A Scheme for Compressed Air Saving in Pneumatic Positioning Systems for High Loads

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Abstract

A problem with pneumatic positioning systems is the low energetic efficiency. This paper presents a structure of a compressed air saving for pneumatic positioning systems with capacity to move high loads. This structure use the loading to save compressed air and thus to increase the energetic efficiency of these systems. This configuration adds a fast switching on/off valve in a feedback between the cylinder chambers. This type of valves is low cost, compared with a proportional valve, but the system can achieve the same performance. Experimental results show that with this type of configuration up to 58 % of compressed air saving can be achieved in comparison to classical pneumatic positioning system.

Keywords: Pneumatic system, fast switching valve, energetic efficiency, position control.

1 Introduction

Pneumatic servo positioning systems are very attractive for industrial applications because they have a good power/weight ratio and easy maintenance, when compared with electric positioning systems, and are low cost when compared to hydraulics. Pneumatic positioning systems can be used, for example, in the metal mechanical industry, in the medical area, in the brakes of heavy vehicles, in the agricultural area and others.

Pneumatic systems are considered as having a low energy-efficiency [1], and also present some control difficulties due to nonlinear characteristics of the system, such as air compressibility, nonlinear behavior mass flow rate through the valve orifice [2] and dead zone [3], besides the friction in the seals of the cylinder [4]. Table 1 presents some advantages and disadvantages between the positioning systems mentioned above.

As mentioned before, one of the major problems in the pneumatics systems is the low energetic efficiency, because the compressed air is exhausted to atmosphere after use. Several authors deals with energy saving or consumption of compressed air in pneumatic systems, what can be seen in [5], [6], [7], [8] and [9].

Specifically, [5] uses an auxiliary on/off valve in a feedback between the chambers of an asymmetrical cylinder. Figure 1 shows this scheme. When the piston moves from the left to the right with constant speed, the pressure p_B in chamber B is greater than the pressure p_A in chamber A. If the auxiliary valve is opened in this case, some exhaust air of p_B will be reused. To guarantee that the system is working in case $p_A > p_B$, a check valve is placed in this path. When the piston moves from the right to the left, the by-pass valve is closed. A PID controller to control the proportional valve in closed-loop is used. The experimental results show that for 10 work cycles 12 % of compressed air is saved using this system strategy

Table 1: Advantages and disadvantages of positioning systems.

Positioning system	Advantages	Disadvantages
Pneumatic	Easy maintenance, clean, high confiability, does not generate sparks, low cost.	Very low energetic efficiency, difficulties in control, separate power supply, low stiffness.
Hydraulic	High force density, accumulators for peaks load, easy control, keep high forces, stiffness, low velocity, does not generate sparks.	Hard installation of power supply, noisy, leakage, fire risk, high cost.
Electric	Easy maintenance, flexibility, easy control, clean.	Low force density, problems with water, sparks, high cost.

In [6], a similar configuration as the one in fig. 1 is presen-

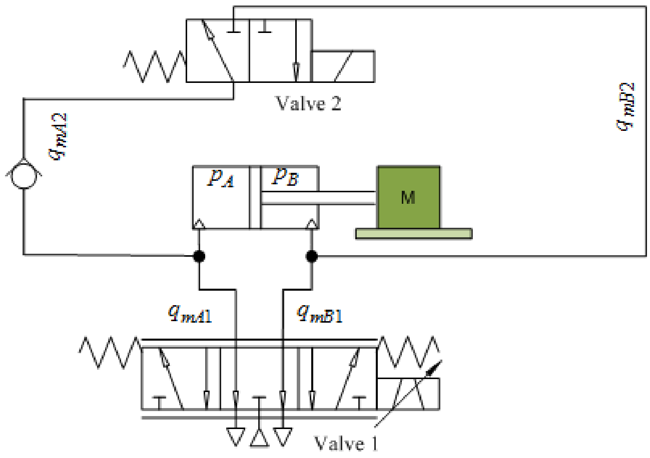

Figure 1: Pneumatic scheme with auxiliary valve (Adapted from [5]).

ted, but in this case the feedback between the chambers of an asymmetrical cylinder is carried out by one proportional valve 2/2 positions (auxiliary valve). An energy saving controller was developed to save compressed air. When the pressure in the depressurized chamber is greater than the pressure in the pressurized chamber, the required mass flow rate can be supplemented via the auxiliary valve. The proposed controller was firstly developed by calculating the control effort required for tracking, and, secondly, by calculating the extent to which the feedback can contribute to the differential mass flow rate. The sliding mode control was utilized to control the two valves. Experimental results present reduced energy consumption by 25-52 % to different desired sinusoidal trajectories and for a step input.

Another concept is proposed in [7]. A typical pneumatic system with a 5/3 proportional valve is replaced by one using two 3/2 proportional valves, such that the mass flow rate on the cylinder chambers can be controlled separately. The idea was to develop two control modes to modified system: an active mode, that uses only compressed air from supply, and a passive mode, that uses only the control of the mass flow rate from the exhaust of the cylinder chamber. Experimental results indicate an energy saving from 10 % to 46 % depending on the desired tracking frequency.

In a similar way, [8] shows a system with two 3/2 proportional valves, using a nonlinear multivariable control law developed for tracking position and energy saving. The system configuration allows to control two different trajectories, position and to control pressure. The control strategy is related to tracking the desired position with minimum energy consumption. The results presented show an improvement of 29 % in terms of energy delivered inside the cylinder with this multivariable nonlinear control.

Finally, [9] shows an analysis not only based on the compressed air consumption of pneumatic systems but also on the balancing of energies. This strategy enables the continuous balancing of all fractions of energy and energy losses. After balancing, as a first step to energy saving, the parameter of optimization is selected and, secondly, the implementation of exhaust-air recovery system is suggested. An exhaust-air recovery system is made by means of installing two additional directional valves as well as a compressed-air reservoir. Thus, the utilization of the exhaust air of a cylinder chamber is possible when the piston moves from the right to the left. The experimental results show energy savings around 23 % in comparison with the energy consumption of the standard and modified system.

The aim of this paper is the study of a structure for pneumatic positioning systems for compressed air saving in systems which need move high loads. This structure consists of adding a fast switching on/off valve in a feedback between the cylinder chambers. These types of valves are low cost, when compared to proportional valves, but the same system performance can be achieved. This scheme will be presented in next section.

This paper is organized as follows. In Section 2 the pneumatic positioning system with and without an auxiliary valve is described. Section 3 is dedicated to the presentation of the technique for compressed air saving and the model for the estimate of the mass of compressed air consume. The control design is shown in Section 4. In Section 5 the results and discussions are presented. The main conclusions are presented in the last section.

2 Description of the pneumatic positioning system

The proposed scheme was implemented in one experimental setup to verify the air compressed saving. This experimental setup was used for tests whose purpose is to control the position of the turbine blades of speed governors of small hydroelectric power plants [10] - [11]. Figure 2 shows a diagram of the typical pneumatic positioning system.

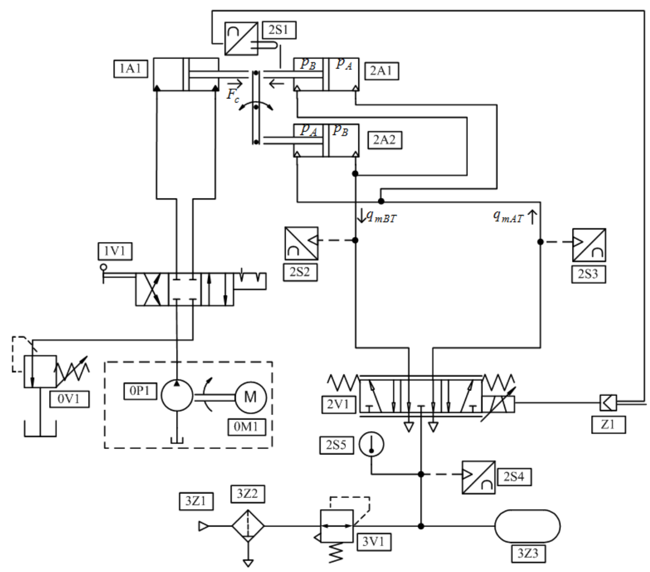

Figure 2: Diagram of the typical test setup (Adapted from [10]).

The components of the typical pneumatic positioning system consist of one 5/3 proportional valve (2V1) for control of mass flow rate, two pneumatic asymmetric cylinders (2A1 and 2A2) in a symmetric configuration, position transducer (2S1), three pressure transducers (2S2, 2S3 and 2S4), thermocouple (2S5), filter (3Z2), compressed air source (3Z1), air reservoir (3Z3), pressure reducing valve (3V1) and valve controller (Z1). The loading system is composed of an asymmetric hydraulic cylinder (1A1), a 4/3 directional hydraulic valve (1V1), a hydraulic power unit (0P1 and 0M1) and a hydraulic pressure reducing valve (0V1).

In this specific study a fast switching on/off valve was add to reuse the compressed air. Figure 3 shows a diagram of the pneumatic positioning system with feedback between chambers with an auxiliary 3/2 fast switching on/off valve configured as a 2/2 valve. In both figures the arrows indicate the positive direction.

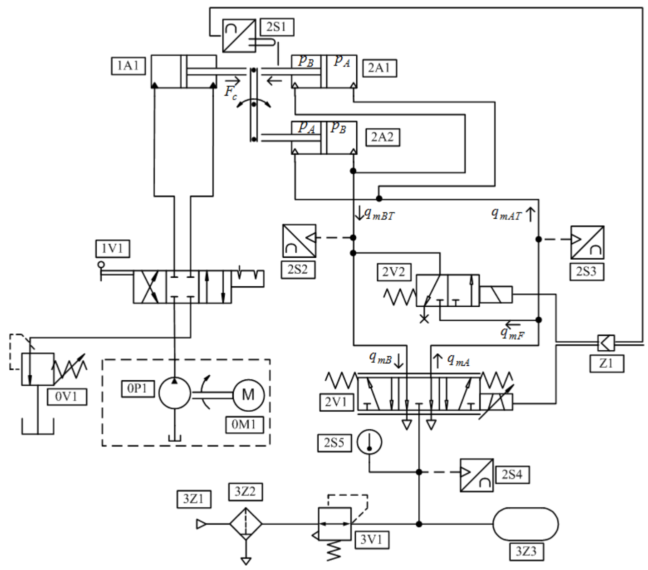

Figure 3: Diagram of the test setup with feedback between the chambers of the cylinder.

The compressed air is supplied to the proportional valve through a pressure reducing valve. The proportional valve spool opens the supplying orifice of compressed air to one chamber of the cylinder and allows the air to be released to the atmosphere. After the variation of the pressure in the cylinder chamber happens, it results in a force that moves the cylinder rod in positive or negative way, depending on the input signal. When the system has a positive load (F_c) and is moving in the opposite direction, compressed air can only be saved using the auxiliary valve, as well for as when the system as a negative load.

3 Compressed Air Saving

As previously described, with a fast switching on/off valve in a feedback between the cylinder chambers, compressed air can be reused. To guarantee the system performance and also provide operating conditions for the system an algorithm was proposed, which is shown in the subsection 3.1. To estimate the mass of air consumption an equation was developed. This equation is presented in subsection 3.2.

3.1 Technique for compressed air saving

The loading system imposes a difference in the pressure in the cylinder chambers. This difference will be used for the operation of the auxiliary valve. Three cases of operation will be considered as shown below:

$$\begin{array}{l}
Case\ \ 1: \\
If\ \ \Delta p > \alpha \\
When\ \ U > 0\ \ 2V1\ \ operating\ \ and\ \ 2V2\ \ closed \\
When\ \ U \leq 0\ \ 2V2\ \ operating\ \ and\ \ 2V1\ \ closed \\
\\
Case\ \ 2: \\
If\ \ -\alpha \leq \Delta p \leq \alpha \\
2V1\ \ operating\ \ and\ \ 2V2\ \ closed \\
\\
Case\ \ 3: \\
If\ \ \Delta p < -\alpha \\
When\ \ U > 0\ \ 2V2\ \ operating\ \ and\ \ 2V1\ \ closed \\
When\ \ U \leq 0\ \ 2V1\ \ operating\ \ and\ \ 2V2\ \ closed
\end{array}$$

where U is the control signal, $2V1$ and $2V2$ are the proportional valve and the auxiliary valve, respectively, Δp is the difference between the pressure in the cylinder chambers, and α is the tolerance of pressure difference, that is the pressure difference required to insure the feedback of the compressed air between the cylinder chambers and also to achieve the performance requirements.

3.2 Mass of compressed air consumed

The mass of compressed air consumed (M_P) can be calculated by the integral of the mass flow rate q_m into the cylinder chamber from the start to stop of piston movement. This is described in the following equation:

$$M_P = \int_{t_i}^{t_f} q_m\, dt. \tag{1}$$

In the pneumatic servo positioning in study, the pneumatic mass of compressed air consumed should be calculated by consumption of total mass flow rate through the supply. When there is flow from S to A or from S to B, there is compressed air consumption. In this way, the mass consumption of pneumatic positioning system can be computed by eq. (2):

$$M_P = \int_{t_i}^{t_f} (q_{mA+} + |q_{mB-}|) dt. \tag{2}$$

In the eq. (2), q_{mA+} and q_{mB-} can be computed by using the following equation, whose parameters are obtained according to ISO 6358 [12]:

$$q_{mA+} = K_u U_{czm} C p_S \rho_0 \omega(\frac{p_A}{p_S}) \qquad if \quad U \geq 0 \tag{3}$$

$$q_{mB-} = K_u U_{czm} C p_S \rho_0 \omega(\frac{p_B}{p_S}) \qquad if \quad U < 0 \tag{4}$$

Replacing qm_{A+} and qm_{B-} in the eq. (2) gives

$$M_P = K_u C \rho_0 p_S \int_{t_i}^{t_f} (U_{czm}\omega(\frac{p_A}{p_S}) + |U_{czm}|\omega(\frac{p_B}{p_S}))dt. \quad (5)$$

The used parameter values are the following constants: $C = 16.5 \times 10^{-9}$ m^5/Ns, $b = 0.12$, $\rho_o = 1.205$ kg/m^3 and $p_S = 8 \times 10^5$ Pa. This equation will be used to calculate the mass of compressed air consumed in pneumatic positioning systems.

4 Control Design

The control strategy that will be used in this system consists in using PI controller. However, since there are two different valves, the control strategy for the proportional valve and the fast switching on/off valve are different. In [13] the authors describe a control scheme with a PWM (pulse width modulation) technique associated with a PI control. Thereby, the gains of the controller do not need to be modified, since that the two PID controllers differentiate only by PWM. The PI controller for both valves will be presented in the next subsection.

4.1 PI Controller

The PI controller is one of the most used controllers in the industry. The PI controller can be seen in [14]. It is a simple method for implementation and the control law is given by:

$$U(t) = k_P e(t) + k_I \int_{t_o}^{t} e(t)dt, \quad (6)$$

where $e(t)$ is the difference between the desired input and the actual output, that is the system error, k_P is the proportional gain to error and k_I is the gain proportional to integrative part of the system error. In general, the proportional gain is used to improve the time performance and the integrative term is responsible for reducing the steady state errors.

This control signal was already used in the subsection 3.1, to give the conditions for compressed air saving in the pneumatic positioning system. At this point, the control signal will be split into two signals, U_V1 and U_V2. One will be used to compose the signal sent to the proportional valve and the other to compose the signal sent to the auxiliary valve. In the signal sent to the proportional valve, a dead zone compensation will be used and the signal sent to fast switching on/off valve shall be applied the PWM technique associated with PI control. In the following subsection the dead zone will be identified and compensated by the methodology employed in [3].

4.2 Dead zone

The dead zone in the proportional valve is an important nonlinearity which should be considered. This nonlinearity happens when the spool width is greater than the orifice valve width. The dead zone presence in a pneumatic positioning system is among the factors that limit the performance of feedback control loops.

Dead zone is a static input-output relationship, that for a range of input values gives no output, where U is the input and U_{zm} is the output. In general, the right (zmd) and left (zme) limits, and the slopes (md and me) are not equal. Figure 4 shows a graphical representation of the dead zone [3].

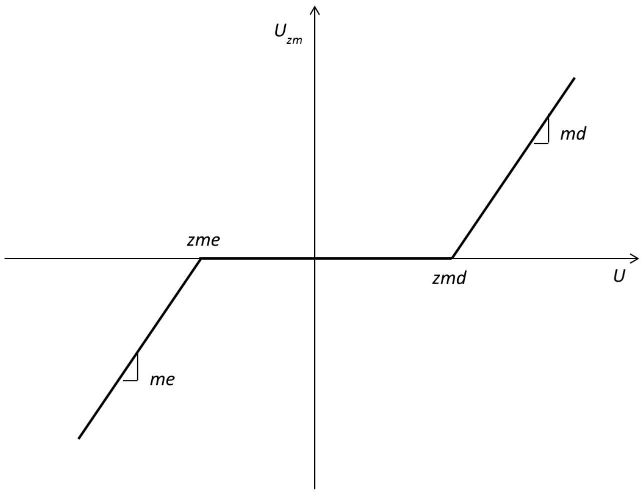

Figure 4: Graphical representation of the dead zone (Adapted from [3]).

The dead zone analytical expression is given by:

$$U_{zm}(t) = \begin{cases} md(U(t) - zmd) & if \quad U(t) \geq zmd \\ 0 \quad if \quad zme < U(t) < zmd \\ me(U(t) - zme) & if \quad U(t) \leq zme \end{cases} . \quad (7)$$

.

The dead zone compensation is carried out through an inverse function. If the inverse is exact and the parameters are known (zmd, zme, md and me) the cancelation of the dead zone would be perfect. Therefore, it is necessary to use the smoothed inverse to avoid the discontinuity as near to the zero position and an abrupt switching between zme and zmd.

Figure 5 shows the graphical representation of the dead zone compensation with smoothing near to zero position,

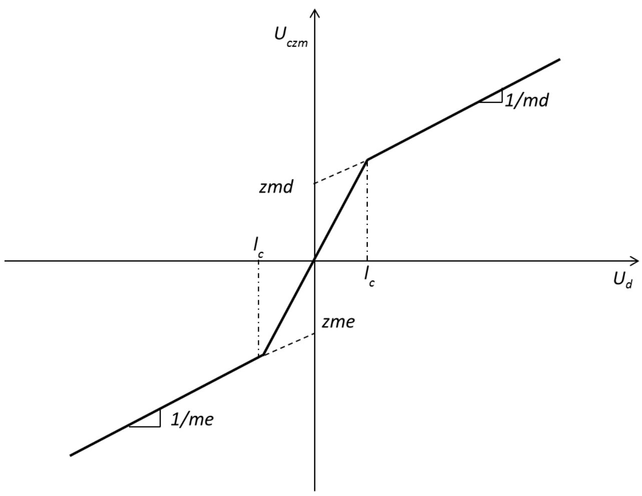

Figure 5: Graphical representation of the dead zone compensation (Adapted from [3]).

where U_d is the desired signal control in the default of the dead zone, U_{czm} is the compensated output signal and l_c is the smoothness width used in compensation. The inverse function used to compensate the dead zone is described by

$$U_{czm} = \begin{cases} \frac{U_d(t)}{md} + zmd & se \;\; U_d(t) \geq lc \\ \frac{U_d(t)}{me} + |zme| & se \;\; U_d(t) \leq -|lc| \\ \left(\frac{zmd + lc/md}{lc}\right) U_d(t) & se \;\; 0 \leq U_d(t) < lc \\ \left(\frac{|zme| + |lc|/me}{|lc|}\right) U_d(t) & se \;\; -|lc| \leq U_d(t) < 0 \end{cases} \tag{8}$$

To identify the dead zone, an open loop test of actuator system (proportional valve and pneumatic actuator) is proposed with a slow sine control signal with 10 volts amplitude and 50 seconds period according to eq. (9). This methodology was presented in [3].

$$u(t) = -10\cos\left(\frac{2\pi}{50}t\right), \tag{9}$$

Experimental tests consist of the acquisition of the pressure and control signal of the sine wave given by eq. (9). The graphical control signal as a function of the pressure is presented in 6. At this point, the pressures variation as a function of control signal are analyzed.

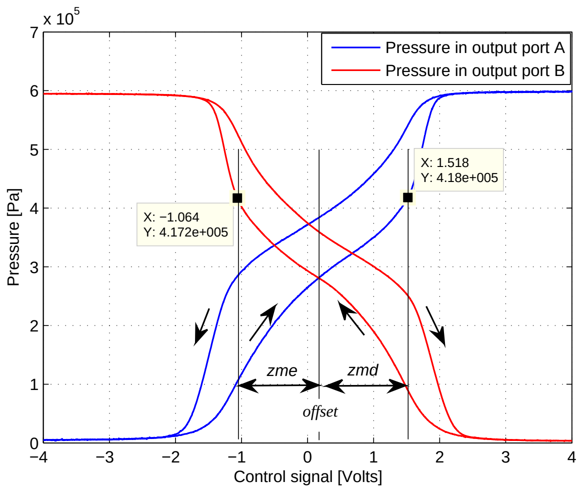

Figure 6: Graphical representation of the dead zone values.

According to fig. 6, for the pressures A and B, when the control signal exceeds the right or left limits of dead zone a pressure change sudden occurs. The offset identification of the proportional valve is through the determination of the midpoint of the pressure variation due to internal leaks. The values achieved from fig. 6 were the right dead zone $zmd = 1.064$ V, the left dead zone $zme = 1.518$ V and the offset was considered $offset = 0$ V because it is very near the center. The values of $md = me = 1$ and $l_c = 0.4$ were used. The l_c value represents a trade-off between control signal quality and effetive dead zone compensation. According to [3], if l_c is large, dead zone compensation is poor. If l_c is too small, oscillations in the control signal can occur near of origin.

4.3 PWM associated with PI control

This previously mentioned control technique can be applied directly to the proportional valve, but to apply to an on/off valve we should associate the PWM technique. The PWM technique consists in the use of a particular frequency switching in the directional control on/off valve and the duty cycle control. The duty cycle is the time that the signal control is "on" in relation to total cycle time [15]. Figure 7 describes the duty cycle through two examples: a system with a duty cycle of 50% and another with 20%, both with a period of 10 s.

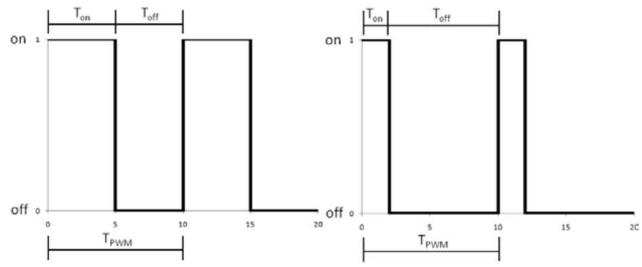

Figure 7: PWM Signal [15].

The control signal from PI control is changed for the duty cycle of the valve according eq. (10)

$$\begin{cases} if \;\; \Delta p > \;\; \alpha \;\; and \;\; U \leq 0 \;\; then \;\; U_{dc} = |U\_V2|/3 \\ if \;\; \Delta p < -\alpha \;\; and \;\; U > 0 \;\; then \;\; U_{dc} = |U\_V2|/3 \end{cases} \tag{10}$$

Figure 8 shows the implementation of the PWM in Matlab/Simulink.

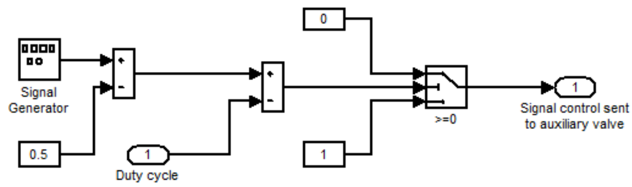

Figure 8: PWM implementation in Simulink.

There is a pulse generator that generates a sawtooth signal with amplitude of -0.5 to 0.5 in a specific frequency (PWM frequency, see fig. 9(a)). This signal is summed with a value of 0.5. This results in a signal with a value ranging from 0 to 1 that is subtracted from the duty cycle (see fig. 9(b)). From the switching block there is a signal sent to the valve (see fig. 9(c)), where 1 is valve "on" and were 0 is valve "off" [15].

5 Results and Discussions

The experiments were based in a test bench of a servo-pneumatic system whose purpose is to control the position

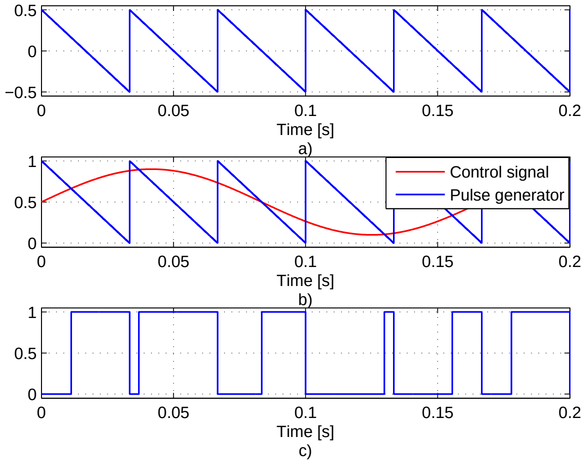

Figure 9: PWM signal generation. a) Pulse generation. b) Pulse and duty cycle. c) Signal sent to the valve (Adapted from [15]).

of the turbine blades used in hydroelectric power plants (see fig. 10). The diagrams of the setup with and without feedback between the chambers of the cylinder were already shown in Chapter 2. Table 2 presented the key components of the test bench.

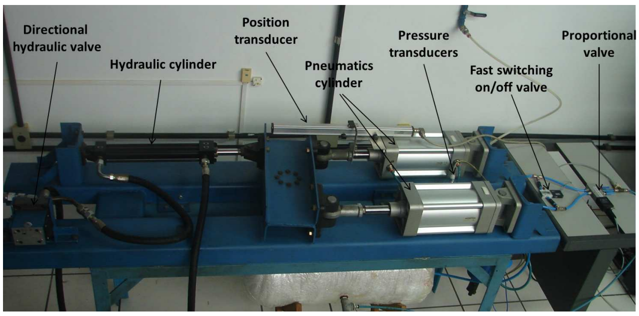

Figure 10: Test bench.

Results for a sinusoidal trajectory with amplitude of 8 mm at a frequency of 0.05 Hz and step inputs of 16 mm and 8 mm will be presented. These trajectories are used in the position control of turbine blades. A dSPACE system was used for data acquisition and control, working together with Simulink/Matlab and ControlDesk software. For the experiments, an integration step of 1×10^{-3} sec and PI controller gains Kp=850 and Ki=40 were used. A period of 0.033 sec for the PWM signal was used. The experiments were obtained with a positive load around 5000 N.

First-order low pass digital filters were used, due to the considerable electromagnetic noise in the position and pressure signals. For the position signal a cut-off frequency of 50 Hz was used and for the pressure signals a 250 Hz were used. The simulink model used in the experimental tests is presented in Appendix A.

Table 2: Main components of test bench.

Component	Maker	Code	Specification
Proportional valve	Festo	MPYE 5-1/8	5/3, 350 l/m.
Fast switching on/off valve	Festo	MHE3-MS1H-3/2G-QS-6	3/2, 200 l/m.
Pneumatics cylinder	Dover	CNGPS125D-B160	Stroke=160 mm, Diameter=125 mm
Hydraulic cylinder	Parker	38.1CBB2HL U29AC-0300	Stroke=300 mm, Diameter=38.1 mm
Directional hydraulic valve	Rexroth	4WMM10 E 10/F	4/3
Pressure transducers	HBM	HDM P8AP	0 - 10 Bar
Position transducer	Balluff	BTL5-A11-M0400-P-S 32	Stroke=400 mm

Figure 11 presents the comparison between the displacement (fig 11(a)) and the error (fig. 11(b)) with and without an auxiliary valve for sinusoidal input. Both responses demonstrate a good trajectory tracking, but both systems present small steps over the sinusoidal trajectory. These small steps are mainly caused by the static friction at the actuator and there are also influences of the flow control valve.

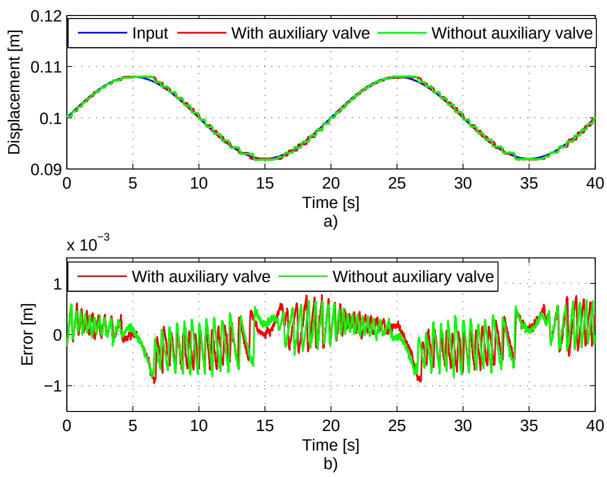

Figure 11: Sinusoidal trajectory tracking (a)) and tracking error (b)).

Both trajectory tracking errors are between ±0.9 mm. Both trajectories, with and without auxiliary valve, did not present significant deviation. The pressures in chambers A and B for the system with and without an auxiliary valve are presented in fig. 12.

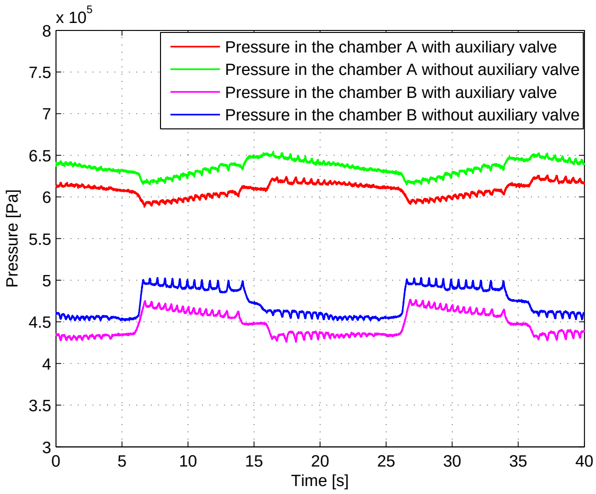

Figure 12: Pressures p_A and p_B for sinusoidal trajectory.

The pressures into the cylinder chambers are relatively lower in the system with an auxiliary valve. This occurs is because when the system has feedback, the supply is closed and the pressures have a slight decrease.

When the pneumatic positioning system is moving in the positive direction, the auxiliary valve is activated and the proportional valve is closed, and thus there is not any consumption of compressed air. Figure 13 shows the mass of compressed air consumed by the system with and without an auxiliary valve related to the sinusoidal input shown in fig. 11(a). In this case, the compressed air saving is around 58 % for two work cycles.

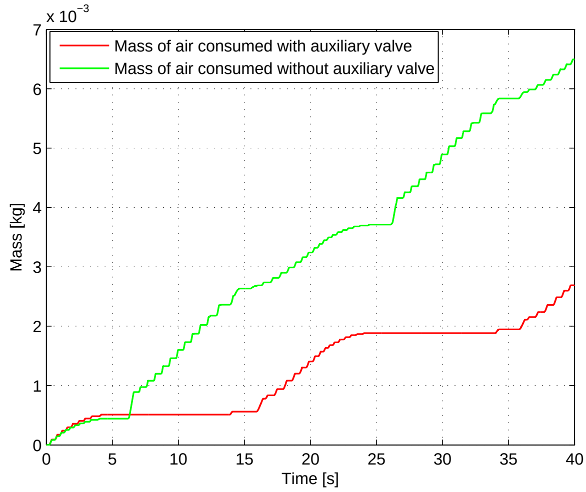

Figure 13: Mass of compressed air consumed for sinusoidal trajectory.

Figure 14 presents the comparison between the displacement (fig. 14(a)) and the error (fig. 14(b)) with and without an auxiliary valve for step inputs.

Both trajectories present position errors that are lower than

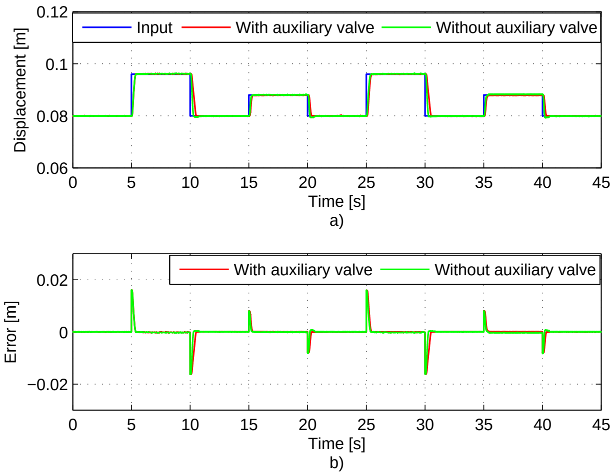

Figure 14: Step responses (a)) and position errors (b)).

± 0.25 mm for the steady state. This steady state behavior can be seen in the fig. 15 that shows a zoom of the position error. The experiment with and without an auxiliary valve did not presented significant deviation. Figure 16 presents the signal control sent to the valves with (fig. 16(a)) and without (fig. 16(b)) auxiliary valve for a steps trajectory.

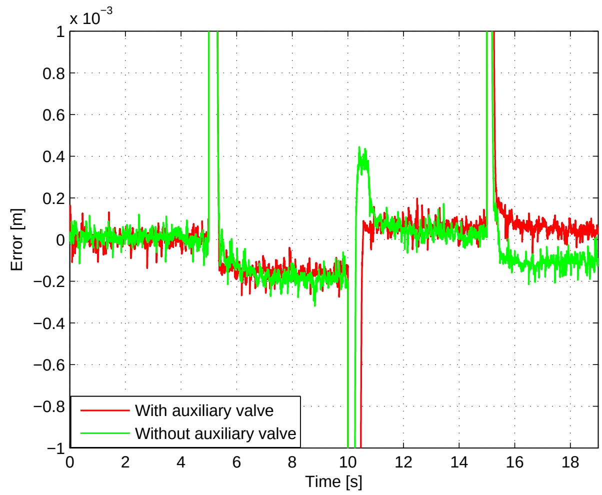

Figure 15: Zoom of the position error.

One can observe the contribution of the auxiliary valve because at the instant that it is acting the proportional valve is closed. This occurs because the load imposes the pressure difference enough to activate the auxiliary fast switching on/off valve. At this moment there is no consumption of compressed air and therefore there is compressed air saving.

Figure 17 shows a zoom of the switching of the on/off valve signal. It can be observed that at the beginning the auxiliary valve is “on" and then it begins to switch. This switch occurs because the piston is decelerating and positioning itself in steady state.

The pressures in the chamber A and B for the system with

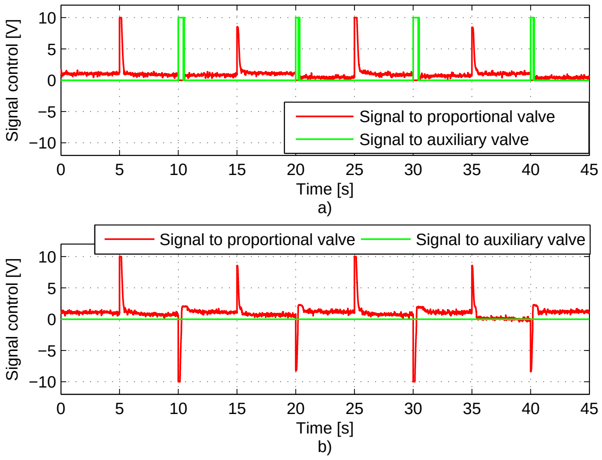

Figure 16: Control signal sent to the valves with and without an auxiliary valve.

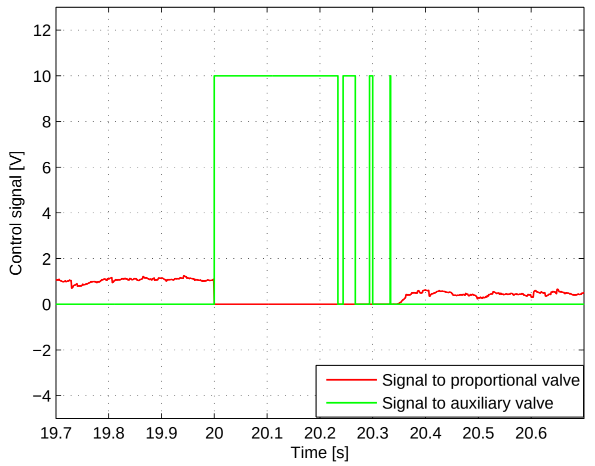

Figure 17: Zoom of the control signal sent valves.

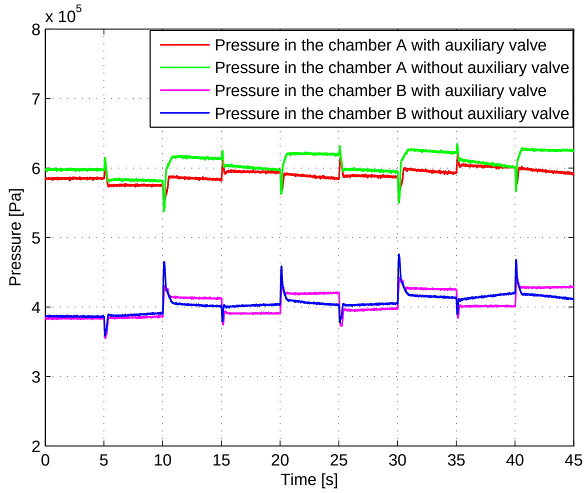

Figure 18: Pressure p_A and p_B for steps input.

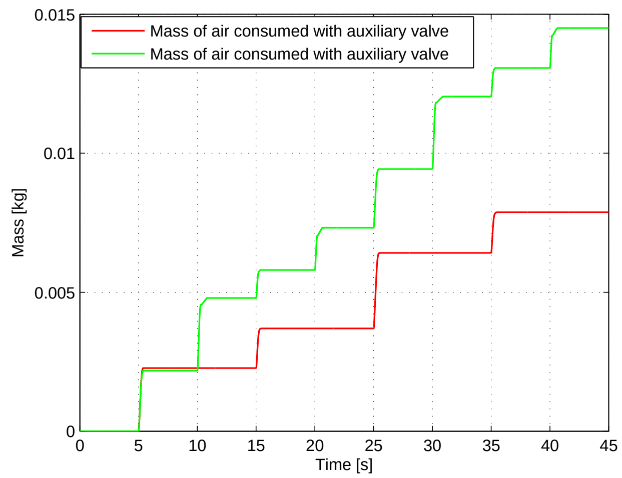

Figure 19: Mass of compressed air consumed for steps input.

and without an auxiliary valve are presented in fig. 18. The pressure in the chambers A and B of the cylinder show similar behaviors and are in the same range values. In the steady state the tendency is to stabilize the pressures through of the supply because the auxiliary valve is closed.

Figure 19 shows the mass of compressed air consumed by system with and without auxiliary valve related to the steps input shown in fig. 14(a). In this case, the compressed air saving is around 46 % for two work cycles.

The feedback between the cylinder chambers helps to increase the energetic efficiency. The equations used to calculate the mass of the compressed air consumed were presented in Section 3.

6 Conclusion

This paper presented a structure for a pneumatic positioning systems for compressed air saving in systems which need to move high loads. A comparison was made between the typical pneumatic positioning system and a system with an auxiliary fast switching valve in a feedback between the cylinder chambers.

For both step and sinusoidal inputs there were low position errors demonstrating that the fast switching on/off valves in a feedback between the cylinder chambers can be a promising solution. Experimental tests show it is possible to combine fast switching on/off valves with proportional valves without difficulties of control and performance losses. Their price can be as low as 10 % when compared to proportional directional valves.

The experimental results showed that with this type of configuration, up to 58 % of compressed air saving can be achieved in comparison to a typical pneumatic positioning system. This increases the energetic efficiency of studied pneumatic positioning system.

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Nomenclature

Designation	Denotation	Unit
C	Sonic condutance	[m^5/N.s]
e	Error	[m]
F_c	Load force	[N]
l_c	Smoothness width	[V]
M_P	Mass of compressed air consumed	[kg]
md	Right slope of output	
me	Left slope of output	
p_A	Pressure in chamber A	[Pa]
p_B	Pressure in chamber B	[Pa]
p_B	Supply pressure	[Pa]
q_m	Mass flow rate	[kg/s]
q_{mA}	Mass flow rate in line A	[kg/s]
q_{mB}	Mass flow rate in line B	[kg/s]
q_{mAT}	Mass flow rate in chamber A	[kg/s]
q_{mBT}	Mass flow rate in chamber B	[kg/s]
q_{mF}	Mass flow rate in feedback between cylinder chambers	[kg/s]
U	Signal control	
U_{czm}	Compensated output signal	[V]
U_d	Desired signal without dead zone	[V]
U_{dc}	Duty cycle at auxiliary valve	
U_{zm}	Signal control with dead zone	[V]
zmd	Right dead zone value	[V]
zme	Left dead zone value	[V]
α	Tolerance of pressure difference	[Pa]
k_P	Proportional gain	
k_I	Integral gain	
ΔP	Pressure difference	[Pa]
ρ_0	Density at the STP [1],	[kg/m^3]

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[1]STP = Standard Condition for Temperature and Pressure adopted by ISO 6358

Appendix A: Simulink model used in the experimental tests