# PRESSURE DROP CALCULATION IN DILUTE PHASE PNEUMATIC CONVEYING SYSTEMS USING A HYDRODYNAMIC MODEL

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***Abstract****. Pressure drop is a key parameter in the design of dilute phase conveying systems. This paper presents a hydrodynamic model to predict the pressure drop in horizontal and vertical pneumatic conveying systems. The pressure drop through the conveying line as a function of the process parameters was obtained from the mass and momentum balances. The flow was considered one-dimensional, steady, incompressible and isothermal. The resultant ordinary differential equations system was expressed as an initial value problem. Some terms of the equations feature empirical correlation. The wall friction and drag force coefficients and terminal velocity correlation were introduced in the model. Finally, the numerical solution of the equation system was obtained by a multiple step method and the calculated results were in accordance with the experimental data available in the literature.*

***Keywords****: Pneumatic Conveying, Pressure drop, Dilute-phase.*

## 1. INTRODUCTION

Pneumatic conveying is frequently employed in industries, such as the chemical, food processing and mining industries. Although such process is well established in these industries, the phenomenon of conveying that occurs inside the pipeline is still of complex nature, since it depends of the gas velocity, the characteristics of the solid particle (size, distribution through the line, density and shape), the gas-solid loading ratio, the feeding system and pipeline configuration. The project of a pneumatic conveying system involves the knowledge of the pressure drop through the pipeline. Thus, many investigations described in the literature and the actual model recommended to predict the pressure drop on a horizontal and vertical pneumatic conveying system in dilute phase consider that the pressure drop is evaluated by adding up the contribution of the components caused by the acceleration of the solid phase, gravity and friction created by the gas-wall and particle-wall interaction.

Hariu and Molstad (1949) presented results of tests made in four different materials in a vertical flow. Measurements of pressure drop, gas velocity and solids mass flow were reported for two pipe diameters. A theoretical analysis of the effects that influences the head loss was accomplished. The equations obtained showed that the total head loss depended on the static height of the solid, the head loss caused by the friction of the gas and the solid on the wall. On the other hand, a balance of forces revealed that the head loss caused by the solid could be obtained through the correlation of the drag. Variations in the coefficient values of solid friction, calculated from the experimental data, pointed to the possibility that the particle still had not reached the equilibrium velocity, so a methodology to estimate the length of the acceleration area was also proposed. However, the simplified hypothesis introduced on the average velocity in two points of test section did not allow for a rigorous treatment.

Arastoopour and Gidaspow (1979), considered the experimental results of Zens (1949) and of Hariu and Molstad (1949) to establish comparisons among four models: pressure drop in both solid and fluid phases (annular flow model) – case A; pressure drop in fluid phase – case B; relative velocity – case C; and the partial pressure drop in both phases – case D. In the work of these researchers, the friction force particle-wall in the model was ignored and, although the investigators had observed that the model of relative velocity (case C) presented better results in comparison with the predictions of the pressure drop, solid velocity and porosity and doubts remained over the fact that the particle-wall friction force is negligible for the dilute phase flow. In the comparative analysis among the models, they concluded that the model with the partial pressure drop in both phases, case D, was inconsistent with Zens's experimental results.

Yang *et al* (1982) developed a hydrodynamic model. Assuming to be a bi-dimensional steady state, based on continuity and momentum equations, they evaluated the pressure drop and saltation velocity for horizontal gas-solid flow. One-dimensional hydrodynamic models proposed in the literature were formulated and analyzed by the authors for two-dimensional cases. The characteristics of the equations as an initial value problem are imaginary. In order to obtain a well-posed equations system, they introduced alternative modifications in the model. Only two of the modified models, the pressure drop in gas phase and the relative velocity model, were well-posed. They compared the calculated values for pressure drop before saltation with Konchesky's experimental data (1975). They verified that the model they developed is unable to generate the sharp increase in pressure drop found experimentally by Konchesky. This behavior could be caused by the improper drag force expression for dense solid transport that occurs after the saltation phenomenon. Neglecting the friction forces between the solid particles and the pipe wall is another possible cause for the low pressure drop values calculated after the saltation in the pipe.

Dzido *et al* (2002), conducted an investigation about the region of acceleration, where the model proposed considers the expressions derivates by Haider and Levenspiel (1989) for calculating the drag in spherical and non-spherical particles. The particle-wall friction coefficient is evaluated by two correlations – one used in the work from Kmiec and Leschonski (1987) and other proposal by Yang (1978). Comments and comparisons are made about the importance of estimating the initial value of the solid velocity over pressure profile through the conveying line. One correlation is presented to estimate the minimum value of the initial solid velocity in the beginning of the acceleration region and the best results are obtained from the B and D classification group of Geldart. The numerical results from this model were compared to the experimental data with good accordance. However, is important to note that this data were the same employed for construction of the drag coefficient and the friction correlations in the acceleration region.

Benitez and Mesquita (1997) proposed and analyzed two extensions of the models of Gidaspow (1979) and Soo (1967) (apud Benitez and Mesquita, 1997) for the gas-liquid flow in dilute phase in the acceleration area. The models included the solid-wall friction where the Yang (1978) correlations were used for the calculation of such parameter. Beyond the modeling and simulation of the pneumatic conveying, they made comparisons with the experimental results obtained by Zens (1949). The comparisons indicated that the extension of the model of Soo (1967) completely underestimated the experimental results while the other one derived from the model of Gidaspow (1979) presented satisfactory results for the pressure drop and acceleration length.

This paper uses the extension of the model of Gidaspow (1979) proposed by Benitez and Mesquita (1997), presenting the mass balance and movement quantity equations in a dimensionless form, introducing in the model the calculation of the drag coefficient for non-spherical particles, and finally, the results for the prediction of the velocity profile and pressure drop are obtained considering the use of two correlations for the calculation of the solid-wall friction, one derived from Yang (1978) and the other from Kerker (1977). All the results are compared to the data available in literature.

## 2. THE MATHEMATICAL MODEL

The pneumatic conveying of solids can be described through the balances of mass and momentum for an isothermal, one-dimensional and permanent flow, considering the dilute phase, the solid phase or the mixture. The equations of continuity for the gas and solid are respectively:

$$\frac{d}{dx}\left(\varepsilon\rho_g v_g\right)=0 \tag{1}$$

$$\frac{d}{dx}\left((1-\varepsilon)\rho_s v_s\right)=0 \tag{2}$$

In Eqs. (1-2), $\varepsilon$ is the void fraction, $v_g$ , $\rho_g$ , and $v_s$ , $\rho_s$ , the velocity and density, respectively, from the gas transport and the solid. The balance of movement quantity for the mixture, taking into account the particle-wall friction, is written as:

$$\left(1-\varepsilon\right)\rho_s v_s\frac{dv_s}{dx}+\varepsilon\rho v_g\frac{dv_g}{dx}+\left(\rho_s\left(1-\varepsilon\right)+\varepsilon\rho\right)g\sin\left(\theta\right)=-\frac{dP}{dx}-F_g-F_s \tag{3}$$

where, $\theta$ is the angle between the pipe axis with the horizontal line, $P$ is the pressure, $F_g$ and $F_p$ are the friction force per unit volume of particles between the gas and the wall and between the particles and the wall, respectively.

In order to determinate the four unknown values ( $v_g$ , $v_s$ , $\varepsilon$ and $P$), we need one more equation. The model adopted in this paper is known in the literature as the relative velocity model, Gidaspow (1979). We included the contribution of solid friction in the model, according to Benitez and Mesquita (1997), and Kimiec and Leschonski (1987). Then, the resulting equation can be written as,

$$-\frac{1}{2}\rho_s\frac{d}{dx}\left(v_g-v_s\right)^2=-F_d-g\rho_s\sin\left(\theta\right)-\frac{F_p}{1-\varepsilon} \tag{4}$$

where, $F_d$ is the drag force exerted by the gas on the particles-per-unit volume of the particles. The friction force resulted from the gas-wall interaction can be expressed through the equation of Fanning

$$F_g=\frac{f_g\rho_g v_g^2}{2D} \tag{5}$$

In the above equation, $f_g$ is the gas friction coefficient, which is a function of the Reynolds number and the relative roughness of the tube. The friction factor, for high Reynolds numbers, can be described from the equation of Colebrook

$$\frac{1}{\sqrt{f_g}} = -0,86\ln\left(\frac{e/D}{3,7} + \frac{2,51}{\mathrm{Re}\sqrt{f_g}}\right) \tag{6}$$

The effect of the friction of solid particles with the wall to the total head loss of a pneumatic conveying system has been studied for many authors (Harius and Molstad, 1949; Yang, 1974; Kerker, 1977; Yang, 1978). The correlations below are used for calculating the friction between the solid particles and the wall. The correlations were obtained by Yang (1978), and Kerker (1977), respectively,

$$f_s = 0.02925\frac{(1-\varepsilon)}{\varepsilon^3}\left[\frac{(1-\varepsilon)v_g}{\sqrt{gD}}\right]^{-1.15} \tag{7}$$

$$f_s = 0.00315\frac{(1-\varepsilon)}{\varepsilon^3}\left[\frac{\mathrm{Re}_t}{\mathrm{Re}_p}\right]^{-0.979} \tag{8}$$

$$f_s = \frac{1}{217(1-\varepsilon)+1}\left\{3.13x10^{-5}Ga^{0.26}(S^{0.5}Fr_p^{-0.25}Ga^{0.16}(DR)+1.55x10^{-3}Fr_u^{0.5})\right\} \tag{9}$$

where, $f_s$ is the solid friction coefficient, $DR$ is the ratio between the tube diameter and the particle diameter, $S$ the relation between the sound velocity in the solid u and the normal particle velocity $V_s$. The Eq. (7) is used in horizontal pipes and the others equations are used in vertical pipes. The Froude ($Fr_p$ and $Fr_u$), Galileo *(Ga)* and Reynolds ($Re_t$ and $Re_p$) numbers are written by the following relations,

$$\begin{aligned} Fr_p &= v_s^2 / gd_p \\ Fr_u &= u^2 / gd_p \\ G_a &= gd_p^3\rho_g^2 / \mu_g^2 \\ \mathrm{Re}_t &= dU_t\rho_g / \mu_g \\ \mathrm{Re}_p &= d(v_g - v_s)\rho_g / \mu_g \end{aligned} \tag{10}$$

Being $U_t$ the particle terminal velocity. The solid friction force between the particle and the wall by volume unit is written as

$$F_p = \frac{f_s\rho_s(1-\varepsilon)v_s^2}{2D} \tag{11}$$

The drag force for spherical and non-spherical particles, respectively, by Dzido *et al* (2002)

$$F_d = \frac{3}{4}C_{Ds}\frac{\rho_g(v_g - v_s)^2}{d_p}\varepsilon^{-2.65} \tag{12}$$

$$F_{d\varphi} = \frac{3}{4}C_{Ds}\frac{\rho_g(v_g - v_s)^2}{\varphi d_v}\varepsilon^{-2.65} \tag{13}$$

where $d_p$ and $d_v$ are the particle diameter and particle volumetric diameter, respectively, and $F$ is the drag force caused by the fluid over the particles per volume unit of particles. $C_{Ds}$ is drag coefficient that is a function of Reynolds number. For Reynolds number greater then 1000, $C_{Ds}$ is constant. Equation (14) is used to determinate $C_{Ds}$

$$\begin{aligned} C_{Ds} &= \frac{24}{\mathrm{Re}_p}(1+0.15\,\mathrm{Re}_p^{0.687}) && \mathrm{Re}_p < 1000 \\ C_{Ds} &= 0.44 && \mathrm{Re}_p > 1000 \end{aligned} \tag{14}$$

or still by Haider and Levenspiel (1989) you can write

$$C_{Ds} = \frac{24}{\mathrm{Re}_p}(1+0.1806\,\mathrm{Re}_p^{0.6459}) + \frac{0.4251}{1+\dfrac{6880.95}{\mathrm{Re}_p}},\ \text{para}\ \mathrm{Re}_p < 2.6x10^5 \tag{15}$$

and for non-spherical particles, Haider and Levenspiel (1989), developed the correlation

$$C_{Ds\varphi} = \frac{24}{\mathrm{Re}_p}\left[1+(8.171\exp(-4.0655\varphi))\,\mathrm{Re}_p^{0.0964+0.5565\varphi}\right] + \frac{73.69\,\mathrm{Re}_p\exp(-5.0748\varphi)}{\mathrm{Re}_p + 5.378\exp(6.2122\varphi)} \tag{16}$$

Eq. (16) is valid for $\varphi > 0.67$. The Reynolds number that appears on the expressions above is written as

$$\mathrm{Re}_p = \frac{\varepsilon\rho_g d_p (v_g - v_s)}{\mu_g} \tag{17}$$

where $\mu_g$ is the viscosity of the fluid.

## 3. NUMERICAL SOLUTION

Eqs. (1 - 2) were integrated and expressed algebraically in function of the solid and gas flow rates, $W_s$ and $W_g$, respectively.

$$(1-\varepsilon)\rho_s v_s = (1-\varepsilon_1)\rho_s v_{s1} = W_s \tag{18}$$

$$\varepsilon\rho_g v_g = \varepsilon_1\rho_{g1} v_{g1} = W_g \tag{19}$$

where, $\varepsilon_1$ is the initial porosity, $v_{s1}$ is the solid initial velocity, $v_{g1}$ is the initial gas velocity and $\rho_{g1}$ is the initial gas density. The gas is by hypothesis considered ideal and the solid and gas phases are considered uncompressible in the state equation. Therefore, from Eqs. (18 – 19) results:

$$\varepsilon = 1-(1-\varepsilon_1)\frac{v_{s1}}{v_s} \tag{20}$$

$$\frac{\varepsilon}{\varepsilon_1}\frac{v_g}{v_{g1}} = \frac{P_1}{P} \tag{21}$$

where $P$ is the initial pressure of the system in the feeding zone. Using the Eqs. (20-21) in the Eqs. (3-4) the system is transformed in a dimensionless form that uses the following definitions.

$$\bar{\varepsilon} = \frac{\varepsilon}{\varepsilon_1},\ \bar{v}_g = \frac{v_g}{v_{g1}},\ \bar{v}_s = \frac{v_s}{v_{s1}},\ \bar{P} = \frac{P}{P_1},\ R = \frac{\rho_s}{\rho_{g1}},\ F = \frac{gL}{v_{g1}^2}\sin(\theta),\ C = \frac{P_1}{\rho_{g1}v_{g1}^2},\ s = \frac{v_{s1}}{v_{g1}} \tag{22}$$

The resulting system is given in the form of an initial value and is represented by the following expressions:

$$\bar{\varepsilon} = \frac{1}{\varepsilon_1}\left(1-\frac{1-\varepsilon_1}{\bar{v}_s}\right) \tag{23}$$

$$\bar{v}_g = \frac{1}{\bar{\varepsilon}\bar{P}} \tag{24}$$

$$\frac{d\bar{v}_s}{d\bar{x}} = A_1\left\{-\left[F\left(R(1-\varepsilon_1\bar{\varepsilon})+\bar{P}\varepsilon_1\bar{\varepsilon}\right)+\frac{2Lf_g\bar{P}\bar{\varepsilon}\varepsilon_1\bar{v}_g^{-2}}{D}+\frac{Lf_sR(1-\bar{\varepsilon}\varepsilon_1)s^2\bar{v}_s^{-2}}{2D}\right]+\right.$$

$$+ (C\overline{\varepsilon}\overline{P}^{2} - \varepsilon_1)\left\{ -\frac{3}{4}\frac{L\varepsilon_1^{-2.65}}{d_p}\frac{C_{Ds}\overline{P}(\overline{v}_g - s\overline{v}_s)}{R}\overline{\varepsilon}^{-2.65} + \frac{F}{(\overline{v}_g - s\overline{v}_s)} + \frac{Ls^2 f_s}{2D(\overline{v}_g - s\overline{v}_s)}\frac{R}{R}\overline{v}_s^{\,2} \right\}\Bigg\} \tag{25}$$

$$\frac{d\overline{P}}{d\overline{x}} = \frac{A_2\left[ F\left( R(1-\overline{\varepsilon}\varepsilon_1) + \overline{P}\overline{\varepsilon}\varepsilon_1 \right) + \frac{2Lf_g\overline{P}\overline{\varepsilon}\varepsilon_1\overline{v}_g^{2}}{D} + \frac{Lf_s R(1-\overline{\varepsilon}\varepsilon_1)s^2\overline{v}_s^{2}}{2D} \right]}{\overline{\varepsilon}\overline{v}_s^{2}(C\overline{\varepsilon}\overline{P}^{2} - \varepsilon_1)} \tag{26}$$

where,

$$A_1 = \frac{\left(\varepsilon_1\overline{\varepsilon}\overline{v}_s^{2}\right)}{\left[ s\varepsilon_1\overline{\varepsilon}\overline{v}_s^{2}\left( Rs(1-\varepsilon_1) - (C\overline{\varepsilon}\overline{P}^{2} - \varepsilon_1) \right) - (1-\varepsilon_1)(C\overline{P}) \right]},$$

$$A_2 = \overline{P}(1-\varepsilon 1)\left(1 - R(\overline{\varepsilon}s\overline{v}_s)^2\,\overline{P}\right)\frac{d\overline{v}_s}{d\overline{x}} - (\overline{\varepsilon}\overline{P}\overline{v}_s)^2 \tag{27}$$

The numerical solution was obtained by using the Adams-Moulton method, where the initial conditions ($v_g = v_{g1}$, $v_s = v_{s1}$, $\varepsilon = \varepsilon_1$, $P = P_1$) on $x = x_0$, must be informed. Some literature presents information about the initial conditions measured in the test sections. This enables us to compare the numerical results with the experimental data, from the acceleration zone to the total developed flow zone.

## 4. RESULTS

Figures 1-9 present the results obtained for this hydrodynamic model. Fig. 1 shows the pressure drop for different values of superficial gas velocity along the vertical tube. In order to compare the results from the present work with with Zenz (1949) experiments, the same initial conditions and the same solid particles were used. For this analysis, it is needed an initial pressure and an initial solid velocity (or concentrations), in addition to the inlet gas velocity. Since such data were not given, reasonable values for initial pressure and inlet void fraction were chosen and used to predict the experimental pressure drop across the lift line. It is clear that this result indicates a point of minimum pressure drop showing the existence of two regions, the dense and dilute phases. The values of pressure drop are evaluated in very good concordance with Zenz's experimental data.

The calculations performed using model A (pressure drop in both solid and fluid phases) were more accurate than those provided by model B and C, and in this case, for superficial gas velocities above 9 m/s, the values are slightly above the experiment.

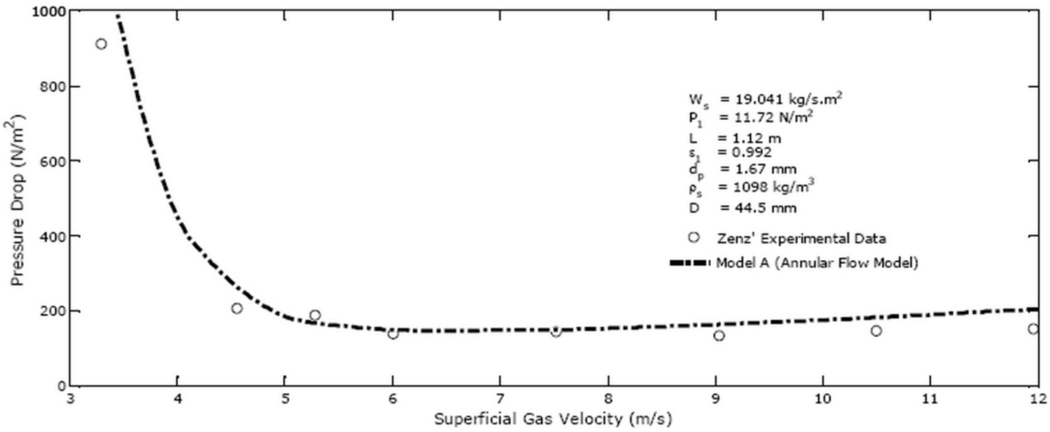


Figure 1 - Comparison between ours results and Zenz' experiment.

Figure 2 shows our calculated results compared to the experiment of Konchesky *et al* (1975). The test section was a 1.22 m straight horizontal pipeline of 0.2 m diameter. The experiments were made with air as the gas phase and crushed coal as the solid phase. The solid-phase density was 2240 kg/m$^3$, and the average particle diameter was 2840 pm. The same initial conditions and the same type of solid particles, gas, and horizontal pipe were used. In the numerical calculations, the phase-velocity distributions and the value and distribution of the gas volume fraction of the inlet points are needed. Because this information is not measured and reported in the experiments, reasonable values and distributions for void fraction and their velocities were chosen. We adopted an inlet pressure of 1.089x105 N/m$^2$ for the flow with a mass flow rate of 27.42 kg/s. m$^2$. The appropriate value of the initial gas volume fraction, $\varepsilon_o$ = 0.99862, was chosen to be in accordance with the experimental data.

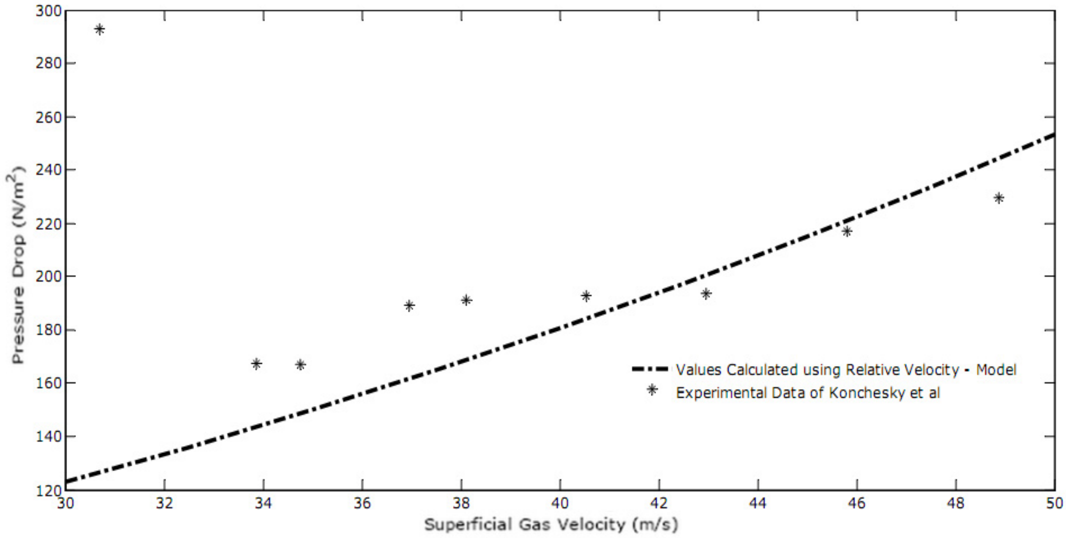


Figure 2 – Drop pressure versus Superficial Gas Velocity.

The calculated values using the relative velocity model show reasonable agreement with Konchesky's experimental data for higher velocities. However, the model is unable to represent the curve in the pressure drop obtained experimentally by Konchesky. This behavior occurs due the fact of the pressure drop related to the weight of the gas-solid mixture in the Mixture Momentum Equations is equal to zero.

Figure 3 shows the numerical results compared with Hariu and Molstad’s experimental data. They measured the total pressure drop in the transport of solid particles by an air stream through a vertical tube with an inside diameter of 6.78 x 10$^{-3}$ m. In ours numerical calculations we used the same solid particles, gas, and vertical tube, and the same initial conditions. The Ottawa Sand Type A with average particle diameter of 0.5 mm and particle density of 2641.65 kg/m$^3$ was chosen. An initial pressure of 1.99 x 10$^5$ Pa, an initial mass flow rate of 10 kg/h and an initial void fraction of 0.955 were used. The predicted values using the relative velocity equation model for the pressure drop were in accordance with the experimental values as long as the values of void fraction were adjusted as function of the mass flow rate , keeping in mind that for each set of air velocity and increasing in the mass flow rate, the void fraction should decrease. By selecting the values for void fraction at different solid mass flow rates, we were able to obtain reasonable values for pressure drop and average solid velocity compared with experimental values. The calculated values corroborate very well the Hariu and Molstad's experimental data.

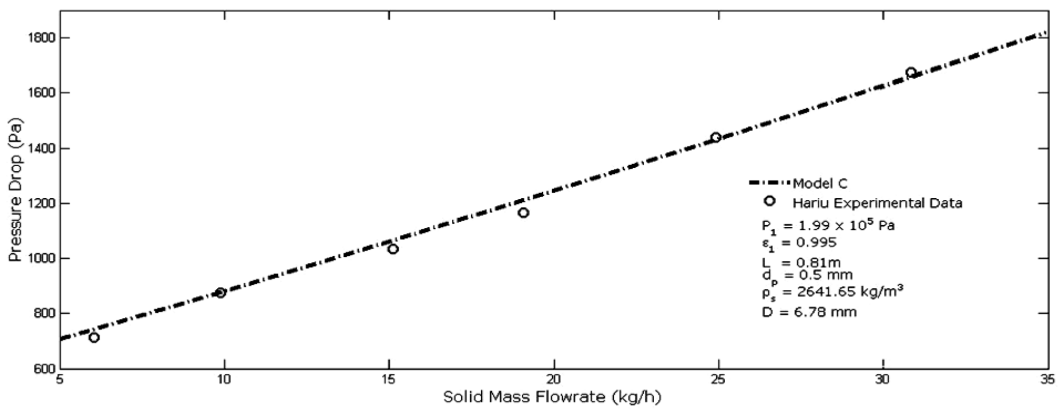


Figure 3 – Comparison of the relative-velocity-model pressure drops with Hariu and Molstad's experimental data for an assumed inlet void fraction of 0.955.

Figure (4-8) shows the numerical results versus Herbreteau and Bouard’s (2000) experimental data. The glass particles used had an average diameter of 0.035, 0.203 and 0.340 mm, respectively. Since the values of inlet void fraction are not given, we chose reasonable values to predict the experimental pressure drop. In order to compare our calculated quantities with Herbreteau and Bouard's experiments, the same initial conditions and the same solid particles, and gas were used. The initial pressure and porosity were 4 x $10^5$ Pa and 0.9, respectively. The experimental setup of Herbreteau and Bouard have both horizontal and vertical sections. The total length of 36 m is divided into two 15-meter horizontal pipelines and one 6-meter vertical section. The inner diameter of the stainless pipe is 80 mm. The total pressure drop per unit length was determined by the sum of the horizontal and vertical pressure drop divided by total length. Since a pipeline presents two bends, the total pressure drop include the contribution from both bends. The three models (A, B and C) are in reasonable concordance with the experimental results (Figs. 5-7). As previously commented, the model cannot represent the curve corresponding the transition between dense and dilute phases. This is because the terms related to weight in the Mixture Momentum Equations are zero.

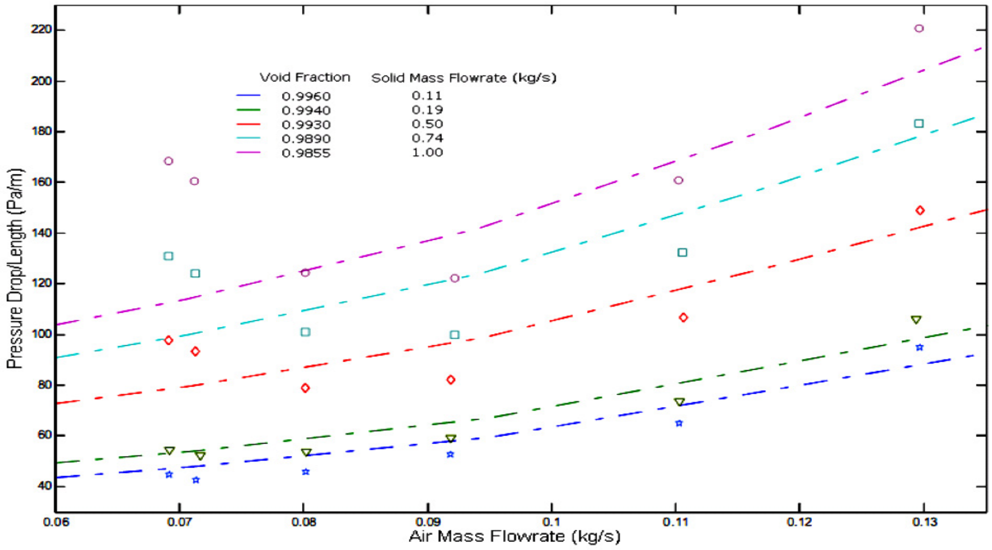


Figure 4. Comparison of the pressure drops with Herbreteau and Bouard's experimental data for 0.035 mm glass beads. Model A.

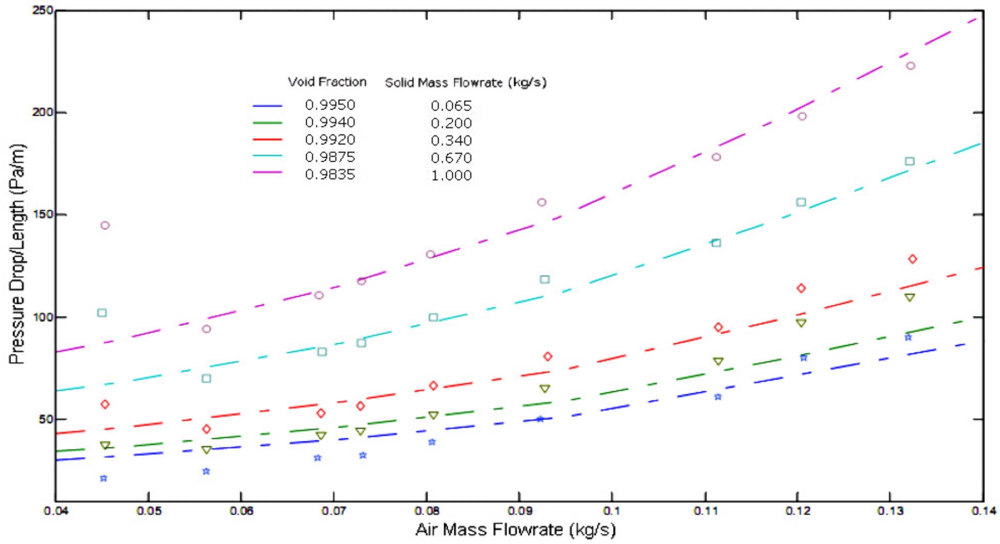


Figure 5. Comparison of the pressure drops with Herbreteau and Bouard's experimental data for 0.203 mm glass beads. Model A.

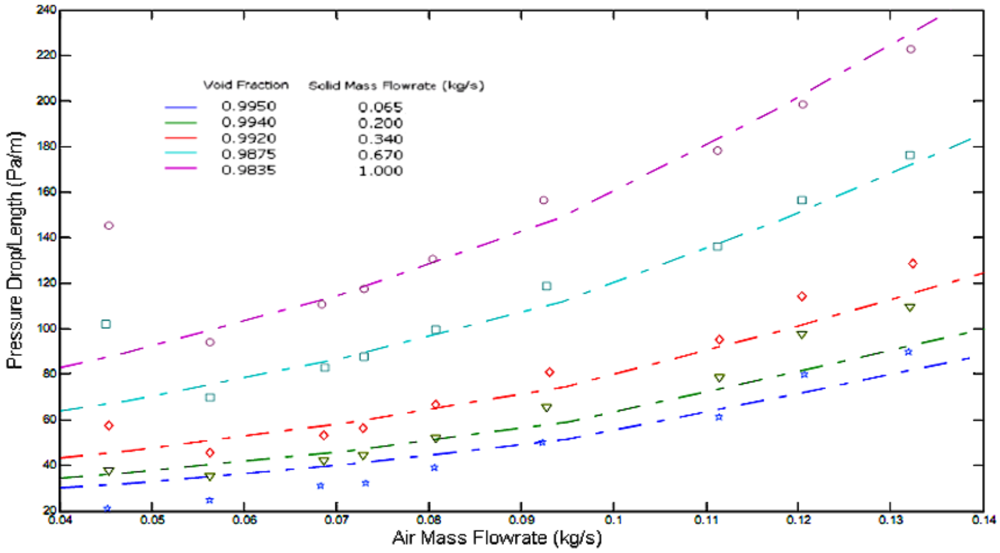


Figure 6. Comparison of the pressure drops with Herbreteau and Bouard's experimental data for 0.203 mm glass beads. Model B.

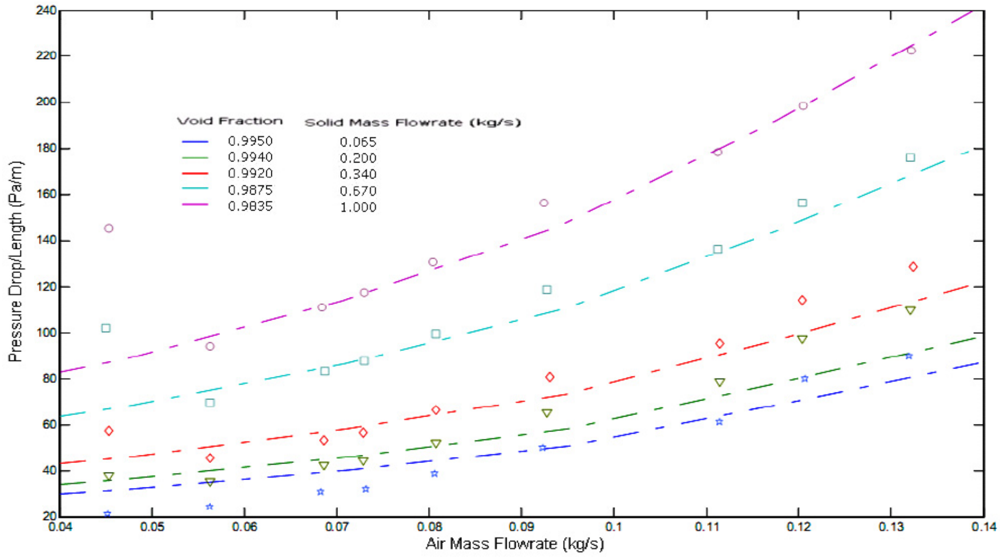


Figure 7. Comparison of the pressure drops with Herbreteau and Bouard's experimental data for 0.203 mm glass beads. Model C.

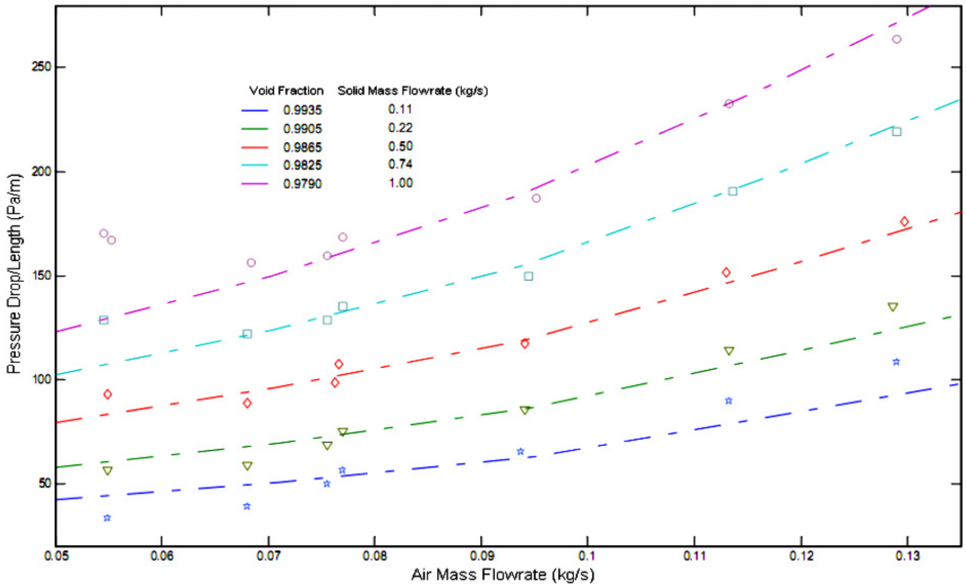


Figure 8. Comparison of the pressure drops with Herbreteau and Bouard's experimental data for 0.340 mm glass beads. Model A.

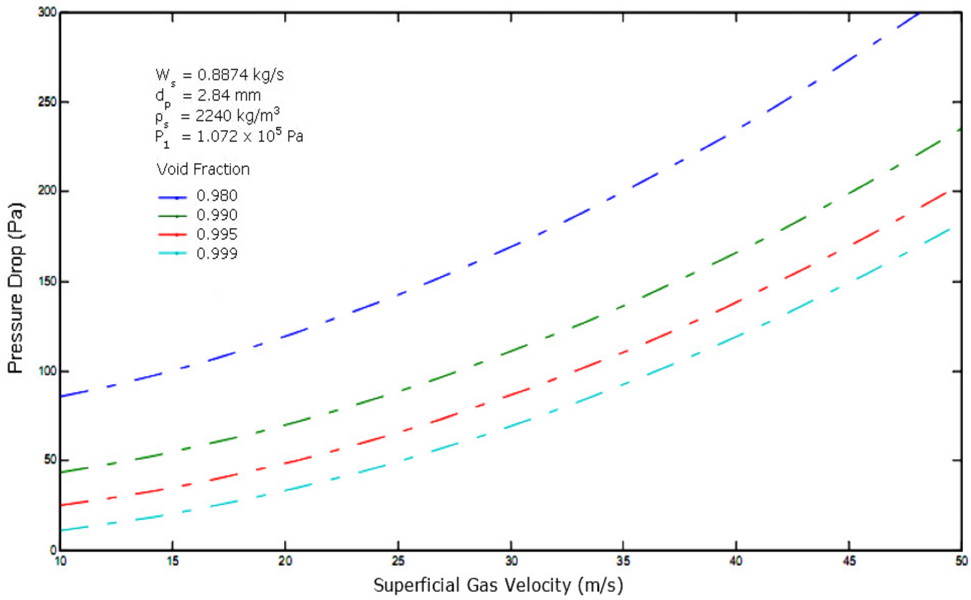


Figure 9. The effect of initial gas volume fraction on the pressure drop for the flow of a solid-gas mixture through a horizontal pipe calculated by using the relative velocity model.

The effect of the inlet gas volume fraction was computed. Figure 9 shows a significant effect on the initial gas volume fraction, $\varepsilon_o$, on the pressure drop along the 1.22 m test section of the pipe at various superficial gas velocities, calculated by using the relative velocity model. The initial system pressure was 1.072 x $10^5$ Pa. A decrease in void fraction results in a decrease in the velocity of the solids, which, in turn, causes higher pressure drop and more settling of the particles. Thus, more gas at the higher velocities is needed to exert a sufficient drag force to sustain a dispersed suspended flow in the line.

## 5. CONCLUSION

Pressure drop in dilute phase horizontal and vertical pneumatic conveying can be predicted by using a hydrodynamic model. Comparison between the model and the experimental data involved the use of standard drag correlations and the fitting of an inlet void fraction which was not measured.

We have observed that the estimate of initial velocity (or initial porosity) is a parameter that strongly influences the theoretical prediction of pressure distribution and there is no methodology that is able to predict the initial conditions in terms of other parameters involved, such as geometric configuration of the feeding device and empirical correlations for calculating the coefficients of drag and friction (gas-solid and particle-wall) in the inlet flow.

Results of numerical analysis for this model, considering different experimental data, demonstrate the ability to obtain accurate predictions of the pressure drop to flow in the horizontal and vertical directions in pneumatic conveying systems. The calculated values are able to explain the physical behavior of the steady-state two-phase transport systems. Therefore, they are suitable to be used as a tool in the design of pneumatic conveying systems.

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