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A New Correlation for Single and Two-Phase Flow Pressure Drop in Round Tubes with Twisted-Tape Inserts

Twisted-tape inserts are frequently used in heat exchangers as a passive and inexpensive heat transfer enhancement method. However, their use results not only heat transfer coefficient increments, but also pressure drop penalties. The present study analyses the literature on single and two-phase frictional pressure drop inside tubes with twisted-tape inserts focusing on the physical mechanism and the effects of the use of twisted-tape in comparison to plain tubes. Experimental data were gathered from the open literature and compared against the available correlations developed in order to predict two-phase frictional pressure drop in tubes containing twisted-tape inserts. It was found that none of the correlations was able to predict such a database accurately. A new correlation to estimate the friction factor for two-phase flows inside tubes with twisted-tape is also proposed. Contrarily to previous studies, the proposed correlation presents reasonable predictions under single and two-phase flow conditions and obeys the trends when the twisted-tape ratio tends to zero and infinite.

Keywords: *twisted-tape, pressure drop, friction factor, swirl flow, flow boiling*

Introduction

The limited availability and rising costs of energy sources have motivated researchers to focus on minimizing energy consumption. In this scenario, great efforts have been made to develop more efficient and compact heat exchangers, according to Akhavan-Behabadi et al. (2009). In terms of heat exchanger efficiency, studies have concentrated on improving the global heat transfer coefficient and minimizing the pumping power. By minimizing the heat exchanger size, both the refrigerant inventory and the material used in its manufacture are reduced resulting in lower initial and operational costs. Moreover, the environmental impact during the system's lifetime is also reduced, since, as pointed out by Ribatski (2008), decreasing the refrigerant inventory implies that the amount of refrigerant leakage decreases in both relative and absolute values.

For compact heat exchangers the external heat transfer coefficient (generally air-side) is usually the predominant thermal resistance. However, according to Reid et al. (1991), this is not always the case, and any increase in the in-tube heat transfer coefficient may lead to a considerable improvement in the global coefficient. Several passive heat transfer enhancement methods have been proposed, and most of them are based solely on modifications of the internal tube surface. According to Shatto and Peterson (1996), the tube surface is altered in order to increase its effective heat transfer area and/or to disturb the flow through the generation of secondary flows and vortices. The available alternatives to increase the heat transfer coefficient through the generation of flow disruption include the use of microfinned tubes, coils, bends and twisted-tape inserts.

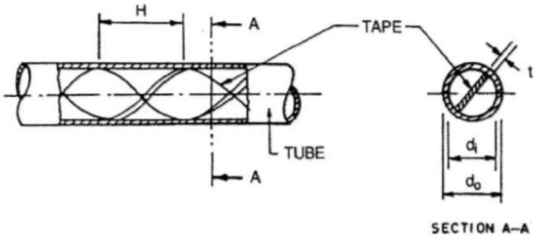

Figure 1. Schematic view of twisted-tape insert inside a tube, Akhavan-Behabadi et al. (2009).

Twisted-tapes are characterized by the twist ratio, defined according to Eq. (1), given by the ratio between the tape turn length of 180^o along its axis and the tube diameter, Akhavan-Behabadi et al. (2009), Figure 1.

$$y = \frac{H}{d_i} \tag{1}$$

Twisted-tape inserts have been used as a heat transfer enhancement technique for over a century, dating back to 1896, according to Manglik and Bergles (1993a). This technique was initially called "retarders" due to the increase in the pressure drop that the insert imposes on the flow, Manglik and Bergles (1993b). Twisted-tape inserts can be found in steam generators, heat recovers from flue gas, domestic heaters, according to Manglik and Bergles (1993b), desalination, according to Shatto and Peterson (1996), and industrial processes in general. The use of twisted-tapes has the following advantages over the other enhancement methods: low fabrication and installation costs, easy maintenance, Akhavan-Behabadi et al. (2009), and the possibility of being installed in heat exchangers already in operation, Thome and Ribatski (2005).

As mentioned above, and similarly to most heat transfer enhancement techniques, the heat transfer coefficient enhancement achieved by twisted-tape inserts is accompanied by a drastic increase in the pressure drop. Therefore, to determine the conditions under which the use of this method is favorable, heat transfer and pressure drop databases covering broad ranges of experimental conditions and a reasonable understanding of the phenomena involved are necessary. Based on both, accurate correlations can be developed and the heat exchangers performance estimated, Manglik and Bergles (1993a).

At this point, it should be highlighted that, for a given heat exchanger capacity, the pressure drop augmentation can be potentially overturned by the possibility of reducing the heat exchanger size due to the improvement in the heat transfer coefficient, which may imply a decrease in the refrigerant pumping power, Shatto and Peterson (1996). The pressure drop penalty and heat transfer enhancement provided by twisted-tape inserts are usually evaluated based on the pressure drop and heat transfer coefficient for a plain tube under similar operational conditions (vapor quality, mass velocity, fluid temperature, refrigerant and internal diameter) as follows:

$$\varepsilon_p = \frac{\Delta p_s}{\Delta p_p} \tag{2}$$

$$\varepsilon_q = \frac{h_s}{h_p} \tag{3}$$

To evaluate the global effectiveness of using twisted-tapes inserts, Akhavan-Behabadi et al. (2009) suggested the ratio between heat transfer enhancement and pressure drop penalty parameters as follows:

$$\varepsilon = \frac{\varepsilon_q}{\varepsilon_p} \tag{4}$$

Alternatively to the use of twisted-tapes along the entire heat exchanger length, researchers have proposed twisted-tape segments regularly spaced inside the tubes, Eiamsa-Ard et al. (2006) and Chang et al. (2009), or only along the initial portion of the flow entry, Eiamsa-Ard et al. (2009).

Lopina and Bergles (1973) developed an experimental study of water flow boiling inside tubes with twisted-tape inserts, and found that the effect of the heat flux on the heat transfer coefficient is negligible under conditions of low bubble generation frequency. Agrawal et al. (1982) and Agrawal and Varma (1991) performed an experimental study of pressure drop for convective boiling of R12 in horizontal pipes under constant heat flux conditions. Jensen et al. (1985) developed a study of pressure drop for two-phase vertical flow of R113 inside tubes with twisted tapes. Reid et al. (1991) compared the performance of microfinned tubes and tubes with twisted-tape inserts and observed that better heat transfer performance is achieved by using microfinned tubes. Manglik and Bergles (1993a, 1993b) presented an extensive review on horizontal single-phase laminar and turbulent flows inside tubes with twisted tapes and described some experimental data for water and ethylene glycol. Shatto and Peterson (1996) also presented a general review on this subject, focusing on two-phase flows. Chakroun and Al-Fahed (1996) studied heat transfer and pressure drop for single-phase laminar flow of oil in tubes with twisted-tape inserts. Eiamsa-Ard et al. (2006) analyzed the single-phase water flow for heat exchangers containing tubes with full length and regularly spaced twisted-tapes. They verified that under certain conditions the distance between the twisted-tape segments may promote a pressure drop reduction without considerable heat transfer coefficient penalties. Akhavan-Behabadi et al. (2009a) and Akhavan-Behabadi et al. (2009b) investigated the influence of the twisted-tape ratio on the flow boiling heat transfer coefficient and pressure drop for R134a.

In the present study, a literature review on the physical mechanisms responsible for the increment of pressure drop and heat transfer coefficient in the presence of twisted-tape inserts inside round tubes is presented. Predictive methods from the literature for pressure drop inside tubes containing twisted-tape inserts are reviewed and critically compared against experimental data from the literature. Additionally, based on the analysis of the literature and databases from independent laboratories, a new correlation to predict flow boiling and single-phase pressure drop inside tubes with twisted tapes is proposed.

Nomenclature

A = *coefficient for Müller-Steinhagen and Heck correlation, Pa/m*
B = *coefficient for Müller-Steinhagen and Heck correlation, Pa/m*
C = *correlation coefficient, dimensionless*
C_i = *constants for correlation being developed, i = 1,2 and 3, dimensionless*
C_{MS} = *coefficient of Müller-Steinhagen and Heck correlation, dimensionless*
C_R = *coefficient for Reddy correlation, dimensionless*
d = *diameter, m*
f = *friction factor, dimensionless*
Fr_h = *Froude number, dimensionless*
g = *gravity acceleration, m/s²*
G = *mass velocity, kg/m²-s*
h = *heat transfer coefficient, W/m²-K*
H = *tape turn length of 180°, m*
L = *tube length, m*
n = *correlation coefficient, dimensionless*
p = *absolute pressure, Pa*
Re = *Reynolds number, dimensionless*
x = *vapor quality, dimensionless*
y = *twist ratio, dimensionless*

Greek Symbol

α = *void fraction, dimensionless*
β = *coefficient in the Müller-Steinhagen and Heck correlation, Pa/m*
Δp = *pressure drop, Pa*
ε = *global efficiency increment, dimensionless*
ε_p = *pressure drop penalty, dimensionless*
ε_q = *heat transfer enhancement, dimensionless*
ϕ_{fo}^2 = *Reddy two-phase multiplier, dimensionless*
μ = *viscosity, Pa.s*
ρ = *density, Kg/m³*
σ = *surface tension, N/m*
θ = *tube inclination relative to vertical, deg*

Subscripts

acc = *accelerational pressure drop*
Cr = *critical property*
f = *frictional pressure drop*
g = *gravitational pressure drop*
H = *homogeneous model*
h = *hydraulic*
i = *internal*
L = *liquid-phase*
p = *plain tube*
s = *swirl flow*
tp = *two-phase*
V = *vapor-phase*

Heat Transfer and Pressure Drop Physical Mechanism

Manglik and Bergles (1993a) presented an analysis of the mechanisms responsible for the heat transfer and pressure drop enhancements in the presence of twisted-tape inserts for single-phase flow. From this analysis they pointed out the following mechanisms:

- Partial obstruction of the cross-section with consequent increase of the fluid velocity;
- Hydraulic diameter reduction: the flow is divided into two hydrodinamically independent flows with smaller hydraulic diameters, but higher wetted perimeter, promoting an increase in both heat transfer coefficient and pressure drop;
- Increase in the flow effective length: the length traversed by the fluid particles is longer than that in a smooth tube, as the fluid is forced to follow the tape profile;
- Swirl Flows: the motion imposed by the tape shape on the fluid particle causes the superposition of centripetal acceleration and longitudinal flow, generating swirl flows. As a result, the fluid velocity near the wall for tubes with twisted-tape inserts is higher than that for a plain tube. This mechanism promotes an increase in both heat transfer coefficient and pressure drop;
- Fin effect: if the contact between the tape and the pipe surface is tight, heat is transferred by conduction through the tape and then to the fluid in such a way that the tape acts as a fin increasing the overall heat transfer area.

According to Shatto and Peterson (1996), in the case of flows under heating conditions, the centripetal acceleration promotes the movement of the colder fluid (with a higher density) from the center of the tube to its periphery replacing the warmer fluid of lower density. Thome and Ribatski (2005) stated that such effect, coupled with the increase in the fluid velocity near the tube wall, favors the detachment of bubbles, intensifying the heat transfer coefficient during convective boiling under nucleate boiling dominated conditions. In the case of cooling applications, the centripetal acceleration acts to segregate the colder and warmer parcels of the fluid near the wall and near the center of the tube, respectively. Therefore, the twisted-tape inserts promote a decrease in the convective effects and are not recommended.

Figure 2 from Manglik and Bergles (1993a) shows schematics of the three single-phase swirl flow modes in the presence of twisted-tape inserts according to Reynolds number and the twisted-tape ratio. The scheme presented in Figure 2(a) corresponds to a condition of low Reynolds number or high twist-ratio, Figure 2(b) corresponds to a condition of high Reynolds number or small twist-ratio, and Figure 2(c) corresponds to a condition of plain tape. Based on this study, Shatto and Peterson (1996) speculated that similar phenomena should be expected during in-tube two-phase flows.

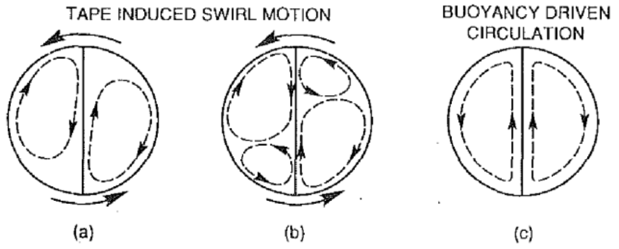

Figure 2. Single-phase flow modes induced by twisted-tape inserts according to Manglik and Bergles (1993a). (a) Low Reynolds number and/or high twist ratio, (b) high Reynolds numbers and/or low twist ratio, (c) infinite twist ratio (flat tape).

For in-tube horizontal flow boiling, swirl generators are advantageous especially under conditions of intermediate vapor qualities and low mass velocities, since for plain tubes and under these circumstances stratified flows are generally observed, according to Shatto and Peterson (1996). The advantage arises from the fact that the twisted-tape insert promotes the transition from stratified to annular flow at low mass velocities. In the case of annular flows, the entire internal tube perimeter is wet; therefore, the effective heat transfer area is much higher than that of stratified flows for which more than half of the tube perimeter is dry. According to Bergles et al. (1971), swirl generators can also be beneficial under mist flow conditions, since the radial acceleration favors the deposition of liquid droplets on the wall increasing the wetted surface length. Based on this behavior, Jensen (1985) suggested the use of twisted-tape segments along the region of high vapor qualities.

Considering only the friction factor behavior with Reynolds number, Manglik and Bergles (1993a) defined four regimes for single-phase flow inside tubes with twisted-tape inserts. The regimes were identified according to the contribution to the pressure drop of viscous, inertial and centrifugal forces, as well as turbulence forces related to the fluid velocity fluctuations. The regimes were described as follows:

- Regime I – <u>Viscous flow</u>: characterized by the predominance of inertial and viscous forces. The pressure drop increment is mainly due to the cross-section partial blockage and the increase in the flow effective length;
- Regime II – <u>Axial and swirl flow</u>: the balance among inertial, viscous and centrifugal forces characterizes the flow behavior. The friction factor increases due to the partial blockage of the cross-section, higher effective flow length, and swirl flows induced by tape;
- Regime III – <u>Swirl-turbulent transition</u>: characterized by the beginning of velocity fluctuations by turbulence and the predominance of swirl flow effects and suppression of turbulent fluctuations due to centrifugal forces. The influence of velocity fluctuations on the pressure drop is smaller than for plain tubes without twisted tapes;
- Regime IV – <u>Turbulent swirl-flow</u>: the velocity field is affected by both velocity fluctuations and centrifugal forces. Cross-section partial blockage and flow mixing by secondary flows are responsible for the pressure drop increment when compared to plain tubes without twisted-tape;

The transition from laminar to turbulent flow is characterized by velocity fluctuations and flow instabilities. According to Manglik and Bergles (1993b), the twisted-tape insert dampens the velocity fluctuations due to the promotion of centrifugal forces; therefore, the transition to turbulent regime is delayed. A flow map based on the Nusselt number as a function of Reynolds number and containing four flow regimes was also presented by Manglik and Bergles (1993a). Manglik and Bergles (1993b) proposed a generalized correlation for single-phase flow pressure drop in tubes with twisted-tape inserts, valid for both laminar and turbulent flow regimes, without discontinuities in the laminar-to-turbulent transitional region.

Pressure Drop Correlations for Two-Phase Flow Inside Tubes Containing Twisted-Tape Inserts

The total pressure drop is given by the sum of the frictional, gravitational and accelerational pressure drop contributions. In the case of two-phase flow inside tubes, the last two are estimated based on analytical models that include the variations of vapor quality and void fraction as follows:

$$\Delta p_g = gH\sin\theta\left[\alpha\rho_V + (1-\alpha)\rho_V\right] \tag{5}$$

$$\Delta p_{acc} = G^2\left\{\left[\frac{(1-x)^2}{\rho_L(1-\alpha)} + \frac{x^2}{\rho_V\alpha}\right]_{out} - \left[\frac{(1-x)^2}{\rho_L(1-\alpha)} + \frac{x^2}{\rho_V\alpha}\right]_{in}\right\} \tag{6}$$

In Equations (5) and (6), the outlet and local vapor qualities are calculated from an energy balance along the heated surface, and an additional correlation (or model) is necessary to estimate the void fraction (α) variation along the tube. To calculate the superficial void fraction for plain horizontal tubes without twisted-tape inserts, Wojtan et al. (2005) suggested the modified correlation of Rouhani and Axelsson (1970) given as follows:

$$\alpha = \frac{x}{\rho_V}\left[(1+0.12(1-x))\left(\frac{x}{\rho_V} + \frac{1-x}{\rho_L}\right) + \frac{1.18(1-x)\left[g\sigma(\rho_L-\rho_V)\right]^{0.25}}{G\rho_L^{0.5}}\right]^{-1} \tag{7}$$

The pressure drop frictional component is usually determined by empirical or semi-empirical correlations based on experimental databases, and includes the effects of transport and thermodynamic properties of the gas and liquid phases, vapor quality, mass velocity, geometry and surface characteristics of the channel, superficial void fraction and flow pattern.

The frictional pressure drop predictive methods for two-phase flow inside tubes with twisted-tape inserts are generally based on the product between a twisted-tape multiplier and the pressure drop for a plain tube with the same diameter. Based on such an approach, Agrawal et al. (1982) proposed the following correlation:

$$\frac{\Delta p_S}{\Delta p_P} = \frac{C}{y^n} \tag{8}$$

where y is the twist ratio, Δp_s is the frictional pressure drop for the tube with twisted-tape inserts, and Δp_p is the pressure drop for plain tube given by Martinelli and Nelson, apud Thome (2004), keeping the same mass velocity and internal diameter. The empirical values of C and n are 5.120 and 0.509, respectively, and were fitted according to their database described in Table 1.

Similarly, Akhavan-Behabadi et al. (2009) compared their two-phase flow pressure drop data inside plain tubes without twisted-tape inserts against predictive methods. According to their data, the Friedel (1979) method was quoted as the best. Therefore, in their predictive method also given by Eq. (8), Δp_p is provided by Friedel (1979) and new values of C and n equal to 5.1 and 0.28, respectively, are adjusted to their database also described in Table 1.

According to the state-of-the-art review by Shatto and Peterson (1996), Blatt obtained values of C and n of 7.36 and 3/5, respectively, based on R11 experimental data.

Jensen et al. (1985) performed an extensive study of two-phase pressure drop inside tubes with twisted-tape inserts using R113 as working fluid. They obtained diabatic and adiabatic pressure drop data for tubes with and without twisted-tape inserts. The experimental conditions are presented in Table 1.

Their method to predict two-phase pressure drop for flow inside tubes with twisted-tape inserts consists of two steps:
Step one: to determine the friction factor for two-phase flow inside plain tubes, as follows:

$$f_{tp} = \Delta p_P \frac{\rho_H d_h}{2G^2 L} \tag{9}$$

where

$$\Delta p_P = \phi_{fo}^2 \Delta p_L \tag{10}$$

with the two-phase multiplier calculated as proposed by Reddy, apud Jensen et al. (1985):

$$\phi_{fo}^2 = 1 + x\left(\frac{\rho_L}{\rho_V} - 1\right)C_R \tag{11}$$

where

$$\begin{aligned} C_R &= 1.17x^{-0.175}G^{-0.45} & p/p_{Cr} &> 0.187 \\ C_R &= 0.41(1+10p/p_{Cr})x^{-0.175}G^{-0.45} & 0.094 \le p/p_{Cr} &\le 0.187 \end{aligned} \tag{12}$$

Step two: to calculate the two-phase pressure drop for the tube with twisted-tape insert with friction factor given by the following equation:

$$\begin{aligned} \frac{f_s}{f_{tp}} &= \frac{2.75}{y^{0.406}} & &for \quad y > 11.5 \\ \frac{f_s}{f_{tp}} &= \frac{\left(4y^2+\pi^2\right)^{3/2}}{8y^3} & &for \quad y \le 11.5 \end{aligned} \tag{13}$$

Table 1. Database descriptions and pressure drop plain tube correlations adopted by the authors in order to develop their two-phase pressure drop correlations for tubes containing twisted-tape inserts.

Author	Fluid	No. of Points	Twist-Ratios	G[kg/m^2-s]	d_i[mm]	Δp_P Method
Agrawal et al. (1982)	R12	120	3.76,5.58,7.37 and 10.15	198-388	10.0	Martinelli and Nelson, apud Thome (2004)
Akhavan-Behabadi et al. (2009)	R134a	160	3,6,9 and 12	54-144	7.5	Friedel (1979)
Jensen et al. (1985)	R113	481	3.94,8.94 and 13.92	120-1600	8.1	Reddy, apud Jensen et al. (1985)

For the characteristic length of the flow, Jensen et al. (1985) adopted the hydraulic diameter calculated assuming the presence of the tape and given by:

$$d_h = \frac{4\left(\pi d_i^2/4 - \delta d_i\right)}{\pi d_i + 2d_i} \tag{14}$$

At this point it is interesting to observe that the values of the constants and exponents presented by Akhavan-Behabadi et al. (2009), Blatt, apud Shatto and Peterson (1996), and Agrawal and Varma (1991) are quite similar, indicating consistency between these works.

A New Pressure Drop Correlation

In general, the flow boiling heat transfer coefficient and pressure drop increase with decreasing of the twisted-tape ratio, Jensen et al. (1985). However, to date there has been no generalized correlation or model for their prediction. Modeling the flow boiling mechanisms inside plain tubes without twisted tapes is still challenging to several researchers over the world, since its solution combines the knowledge of nucleate boiling mechanisms, turbulence, interfacial phenomena, two-phase flow, thermal instabilities, and others. All these subjects are also not completely understood and have been themes of several scientific papers published every year. Additional difficulties appear in the case of flow boiling in tubes containing twisted-tape inserts, due to the presence of two-phase swirl flow. Moreover, only a limited amount of experimental data concerning heat transfer, pressure drop and flow pattern is available. Based on these aspects, instead of developing a mechanistic model, this paper proposes a new empirical correlation for the frictional pressure drop during single and two-phase flow inside tubes with twisted-tape inserts. The correlation was developed based on experimental data for R134a flow boiling and air and water single-phase flows gathered in the literature, totalizing 180 data. The database is described in Table 2 and includes data from independent laboratories. It was not possible to gather data from other studies since they were not available or some of their experimental conditions were missing.

Table 2. Data of pressure drop collected for the development of a new correlation.

Author	No. of Data Points	Fluid	Internal Diameter	Twist-Ratio	Inlet Vapor Quality Range	Mass Velocity
Akhavan-Behabadi et al. (2009)	116	R134a	7.5	3,6,9 and 12	0.2-0.9	54-144
Eiamsa-Ard et al. (2006)	34	Water	47.5	6 and 8	–	45-227.2
Naphon (2006)	20	Water	8.1	3.08 and 3.7	–	241-1209
Promvonge (2008)	10	Air	47.5	4	–	1.284-6.227

Hernandes (2010) performed a comparison between the leading plain-tube frictional pressure drop predictive methods and a database gathered from the literature containing experimental results from independent laboratories. From this study, he found that the predictive method by Müller-Steinhagen and Heck (1986) provided the best agreement with his database. Figure 3 illustrates a comparison between the experimental data for R134a of Saiz-Jabardo and Bandarra Filho (2006), Moreno-Quibén (2005) and Colombo et al. (2008) with internal diameters of 8.76, 8.92 and 13.8 mm, respectively, and the predicted data according to Müller-Steinhagen and Heck (1986) correlation. The figure shows that most of the experimental data are predicted by this method within an error range of ± 30%.

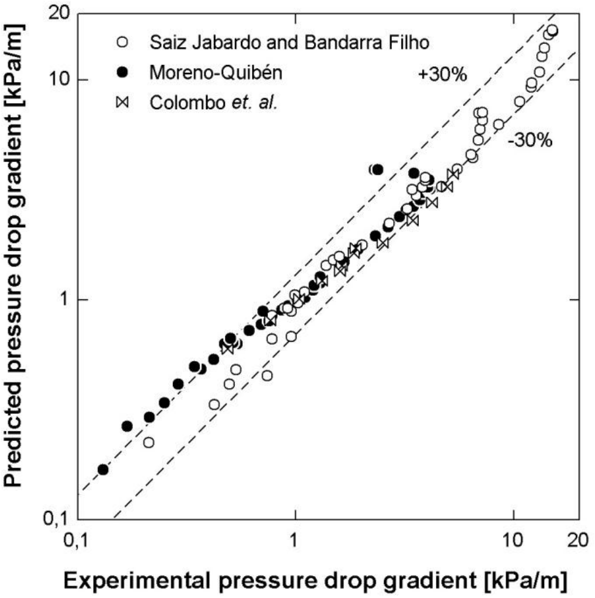

Figure 3. Comparison between Müller-Steinhagen and Heck (1986) correlation and data from Saiz Jabardo and Bandarra Filho (2006), Moreno-Quibén (2005) and Colombo et al. (2008).

According to the Müller-Steinhagen and Heck (1986) method, the two-phase frictional pressure drop is basically given as an empirical interpolation between liquid and vapor frictional pressure drop given by Blasius method. Therefore, the correlation can also be used under single-phase flow conditions.

The Müller-Steinhagen and Heck (1986) correlation is given as follows:

$$\left(\frac{dp}{dL}\right)_{f,tp} = \beta(1-x)^{1/C_{MS}} + Bx^{C_{MS}} \tag{15}$$

where C_{MS} is 3 and β is given by

$$\beta = A + 2(B-A)x \tag{16}$$

with

$$\left(\frac{dp}{dL}\right)_{f,L} = f_L\frac{2G^2}{\rho_L d_h} = A \tag{17}$$

$$\left(\frac{dp}{dL}\right)_{f,V} = f_V\frac{2G^2}{\rho_V d_h} = B \tag{18}$$

and

$$f_L = \frac{16}{\mathrm{Re}_L},\ f_V = \frac{16}{\mathrm{Re}_V},\ \text{for } \mathrm{Re}_L,\ \mathrm{Re}_V\ \le 1187$$

$$f_L = \frac{0.079}{\mathrm{Re}_L^{1/4}},\ f_V = \frac{0.079}{\mathrm{Re}_V^{1/4}},\ \text{for } \mathrm{Re}_L,\ \mathrm{Re}_V\ > 1187$$

$$\mathrm{Re}_L = \frac{Gd_h}{\mu_L}\ ,\ \mathrm{Re}_V = \frac{Gd_h}{\mu_V}$$

Taking into account that the method of Müller-Steinhagen and Heck (1986) is simple to be implemented and worked the best according to Hernandes (2010) comparisons, this method was adopted in the present study to estimate the plain tube friction factor and to develop a correlation for frictional pressure drop inside tubes containing twisted-tape inserts. In our proposed method, the plain tube friction factor is calculated as follows:

$$f_{tp} = \Delta p_p \frac{\rho_H d_h}{2G^2 L} \tag{19}$$

where G is based on the internal diameter, Δp_p is the plain tube pressure drop given by Müller-Steinhagen and Heck (1986) correlation, and ρ_H is the homogeneous two-phase density.

$$\rho_H = \left[\frac{(1-x)}{\rho_L} + \frac{x}{\rho_V}\right]^{-1} \tag{20}$$

In order not to take into account the tape thickness influence on the correlation to be developed, a hydraulic diameter neglecting the tape thickness was assumed in Eq. (14) as follows:

$$d_h = d_i \frac{\pi}{\pi+2} \tag{21}$$

Therefore, the friction factors for swirl and plain tube flows are calculated with the hydraulic diameter given by Eq. (21) and keeping the same mass velocity.

Based on these assumptions, friction factor ratios based on the experimental data for tubes with twisted-tape inserts and on the Müller-Steinhagen and Heck (1986) correlation for plain tubes were calculated. These results are displayed in Figure 4, according to which the friction factor ratio has a strong dependence on the mass velocity, in this case given through the Froude number (Eq. (24)). It can be also speculated that gravitational effects can play a role in the frictional pressure drop since they are related to the transition from stratified to annular flow. Therefore, the Froude number was considered in order to capture inertial and gravitational effects. Taking into account these effects, a relationship in the form of Eq. (22) was considered for the development of the new pressure drop correlation for single and two-phase flows inside tubes with twisted-tape inserts.

$$\frac{f_s}{f_p} = \left(1 + C_1 y^{C_2} Fr_h^{C_3}\right)^{C_4} \tag{22}$$

Using the least square regression analysis, the database abovementioned was correlated as follows:

$$\frac{f_s}{f_p} = \left(1 + 2y^{-0.4} Fr_h^{-0.1}\right)^{1.5} \tag{23}$$

with the Froude number given by:

$$Fr_h = \frac{G^2}{g d_h \rho_H^2} \tag{24}$$

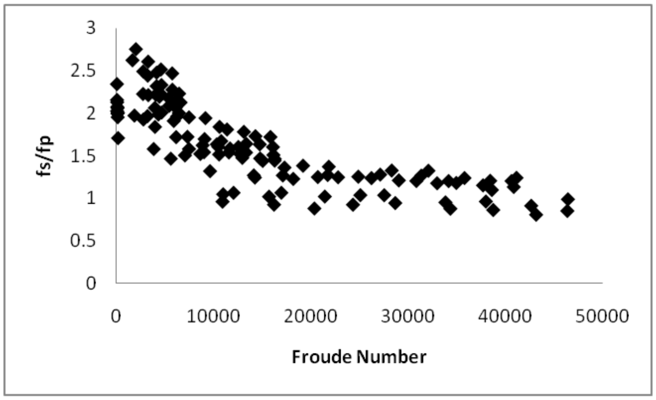

Figure 4. Dependence between friction factor ratio and Froude number, for twist ratio from 4 to 15, considering data from studies described in Table 2.

In the right hand-side of Eq. (22), the unity was added to predict the following limit cases: (i) plain tape corresponding to an infinity twist ratio when the friction factor ratio should be equal to one; (ii) blocked tube, corresponding to a twist ratio equal to zero when the friction factor ratio should be infinity. The fact that both exponents (of Froude number and twist ratio) are negative in Eq. (23) is in agreement with the following aspects: (i) the swirl effect decreases with increasing mass velocity approaching the behavior of tubes without twisted-tape inserts; (ii) by increasing the twist ratio, the effects of the twisted tape on the two-phase flow tend to become negligible. Figure 5 shows that, according to the proposed correlation, the ratio of the friction factors for a tube with and without twisted tape inserts tends asymptotically to 1 by increasing the twist ratio.

Finally, the two-phase pressure drop for tubes with twisted tape inserts is given as follows:

$$\Delta p_s = f_s \frac{2G^2 L}{\rho_H d_h} \tag{25}$$

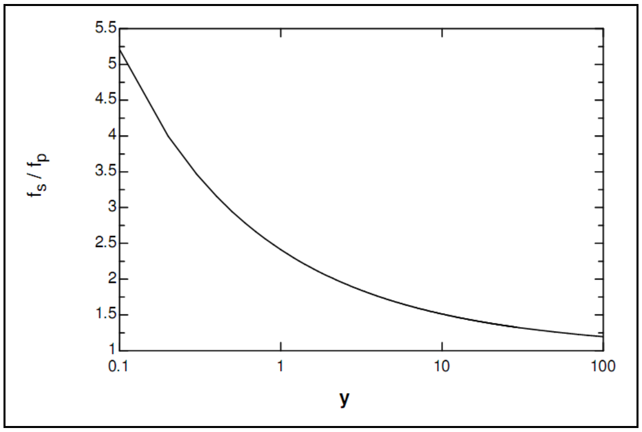

Figure 5. Illustration of correlation trend for cases of twist ratio tending to zero and infinite, for R134a, T_{sat} of 12°C, internal diameter of 7.5 mm, G of 200 kg/m²-s, and vapor quality of 0.1.

The proposed correlation shows reasonable agreement with the data points considered in the present study since 81.5% of the results were predicted within ±30% of the experimental data (see Figure 6). The best agreement was obtained for the single-phase flow, with 98% of the predictions within ±30% of the experimental data, while the two-phase flow presented 73.3% within this accuracy range.

Considering the database of Akhavan-Behabadi et al. (2009), a comparison between the proposed and prior correlations is displayed in Figure 7. It can be observed that the proposed correlation presents good agreement with the experimental data, as well as the correlation of Akhavan-Behabadi et al. (2009). Agrawal et al. (1982) correlation overpredicted the friction factor, and Jensen et al. (1985) and Blatt, apud Shatto and Peterson (1996) underpredicted it.

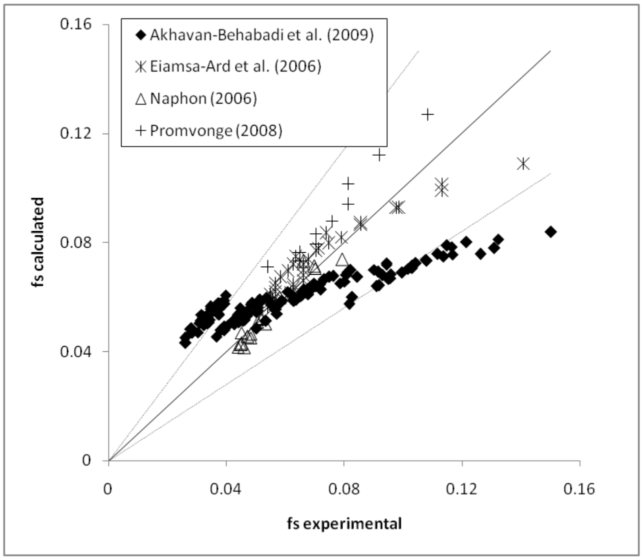

Figure 6. Predicted and experimental friction factor for single and two-phase flow inside tubes with twisted-tape inserts. Dotted lines for ±30% deviation, and solid line for null deviation.

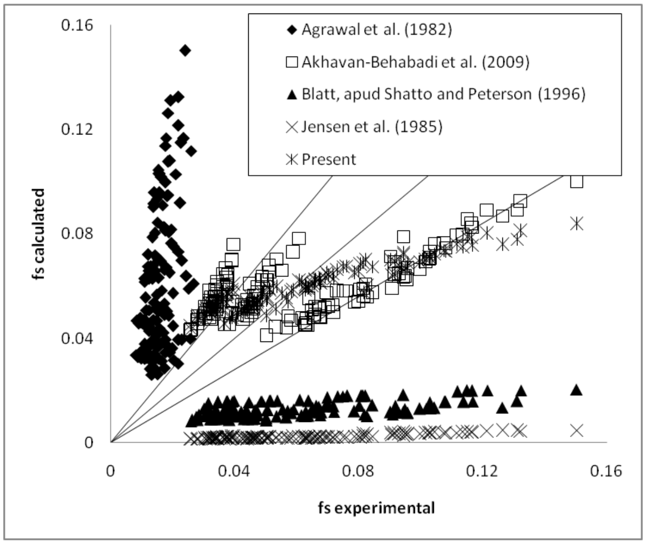

Figure 7. Calculated and experimental friction factor for Akhavan-Behabadi et al. (2009) data points.

Finally, it must be highlighted that, contrarily to the previous predictive methods, the proposed correlation has the advantages of predicting the friction factor for both, single and two-phase flow. Additionally, the correlation obeys the limits when the twist ratio tends to 0 and infinity.

Conclusions

Based on the present study, the following conclusions can be drawn:

- An extensive review of single and two-phase flow inside tubes with twisted tape has been presented, addressing mechanisms and working parameters that may influence pressure drop and heat transfer coefficient. As a result, correlations for the frictional pressure drop available in the literature for two-phase flow inside tubes with twisted-tape inserts have been presented and discussed and seem to have failed when predicting independent databases;
- The technique of twisted tape inside tubes allows enhancing the heat transfer coefficient with reduced cost, and its application is possible in heat exchangers already in use;
- A new correlation for the friction factor valid for single and two-phase flows inside tubes with twisted-tape inserts has been proposed.

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