

Hydro – Pneumatic Linear Position Controller

Sidney Nogueira Pereira de Jesus
University of São Paulo
sidneyjpn@ig.com.br

Linilson R. Padovese
University of São Paulo
Lrpadove@usp.br

Abstract. This work presents one study of a Linear Position Controller operated by Hydraulic and Pneumatic fluids, using two electrical conventional Directions Valves normally denominated type On – Off. In the input side the pneumatic pressure is intensified using a Booster obtaining high hydraulic pressure in the output side of the device. With two systems like this operating in opposite displacements, we can move one output shaft in a linear direction promoting the movement of the external part of the machine. The main objective is to control the position of the mechanical part using exactly the two Directional Valves On – Off operating by a PWM input signal and an analogue transducer for the feedback signal about the location of the mechanism being controlled. The methodology used was one model based in the state equation of the perfect gases in an isentropic process without the friction of internal forces, and considering fluids without viscosity with null leakage between the piston and the corresponding bore. The conclusion indicates a good performance about the time to accommodate an external load of 30.000 N after 0,30 seconds after the controller has been acioned. These results were obtained by numerical simulation in a computer using Simulink MatLab software.

Keywords. hydraulic controller; pneumatic controller; position controller; automation controller; controller on – off .

1. Introduction

Normally pneumatic cylinders (Andersen, 1967) have low force capability, high velocity but are very bad to stop in a certain position. On the other hand, hydraulic cylinders (McCloy, 1998) have high force capability, a good performance to stop at one precision position but works in low velocities.

This device called “ **Hydro-Pneumatic Linear Position Controller** “ combines the two systems above into one, which moves in high velocity and is capable to push or pull high loads applied against it. Using a big ratio between the frontal areas of the pneumatic and hydraulic pistons respectively, we have one system that receive air under low pressure and obtain, in the opposite side, oil submitted to a greater pressure value. This is the basic principle in which operate a Booster of pressure.

Looking the figure (1) below , we have a hybrid system in which on the down side, the compressed air flows (Andersen, 1967) into a pneumatic cylinder with a great frontal area, and on the top side the hydraulic oil is “pressed “ by using a small frontal area, and thus it will be in a high pressure intensity . The movement of the two pistons (pneumatic and hydraulic) are linear and simultaneous, because they are linked with one rigid shaft.

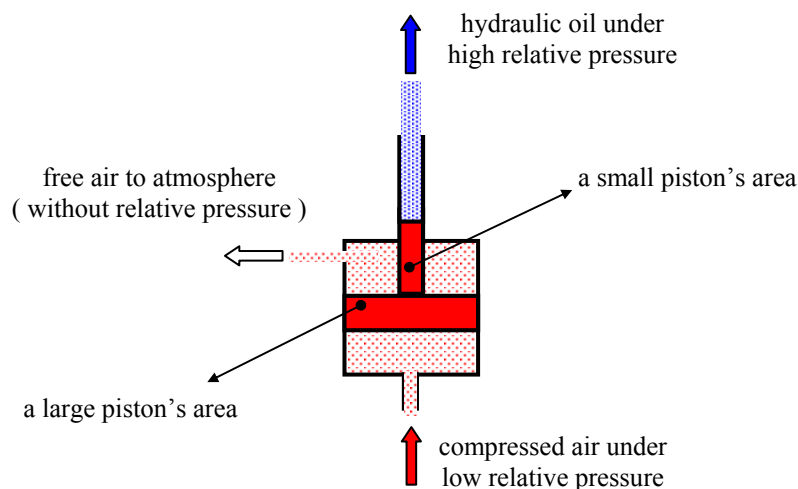


Figure 1: A sketch of an intensify of pressure (Booster)

The hydraulic piston can move from the left to the right positions with respect your initial point, respectively negative and positive measurements, we have two groups like one described above in the Fig. (1) working in a parallel direction but in opposite movements, thus when one moves to its forward direction, the other executes the same stroke in the inverse displacement. If the hydraulic oil (McCloy, 1998) is submitted to a high pressure to leave at a big ring area of one piston, we can produce enough force to move a piece or part of one machine, until this component find its exact position necessary for that purpose. The Figure (2) presents a longitudinal section in which we can see the names of all components which form this controller.

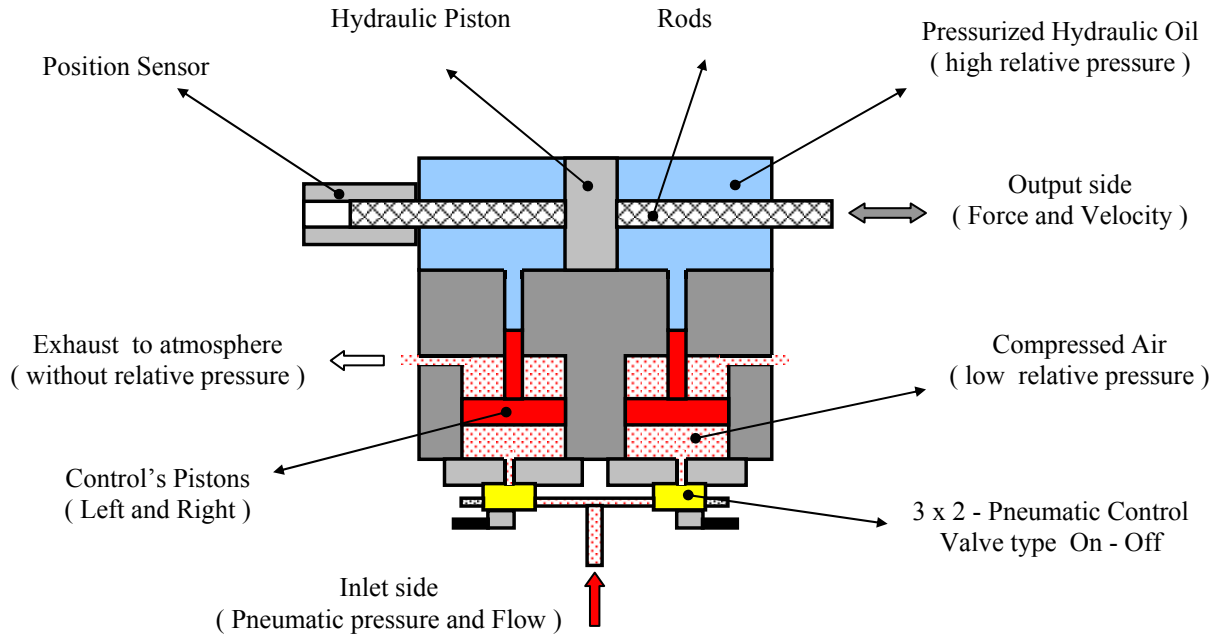


Figure 2 : A simple view of the main parts / components of the Hydro-Pneumatic Linear Position Controller.

For complementing the information about this Controller, the next Figure (3) shows the main parameters that will be used for the mathematical deductions, and thus to get a reasonable model which better reproduces the real behaviour of this device.

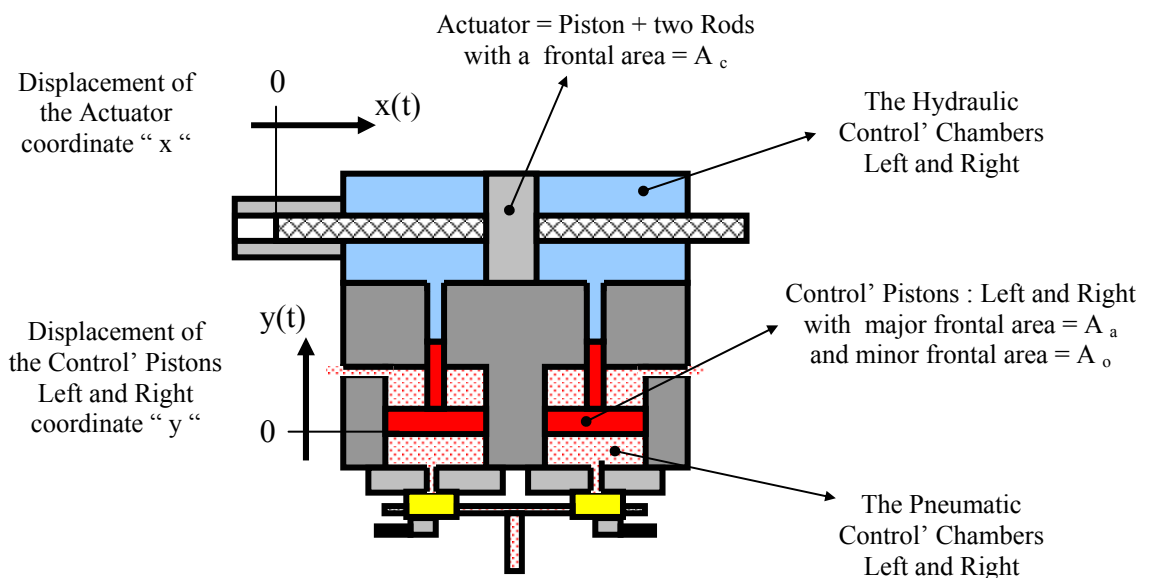


Figure 3 : Displacements of the internal / external parts of the Controller

2. Modeling the System

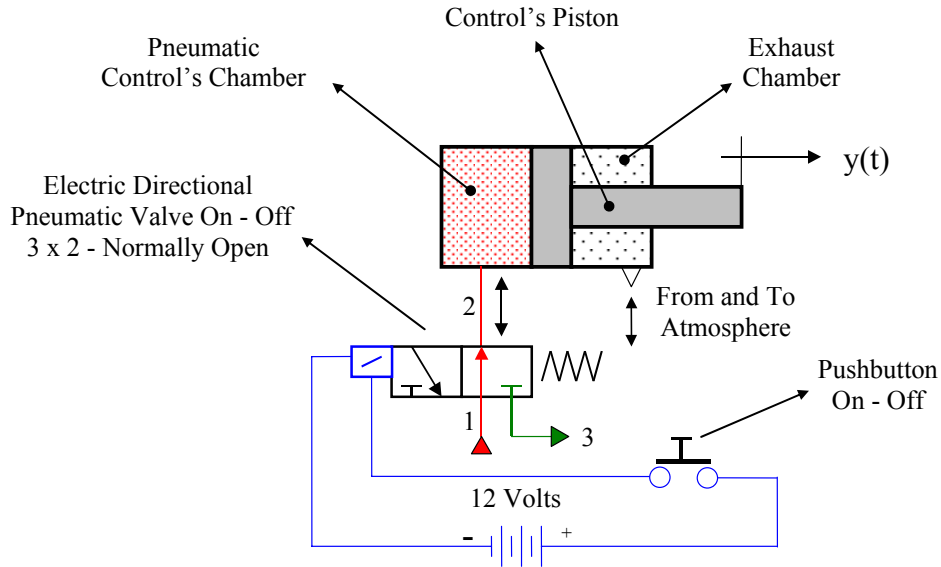


Figure 4 : Movement of the Control's Piston, by compressed air being injected into your chamber .

In the Figure (4), we represent the interactive action between the directional pneumatic control valve 3x2 – NC – On / Off type (Andersen, 1967), and the Control's Piston. It is very important to deduce one simple model that represents the evolution of the pneumatic pressure into this volume with respect to the time. This whole represents the basic principle of the operation of the Hydro-Pneumatic Linear Position Controller., because according to the intensity of the pressure into this volume at a generic instant, the Actuator of the Controller will have one capability to move loads coupled at the end of its extremity .

2.1 Model of the pressure evolution into the Control's Chambers

We start the modeling using the state's equation for the perfect gases (Tonyan, 1997) :

$$p = \rho.R.T \quad [N / m^2] \quad (1)$$

p = absolute pressure of air [N / m²];
 ρ = specific mass of air [Kg / m³];
 R = constant of air = 286,9 [J / Kg . K];
 T = absolute temperature of air [K];

For definition we know the follow relation :

$$\rho = \frac{m}{v} \quad [Kg / m^3] \quad (2)$$

m = mass of air [Kg];
 v = volume of air [m³];

Combining the Equations (1) and (2) and making the derivative of the pressure with respect to the time, we obtain the expression (Phillips & Harbor , 2000):

$$\frac{\dot{p}}{p} = \frac{\dot{m}}{m} - \frac{\dot{v}}{v} + \frac{\dot{T}}{T} \quad (3)$$

Assumption an isentropic's process (Munson et all, 1985) (adiabatic and without friction) when the air is compressed and / or depressed in the Control's Chambers, there is one relation valid to the perfect gases, according to the expression below

$$\frac{\dot{T}}{T} = \frac{k-1}{k} \cdot \frac{\dot{p}}{p} \quad (4)$$

Combining the Equations (3) and (4), after some algebraic artifices we get the expression (Andersen, 1967):

$$p_c(t) = \frac{p_0}{\rho_0^k} \cdot \left(\frac{m(t)}{vc(t)} \right)^k \quad (5)$$

$p_c(t)$ = the pressure in the Control's Chamber at instant 't' [N / m²] ;
 p_0 = the pressure in the Control's Chamber at instant t_0 , when the process start [N / m²] ;
 ρ_0 = the specific mass of the air under initial conditions [Kg / m³] ;
 $m(t)$ = liquid mass of air, into the Control's Chamber at instant 't' [Kg] ;
 $vc(t)$ = volume of the Control's Chamber at instant 't' after the begin of the process [m³] ;
 k = air's constant, equal 1,4 .

The Equation (5) indicates how the pressure into the Control's Chamber changes according to the increasing or decreasing of the liquid mass of air into this volume. In this expression is expressed also, what is the influence due to variation of the volume of the chamber, when the Control's Piston moves. However the liquid mass of air contained in the Control's Chamber at a generic instant, can be expressed by :

$$m(t) = m_e(t) - m_s(t) \quad [Kg] \quad (6)$$

$m_e(t)$ = amount of air that goes in the Control's Chamber, at instant 't' [Kg] ;
 $m_s(t)$ = amount of air that goes out the Control's Chamber, at instant 't' [Kg] ;

By the way, each term(s) of the second member of the Eq. (2), can be obtained by using the Bernoulli's equation (Munson et all, 1985) applied to one circular orifice to represent what is the mass's flow that pass through for it, which results are presented in these expressions :

$$m_e(t) = Cd_{oe} \cdot A_{oe} \cdot \sqrt{\frac{2 \cdot p_0}{(p_0)^{\frac{1}{k}}}} \cdot \int_0^t \left[\sqrt{p_c(t)^{\frac{1}{k}} \cdot [P_s - p_c(t)]} \right] \cdot dt \quad (7)$$

$$m_s(t) = Cd_{os} \cdot A_{os} \cdot \sqrt{\frac{2 \cdot p_0}{(p_0)^{\frac{1}{k}}}} \cdot \int_0^t \left[\sqrt{p_c(t)^{\frac{1}{k}} \cdot [P_s - p_c(t)]} \right] \cdot dt \quad (8)$$

$m_e(t)$ = mass of compressed air that goes into the Control's Chamber [kg] ;
 $m_s(t)$ = mass of compressed air that goes away the Control's Chamber [Kg] ;
 P_s = absolute pressure from the pneumatic source, considered constant during all the process [N / m²] ;
 $p_c(t)$ = absolute pressure into the Control's Chamber, at any instant 't' after the process has been started [N / m²] ;
 Cd_{oe} = discharge's coefficient of the orifice in which enter the compressed air form pneumatic source ;
 Cd_{os} = discharge's coefficient of the orifice in which the compressed air goes to atmosphere ;

Those orifices indicated above, represent one gicleur that exists on the inlet and outlet of the pneumatic directional control valves On-Off, which inject and exhaust air of the Control's Chamber to move the Control's Piston .

The internal volume $vc(t)$ of the Control's Chambers is variable during the functioning of the device, because when the actuator moves to its positive displacements $x(t)$, the left increases its value exactly the same quantity of the right decreases an vice-versa , thus this variable volume can be represented by de expressions below :

$$v_c(t) = VM + \frac{1}{2} \cdot A_a \cdot C + A_a \cdot y(t) \quad [m^3] \quad (9)$$

VM = residual volume of the Control's Chamber [m^3] ;
 C = total stroke of the Control's Piston [m] ;
 A_a = frontal large circular area of the Control's Piston [m^2] ;
 $y(t)$ = position of the Control's Piston at any instant of the process [m] ;

Using the Equations (5) to (9), we can realize one numerical computer simulation (Hanselman & Littlefield, 1999) of the evaluation of the pressure into the Control's Chamber with respect the time., during the mass of air flowing through it, as showed in the Fig. (5) below :

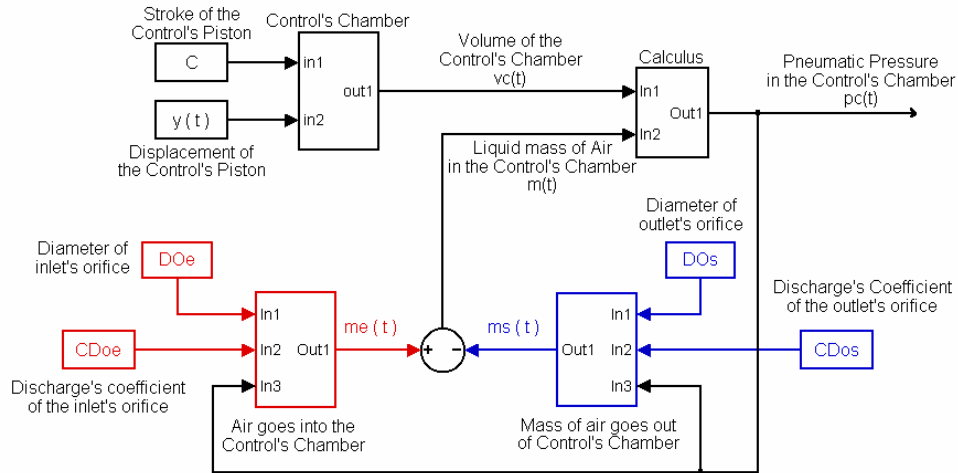


Figure 5 : Simulink – Model for the variations of the pressure into the Control's Chamber versus the time .

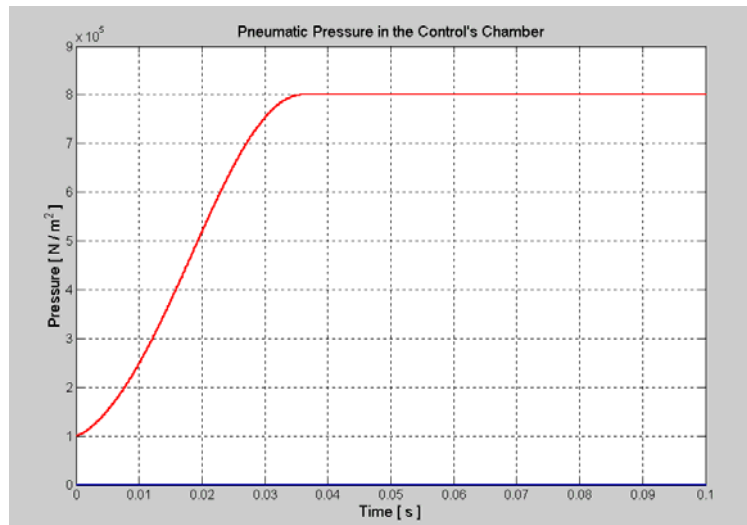


Figure 6 : Evaluation of the pressure into the Control's Chamber when compressed air is injected on it .

The Figure (6) was obtained with the Simulink – MatLab Software showed in Fig. (5) above, in which the follow assumptions and parameters were considered :

Inlet's Pressure : $P_s = 8.10^5$ N / m (absolute) ;
 Outlet's Pressure (atmosphere) : $P_a = 1.10^5$ N / m² (absolute) ;
 Initial temperature : $T_0 = 288$ K (absolute) ;
 specific mass of air : $\rho_0 = 1,225$ Kg / m³ ;
 altitude of test = level of oceanic ;
 Large diameter of Control's Piston : $D_a = 0.100$ m ;
 Diameter of inlet's orifice : $D_{oe} = 0,005$ m ;

Diameter of outlet's orifice : $D_{os} = 0,003 \text{ m}$;
 Death's volume of Control's Chamber : $VM = 78.5 \cdot 10^{-6} \text{ m}^3$;
 Fixed distance of the Control's Piston : $y(t) = 0.050 \text{ m}$;
 Total's volume of Control's Chamber : 0.0008639 m^3 ;
 Inlet's and Outlet's discharge's coefficient : $CDe = CD_s = 1$ (ideal) ;

2.2 Modeling the Hydro-Pneumatic Linear Position Controller

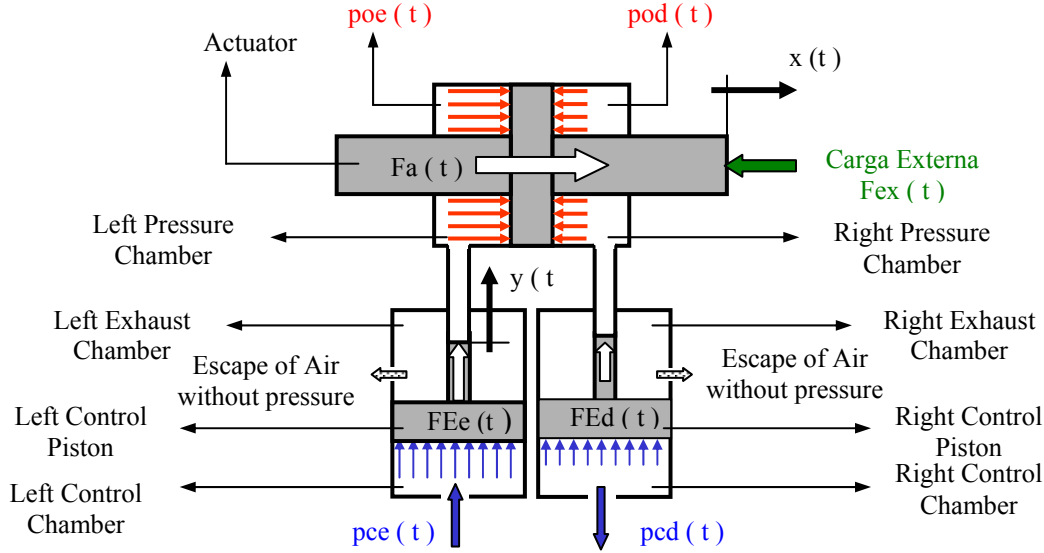


Figure 7 : Pressures and Forces actuating on the internal parts of the Hydro-Pneumatic Linear Position Controller

$poe(t)$ = pressure of the hydraulic oil, in the Left Pressure Chamber [N / m^2] ;
 $pod(t)$ = pressure of the hydraulic oil, in the Right Pressure Chamber [N / m^2] ;
 $x(t)$ = external displacement of the Actuator [m] ;
 $Fa(t)$ = force due to the difference between pressures $poe(t)$ and $pod(t)$ [N] ;
 $y(t)$ = internal displacement of the two Control Pistons (left and right) [m] ;
 $pce(t)$ = pressure of compressed air, in the Left Pressure Chamber [N / m^2] ;
 $pcd(t)$ = pressure of compressed air, in the Right Pressure Chamber [N / m^2] ;
 $FEE(t)$ = force actuating on the left Control Piston, due the pressure $pce(t)$ [N] ;
 $FED(t)$ = force actuating on the right Control Piston, due the pressure $pcd(t)$ [N] ;

Analyzing the Figure (7) above, we can view the pressures existing into each important chamber, and the forces produced due to action of the fluid being applied on the respective frontal areas of these components, which have relative movement during the functioning of the device in study.

Therefore this picture get us a better knowledge about the operational principle of the Hydro-Pneumatic Linear Position Controller, how we obtain the big resultant external force in the Actuator, starting with a low pressure of the compressed air flowing In and Out of its corresponding Control's Chamber : the Left and the Right .

Looking at the Figure (7), we can separate each mobile component, i. e. , the two Control's Piston (Left and Right) and the Actuator , and making the corresponding free diagram of forces for these elements, we obtain one system of equations, which after resolved leave us to get the follow expression :

$$A_c \cdot \frac{A_a}{A_o} \cdot [pce(t) - pcd(t)] - FEx(t) = Ma \cdot \ddot{x}(t) \quad (10)$$

A_c = ring frontal area of the Actuator; the same in both sides [m^2] ;
 A_a = large frontal area of the Control Pistons same to both, the left and right [m^2] ;
 A_o = small area of the Control Piston same to both, left and right [m^2] ;
 Ma = mass of the Actuator [Kg] ;
 $\ddot{x}(t)$ = the acceleration of the Actuator [m / s^2] ;

But the relation between $x(t)$ and $y(t)$, respectively the vertical displacement of the Control's Pistons and the horizontal displacement of the Actuator, can be expressed by the follow equation :

$$y(t) = \frac{A_a}{A_o} \cdot x(t) \quad [m] \quad (11)$$

Substituting Equation (6) into (5), and applying the fact that when Control's Chambers growth your volume the other reduce its at the same value, we obtain these equations for the Left and Right Control's Chambers, like showed below :

$$vce(t) = VM + \frac{1}{2} \cdot Aa \cdot C + Aa \cdot \frac{Ac}{Ao} \cdot x(t) \quad [m^3] \quad (12)$$

$$vcd(t) = VM + \frac{1}{2} \cdot Aa \cdot C - Aa \cdot \frac{Ac}{Ao} \cdot x(t) \quad [m^3] \quad (13)$$

$vce(t)$ = internal volume of the Left Control Chamber [m³] ;
 $vcd(t)$ = internal volume of the Right Control Chamber [m³] ;

Substituting in the Equation (10) the others (11), (12) and (13), we obtain the Mathematical Model of the Hydro-Pneumatic Linear Position Controller .

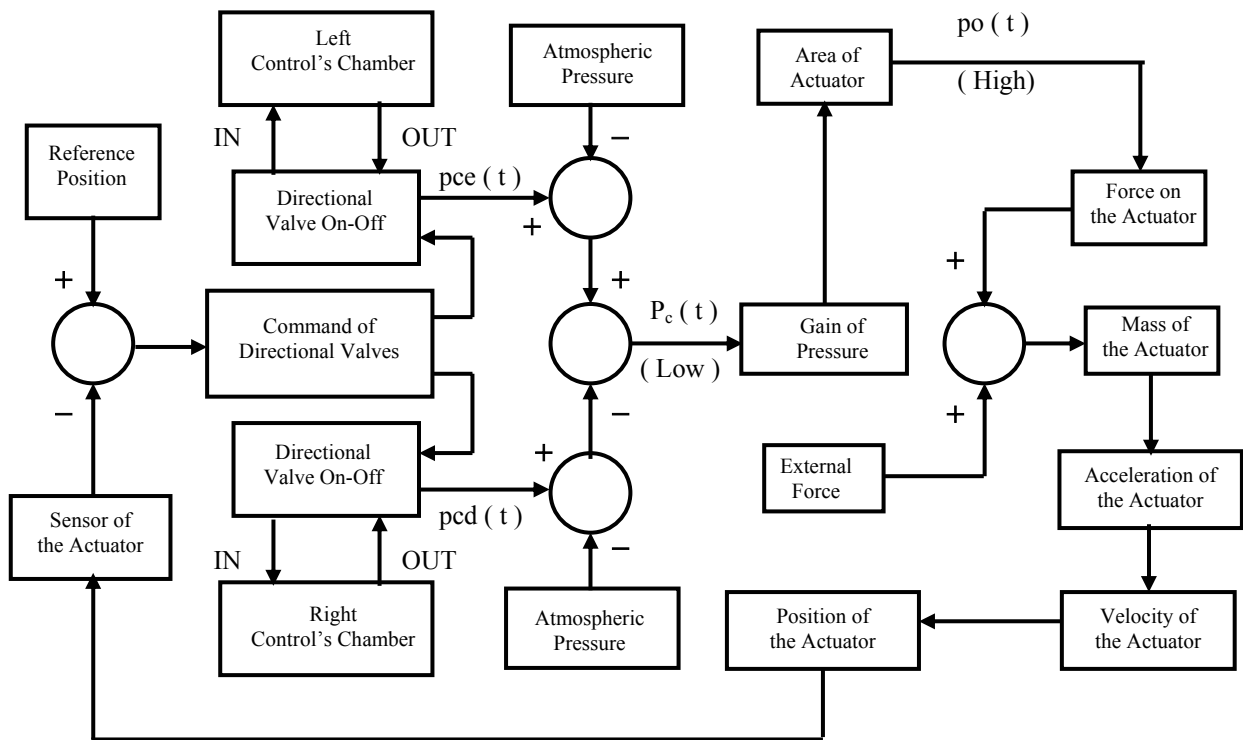


Figure 8 : Block's Diagram of the flow signal in the Hydro-Pneumatic Linear Position Controller .

The Figure (8) above illustrates the appearance of several components there exist in the Hydro-Pneumatic Linear Position Controller, and also the relationship among them.

We define a determined precision position necessary for the Actuator obtain, according to one specific application, using for this one potentiometer. This is the ' Reference Position ' in the diagram. The systems start and the movement of the Actuator is promoted by the simultaneous actions of the two pneumatic directional valves type On – Off (Left and Right) being correctly operated using one electronic controller Proportional and Integrative (Ogata, 1998; Distefano, et all, 2001) (PI) plus a Pulse Width Modulation Signal in high frequency (PWM) (Junior, 1997) . In these conditions each solenoid will be energized during some time letting that some quantity of mass of air goes into that the respective Control' Chamber, increasing the pressure into this volume, which produces enough force on the corresponding Control's Piston. This force distributed in the small circular area in the oil side, produces a big hydraulic pressure, which applied on the ring area of the Actuator results in one high axial force. When this force is greater than the mechanical load that exists actuating against the free extremity of the Actuator then, at this moment, it will be moved forward to its position in an accelerated motion.

Thus, while one Control's Piston has a positive displacement $y(t)$, the other functions exactly in the opposite distance / negative coordinate. According to the Actuator's mass it will have a determined acceleration, and its velocity increases.

When the position real $x(t)$ of the Actuator get the same value that was adjusted by reference, the controller PI maintain one output signal alternately on each solenoids On – Off, guaranteeing that position during a long time, independently of the variations in the external force.

At this moment, we will be able to use the same configuration / parameters that produced the Fig. (6) before, and by a numerical computing with the Simulink – MatLab , which reproduces graphically the Model of the Hydro-Pneumatic Linear Position Controller, mathematically represented by the Equations (10), (11), (12) and (13), and thus to get one “artificial history” about the behaviour of this hybrid controller, knowing the time response of it.

For this, some assumption were realized : the values for the $C_{d_{oe}}$ and $C_{d_{os}}$ coefficients are equal 1 (ideal flowing), forces provided by desalignment between mechanical parts or friction with others pieces, viscous forces from the particles of the fluids (Air and Hydraulic Oil) in contact with all internal components, any influence due to temperature variations during the process, and finally the compressibility of the hydraulic oil were ignored.

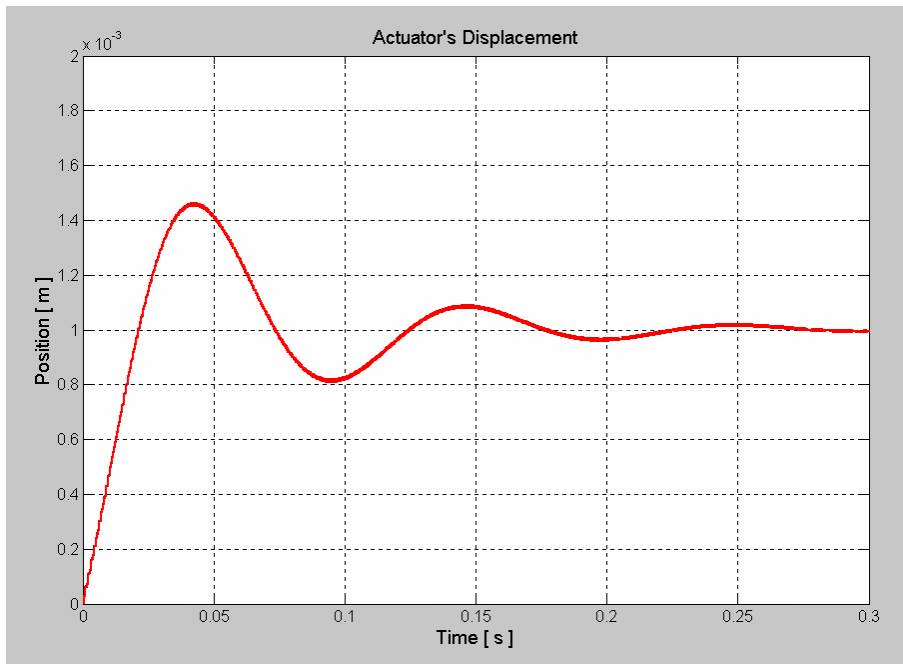


Figure 9 : Response of the hybrid Controller when fixed the position +1mm for the Actuator gets it .

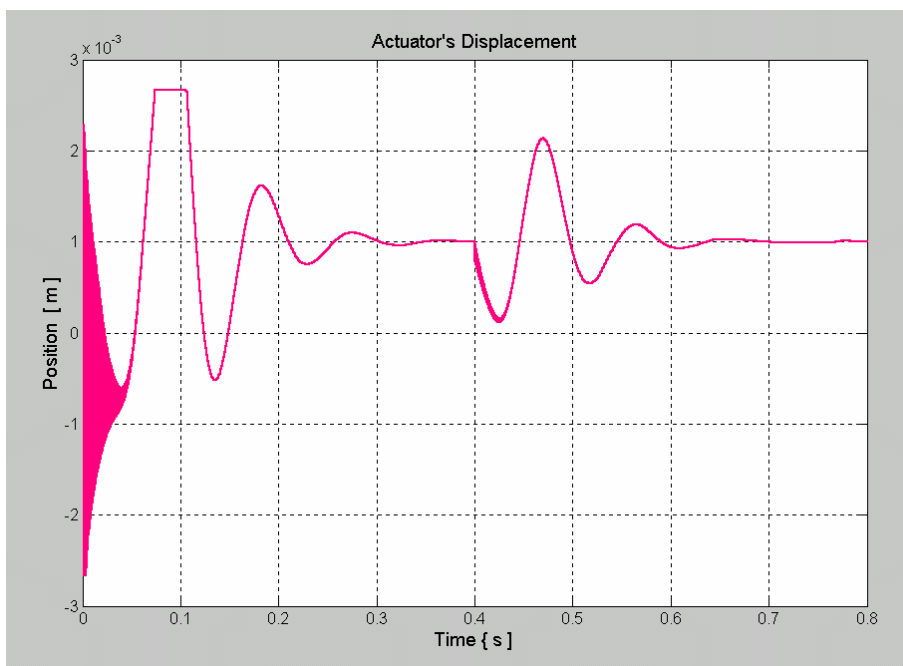


Figure 10 : Response of the hybrid Controller when submitted with one Step Load equals 30000 N after 0,4secs.

3. Conclusions

The main objective of this study is to develop a Model of one Hybrid Controller denominated ‘ Hydro-Pneumatic Linear Position Controller ’ which uses on the inlet side a low pneumatic pressure source, and produces on the outlet side a force high enough to move big loads fixed on the end of its free shaft, although limited to small displacements .

We can encounter various applications where this device could be applied, for example : controlling the position of one laminator’s cylinder, used by several industries like Siderurgicals , Foods, Beverages, etc .

In Figure (9) we can view the behaviour of this device, when it moves one constant load of 10000 N up to the Actuator obtains the distance of +1 mm with respect the original position, considered equal $x(t) = 0$ at the start of the process . This result shows one time approximately of 0,30 seconds for to have this positioning .

In another Figure (10) we simulated the same application of the Fig. (9), however in this case after 0,4 seconds was applied one step constant load of 30000 N. We observe that the Actuator has a lead time of 0,30 seconds approximately, for returning up to the coordinate $x(t) = +1\text{mm}$ after this action .

Thus, based on these results we think that the response time of this Controller was reasonable if compared to one real device, although were considered some ideal assumptions related above, during the mathematical modeling .

In these terms the initial purpose of this work was obtained, because combining two types of fluids (air and hydraulic oil) we get a good performance of positioning, a satisfactory accommodation time and a high capacity to move big loads, which comments were realized in that moment.

We agree that just using one prototype of this device, could have confirmed the real performance about this equipment, which we already know, that it will never substitute specially components like proportional (Gouveia, 1996) and /or servo directional valves, because this will be the better choice for one application that wants to have a precision position for some part of one machine, mainly when we have a large distance to move the load.

However, this suggestion opens a large field in which the movements occur with small strokes and big forces, and in this conditions the profile of this device represents a better choice than those conventional solutions (proportional and servo valves), mainly the aspect cost is an important parameter in discussion .

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