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A scheme of correlation for frictional pressure drop in steam-water two-phase flow in helicoidal tubes

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ABSTRACT

In the nuclear field, helically coiled tube Steam Generators (SGs) are considered as a primary option for different nuclear reactor projects of Generation III+ and Generation IV. For their characteristics, in particular compactness of the component design, higher heat transfer rates and better capability to accommodate thermal expansion, they are especially attractive for Small-medium Modular Reactors (SMRs) of Generation III+.

In this paper, starting from two existing databases, a new correlation is developed for the determination of the two-phase frictional pressure drop. The experimental data cover the ranges 5 to 65 bar for the pressure, 200 to 800 kg/m^{2}s for the mass flux and 0 to 1 for the quality. Two coil diameters have been considered, namely 0.292 m and 1.0 m. The coil diameter in particular is crucial for a correct estimation of the two-phase frictional pressure drop. Actually, no general correlation reliable in a wide range of coil geometries is available at the moment. Starting from the noteworthy correlation of Lockhart and Martinelli, corrective parameters are included to account for the effect of the centrifugal force, introduced by the helical geometry, and the system pressure. The correlation is developed with the aim to obtain a form of general validity, while keeping as low as possible the number of empirical coefficients involved.

The average relative deviation between the correlation and the experimental data is about 12.9 % on the whole database, which results the best among numerous literature correlations. In addition, the new correlation is characterized by an extended range of validity, in particular for the diameter of the coil.

KEYWORDS: Helical pipes; Two-phase flow; Frictional pressure drop; Empirical correlation; Lockhart-Martinelli correlation.

1 INTRODUCTION

Helical pipes and heat exchangers with helical coils are extensively used in different industrial fields and applications, including hot water heaters, chemical process reactors, industrial and marine boilers, cooling systems and blood oxygenators, among many others (Bejan and Kraus, 2003). They provide a substantial improvement in heat and mass transfer rates and, most important for boiling and evaporation, a significant enhancement of the critical heat flux (Bejan and Kraus, 2003). Despite helical pipes have been applied in the past for Steam Generators (SGs) in nuclear power plants (Advanced Gas Reactor (AGR), Fort St. Vrain HTGR, THTR 300, Otto Hahn nuclear ship), at the moment they are experiencing a renewed interest in the nuclear field. In fact helical pipes are considered as a primary option for SGs of different nuclear reactor projects of Generation III+ and Generation IV, aiming at improving safety, sustainability, performance and costs of nuclear power plants through the adoption of new technological solutions and the improvement of single plant components. Helical tube heat exchangers offer better heat transfer characteristics, improved capability to accommodate tube thermal expansion and compactness of the design (Cinotti et al., 2002). Among other projects, helical pipes are very attractive for Small-medium Modular Reactors (SMRs) of Generation III+, which require in particular compactness as all the primary system components are located inside the reactor vessel (Kim et al., 2003; Carelli et al., 2004).

Many researchers have focused their attention on the thermal hydraulic characteristics of the flow inside helical pipes: the various works available in literature have been discussed in different comprehensive reviews (Berger et al., 1983; Shah and Joshi, 1987; Naphon and Wongwises, 2006). Concerning the single-phase flow, advances have been made in the understanding of the physical phenomena that characterize the fluid dynamics and the heat transfer. As a matter of fact, the literature provides different tools able to predict with a high degree of accuracy the heat transfer coefficients and the pressure drop. Differently, for the two-phase flow the subject is much more complex. Consequently, more research is needed to improve the anyway remarkable results already achieved on fields such as the two-phase pressure drop, the two-phase heat transfer and the thermal crisis. In particular, this paper is focused on the prediction of the two-phase frictional pressure drop. As already reported by Santini et al. (2008), although a significant number of correlations have already been developed, no general correlation reliable in a wide range of geometrical parameters and operating conditions is available at the moment. On the contrary, several of them are only valid within a limited range of parameters. In addition, many show a complicated form, including numerous empirical coefficients determined by multivariable regressions (e.g. Ruffel (1974), Guo et al. (2001) and Zhang et al. (2003)).

In this paper, a new correlation for the two-phase frictional pressure drop is developed, with the aim to expand its range of validity with respect to existing correlations, reaching a satisfactory accuracy in an extended range of conditions. The starting point is the noteworthy Lockhart-Martinelli (Lockhart and Martinelli, 1949) correlation, developed for straight horizontal tubes, since it has been used successfully in many circumstances and has been reported by several authors to work fairly well also in helical geometry. Proper modifications are introduced starting from the analysis of the experimental results and their physical interpretation, while keeping the number of the needed empirical constants as lowest as possible, to avoid complicated formulations. The new correlation is first applied to the same database reported by Santini et al. (2008) and collected in an experimental facility installed at SIET laboratories, in Piacenza, Italy. As a first check, also the correlation proposed by Friedel (1979) for straight vertical pipes is modified with the same scheme of correlation. At a later stage, also the experimental data from the work of Zhao et al. (2003) are included in the database. The global database includes measurements from different values of the coil diameter, essential to develop a correlation applicable in an extended range of conditions and able to reproduce the influence of the helical geometry. The accuracy of the new correlation is finally compared with the results of other correlations available in literature.

2 LITERATURE REVIEW

In the past, different authors handled the problem of the two-phase frictional pressure drop in helical pipes, starting from own experimental results to derive proper correlations. Early works on the subject reported in literature, due to Rippel et al. (1966), Owhadi et al. (1968), Banerjee et al. (1969), Akagawa et al. (1971) and Katsuri and Stepanek (1972), found a satisfactory agreement between their experimental data and the original or slightly modified correlation of Lockhart and Martinelli, developed for horizontal straight pipes. All the authors reported maximum relative errors always lower than 50% and average relative errors between 20% and 40%, in a large number of different coils and operating conditions. Other authors started their analysis from models and correlations developed for straight pipes. Awwad et al. (1995) and Xin et al. (1996; 1997) modified the original form of the Lockhart-Martinelli multiplier as a function of the Froude number *Fr* and the pipe diameter to coil diameter ratio *(d/D)*. They reported maximum deviations always lower than 35% for circular and annular channels, with both vertical and horizontal orientations. Nariai et al. (1982) adopted the correlation for straight tube due to Martinelli and Nelson (1948), evaluating the single-phase pressure drop with the Ito correlation (1959). Comparison with data from a test rig of an integrated type marine water reactor showed agreement within 30%. Slightly better results were obtained using a modified version of the correlation proposed by Kozeki et al. (1970), based on a different form of the two-phase pressure drop multiplier. Czop et al. (1994) considered experimental data obtained with adiabatic water-SF6 mixture in a helically coiled tube and found better agreement with the Chisholm correlation (Chisholm and Sutherland, 1969) than using the Lockhart and Martinelli method. Other models proposed for straight tubes have been tested by different authors, in particular the Dukler approach (Dukler et al., 1964) in Katsuri and Stepanek (1972) and Baroczy (1965) and Thom (1964) correlations in Nariai et al. (1982), finding always higher errors with respect to the Lockhart and Martinelli correlation.

Among the correlations developed without referring to straight channel models, Ruffel (1974) proposed a different form of the liquid only friction multiplier, based on experimental results in three different coils tested to study the AGR secondary system. Unal et al. (1981) developed a model to calculate frictional pressure losses after testing three different coils heated by a sodium flow, in order to investigate the behaviour of a Liquid Metal Fast Breeder Reactor. Chen and Zhou (1981), based on a steam-water mixture flowing in three different helical coils, obtained a relation for the liquid only friction multiplier including the effect of the void fraction α and the *d/D* ratio. Guo et al. (2001) proposed a correlation for the liquid only friction multiplier based on the data from two helical tubes with four different axial inclinations of the helix. Authors reported maximum relative errors always lower than 40%. A liquid only friction multiplier correlation was developed for horizontal helically coiled pipes by Zhao et al. (2003), reaching an average relative error within 15%. Mandal and Das (2003), starting from data relative to coils with different geometrical parameters, proposed an empirical correlation based on various dimensionless quantities to reach an average relative error of about 15%. Santini and co-authors (2008) completed an experimental investigation on the helically coiled tube of a Generation III+ SMR project, the IRIS reactor (Carelli et al., (2004). The empirical correlation proposed correlates the frictional pressure drop with flow rate, mixture density and tube diameter. In addition, it accounts for the mixture quality by a cubic function. The authors indicate an average error of about 9%, with about 95% of the data within the range ±15%. However, the correlation does not account for the effect of the coil curvature, therefore it seems difficult to extend its range of applicability, in particular with respect to the coil diameter. Actually, it is a best-fit of experimental data, coming from an engineering approach useful for the design of the reactor. Since the large number of considered correlations are representative of many different experimental conditions (from air-water two component flow in atmospheric conditions to steam-water two-phase flow at high pressure), their ranges of applicability are summarized in Table 1.

3 EXPERIMENTAL MEASUREMENTS AND DATA ANALYSIS

3.1 Experiments

For the development and the validation of the new empirical correlation, two experimental databases have been considered, relative to the work of Santini et al. (2008) and Zhao et al. (2003). Experiments in the work of Santini et al. were made in the test facility of Politecnico di Milano, installed at SIET laboratories, in Piacenza, Italy. The open loop, full scale test facility simulates the helically coiled tube of the IRIS reactor SG (Carelli et al., 2004) and was designed to reproduce the prototypical thermal hydraulic conditions during reactor operation. A constant heat flux boundary condition was imposed through electrical power supply, instead of the real controlled temperature boundary. Coil diameter D is 1 m, inner tube diameter d is 0.01253 m and coil pitch per helix turn is 0.8 m. Tube length L is 32 m, while facility height H is 8 m. A detailed description of test facility, experimental instrumentation and experimental procedure has been reported by Santini et al. (2008). To facilitate the reader, a schematic representation of the plant is reported also here in Figure 1.

Zhao et al. (2003) studied a small horizontal helically coiled once-through steam generator. The test section included a 0.009 m inner diameter stainless steel tube, with a coil diameter D equal to 0.292 m and a coil pitch of 0.03 m. Imposed thermal flux was supplied through electrical power.

3.2 Data analysis

From the work of Santini et al. (2008), the frictional pressure drops have been calculated starting from the experimental measurements of the total pressure drop. In every experimental run, a total of 8 values of pressure drop were measured through the 9 pressure taps installed on the test section. Differently from the original paper, here the frictional pressure drop is calculated from a mixture momentum balance. For steady-state, one dimensional two-phase flow, this reads (Todreas and Kazimi, 1993):

$$\frac{dp}{dz} = -\tau_w \frac{P}{A} - g[\alpha\rho_v + (1-\alpha)\rho_l]sin\beta - G^2 \frac{d}{dz}\left[\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{(1-\alpha)}\right] \quad (1)$$

In the hypothesis of homogeneous flow, that implies thermal equilibrium between the phases and equal liquid and vapor velocities, Eq.(1) greatly simplifies as void fraction, mixture density and quality become:

$$\alpha = \frac{1}{1 + \frac{(1-x)}{x}\frac{\rho_v}{\rho_l}} \quad (2)$$

$$x = \frac{h_m - h_l}{h_v - h_l} \quad (3)$$

$$\rho_m = \left[\frac{x}{\rho_v} + \frac{(1-x)}{\rho_l}\right]^{-1} \quad (4)$$

Furthermore, if the assumptions of linear quality increase with the axial coordinate and constant saturation properties are made, Eq.(1) becomes:

$$\frac{dp}{dz} = -\tau_w \frac{P}{A} - g\rho_m sin\beta - G^2 \frac{dv_m}{dz} \quad (5)$$

After integration over the path between two consecutive pressure taps, the experimental frictional pressure loss per unit length is obtained as:

$$\frac{\Delta p_F}{L} = \frac{(p_1 - p_2)}{L} - g\rho_m sin\beta - G^2(v_2 - v_1) \quad (6)$$

where the subscript 2 refers to the outlet section and the subscript 1 to the inlet section. Reference pressure and thermodynamic quality, necessary to calculate saturation properties, local operating conditions and mixture properties, have been calculated as the arithmetic mean between the values corresponding to two consecutive pressure taps. A linear error analysis (Moffat, 1988) reported in the Appendix resulted in an average uncertainty for the experimental values of frictional pressure drop of about 1%, with maximum uncertainties of about 10% for a small number of data at high pressure and low mass flux. Since the instrumental uncertainty associated to the pressure drop measurement is related to the total pressure drop, an higher uncertainty characterizes conditions where the frictional pressure drop is a small fraction of the total value. The latter occurs for some experimental measurements at low flow quality, low mass flow rate and high pressure.

From the work of Zhao et al. (2003), the value of the liquid-only two-phase frictional multiplier is known. The liquid-only multiplier relates the two-phase frictional pressure drop to the single-phase frictional pressure drop at the same mass flux, considering the fluid as entirely liquid:

$$\left(\frac{\Delta p}{L}\right)_{tp} = \Phi_{lo}^2 \left(\frac{\Delta p}{L}\right)_{lo} \tag{7}$$

Making explicit the liquid-only frictional pressure drop, the two-phase frictional pressure drop is calculated by:

$$\left(\frac{\Delta p}{L}\right)_{tp} = \Phi_{lo}^2 \frac{f_{lo}}{2} \frac{G^2}{\rho_l d} \tag{8}$$

For the evaluation of the friction coefficient, the correlation by White (1932) is suggested by the authors:

$$f = 0.32Re^{-0.25} + 0.048\sqrt{\frac{d}{D}} \tag{9}$$

3.3 Frictional pressure losses

Experimental data from Santini et al. (2008) were collected in the ranges 10 to 65 bar for the pressure, 200 to 800 kg/m^2s for the mass flux and from 0 to 1 for the quality. The database from Zhao et al. (2003) includes experiments in a range of quality up to 0.95, 5 to 35 bar for the pressure, 236 to 943 kg/m^2s for the mass flux and 0-900 kW/m^2 for the heat flux. Although, in this work only experimental measurements up to a value of mass flux equal to 800 kg/m^2s have been considered. A global resume of experimental ranges is reported in Table 2. In this paper, only the pressure drops due to friction for a single value of the mass flux (400 kg/m^2s) of the database from Santini et al. (2008) are reported (Figure 2), to recall general considerations on the influence of significant operating parameters. It is seen in Figure 2 that an increase in the pressure always decreases pressure drops, as the density ratio is reduced. Concerning the dependence on the quality, the pressure drop exhibits a maximum for $0.7 < x < 0.8$, then it decreases to approach the single-phase (vapor) value.

4 CORRELATION WITH THE LOCKHART-MARTINELLI METHOD

A first attempt was made to predict frictional pressure drops from the Santini et al. (2008) database by the well known correlation of Lockhart and Martinelli, as suggested for example in Owhadi et al. (1968). The two-phase frictional pressure drop can be related to the single-phase pressure drop of the liquid or the vapor phase as flowing alone in the channel at their actual flow rate:

$$\left(\frac{\Delta p}{L}\right)_{tp} = \Phi_l^2 \left(\frac{\Delta p}{L}\right)_l \tag{10}$$

where Φ_l^2 is the only-liquid friction multiplier:

$$\Phi_l^2 = \frac{f_{tp}}{f_l} \frac{\rho_l}{\rho_m} \frac{1}{(1-x)^2} \tag{11}$$

A simple relation exists between the only-liquid multiplier and the liquid-only multiplier (Eq.(7)), which considers the two-phase frictional pressure drop function of the single-phase pressure drop of the liquid flowing at the same mass flux of the total two-phase as a mixture (Todreas and Kazimi, 1993):

$$\Phi_{lo}^2 = \Phi_l^2 (1-x)^{1.8} \tag{12}$$

The only-liquid friction multiplier can be correlated to the Martinelli parameter (Lockhart and Martinelli, 1949), i.e. the ratio of the single-phase liquid pressure drop to the single-phase vapor pressure drop:

$$\chi = \frac{\left(\frac{\Delta p}{L}\right)_l}{\left(\frac{\Delta p}{L}\right)_v} = \left[\frac{(1-x)}{x}\right]^{1.8} \frac{\rho_v}{\rho_l} \left(\frac{\mu_l}{\mu_v}\right)^{0.2} \tag{13}$$

Lockhart and Martinelli (1949) proposed:

$$\Phi_l^2 = 1 + \frac{C}{\chi} + \frac{1}{\chi^2} \tag{14}$$

A value of *C* equal to 20 was chosen for turbulent flow of both liquid and vapor phases (Lockhart and Martinelli, 1949). The pressure drop due to the liquid phase as flowing alone in the channel has been calculated as:

$$\left(\frac{\Delta p_F}{L}\right)_l = \frac{f_l}{2} \frac{G^2 (1-x)^2}{\rho_l d} \tag{15}$$

For the calculation of the single-phase friction coefficient in the SIET test section, the Ito correlation for turbulent flow in helical pipes has been selected, as suggested by Colombo et al. (2012):

$$f = 0.304Re^{-0.25} + 0.029\sqrt{\frac{d}{D}} \tag{16}$$

The Lockhart and Martinelli correlation returned an average error of about 35%, with nearly 70% of the data within ±40%, in agreement with the results reported in literature (Rippel et al., 1966; Owhadi et al., 1968; Banerjee et al., 1969; Akagawa et al., 1971; Katsuri and Stepanek, 1972). The comparison between experimental data and predictions is shown in Figure 3 for the three tested values of the system pressure. Although data point are quite scattered, generally the Lockhart-Martinelli correlation tends to underestimate the pressure drop at the lower value of the system pressure, whereas it overestimates it at the higher pressure. In addition to the effect of the system pressure, an effect of mass flow rate, already noticed by Lockhart and Martinelli (1949), is clearly visible.

As the liquid friction multiplier is adopted, the experimental values can be calculated with Eq.(11), replacing:

$$f_{tp} = \frac{2\left(\frac{\Delta p}{L}\right)_{exp} \rho_m d}{G^2} \tag{17}$$

It is interesting to compare the experimental friction multiplier as a function of the Martinelli parameter χ with the predictions given by Eq.(14), as shown in Figure 4. The Lockhart-Martinelli correlation returns a lower value of the multiplier with respect to experiments for low values of χ, that is at high quality. The situation is reversed for χ close to unity, that is at low quality, where the highest errors appear and the friction multiplier is always overestimated with respect to the experimental data. Larger error for χ close to unity was also reported by Akagawa et al. (1971) and it has been taken into account in the development of the modified correlation.

5 DEVELOPMENT OF THE NEW CORRELATION

Since the Lockhart-Martinelli correlation was developed for two-phase pressure drop in horizontal straight tubes, some modifications are necessary to attain a satisfactory agreement with the experimental data from helical geometry. In particular, as in the majority of the correlations valid for the single-phase flow, the action of the centrifugal force could be taken into account by the Dean number, defined as:

$$De = Re\sqrt{\frac{d}{D}} \tag{18}$$

which includes the tube diameter to coil diameter ratio. It is expected that centrifugal force, being proportional to fluid density, affects mainly the liquid-phase, as long as density ratio is higher than slip ratio. Consequently, the liquid is drawn from the center to the wall of the tube, as confirmed by visual inspections (Lacey, 1967) and more recently by backlight imaging tomography (Murai et al., 2005). This seems also to be suggested by the higher deviation between Lockhart-Martinelli multiplier and experimental data at low quality, as shown in the previous section. For this reason, the Dean number of the liquid-phase as flowing alone in the channel has been considered:

$$De_l = Re_l\sqrt{\frac{d}{D}} = \frac{G(1-x)d}{\mu_l}\sqrt{\frac{d}{D}} \tag{19}$$

In a similar manner, also Awwad et al. (1995) and Xin et al. (1997) included the pipe to coil diameter ratio, but using the Froude number instead of the Reynolds number. Among others, also Ruffel (1974) and Guo et al. (2001) added the tube diameter to coil diameter ratio, including it in different forms in their correlations. In addition, in this work a value of C equal to 10 has been adopted in the original form of the Lockhart-Martinelli friction multiplier (Eq.(14)). It is suggested by Chenoweth and Martin (1955) to correlate data at higher pressure with respect to the conditions considered in the original work of Lockhart and Martinelli (1949). Indeed, the Lockhart and Martinelli model was developed for data at atmospheric or slightly higher pressure. To further account for the effect of the system pressure, noticed in the results in Figure 3, a density ratio has been introduced in the correlation, as frequently made in other studies of different two-phase flow phenomena (Ishii and Zuber, 1970). In particular, the ratio ρ_m / ρ_l has been introduced. The mixture density calculated with Eq.(4) has been used, rather than the vapor density, to include also the dependence of the frictional pressure drop on the mixture quality. Density ratio has been also used by other authors, as for example Guo et al. (2001) and Zhao et al. (2003), albeit in the form of a vapor to liquid density ratio. These authors included also some functions of the vapor quality, to mimic its influence on the frictional pressure drop. A cubic function of the quality has been adopted also in Santini et al. (2008). Differently, in this work the effect of the quality is already included in the mixture density. In addition to an improved accuracy of the correlation, the latter is expected to avoid a too complicated formulation. Finally, the correlation has been set in the form:

$$\Phi_l^2 = \Phi_{LM}^2 a_1 De_l^{a_2}\left(\frac{\rho_m}{\rho_l}\right)^{a_3} \tag{20}$$

The values of the three empirical coefficients a_1,a_2 and a_3 has been determined with a multivariable regression, to minimize deviations with experimental data through the least square method:

$$\frac{\partial}{\partial a_k}\sum_i\left[\Phi^2_{exp,i}-\Phi^2_{LM,i}a_1De_{l,i}^{a_2}\left(\frac{\rho_{m,i}}{\rho_{l,i}}\right)^{a_3}\right]^2 \tag{21}$$

Resolution of Eq.(21) for the three empirical coefficients allowed to define the final form of the new correlation for the two-phase friction multiplier:

$$\Phi_l^2 = 0.13\Phi_{LM}^2 De_l^{0.15}\left(\frac{\rho_m}{\rho_l}\right)^{-0.37} \tag{22}$$

Mean absolute percentage error (MAPE) has been used to quantify the accuracy of the correlations:

$$\mathrm{MAPE} = \frac{1}{N}\sum_{i=1}^{N}\frac{\left|\Delta p/L_{pred}-\Delta p/L_{exp}\right|}{\Delta p/L_{exp}}\cdot 100\% \tag{23}$$

Eq. (22) returns a mean absolute percentage error of 11.6% with respect to the experimental data, with more than 75% of the data within ±15% and 88% of the data within ±20%. Comparison between experimental data and Eq.(22) is reported in figures 5 to 7 for the three system pressures. It is worth noting that, differently from the results for the Lockhart-Martinelli correlation itself in Section 4, the introduction of the density ratio into the correlation actually allows to properly account for the effect of pressure. It is also mentioned that comparable errors have been obtained starting from a single-phase friction factor correlation valid for straight tube instead of the Ito correlation (Eq.(16)). Thus it seems more important to account for the effect of the centrifugal force within the two-phase multiplier (i.e. with the introduction of the Dean number) than in the prediction of the single-phase pressure drop.

As a first verification of the effectiveness of the correlation scheme adopted with Eq.(20), it has been applied also to the Friedel correlation (1979). The correlation was developed for two-phase flow in straight pipes and it includes the Weber number, to account for interfacial effects, the Froude number, to account for gravity and other empirical coefficients to best fit data. In addition, differently from the Lockhart-Martinelli model, the Friedel correlation is based on the liquid-only approach:

$$\Phi_{lo}^2 = A_1 + \frac{3.24A_2A_3}{Fr^{0.045}We^{0.035}} \tag{24}$$

$$A_1 = (1-x)^2 + x^2\left(\frac{\rho_l f_{vo}}{\rho_v f_{lo}}\right) \tag{25}$$

$$A_2 = x^{0.78}(1-x)^{0.224} \tag{26}$$

$$A_3 = \left(\frac{\rho_l}{\rho_v}\right)^{0.91}\left(\frac{\mu_v}{\mu_l}\right)^{0.19}\left(1-\frac{\mu_v}{\mu_l}\right)^{0.7} \tag{27}$$

Application of Eq.(24) resulted in an error of about 29%, with a general underestimation of the pressure drop (Figure 8a). As in the case of the Lockhart-Martinelli correlation, the inclusion of both the Dean number and the mixture-to-liquid density ratio have determined a remarkable improvement of the correlation accuracy. The modified form of the Friedel correlation is:

$$\Phi_{lo}^2 = 0.12\Phi_{FR}^2 De_l^{0.21}\left(\frac{\rho_m}{\rho_l}\right)^{-0.26} \tag{28}$$

An average deviation from experimental data of about 12.6% has been obtained, slightly higher than the modified Lockhart-Martinelli correlation. In this case, 83% of the experimental data are predicted within the range ±20%, as shown in Figure 8b. In summary, the introduction of the same scheme of correlation allows for both the correlations to reach satisfactory and comparable agreement with the experimental data.

Figure 9 compares the frictional pressure drop per unit length calculated with Eq.(22) and Eq.(28) with the correlations of Lockhart-Martinelli and Friedel. The experimental data relative to 400 kg/m^2s and 20, 40 and 60 bar are reported for comparison. Eq.(22), developed modifying the original Lockhart and Martinelli correlation, shows a satisfactory accuracy for all the three pressure levels considered, while Eq.(28) provides globally worse results, although they can be also considered satisfactory. A remarkable improvement is obtained with respect to the Lockhart-Martinelli correlation and the Friedel correlation, which are unable to predict the behavior of the pressure drop in the helical tube, being developed for straight geometry. In particular, the Friedel correlation underestimates the pressure drop, while the Lockhart-Martinelli correlation overestimates the pressure drop at low quality and underestimates it at high quality. Most important, the new scheme of correlation allows to reproduce the right position of the peak in the pressure drop versus quality curve, differently from the original Lockhart-Martinelli and Friedel correlations.

Friedel correlation has been selected here only for a further application of the corrective parameters used in the development of Eq.(22), which will be considered hereinafter. Indeed, Eq.(22), obtained with a modification of the Lockhart-Martinelli correlation, shows a better global accuracy and takes the advantage of a simpler formulation.

6 COMPARISON WITH LITERATURE

In this section, the developed correlation (Eq.(22)) is compared to some of the most representative correlations available in literature. The correlations of Ruffel (1974), Awwad et al. (1995), Xin et al. (1997), Guo et al. (2001), Zhao et al. (2003) and Santini et al. (2008) have been considered. They are summarized in Table 3.Table 4 reports the average deviations from the experimental data for all the correlations, while figures 10 to 12 show the comparison at 400 kg/m^2s for all the system pressures considered in the experiments.

Apart from the Santini et al. (2008) correlation, which as pointed out in the introduction arises from a best-fit of experimental data, the other correlations return significant errors, some of them even higher than those obtained in the previous section with Lockhart-Martinelli and Friedel correlations. Best predictions are given by the Ruffel (1974) and the Awwad et al. (1995) correlations, the latter developed for horizontal helically coiled pipes. However, both the correlations are unable to reproduce the peak in the pressure drop versus quality curve. The existence of the maximum value of the pressure drop is predicted by the Zhao et al. (2003) correlation. In these respects, the peak and the pressure drop dependence on the vapor quality is correctly reproduced by the developed correlation. Therefore, the inclusion of the mixture to vapor density ratio in Eq. 20 lead to an improved accuracy without using a complex function of the vapor quality. In addition, also the correlation by Xin et al. (1997) reproduces the peak, although in general it overestimates the pressure drop. Both Awwad et al. (1995) and Xin et al. (1997) considered the effect of curvature through a tube diameter to coil diameter ratio added to the Froude number. In view of the improved accuracy obtained in this work, the use of the Reynolds number seems more appropriate, in particular at high values of the Froude number where effect of gravity is negligible. Although, it's difficult to draw a definite conclusion since many factors influence the results. Actually, both Awwad et al. (1995) and Xin et al. (1997) does not include any explicit dependence on the quality or corrective parameter for the system pressure, being derived with data at atmospheric pressure. The large differences found between the results of the correlations demonstrate the difficulty to predict with a high degree of accuracy the two-phase flow frictional pressure drop, in particular in a wide range of operating conditions. However, the correlation scheme seem appropriate to reproduce the influence of system parameters such as the pressure and the mass flow rate as well as the effect of the coil curvature, which characterizes the helical geometry.

Since the databank from Santini et al. (2008) is representative of only one coil diameter, the addition of the data of Zhao et al. (2003) allowed a verification and a further development of the correlation considering a different value of the coil diameter. Looking at Table 1, it is interesting to note that the experimental conditions of the data of Zhao et al. (2003) are closer to the ranges of validity of the other correlations, generally limited to smaller diameter coils with respect to the SIET facility. In particular for the coil diameter, although only a single value is added to the database, it is important to notice that it is quite different with respect to coil diameter of the SIET facility, making the database representative of an extended range. In addition, not so many works include measurements at high pressure, numerous being related to an air-water two-phase flow at atmospheric pressure. The results for the data of Zhao et al. (2003) from all the correlations considered previously are summarized in Table 5. Except for the correlation developed by the authors, that obviously shows the better accuracy, the new correlation (Eq.(22)) returns the lowest mean absolute percentage error, equal to 25.3%. All the other correlations show errors higher than 30%. In particular, the correlation of Santini et al. (2008) shows an error of almost 40%. Actually, it could have been expected, since no attempt to take into account the effect of the coil curvature was made in Santini et al. (2008). Although the results cannot be considered satisfactory and the differences from the other correlations are not substantial, the corrective parameters seem again reliable for an improved correlation of the experimental data, in particular for the effect of the coil curvature.

Therefore, starting from the scheme of correlation of Eq.(20), a proper form of the correlation has been found also for the databank from Zhao et al. (2003):

$$\Phi_l^2 = 0.032\Phi_{LM}^2 De_l^{0.305}\left(\frac{\rho_m}{\rho_l}\right)^{-0.51} \quad (29)$$

Eq.(29) returns an mean absolute percentage error equal to 15.6 %. Although a little higher, it is comparable with respect to the correlation provided by the authors. The above result confirms the reliability of Eq.(20) as an appropriate scheme of correlation for the frictional pressure drop in helical pipes. Consequently, it has been also applied to the whole databank to obtain the final form of the correlation, provided with extended ranges of validity.

7 FINAL IMPROVED FORM OF THE CORRELATION

The final form of the new correlation is obtained considering the whole databank, including the experimental data from Santini et al. (2008) and Zhao et al. (2003):

$$\Phi_l^2 = 0.0986\Phi_{LM}^2 De_l^{0.19}\left(\frac{\rho_m}{\rho_l}\right)^{-0.40} \quad (30)$$

The correlation predicts the frictional pressure drop with a satisfactory 12.9 % mean absolute percentage error. For the Santini et al. (2008) database, the accuracy is slightly worsened, as the error increases to 12.0 %, starting from the 11.6 % of Eq.(22). Focusing only on the new data (Zhao et al., 2003), Figure 13 show a comparison with the other correlations considered. The two-phase multiplier is estimated with an error that is the 17.8 % on average, which is a little higher only with respect to the correlation derived by the authors, while all the other correlations generally underestimate the frictional pressure drop. The accuracy of all the correlations with respect to the whole database is shown in Figure 14 and Figure 15. The two correlations of Santini et al. (2008) and Zhao et al. (2003), which are the only to outperform Eq.(30) limited to their respective databases, result unable to provide a satisfactory accuracy with a different coil diameter. This is a direct consequence of the choice of both authors not to include the effect of the coil diameter in their correlations. In particular, the correlation of Santini et al. (2008) underestimates the frictional pressure drop from the database of Zhao et al. (2003), showing an average error of almost 40 % (Figure 15a). On the contrary, the correlation of Zhao et al. (2003) overestimates of more than 40 % the data from Santini et al. (2008) (Figure 15b). Being the data of Zhao et al. (2003) related to a significantly lower coil diameter, the above result is a confirmation of the importance of a correct estimation of the effect of the coil diameter, which increases the frictional pressure drop through an increase of the centrifugal force. Concerning the coil diameter, the new correlation shows a significantly extended range of validity, without a significant loss of accuracy on the individual databases. The other correlations show significantly higher errors (higher than 25-30 % at best) with respect to Eq.(30), as they generally tend to underestimate the frictional pressure drop (Figure 15).

8 CONCLUSIONS

A new scheme of correlation for the prediction of the two-phase frictional pressure drop in helically coiled tubes has been developed. Numerous correlations are available in literature, although for the additional complexity introduced by the effect of the coil diameter none of them can be considered either of general validity nor applied to a wide range of geometrical parameters and operating conditions.

The model is based on the well-known work of Lockhart and Martinelli (1949), which was reported by different authors to show satisfactory agreement also for helical tubes. Proper modifications have been introduced to extend the range of validity while keeping as low as possible the number of empirical constants. In particular, the effect of the coil curvature has been introduced by the Dean number of the liquid phase whereas the mixture-to-liquid density ratio accounts for the effect of the system pressure.

The two databases of Santini et al. (2008) and Zhao et al. (2003) have been considered for the validation. (more than 1000 experimental points). The final form of the correlation returns a mean absolute percentage error of about 12.9 % on the whole database, with a value of 12.0 % on the data of Santini et al. (2008) and 17.8 % on the data of Zhao et al. (2003). Further comparison has been made with numerous correlations available in literature. The proposed correlation, differently from the others, shows the highest accuracy on the whole database without a significant loss of accuracy on the individual sets of conditions.

APPENDIX

During experiments, some physical quantities are directly measured, whereas many other are derived from the measured ones. Their uncertainties, relative to the available experimental measurements of Santini et al. (2008), have been evaluated through the linear error propagation technique (Moffat, 1998) summarized in this appendix. On the other hand, uncertainties of the measured quantities have been reported in Table 6.

For the error analysis of the two-phase frictional pressure drop, the uncertainties of frictional pressure drop and quality have to be determined. The quality is calculated by an energy balance (Eq.(3)), so its uncertainty depends on the uncertainty of the enthalpy of the mixture:

$$h_m = h_{in} + 4\frac{q^{''}L}{Gd} = h_{in} + 4\frac{q^{''}L}{Gd} \quad \text{(A.1)}$$

Therefore, uncertainties of the quality is dependent on inlet enthalpy, mass flow rate and thermal power supplied to the mixture. Uncertainties of inlet enthalpy as well as of fluid and saturation properties have been evaluated as negligible with respect to other quantities. The thermal power supplied to the fluid is obtained as the difference between the measured electrical power and the thermal losses. Therefore its uncertainty is obtained from the root sum square of both contributions:

$$\delta q = \sqrt{(\delta q_{el})^2 + (\delta q_{loss})^2} \tag{A.2}$$

The uncertainty of the electrical power is found in Table 6, whereas the uncertainty of thermal losses has been evaluated in Santini et al. (2008) as approximately 15 %. Uncertainty of the mass flux is known from measured values. Eventually, relative uncertainty for quality is:

$$\frac{\delta x}{x} = \sqrt{\left(\frac{\delta q}{q}\right)^2 + \left(\frac{\delta G}{G}\right)^2} \tag{A.3}$$

The frictional pressure drop is calculated subtracting the gravitational and the accelerative pressure drop from the total measured pressure drop (Eq.(6)), thus its uncertainty reads:

$$\delta\Delta p_{fr} = \sqrt{\left(\delta\Delta p_{exp}\right)^2 + \left(\delta\Delta p_{grav}\right)^2 + (\delta\Delta p_{acc})^2} \tag{A.4}$$

The uncertainty in the measured pressure drop is obtained from the relative uncertainty in Table 6. The uncertainty in the gravitational pressure drop is due to mixture density (Eq.(4)), which is function of uncertainty in the quality:

$$\frac{\delta\Delta p_{grav}}{\Delta p_{grav}} = \frac{\delta x}{x} \tag{A.5}$$

Finally, the uncertainty of the accelerative pressure drop is obtained from the uncertainty of the mass flux and the uncertainty of inlet and outlet qualities:

$$\frac{\delta\Delta p_{acc}}{\Delta p_{acc}} = \sqrt{\left(2\frac{\delta G}{G}\right)^2 + \left(\frac{\delta(x_2 - x_1)}{x_2 - x_1}\right)^2} = \sqrt{2\left(\frac{\delta G}{G}\right)^2 + \left(\frac{\sqrt{(\delta x_2)^2 + (\delta x_1)^2}}{x_2 - x_1}\right)^2} \tag{A.6}$$

From the linear error analysis, uncertainties under 10 % were found for all the frictional pressure drop measurements, with maximum uncertainties of about 10% for a small number of data at high pressure and low mass flux.

ACRONYMS

AGR	Advanced Gas Reactor
HTGR	High Temperature Gas-cooled Reactor
IRIS	International Reactor Innovative and Secure
SG	Steam Generator
SMR	Small-medium Modular Reactor
THTR	Torium High Temperature Reactor

NOMENCLATURE

A	pipe cross-section [m^2]
A_1,A_2,A_3	empirical constants [-]
a_1,a_2,a_3	empirical constants [-]
C	empirical constant [-]
D	coil diameter [m]
d	pipe diameter [m]
De	Dean number [-]
f	friction factor [-]
Fr	Froude number [-]
G	mass flux [kg/m^2s]
g	gravity acceleration [m/s^2]
H	height [m]
h	specific enthalpy [J/kg]
j	superficial velocity [m/s]
L	length [m]
P	perimeter [m]
p	pressure [Pa]
Q	volumetric flow rate [m^3/s]
q	thermal power [W]
q''	thermal flux [W/m^2]
Re	Reynolds number [-]
v	specific volume [m^3/kg]
We	Weber number [-]
x	quality [-]
z	axial coordinate [m]

Greek symbols

α	void fraction [-]

β	helix inclination angle [°]
Γ	mass flow rate [kg/s]
Δ	difference [-]
μ	dynamic viscosity [Pa·s]
ρ	density [kg/m^3]
τ	shear stress [Pa/m^2]
Φ	two-phase friction multiplier [-]
χ	Martinelli parameter [-]

Subscripts

cr	critical
D	Darcy
exp	experimental
f	frictional
Fr	Friedel
g	gas
l	liquid
lo	liquid-only
LM	Lockhart-Martinelli
m	mixture
pred	predicted
tp	two-phase
v	vapor
w	wall

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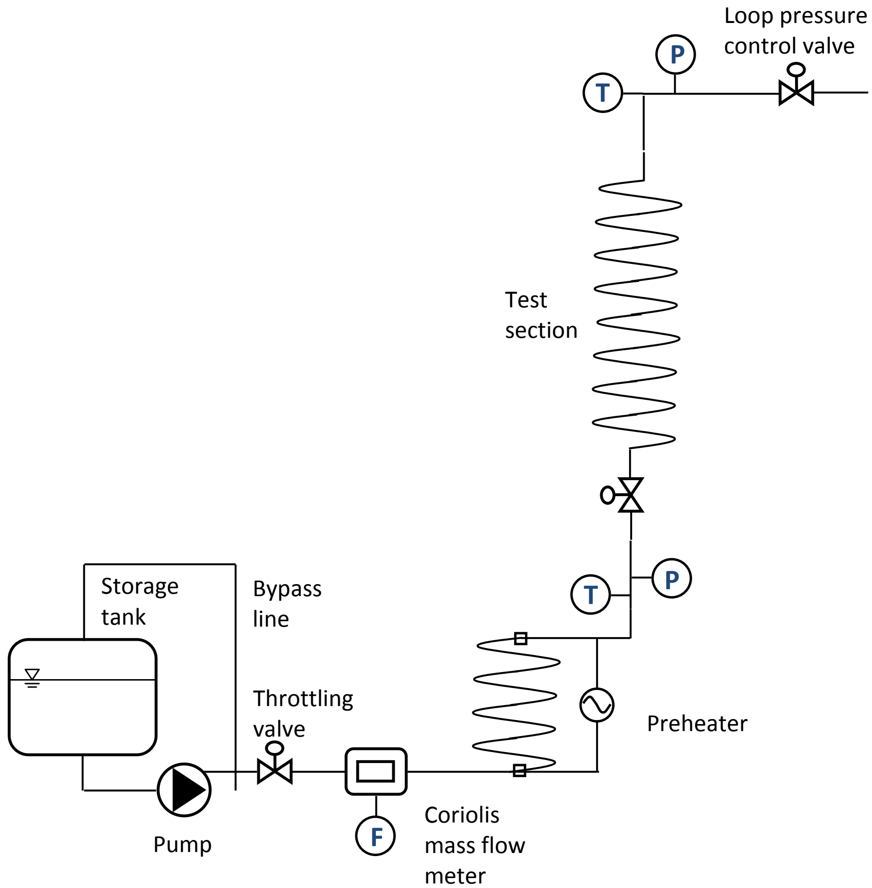

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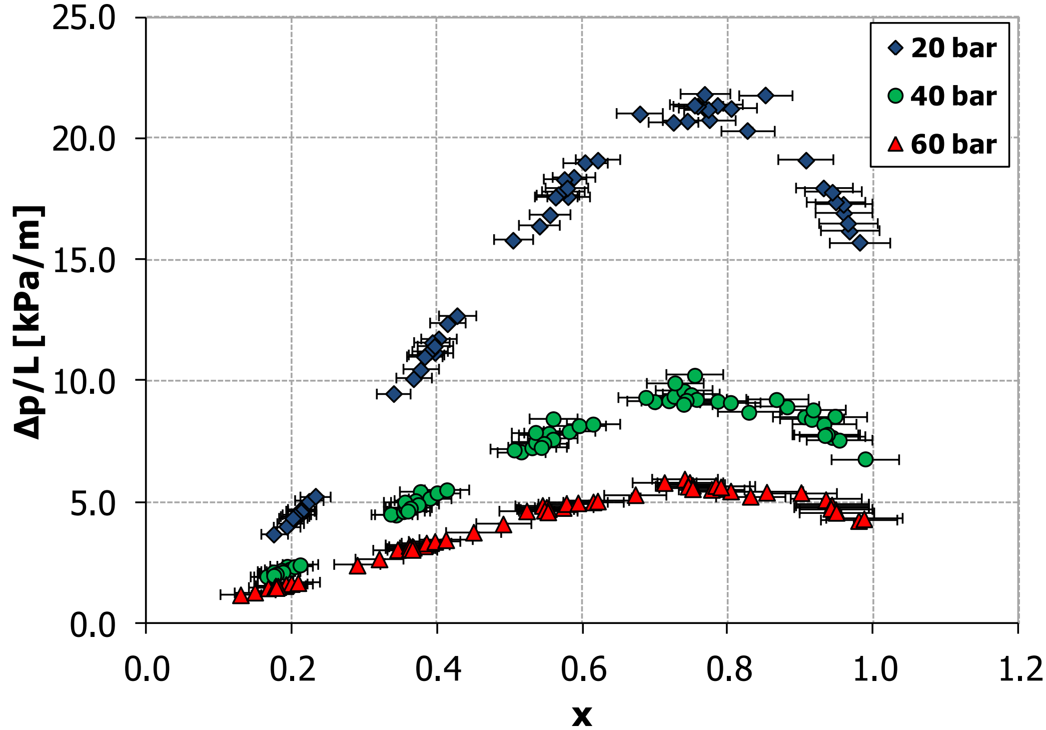

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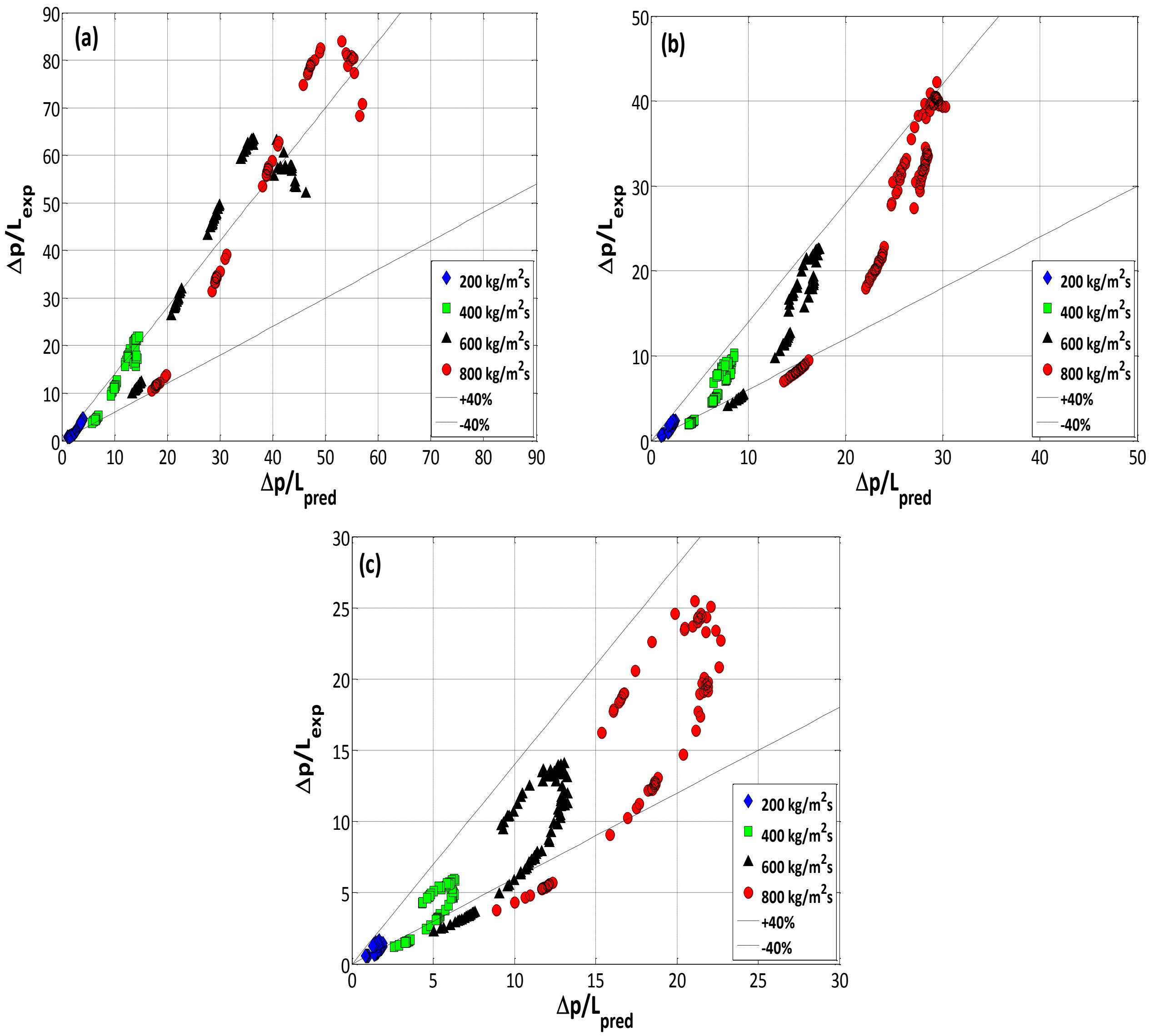

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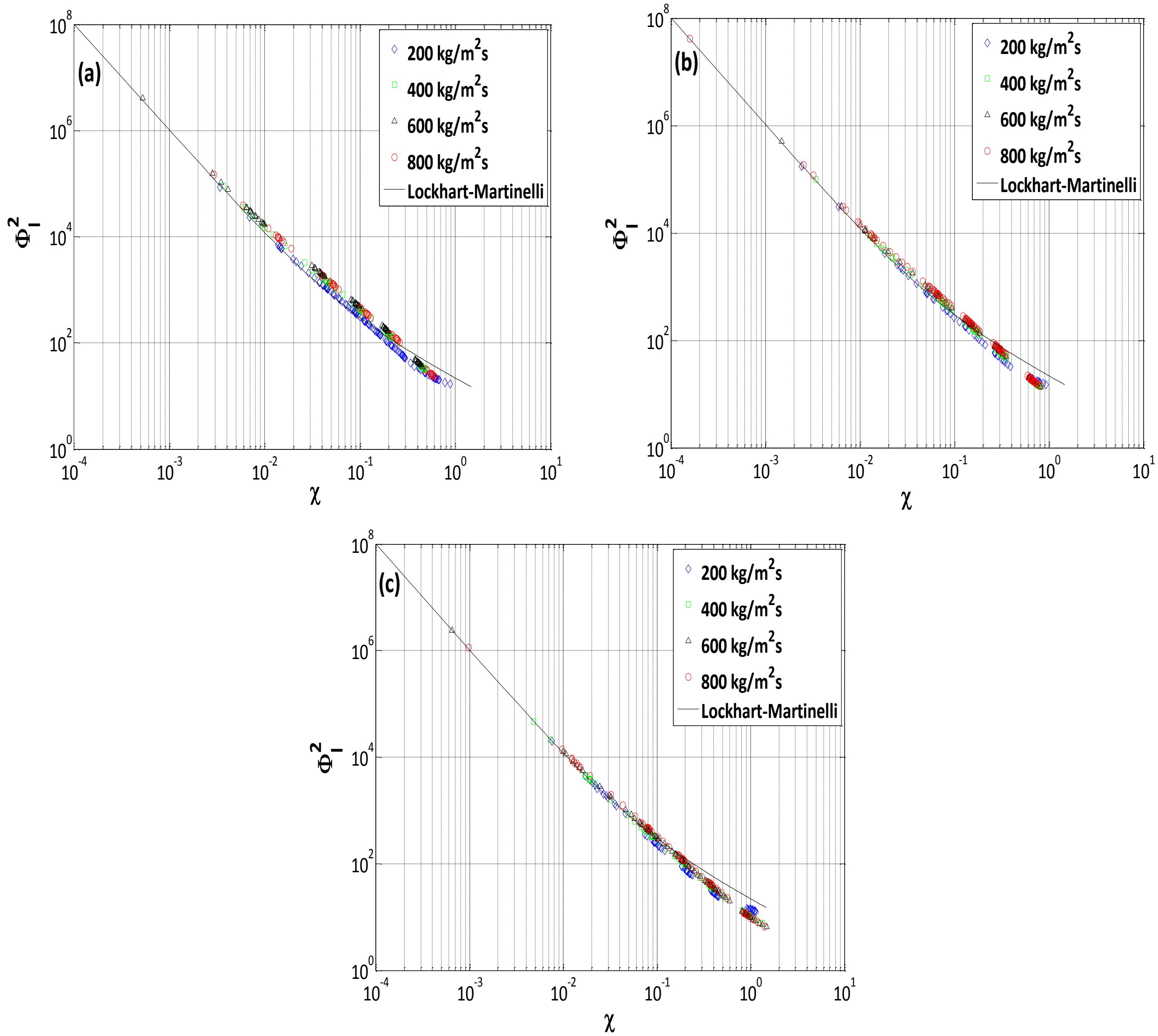

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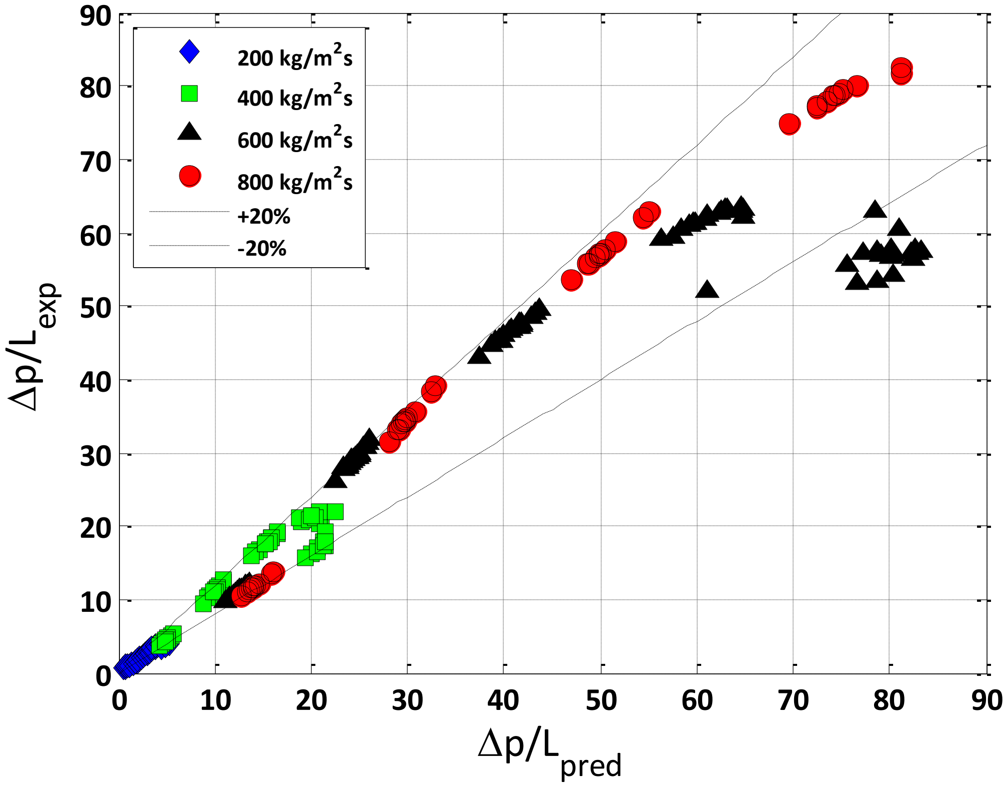

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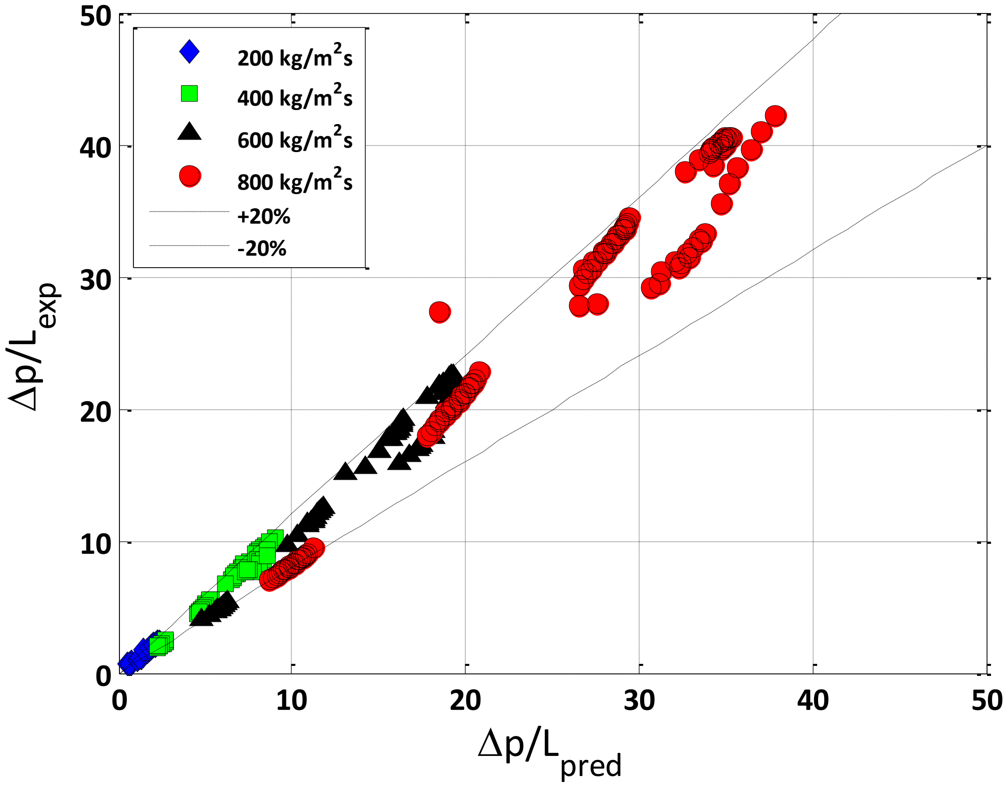

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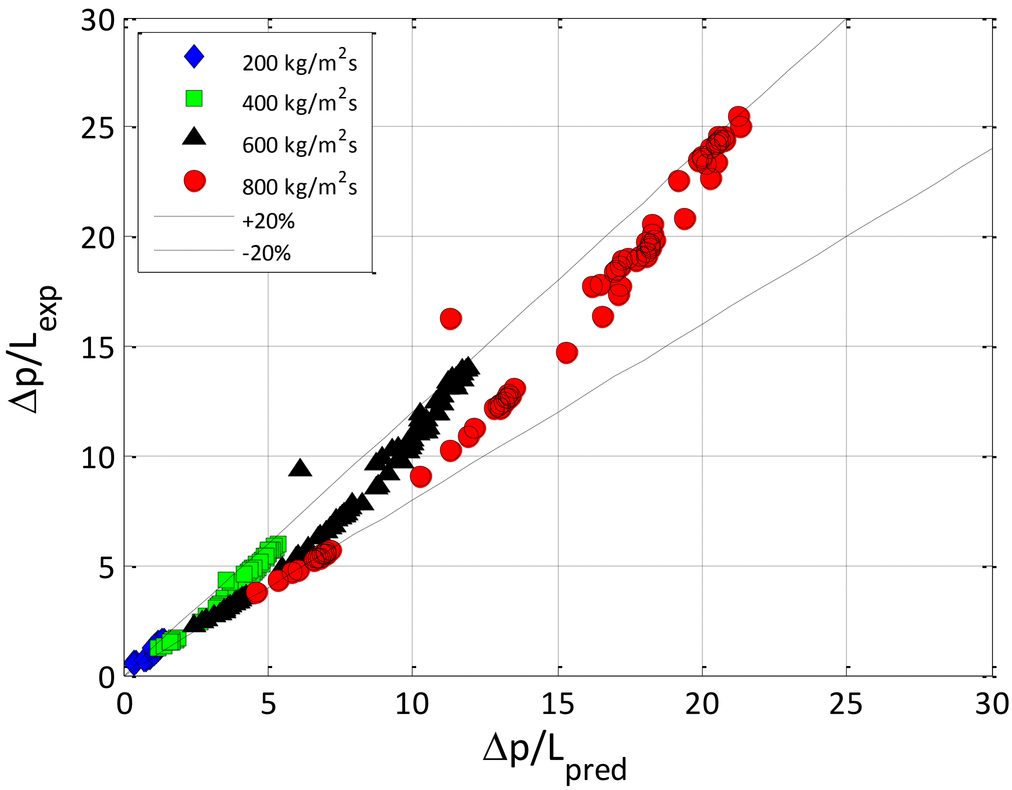

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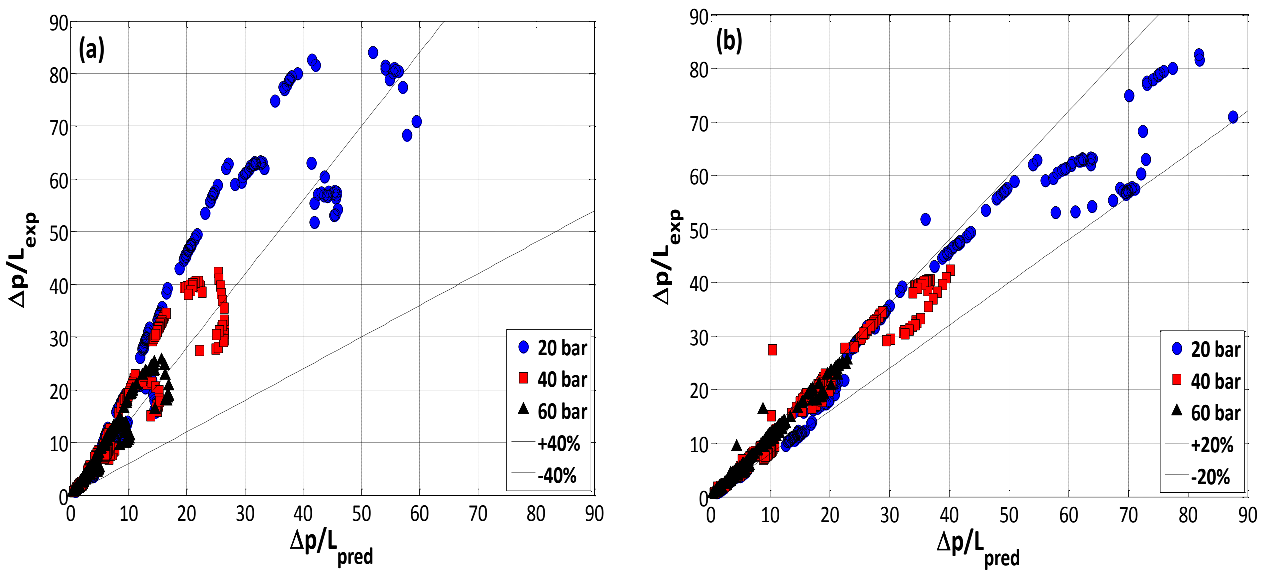

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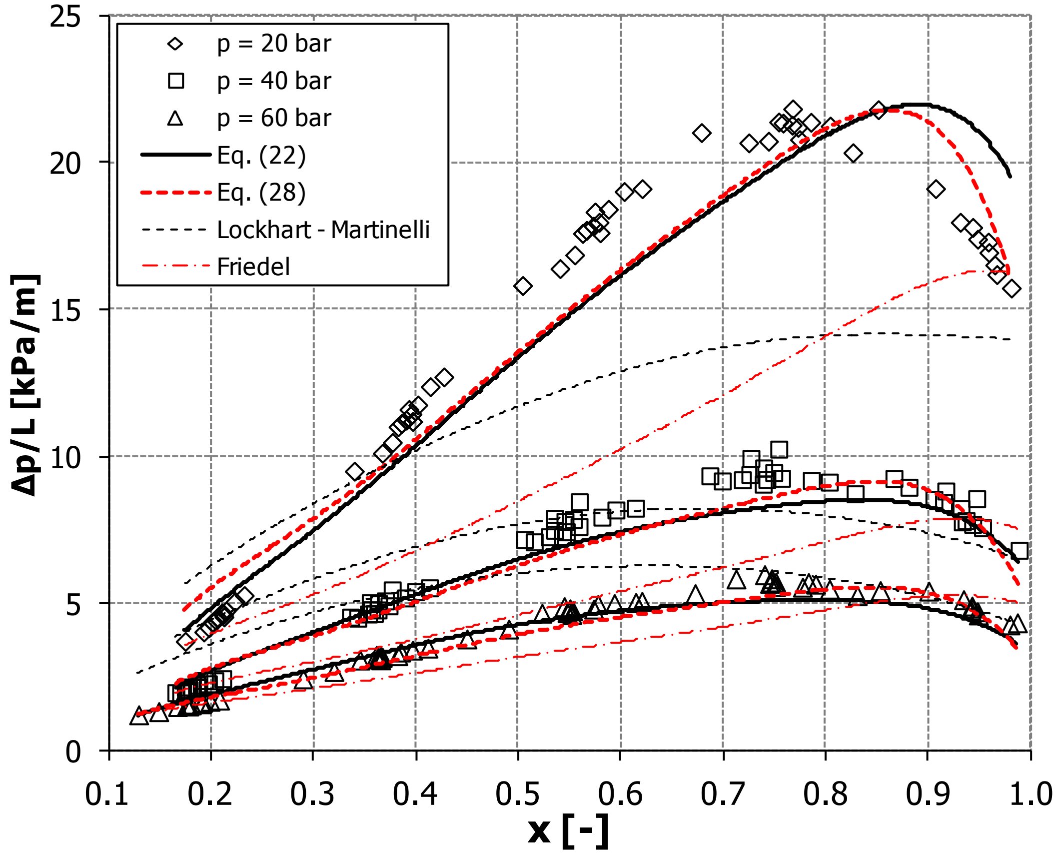

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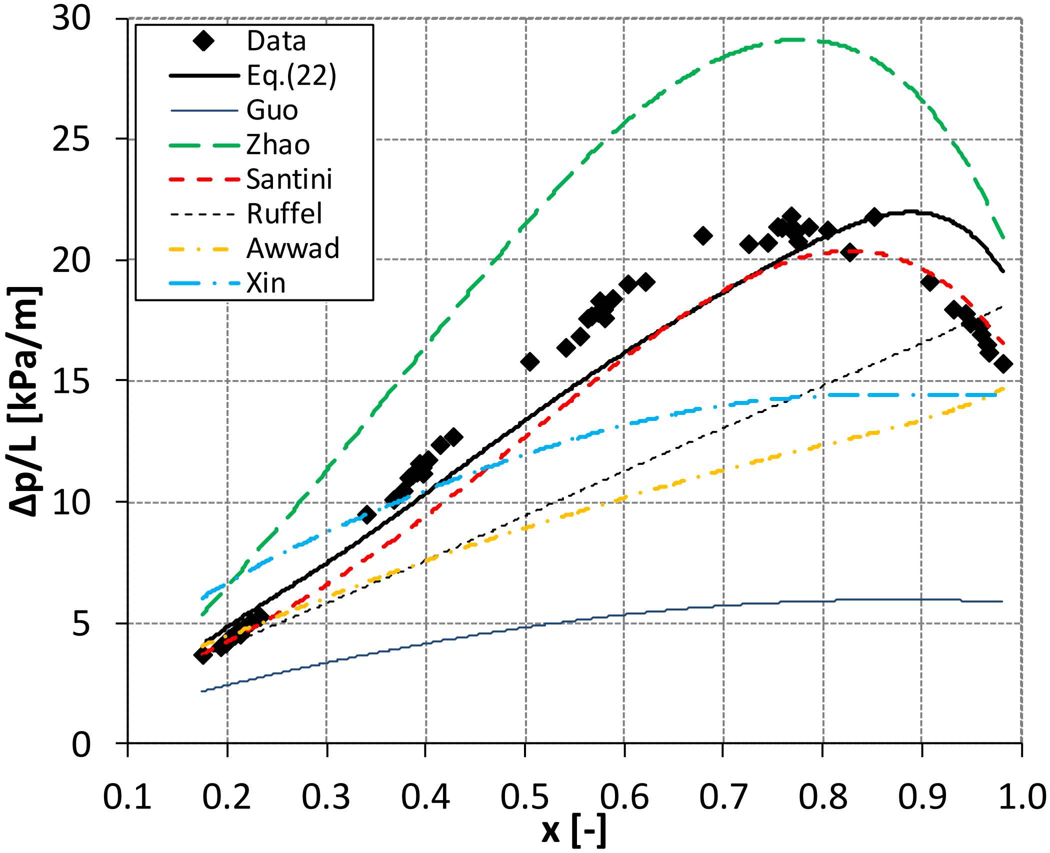

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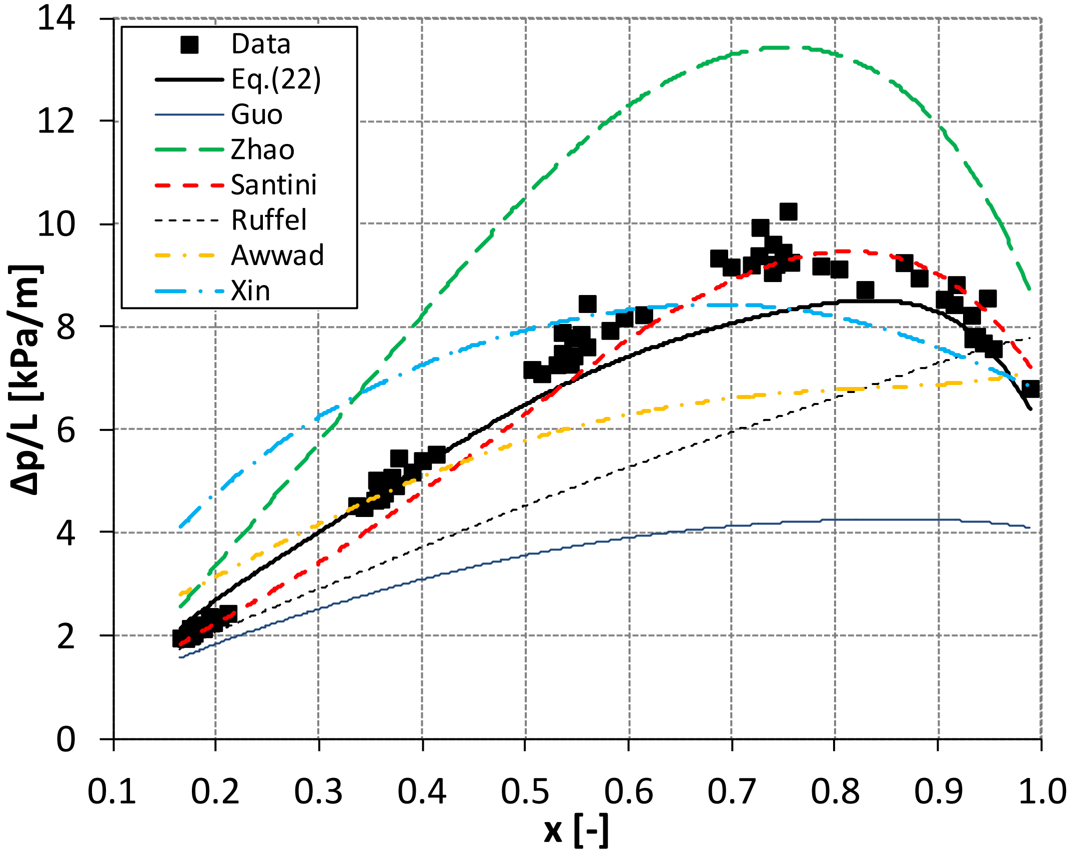

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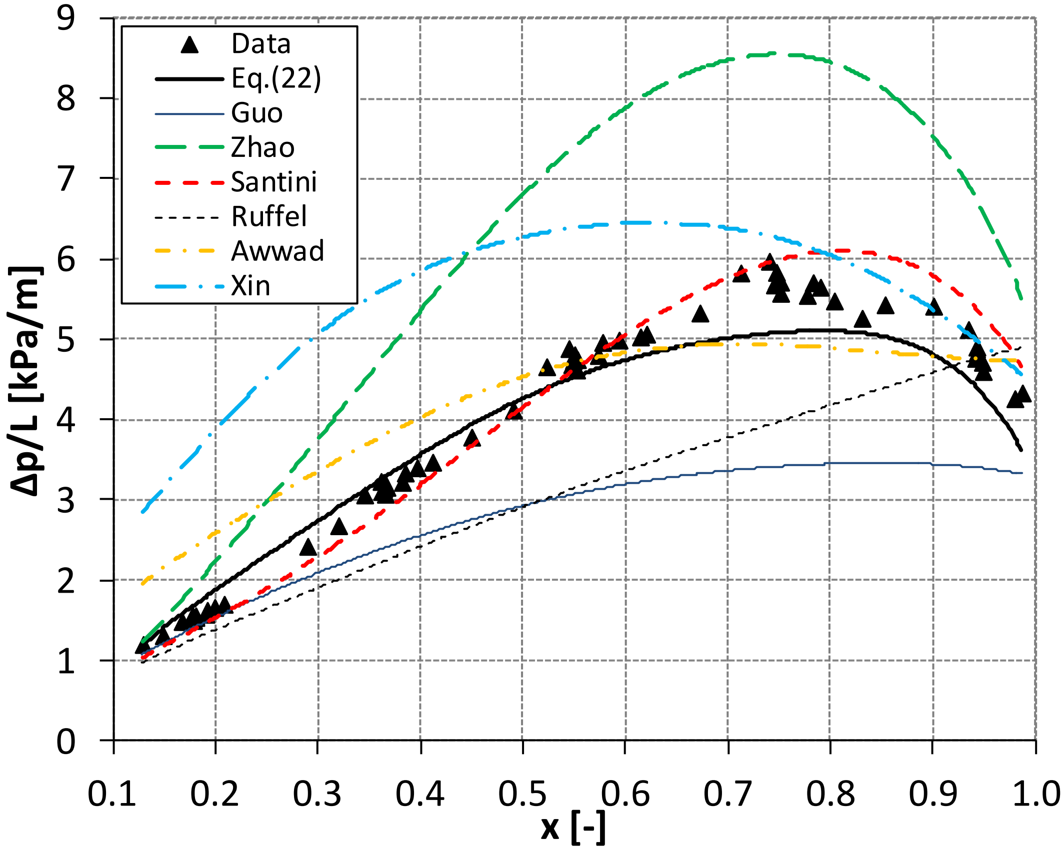

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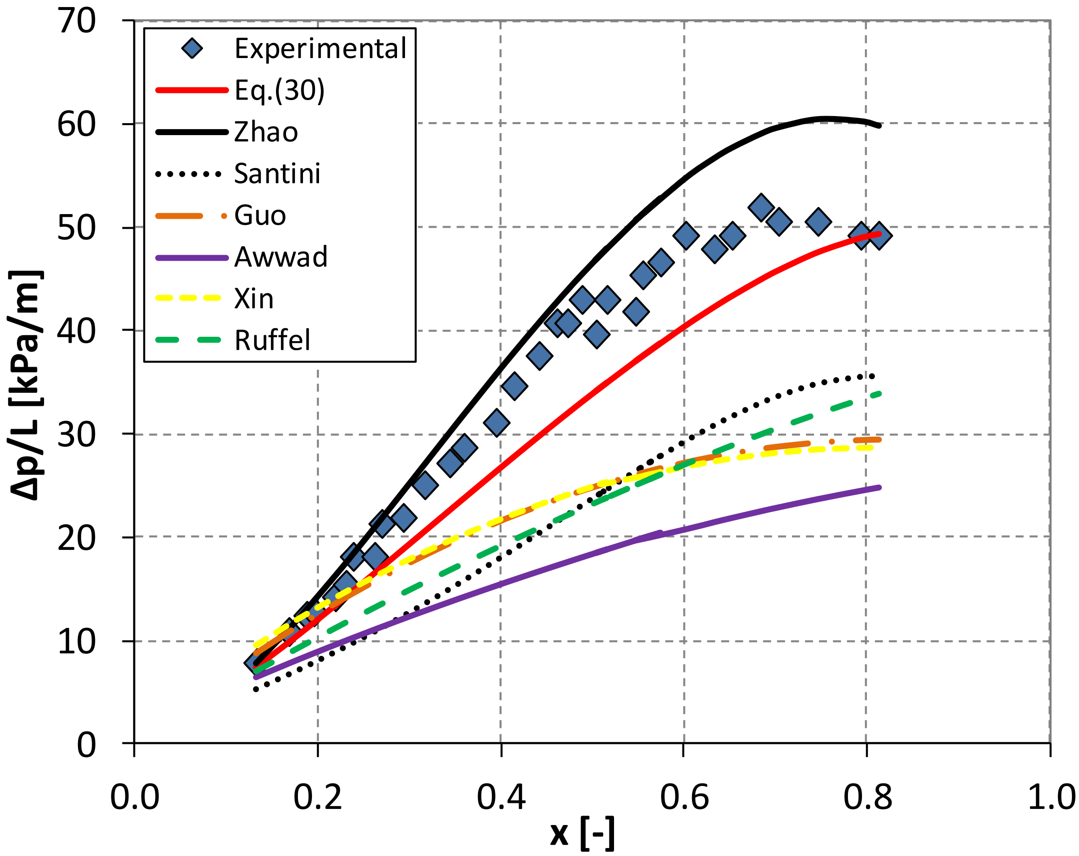

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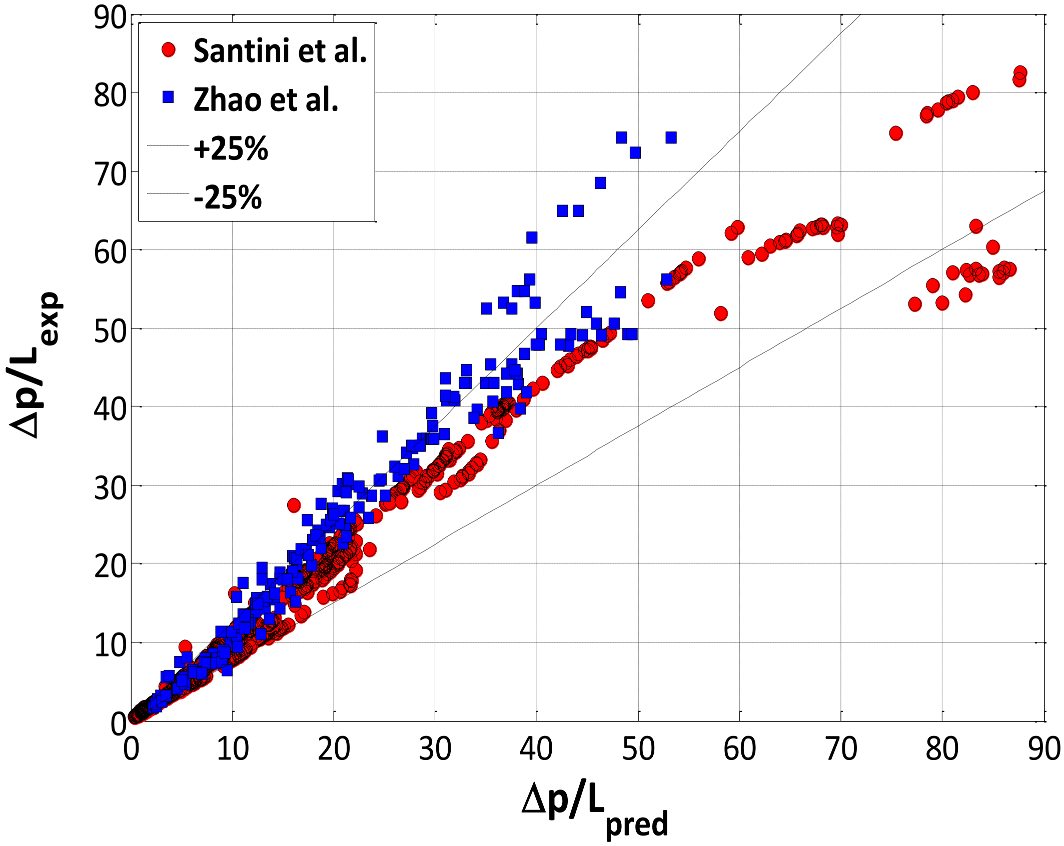

Figure 14 Comparison between the developed correlation (Eq.(30)) and the experimental data from the whole databank.

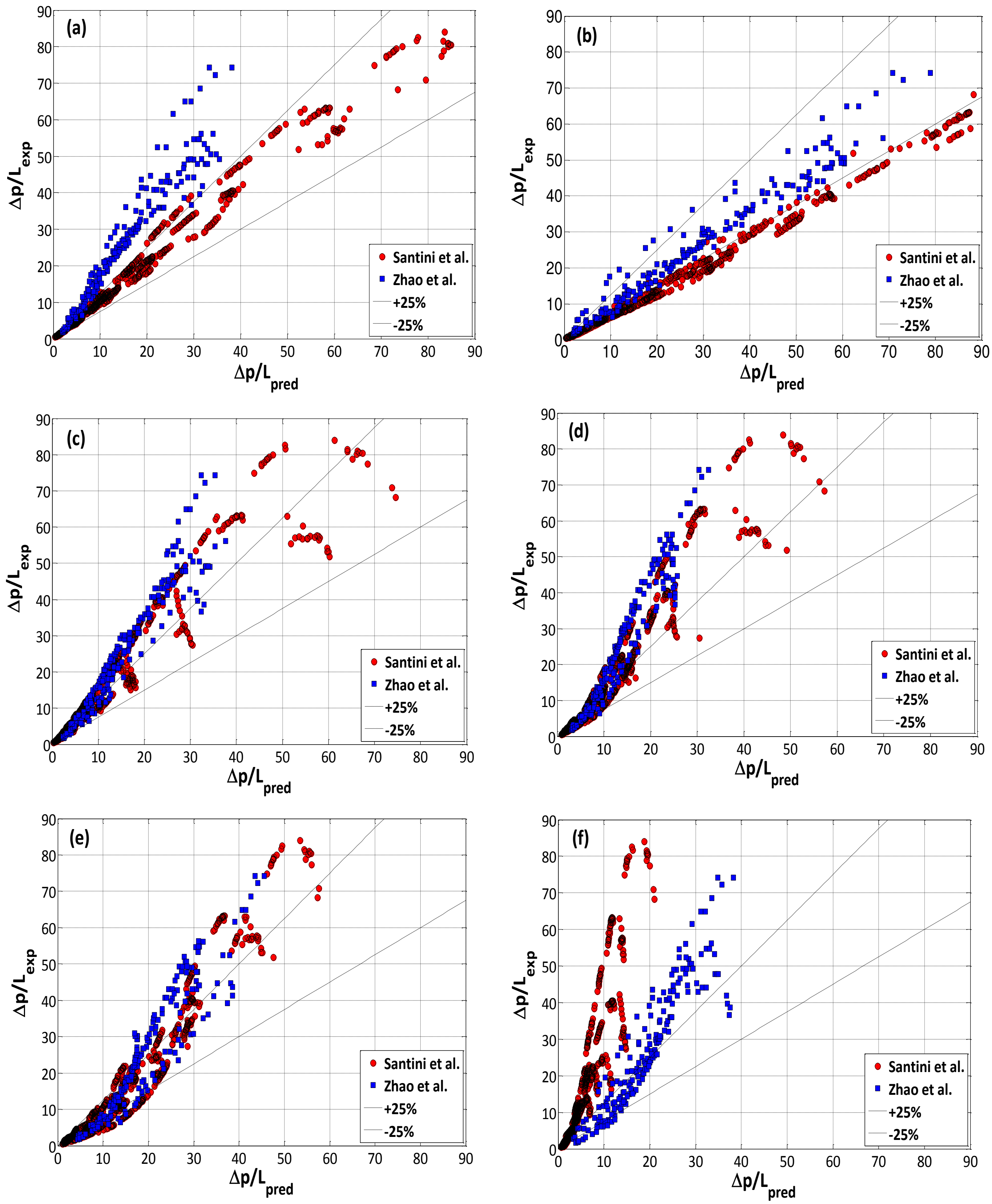

Figure 15 Comparison between correlations and the experimental data from the whole databank. (a) Santini et al. (2008); (b) Zhao et al. (2003); (c) Ruffel (1974); (d) Awwad et al. (1995); (e) Xin et al. (1997); (f) Guo et al. (2001).

Table 1 Summary of the correlations available for frictional two-phase pressure drops in helical pipes and their ranges of applicability. Abbreviations: V = vertical, H = horizontal, I = inclined, A = ascending, D = descending.

Study	Orientation	d [mm]	D [m]	β [°]	Operating conditions
Rippel et al. (1966)	V; D	12.7	0.2032	-	air-water, He-water and Freon 12-water in atmospheric conditions
Owhadi et al. (1968)	V; A	12.5	0.25	7.2	steam-water at atmospheric pressure Γ=35-139 kg/h, q”=60-256 kW/m^2
Banerjee et al. (1969)	V; A	15.9-54.8	0.152-0.610	2-8	air-water in atmospheric conditions Re_l=100-8000, Re_g=550-40000
Akagawa et al. (1971)	V; A	9.92	0.109, 0.225	1.2, 2.5	air-water in atmospheric conditions j_l=0.35-1.16 m/s, j_g=0-5 m/s
Katsuri and Stepanek (1972)	V; A	12.5	0.665	-	air-water in atmospheric conditions, Γ_l=0.01-0.22 l/s, Γ_g=0.1-8.5 l/s
Ruffel (1974)	V; A	10.7-18.6	0.0054-0.16	-	steam-water p=6-18 MPa, G=300-1800 kg/m^2s
Unal et al. (1981)	V; A	18	0.7-1.5	-	steam-water p=14.9-20.1 MPa, G=296-1829 kg/m^2s, x=0.15-1.0
Chen and Zhou (1981)	V; A	18	0.236, 0.450, 0.909	-	steam-water p=4.2-22 MPa, G=400-2000 kg/m^2s
Nariai et al. (1982)	V; A	14.3	0.595	-	steam-water p=2-5 MPa, G=150-850 kg/m^2s
Czop et al. (1994)	V; A	19.8	1.17	7.27	water-SF6 p=1-13.5 bar, G=500-3000 kg/m^2s, x=0.04-0.6
Awwad et al. (1995)	H	12.7-38.1	0.330-0.670	0.5-20	air-water in atmospheric conditions, j_l=0.008-2.2 m/s, j_g=0.2-50 m/s
Xin et al. (1996)	V; A	12.7-38.1	0.305-0.609	0.5-10	air-water in atmospheric conditions, j_l=0.008-2.2 m/s, j_g=0.2-50 m/s
Guo et al. (2001)	V, H, I	10, 11	0.132, 0.256	4.27, 5.36	steam-water p=0.5-3.5 MPa G=150-1760 kg/m^2s q”=0-540 kW/m^2, x=0.01-1-2
Zhao et al. (2003)	H	9	0.292	1.9	steam-water p=0.5-3.5 MPa, G=236-943 kg/m^2s q”=0-900 kW/m^2, x=0-0.95
Mandal and Das (2003)	V; A	10, 13	0.131-0.222	0-12	air-water in atmospheric conditions, Q_l=0.13-5.25·10^{-4} m^3/s, Q_g=3.65-14.2·10^{-5} m^3/s

Table 2 Experimental operating conditions considered for the correlation development.

Operating Conditions	**Global**	**Santini et al. (2008)**	**Zhao et al. (2003)**
p [bar]	5 – 65	10 – 65	5 - 35
G [kg/m^{2}s]	200 – 800	200 – 800	200 – 945
x [-]	0.0 – 1.0	0.0 – 1.0	0.0 – 1.0
q” [kW/m^2]	0 - 900	50 - 200	0 - 900

Table 3 Summary of the literature correlations used for the comparison.

Authors	**Correlation**
Ruffel (1974)	$\Phi_{lo}^2 = (1+F)\frac{\rho_l}{\rho_m}$ $F = \sin\left(\frac{1.16G}{10^3}\right)\left[0.875 - 0.314y - 0.74\frac{G}{10^3}(0.152 - 0.07y) - x\left(\frac{0.155G}{10^3} + 0.7 - 0.19y\right)\right] \times [1 - 12(x-0.3)(x-0.4)(x-0.5)(x-0.6)]$ $y = \frac{D}{100d}$
Awwad et al. (1995)	$\Phi_l = \left[1 + \frac{\chi}{CF_d^n}\right]\left[1 + \frac{12}{\chi} + \frac{1}{\chi^2}\right]^{0.5}$ $F_d = Fr\left(\frac{d}{D}\right)^{0.1}$; $F_d \leq 0.3$: $C = 7.79, n = 0.576$; $F_d > 0.3$: $C = 13.56, n = 1.3$
Xin et al. (1997)	$\Phi_l = \left[1 + \frac{\chi}{CF_d^n}\right]\left[1 + \frac{20}{\chi} + \frac{1}{\chi^2}\right]^{0.5}$ $F_d = Fr\left(\frac{d}{D}\right)^{0.5}(1 + \tan\beta)^{0.2}$ $F_d \leq 0.1$: $C = 65.45, n = 0.6$; $F_d > 0.1$: $C = 434.8, n = 1.7$
Guo et al. (2001)	$\Phi_{lo}^2 = 142.2\psi\left(\frac{p}{p_{cr}}\right)^{0.62}\left(\frac{d}{D}\right)^{1.04}\left[1 + x\left(\frac{\rho_l}{\rho_g} - 1\right)\right]$ $G \leq 1000\ \frac{kg}{m^2s}, \psi = 1 + \frac{x(1-x)\left(\frac{1000}{G} - 1\right)\left(\frac{\rho_l}{\rho_g}\right)}{1 + x\left(\frac{\rho_l}{\rho_g} - 1\right)}$ $G > 1000\ \frac{kg}{m^2s}, \psi = 1 + \frac{x(1-x)\left(\frac{1000}{G} - 1\right)\left(\frac{\rho_l}{\rho_g}\right)}{1 + (1-x)\left(\frac{\rho_l}{\rho_g} - 1\right)}$
Zhao et al. (2003)	$\Phi_{lo}^2 = 1 + \left(\frac{\rho_l}{\rho_g} - 1\right)[0.303x^{1.63}(1-x)^{0.885}Re_{lo}^{0.282} + x^2]$
Santini et al. (2008)	$\frac{dp_f}{dL} = K(x)\frac{G^{1.91}v_m}{d^{1.2}}$ $K(x) = -0.0373x^3 + 0.0387x^2 - 0.00479x + 0.0108$

Table 4 Average relative error between the compared correlations and the experimental data of Santini et al. (2008).

Correlation	**Average Deviation [%]**	**Correlation**	**Average Deviation [%]**
Eq. (22)*	11.6	Ruffel	23.5
Eq. (28)+	12.6	Awwad	27.5
Friedel	29	Xin	39.6
Lockhart – Martinelli	35.6	Guo	48.8
Santini	8.4	Zhao	47.8

*Obtained from modification of the Lockhart and Martinelli correlation

+Obtained from modification of the Friedel correlation

Table 5 Average relative error between the compared correlations and the experimental data of Zhao et al. (2003).

Correlation	**Average Error [%]**	**Correlation**	**Average Error [%]**
Eq.(22)*	25.3	Ruffel	34.7
Eq.(29)+	15.6	Awwad	40.0
Santini	39.4	Xin	32.5
Zhao	14.7	Guo	31.8

*Fitted on the Santini et al. (2008) database only

+Fitted on the Zhao et al. (2003) database only

Table 6 Summary of the uncertainties of the experimental measurements.

Water flow rate	± 1%
Fluid bulk and wall temperature	± 0.7 °C
Absolute pressure	± 0.1%
Differential pressure	± 0.4%
Supplied electrical power	± 2.5%