5. Conclusions

Drag reduction, as a function of air flow rate, during the piping multiphase flow of a lubricating grease/air mixture, significantly differs from that found in the literature for other non-Newtonian fluid/air mixtures with viscosities of around 200 times lower than these highly viscous pastes. The Sisko model has been used to predict the pressure drop gradient. In general, drag reduction appears to be dramatic by injecting relatively low flow rates of air, even more as liquid flow rate decreases, although it is dampened by increasing the volumetric flow rate of air. Sisko's parameters, obtained from the fitting of the viscous flow curve measured with roughened tools, must be used to obtain the drag ratio when grease/air mixture flows along a pipe with internal rough surfaces, in order to avoid wall slip phenomena. Drag reduction increases with pipe diameter, especially at high superficial air velocities.

Experimental data deviate from the theoretical general expression for the drag ratio under idealised plug flow conditions in two different manners. At low values of $Re'_{\mathrm{L}}/Re'_{\mathrm{TP}}$, this expression underestimates the experimental drag ratio, as is typically reported in the literature. On the other hand, for $Re'_{\mathrm{L}}/Re'_{\mathrm{TP}}$ close to 1, the opposite deviation has been observed. Thus, the experimental data are overestimated due to the dramatic decrease in pressure loss for extremely low air flow rates. The pressure drop gradient can be predicted by modifying the classical approach of Lockhart and Martinelli with an empirical correction factor. An empirical model, with a combination of power-law and sigmoidal-type equations, has been proposed to describe the experimental evolution of the drag ratio with $Re'_{\mathrm{L}}/Re'_{\mathrm{TP}}$. The accuracy of the proposed model has been tested by estimating the classical Fanning friction factor for a non-Newtonian fluid, $f = 16/Re'$, once the pressure loss has been corrected with the drag ratio previously obtained. In order to propose a more general model taking into account the hydrodynamics phenomena of pipeline flow, further investigation should be addressed. In this sense, some experiments involving non-Newtonian fluids within a wide range of viscosities are being developed presently.

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