COBEM 99
XV Congresso Brasileiro de Engenharia Mecânica
15th Brazilian Congress of Mechanical Engineering
22 - 26 de Novembro de 1999 / November 22 - 26, 1999 Águas de Lindóia, São Paulo,

STUDY OF ABRUPT CONTACT FORCES VARIATION ON THE DYNAMICS OF PIGS MOVING THROUGH MULTISIZE PIPELINES

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Abstract. *Simulation of the dynamics of pig through gas pipelines with variable contact force is presented. The differential mass and linear momentum equations were numerically solved by a finite difference numerical scheme, for compressible flow through pipelines. The fluid flow equations were combined with an equation representing a force balance on the pig. Pressure forces developed due to flow through by-pass holes in the pig, pig acceleration and pig/pipe contact forces were considered. A stick/slip model was developed to account for the distinct friction regimes that prevail depending on whether the pig is stopped or in motion. An adaptive grid technique was employed to account for the moving pig. Very often, pipelines are built with different pipe sets (pipes with different diameters, roughness, etc). As a result, severe contact force changes can be encountered by pig along its movement inside the pipeline, in which a cleaning operation must be carry on. Further, flanges also induce great contact force variation. At the present work, two cases are examined. The first case presents an analysis of the effect of pipeline flanges in the pig motion. The second case examines the pig movement along a pipeline, with different pipe sets. The numerical solution examined helps to understand the pig dynamics, to prevent situations in which the pig gets stuck inside the pipeline.*

Key-words: *Pipeline Pigging, Numerical Simulation, Compressible/Incompressible Flow*

1. INTRODUCTION

Multi-diameter pipelines are frequently encountered in oil and gas operations. Sub-sea oil production lines, for instance, are normally constructed of multidiameter pipes, in order to decrease its initial cost. It is not uncommon that the transition from one diameter to the other be implemented in a relatively short duct length, giving rise to an abrupt pipeline area change.

Pigs are normally utilized in different stages of the pipeline life to perform operations such as, drying, cleaning or internal inspection for damage or corrosion spots. In general terms, a pig is a solid cylindrical plug driven through the pipeline by the flowing fluid. Contact forces between the pig and the pipe wall are developed due to the oversize of the pig and should be overcome by the driving pressure provided by the flow. In the case of multidiameter lines, the abrupt area changes located at the diameter transition section give rise to potentially elevated values of the contact forces. These localized, higher values of the contact forces may force the pig to slow down or even stall. Depending of the flow operating conditions, the pressure upstream of the pig can be insufficient to set it back into motion, causing the loss of the pipeline. Conversely, if the available upstream pressure is sufficiently high, the pig will normally be set in motion with high acceleration, leading to elevated pig velocities with potential risks for the facilities and personal, specially in the case of pigs with elevated mass.

As exemplified in the above, the passage of a pig through a pipeline is a complex operation. Nevertheless, today the vast majority of pigging operation are still designed based solely on field experience, without a solid engineering background (Cordell, 1986).

Few works can be found in the literature about the motion of pigs in pipeline. Haun (1986) treat the dynamics of pigs in gas lines. Modeling of the contact forces is presented by Gomes (1994). Azevedo et al. in 1996 presented a simple model to predict the pig motion and in 1997 analyzed the by-pass flow and contact force. Burt and MacDonald, 1997 investigated how to track the pig in a pipeline. Recently, Vianes and Rachid (1997) and Monteiro et al., 1998, studied the pig motion through pipelines and Nieckele et al. (1998) investigated the pig performance in dewatering operations.

The present work is part of a broader project aimed at simulating pigging operations based on fundamental mechanical principles (Braga et al., 1998). The equations governing the conservation of mass, linear momentum for the fluid are numerically solved, coupled with an equation that describes the pig dynamics. Models for the prediction of the contact forces developed between the pig and the pipe wall were also devised. In particular, for pigs made from assemblies of flat poliurethane discs, a parametric study, based on finite element simulations, resulted in a comprehensive database to be employed in fast estimates of their driving pressures.

In the present paper, attention is focused on the prediction of the fluid flow and pig dynamics characteristics of pigging operations in multidiameter pipelines where abrupt changes on the magnitude of the pig/wall contact force are encountered. In particular, two cases are studied: pigging of gas lines with flanges protruding inwards into the pipe, and pigging of a pipeline with severe area changes. These two representative cases provide insight into various phenomena that occur in field operations.

2. MATHEMATICAL MODEL

The motion of a pig inside a pipe can be obtained by the solution of the fluid flow problem, coupled with a model to predict the pig motion. At the present work the gas flowing in the pipeline is considered to be Newtonian and isothermal. Thus, the flow problem is governed by the conservation of mass and linear momentum equations.

It is assumed that the flow is one-dimensional, however, the centerline of the duct can be inclined with the horizontal at an angle α. Pipe deformation effects due to pressure variations along the fluid are considered. The mass conservation equation can be written as (Wylie and Streeter, 1978)

$$\frac{\partial p}{\partial t} + V \frac{\partial p}{\partial x} + \frac{\rho a^2}{\xi} \frac{\partial V}{\partial x} + \frac{\rho a^2}{\xi} \frac{V}{A} \frac{\partial A}{\partial x} = 0 \qquad (1)$$

where ρ, V, P, a, A are the density, velocity, pressure, speed of sound and area, respectively. ξ is given by $\xi = 1 + \rho a^2 \; 2 C_D \, (D / D_{ref})$ where D and D_{ref} are the pipeline diameter and the reference diameter determined at atmospheric pressure p_{atm}. The pipe deformation due to pressure is accounted by the coefficient C_D, given by $C_D = (1 - \mu^2)\, D_{ref} \,/(2\, e\, E)$, where e is the pipe wall thickness, E the Young's modulus of elasticity of the pipe material, μ the Poisson's ratio. The cross sectional area can be determined based on the reference diameter D_{ref} as

$$A = \pi \frac{D^2}{4} \qquad ; \qquad D = \frac{D_{ref}}{\left[1 - C_D \left(p - p_{atm}\right)\right]} \tag{2}$$

The momentum conservation equation can be written as

$$\frac{\partial U}{\partial t} + U \frac{\partial U}{\partial x} = -\frac{1}{\rho}\frac{\partial p}{\partial x} - \frac{f}{2}\frac{|U| U}{D} - g \sin \alpha \tag{3}$$

where g is gravity and f the friction coefficient. The friction factor depends on the Reynolds number Re,

$$Re = \rho\, V\, D / \mu_f \tag{4}$$

where μ_f is the absolute viscosity. In the turbulent regime, the friction factor is also a function of the relative pipe roughness ε/D. It can be approximated by its fully developed expression. (Fox and McDonald, 1995)

$$f = \frac{64}{Re} \quad \text{for } Re \leq 2300 \quad \text{and} \quad f = 0.25 \left[\log\left(\frac{\varepsilon / D}{3.7} + \frac{5.74}{Re^{0.9}} \right) \right]^{-2} \quad \text{for } Re > 2300 \tag{4}$$

The coupling of the pig motion with the fluid flow in the pipeline is obtained through a balance of forces acting on the pig (Azevedo et al., 1996) which can be written as

$$m \frac{dV_p}{dt} = (p_1 - p_2)\, A - m\, g \sin \alpha - F_{at}(V_p) \tag{5}$$

where, V_p is the pig velocity, m the pig mass, p_1 and p_2 the pressure on the upstream and downstream faces of the pig, α is the angle of the pipe axis with the horizontal.

The term $F_a\,(V_p)$ represents the contact force between the pig and the pipe wall. When the pig is not in motion, the contact force varies from zero to the maximum static force, F_{stat} in order to balance the pressure force due to the fluid flow. In case the pressure gradient is negative, this maximum force is F_{stat}^{neg}. If the pressure gradient is positive, the maximum force is F_{stat}^{pos}. These two values of forces are not necessarily equal since the pig may resist differently to being pushed forward or backward.

Once the pig is set in motion by the flow, the contact force assumes the constant value, F_{dyn}, representing the dynamic friction force that is generally different from the static force. As in the previous situation, two different values for the dynamic contact force are allowed, F_{dyn}^{neg} and F_{dyn}^{pos}, depending on the direction of the pig motion.

The contact force depends on x_p, the pig axial coordinate, indicating that it can vary along the pipe length. The values assumed by the contact force can be summarized as follows,

$$F_{at}(V_p) = \begin{cases} -F_{din}^{neg}(x_p) & \text{if} \quad V_p < 0 \\ F(x_p) & \text{if} \quad V_p = 0 \quad \text{where } -F_{stat}^{neg}(x_p) \le F(x_p) \le F_{stat}^{pos}(x_p) \\ F_{din}^{pos}(x_p) & \text{if} \quad V_p > 0 \end{cases} \tag{6}$$

The gas is considered to behave as an ideal gas. Therefore for an isothermal flow,

$$\rho = \frac{p}{a^2} \qquad ; \qquad a^2 = z\ R_{gas}\ T_{ref} \tag{7}$$

where R_{gas} is the gas constant, T_{ref} the reference temperature and z the compressibility factor. The fluid absolute viscosity is considered constant for the present analysis.

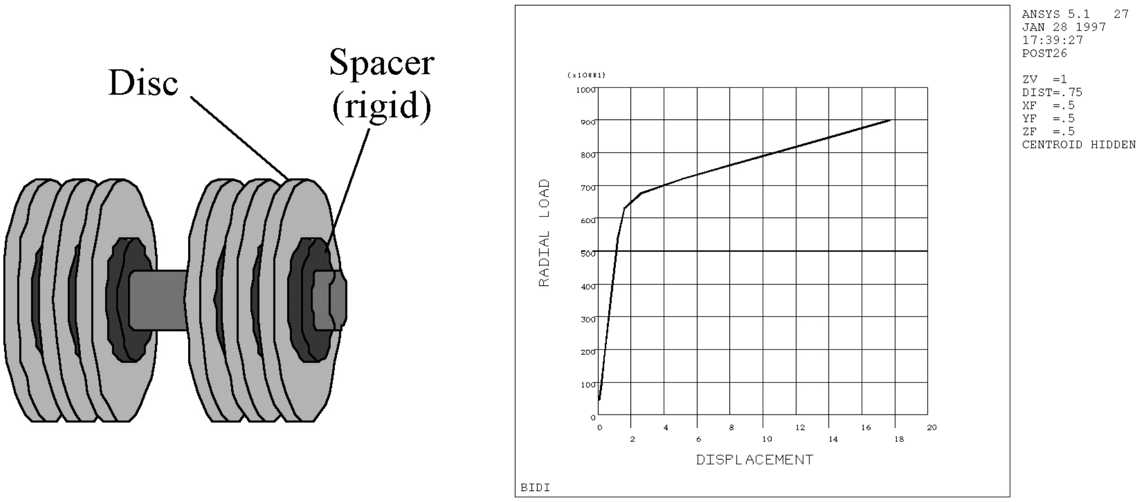
3. NUMERICAL METHOD

The set formed by equations (1), (3) and (5), together with the appropriate boundary and initial conditions, require a numerical method to obtain the desired time-dependent pressure and velocity fields. These equations were discretized by a finite difference method. A staggered mesh distribution was selected to avoid unrealistic oscillatory solutions, as recommended by Patankar, (1980). The equations where integrated in time using a semi implicit method, that is, the equations are integrated by a totally implicit method, but the coefficients are locally linearized. The space derivatives were approximated by the central difference method around the mesh point. The resulting coefficient matrix is penta-diagonal, and can be easily solved a direct penta-diagonal algorithm.

The total number of grid points inside the pipe was maintained constant in the numerical calculations of the flow field upstream and downstream of the pig, as well as for the pig dynamics calculations. However, as the pig moves along the pipe, it is convenient to rearrange the node distribution. The number of grid points upstream and downstream of the pig was made proportional to the length of the pipe at each side of the pig.

4. CONTACT FORCE MODEL FOR FLAT DISCS

Figure 1(a) shows the sketch of a *flat disc* pig, a tool that is ordinarily used in a number of conventional pigging operations. As discussed above, contact forces due to oversize are some of the key input parameters in simulations of pig dynamics. In the case of discs such as those depicted in Figure 1(a), the radial and friction (axial) contact forces are high enough to make them buckle to fit into the pipe. Therefore, approximate models used to estimate the driving pressure for disc pigs must take their geometrically nonlinear, postbuckling behavior into account. Typical results from finite element simulation of a disc postbuckling response are reproduced in Figure 1(b). One notices that the load *vs*. displacement response may be approximated by a bilinear curve. This, in fact, is the basis for the parametric model developed for fast estimates of contact forces on disc pigs. The important nondimensional parameters are the thickness to diameter and the thickness to spacer diameter ratios, as well as the friction and Poisson coefficients. For different values of these parameters, over 200 finite element runs yielded the nondimensional, approximately bilinear, radial load *vs*. displacement curves that have been stored in a database. These data may be employed, as in the first case study reported here, in order to provide fast estimates of contact forces for a wide range of



(a) (b)

Figure 1 – (a) Typical disc pig; (b) Result of finite element, postbuckling simulation.

commercially available discs. Results of such estimates have been validated in experiments conducted at the Petrobras R&D Center (Braga *et al*., 1988).

5. RESULTS

The present analysis was obtained with the numerical code PIGSIM (Nieckele et al., 1998), which has been extensively tested, presenting good agreement with several tests problems available in the literature.

Two different situations are investigated here. The first one consists of a pigging operation in a gas line with flanges protruding inwards into the pipe. The second one presents the pigging operation in a pipeline with severe area change.

5.1 Pigging in a gas line with flanges

The pipeline under consideration is 40 meters long and horizontal. It is non-rigid with a reference diameter equal to 10 cm. It has two internal flanges, with thickness t_f = 1 cm and internal diameter D_f, = 9 cm, as illustrated in Figure 2. The wall thickness e has 2.5 mm, and its roughness ε is 0.05 mm. The pipe Young's modulus of elasticity, E is 2×10^{11} MPa and its Poisson's coefficient, μ is 0.3.

Initially there is no flow, then at time equal to zero, air is pumped at the inlet, taking 0.6 seconds to achieved a mass flow rate of 0.03 kg/s, which is maintained constant. The

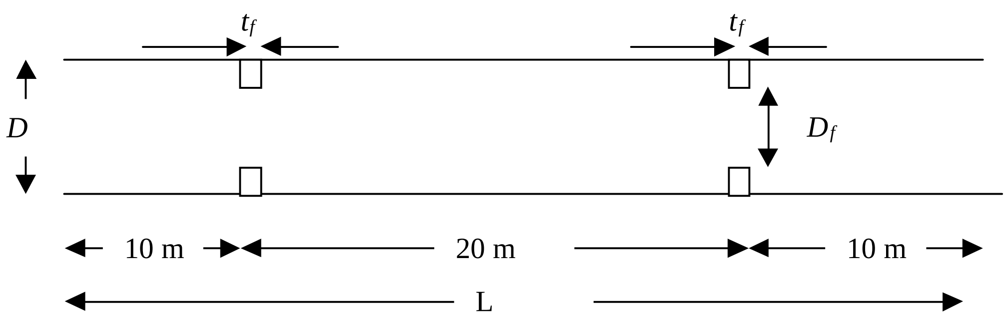


Figure 2 - Pipeline with flanges

discharge pressure is kept constant and equal to the atmospheric pressure.

Air was considered as an ideal gas. Its gas constant was set as R = 287 N m /(Kg K) and the compressibility factor was specified as z = 1.04. The temperature was maintained at 294K and the absolute viscosity μ_f was kept constant and equal to 1.9×10^{-5} Kg/ (m s).

The pig had a mass equal to 500 g. At time equal to zero, the pig was introduced in the entrance of the pipeline with zero velocity. The positive and negative dynamic forces were: $F_{dyn}^{pos} = F_{dyn}^{neg} = 230\ N$ for the pipeline and $F_{dyn}^{pos} = F_{dyn}^{neg} = 1000\ N$ for the flanges. The static forces $F_{stat}^{pos} = F_{stat}^{neg} = 250\ N$ were for the pipeline and $F_{stat}^{pos} = F_{sta}^{neg} = 1050\ N$ for the flanges.

Figure 3 illustrates the pig velocity versus the pig position, while figure 4 presents the variation of the pig position with time. Figure 5 presents the pressure distribution along the pipeline for several time instants. Figure 5a corresponds to the following time instants: 2.5 s; 5 s, 7.5 s and 10 s., while Figure 5b, corresponds to 12.5 s, 17.5 s, 22s and 24.5 s

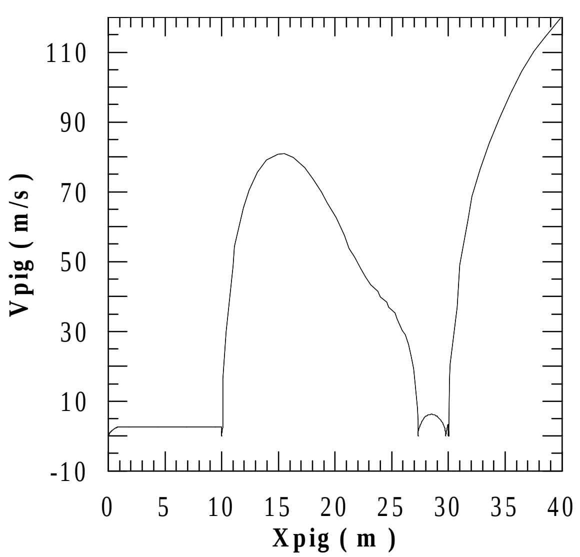


Figure 3 – Pig velocity versus pig position.

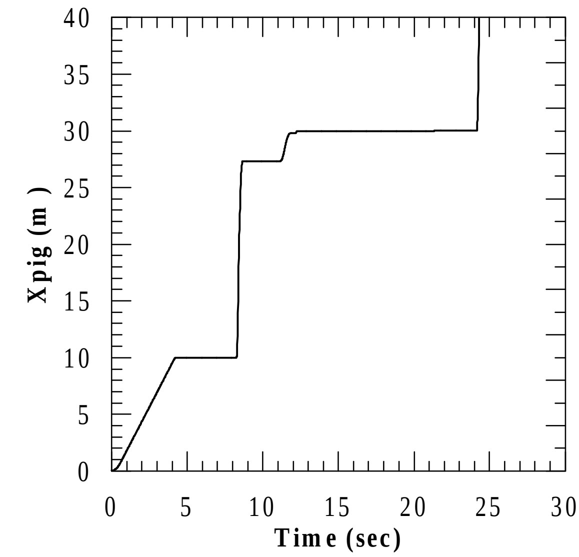
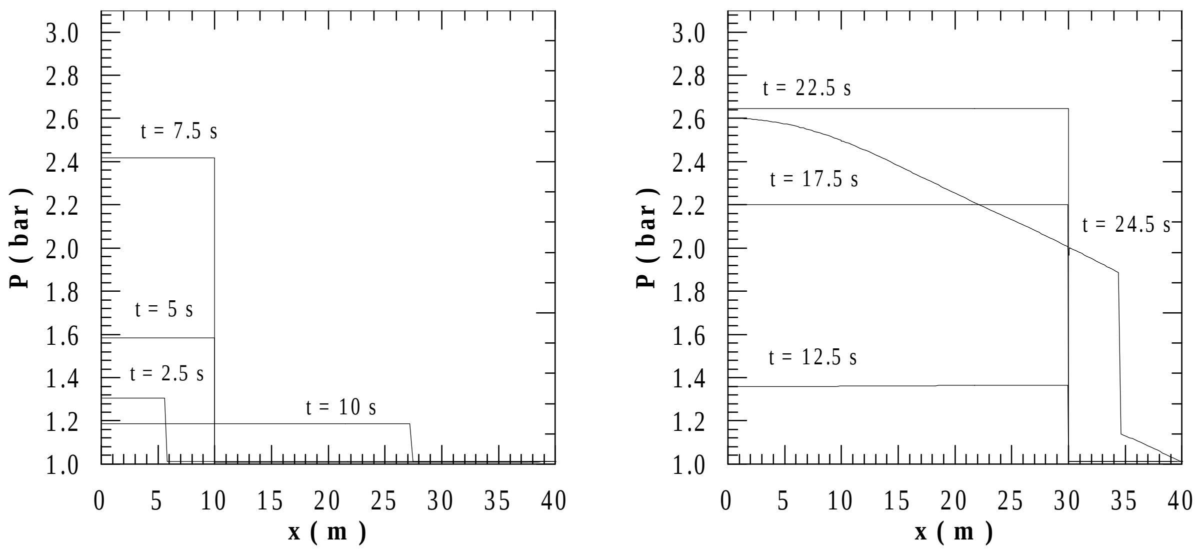


Figure 4 – Time variation of pig position



(a) (b)

Figure 5 – Pressure distribution along pipeline at different time steps. Pipeline with flanges.

By examining all these figures simultaneously, it can be seen that the pig, which is initially at rest, starts moving as air is injected into the pipeline. It rapidly reaches constant velocity, moving until it reaches the first flange. There, the pig stops moving, and pressure upstream of the pig starts to build up (4 s$\leq$ t $\leq$ 8s). After a few seconds, the pressure force across the pig reaches the value corresponding to the static contact force at the flange and the pig begins to move. As the pig enters the central section of the pipeline, there is a substantial drop in the contact force and the pig attains a very high velocity level, due to the high pressure difference across it, which was developed while the pig was trapped at the flange region. As the pig moves along the middle section of the pipeline, there is an expansion of the gas, resulting in a significant pressure drop. The pipeline/pig contact force slows the pig, which comes to a stop before reaching the second flange. The pressure builds up again and the pig is driven to the second flange where it is, once more, stopped by the higher contact force. A third pressure build up period is verified (12 s$\leq$ t $\leq$ 24s) until the pig overcomes the flange contact force. After released form the second flange, the pig reaches an even higher velocity level, because the volume of pressurized gas upstream of the pig is now higher than when the pig was released from the first flange. It should be noted that the velocity levels attained by the pig represent a severe risk of damage to the pig receiving facility or to the operating personal. As the pig moves along the last section of the pipeline, the upstream gas pressure is seen to decrease.

5.2 Pipeline with severe area changes

Figure 6 illustrates the pipeline for this case. The upstream and downstream portions of the pipeline have a diameter equal to 85 cm while the central part has a diameter of 86.5 cm. The roughness ε of the whole pipe is 1.8 x 10^{-3} mm. The wall thickness e is equal 2.54 mm at inlet and outlet pipe sections and equal to 2.5 mm at mid section. The pipe Young's modulus of elasticity, E is 2×10^{11} MPa and its Poisson's coefficient, μ is 0.3.

Nitrogen was injected through the pipeline at the constant temperature of 298 K. Its gas constant was set as R = 296.9 N m /(Kg K) and the compressibility factor was specified as z = 1.04. The absolute viscosity μ_f was kept constant and equal to 1.5×10^{-5} Kg/ (m s).

Initially the pressure distribution was uniform, equal to 1.5 bar, and there was no flow. Then the pressure at the inlet was raised to 7 bar in 20 seconds. At the outlet there was a valve connected to a tank with pressure p_{res} equal to 1.5 bar. The mass flow rate at the outlet is given by

$$\dot{m} = \rho\ (C_d A)_o\ \beta\ \sqrt{(p - p_{res})/\rho} \tag{8}$$

where $(C_d A)_o$ is the product of the discharge coefficient times the area of the valve totally

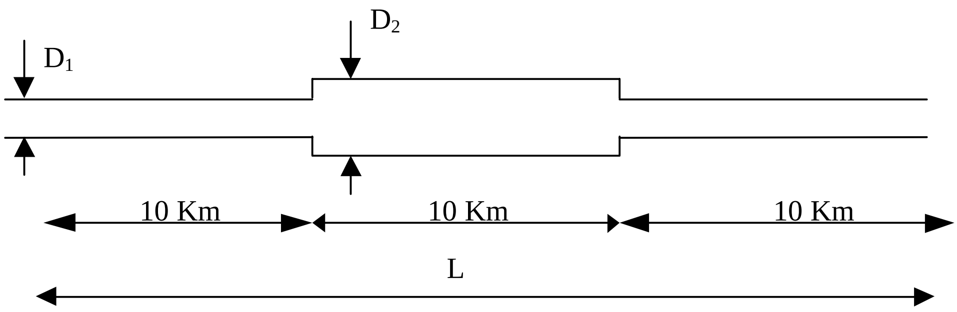


Figure 6 - Pipeline with severe area change

opened and β is the percentage of the total area of the valve that is opened. The following values were specified: $(C_d A)_o = 0.15$ m^2 and $\beta = 30\%$.

A pig with a mass equal to 100 Kg was introduced at the entrance of the pipeline with zero velocity at time equal to zero. The pig was considered symmetric, therefore the positive and negative dynamic contact forces were the same. The positive and negative static contact forces were also equal. At the inlet and outlet sections, the dynamic contact forces and the static forces were $F_{dyn}^{pos} = F_{dyn}^{neg} = 1.471 \times 10^5 N$ and $F_{stat}^{pos} = F_{sta}^{neg} = 1.724 \times 10^5 \; N$. At the central section, where the diameter was larger, the dynamic and static forces were $F_{dyn}^{pos} = F_{dyn}^{neg} = 3.971 \times 10^4 N$ and $F_{stat}^{pos} = F_{sta}^{neg} = 4.766 \times 10^4 \; N$. All these values were calculated following the procedures described in section 4 of this paper.

Figure 7 and 8 illustrate the pig velocity versus pig position and pig position versus time, respectively. It can be seen that the pig velocity shoots out at the beginning due to the pressure increase from 1.5 bar to 7 bar. After the inlet pressure stabilizes, the pig slows down as a result of the action of the contact force. When the pig reaches the central section, which is larger with a smaller contact forces, the pig accelerates attaining a very high velocity in very short time. Then, due to the contact force, the pig decelerates. The pig stops at the entrance of the outlet section of the pipeline, due to its smaller diameter, and higher dynamic and static forces. After that, the upstream pressures builds up, until the pressure force is enough to drive the pig into the smaller diameter pipe. The moves in this pipe section with an aproximately constant velocity.

Figure 9 illustrates the pressure distribution along the pipeline for different time instants. After 200 seconds, the pig is in the first section of the pipeline. It can be clearly seen the pressure drop across it. At time equal to 500 seconds, the pig is approximately at the center of the central section. Note that, since the dynamic force is much smaller, the pressure drop across the pig is also smaller. After 1000 seconds, the pig is moving along the outlet section. It can be seen that since the dynamic force increases, the pressure drop across the pig also increases. Note also, that the pressure variation is almost linear upstream of the pig. Finally, it can be seen that a few seconds after 1500 sec, the pig leaves the pipeline.

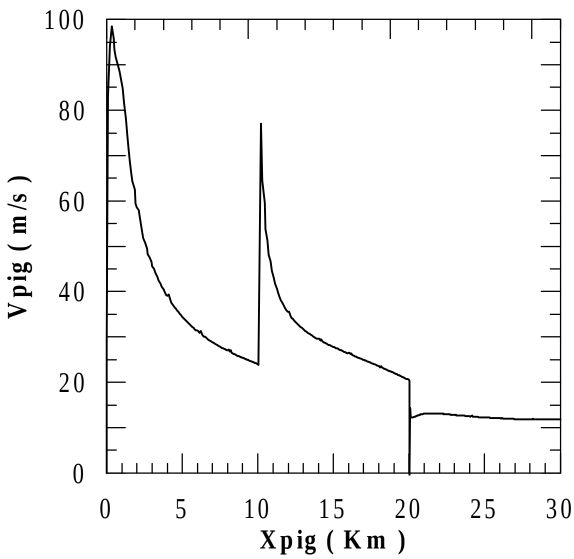


Figure 7 – Pig velocity versus pig position

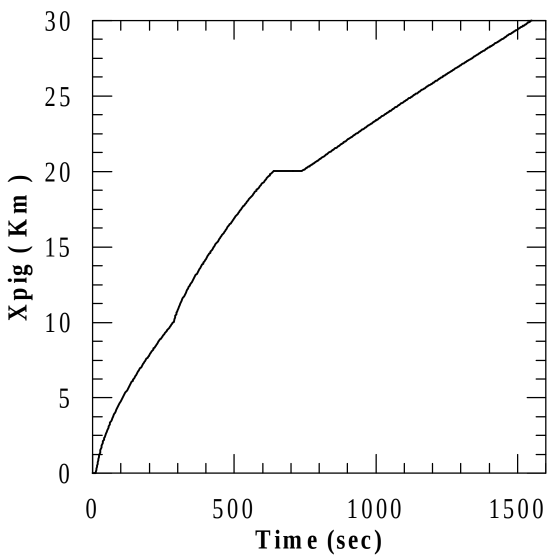


Figure 8 – Pig position with time.

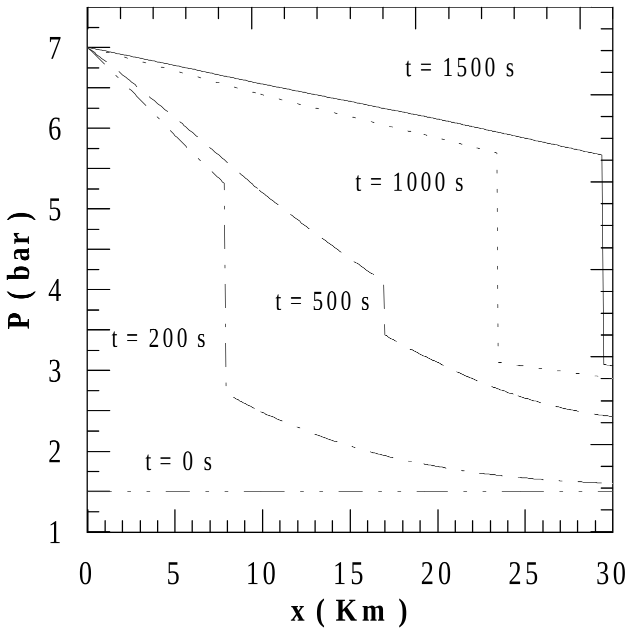


Figure 9 – Pressure distribution along pipeline at different time steps.

5. CONCLUSION

The present paper presented a simulation of the fluid flow and pig dynamics of pipeline with severe contact forces variations. The basic equations governing conservation of mass and linear momentum were numerically solved, coupled with an equation for the pig movement. The results obtained showed the influence of the contact forces on the pig dynamics and demonstrated the capability of the numerical method developed to handle severe changes in contact forces. High pig velocities are obtained when there is a significant drop in the contact forces. On the other hand, an increase in the contact forces brings the pig to a stop. The knowledge of the pig dynamics along pipelines can be an useful information, not only to improve its design, but to control pig operation procedures along pipelines.

Acknowledgment

The present research was jointly supported by the following companies: PETROBRAS from Brasil, STATOIL, NORSK HYDRO and SAGA from Norway, ANCAP from Uruguay, HALLIBURTON, EXXON, TEXACO AND PIPETRONIX from the USA and ECOPETROL from Colombia. The authors gratefully acknowledge the support awarded to the research.

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