EXPERIMENTAL IDENTIFICATION OF THE DEAD ZONE IN PROPORTIONAL DIRECTIONAL PNEUMATIC VALVES

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Abstract

This work presents a new methodology for dead zone nonlinearity identification in proportional directional pneumatic valves. It is based on observing the dynamic behaviour of the pressure in the valve gaps. Dead zone is common in hydraulic and pneumatic valves because the spool blocks valve orifices with some overlap, so that for a range of spool positions there is no fluid flow. The dead zone nonlinearity is a key factor that limits both static and dynamic performance in feedback control of fluid power systems. The usual method to cancel the harmful effects of dead zone is to add its fixed inverse function into the controller. This inverse is modelled by a set of parameters that need to be identified. The classic dead zone parameter identification uses expensive flow transducers and special test rig, while our proposed methodology needs only pressure transducers. Experimental results illustrate the efficacy of this methodology that is cheaper and faster.

Keywords: dead zone nonlinearity, pneumatics, proportional directional valves, dead zone identification

1 Introduction

This work presents a new methodology for identification of dead zone nonlinearity in proportional directional pneumatic valves, which is based on the study of pressure dynamic behaviour in valve gaps. This nonlinearity is a common imperfection of mechanical system components and mainly of closed centre valves when the land width is greater than the port width at neutral spool position (Virvalo, 1997).

The dead zone presence in a system is among the factors that limit the performance of feedback control loops (Sobczyk, 2000 and Valdiero, 2005), but components without such imperfections are expensive to manufacture and usually require specialized personnel to maintain (Tao and Kokotovic, 1996). Besides, dead zone can be desired in some cases, as for example the application of hydraulic valves in automotive suspension systems, where valve dead zone prevents internal leakages and maintains the height when the car is parked and the engine is turned off. However, in this same example, when the suspension is active, the effect of the dead zone is harmful and needs to be "removed" by an adequate compensation in control scheme (Bu and Yao, 2000, Corteville et al., 2005 and Valdiero et al., 2007).

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The conventional method used for dead zone compensation is to add an inverse dead zone function to the output signal of the controller, but this requires that the magnitude of the dead zone is known or accurately estimated (Turner, 2006) and the valve dynamics are fast enough to be neglected (Liu and Yao, 2004). The classical procedure used to identify dead zone in proportional valves is carried out in laboratory using expensive flow transducers and requires a long time. This proposed new methodology uses pressure transducers only, is cheaper and faster and can be carried out without special test apparatus.

Other researchers have proposed new methods for dead zone compensation as adaptive control strategies (Ibrir et al., 2006), fuzzy logic (Chiang and Yang, 2006) and sliding mode controller (Corradini and Orlando, 2002). The main problem with these schemes is their complexity, because adaptive control and fuzzy logic are not cheap from a computational point of view. In addition, these techniques are often viewed with some apprehension and distrust by industrialists (Turner, 2006).

According to Xiang (2001), the servo valve dead zone, in an individual form, was almost not studied in pneumatics. However, after determining its parameters, it has a simple compensation strategy with good results in trajectory tracking control.

This paper begins with the presentation of mathematical modelling used to depict dead zone nonlinearity. Next, section 3 shows the primary cause for dead zone nonlinearity in proportional directional pneumatic valves. In section 4, the new methodology for identification of dead zone in pneumatic valves is described including the test rig used. An application of dead zone compensation in a pneumatic servo system is depicted in section 5 using parameters obtained from application of this new identification methodology and their experimental results are used to validation. Conclusions are outlined in section 6.

2 Dead Zone Model

In this section we present the mathematical model for dead zone nonlinearity and its graphical representation that makes it easy to understand. This model is the same used by Tao and Kokotovic (1996). Dead zone is a static input-output relationship which for a range of input values gives no output. Figure 1 shows a graphical representation of the dead zone, where u is the input and u_{zm} is the output. In general, neither the break-points ($zmd \geq 0$, $zme \leq 0$) nor the slopes ($md > 0$, $me > 0$) are equal.

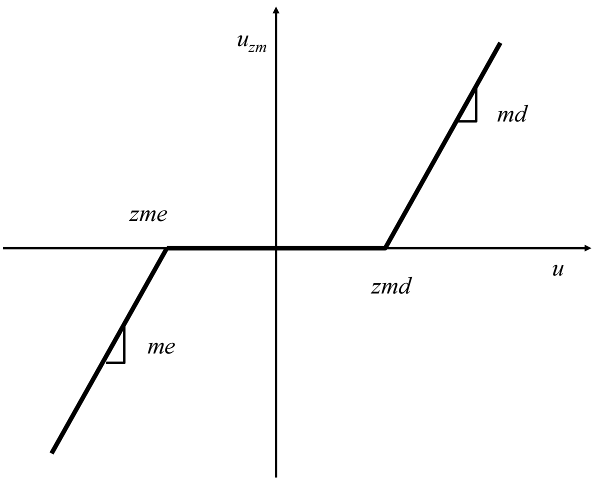

Fig. 1: *Graphical representation of the dead zone*

The dead zone analytical expression is given by Eq. 1.

$$u_{zm}(t) = \begin{cases} md \ (u \ (t) - zmd) & if \ u \ (t) \geq zmd \\ 0 & if \ zme < u \ (t) < zmd \\ me \ (u \ (t) - zme) & if \ u \ (t) \leq zme \end{cases} \quad (1)$$

3 Dead Zone in Proportional Directional Pneumatic Valves

In proportional pneumatic valves of directional control, dead zone is located at the dynamic system as a block diagram shown in Fig. 2. To understand this phenomenon better will be presented a detailed depiction of three-lands-five-ways spool valve components and its working.

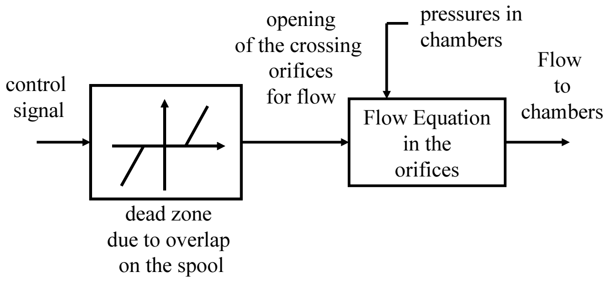

Fig. 2: *Block diagram of the proportional valve with input dead zone*

Figure 3 shows a sectional view sketch of a typical spool valve with main mechanical elements that can be used as a proportional valve. The control signal *u* energizes valve's solenoids so that a resulting magnetic force is applied in the valve's spool. An example of industrial spool proportional pneumatic valve is Festo MPYE-5 (Festo, 1996).

In closed centre or overlapped valves, the land width is greater than the port width when the spool is at null position, resulting the presence of the dead zone nonlinearity (Merrit, 1967). The dead zone nonlinearity is among the key factors causing delay and error in the system's response.

In proportional valves, it is a good idea that the relation between flow rate and control signal is linear. This relation is also damaged by the presence of the dead zone if there isn't compensation.

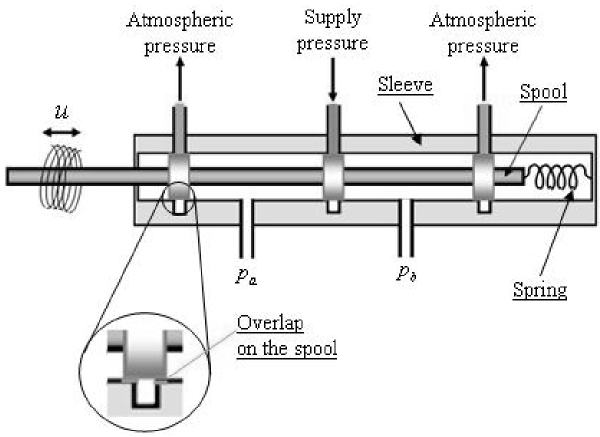

Fig. 3: *Sectional view sketch of typical spool valve with main mechanical elements of the proportional valve with input dead zone*

4 Experimental Identification of the Dead Zone Nonlinearity in Pneumatic Valves

This section describes the proposed methodology to dead zone identification in overlapped proportional valves. It is an alternative for classical procedure depicted by ISO 10770-1 standard (ISO, 1998), where the dead zone identification in spool valves can be experimentally made in test installation with flow rate transducer. Such flow rate transducer is expensive and these experimental tests can result in unacceptable cost for many applications.

This new methodology is an experimental procedure formed by some steps based in dynamic analysis of pressure's behaviour in actuator system. Proposed tests can be taken without special test installation and expensive instrumentation. They request only pressure transducers that have more acceptable prices. Also, in many times, they already are available in feedback control loops.

Our experimental tests were carried out in Laboratory of Fluid Power (NAPME) of the Regional University of North-western Rio Grande do Sul State (UNIJUÍ), where the test rig is described in section 4.1. The proposed methodology for identification of valve's dead zone is presented in section 4.2, where a detailed analysis of pressure's behaviour in actuator system is shown too.

4.1 Test Rig

The test rig used for the purpose of investing the dead zone in pneumatic valves is depicted in Fig. 4, which presents the main components of experimental setup. It is formed by one acquisition and control system mounted in a PC microcomputer and one pneumatic system, that is composed by one rodless pneumatic actuator (2) and one proportional directional pneumatic valve (4). Sensors permit measure air system inlet pressure (1), the actuator position (3) and actuator chamber pressures (p_a and p_b), (5) and (6). The acquisition and control system used is a dSPACE DS 1102 board. It is composed by 4 analog inputs (ADCs) and 4 analog outputs (DACs) as shown in Dspace (1996). Table 1 presents the main components of experimental system.

Table 1: Main components from experimental test apparatus

Component	Maker	Catalogue code
Pneumatic rodless cylinder	Rexroth	502 602 020 0
Proportional directional pneumatic valve	Festo	MPYE-5-1/8
Pressure sensors	Gefran	TKG E 1 M 1D M
Position transducer	Festo	MLO-POT-500-TLF
Compressed air reservoir	Pró-Ar	RA 080.500.1

4.2 New Identification Methodology of the Dead Zone

The proposed methodology is composed by open-loop tests of actuator system (valve and pneumatic actuator) with a slow sine control signal (10 volts amplitude and 100 seconds period, according to Eq. 2), pressure measurements, and analysis of their dynamic behavior as functions of the control signal $u(t)$.

$$u(t) = 10\sin(2\pi t/100) \tag{2}$$

The slopes of dead zone (md and me, as shown in Fig. 1) can be regulated such as $md = me = 1$ in a compensation block in the controller as well as the control input signal set in a -10 to 10 volts range. The open loops used to identify valve dead zone through this new methodology are carried out in three steps as follows.

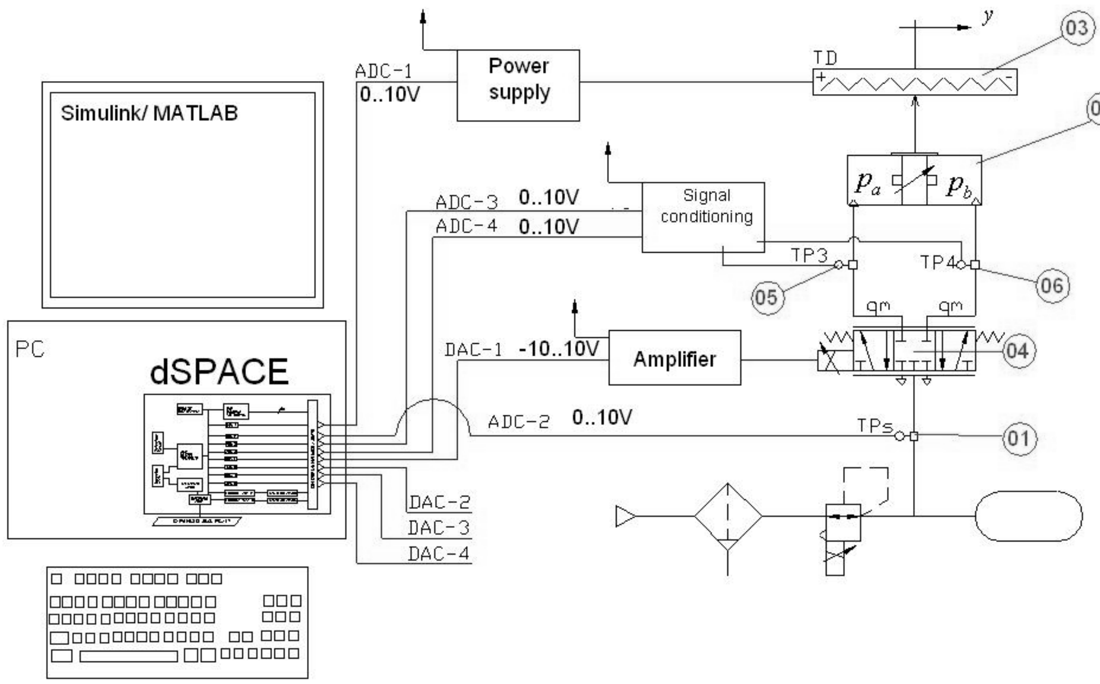

Fig. 4: *Experimental setup with main components*

Table 2: Behaviour description of the pneumatic system's elements for the enumerated pressure branches in Fig. 7

Branches	**Control signal**	**Mass flow rates**	**Pressures**	**Cylinder's piston**
(1)	$zmd < u < 3.2$	Cross cylinder chambers	Maintain necessary difference to movement	Travel to positive position
(2)	$3.2 \le u \le 10$ and $10 > u \ge 1.7$	There aren't flow rates to the chambers and the leakages aren't considerable.	Become equal to atmospheric pressure and supply pressure	Remain stopped at the end of its stroke.
(3) and (4)	$zmd > u > zme$	Leakages are considerable	Vary due to the leakages	Remain stopped
(5)	$zme > u > -3.0$	Cross cylinder chambers	Maintain necessary difference to movement	Travel to negative position
(6)	$-3.0 \ge u \ge -10$ and $-10 < u \le -1.5$	There aren't flow rates to the chambers and the leakages aren't considerable.	Become equal to atmospheric pressure and supply's pressure	Remain stopped at the end of its stroke.
(7) and (8)	$zme < u < zmd$	Leakages are considerable	Vary due to the leakages	Remain stopped

At the first step, it is observed the p_a pressure curve in the valve gap for the u control signal range from -10 to 10 volts (Fig. 5), where is possible to estimate the right dead zone value (right break-point *zmd*) with the knowledge of the pressure's dynamic behaviour. In the range where the control signal varies from -10 V to ≈ 1.5 V, the valve is opened so that its "a" port is connected to the atmosphere, the pressure in this port is $p_a = 0$ bar and piston remains in retracting position without movement. When the valve begins to block the control orifices and the control signal is next to null value, the valve's leakages are considerable and they have a smoothing influence on the pressure variation. At the moment that control signal (u = 1.2 V) crosses the right dead zone value (right break-point, *zmd*), there is a sudden pressure's variation as it is seen in Fig. 5.

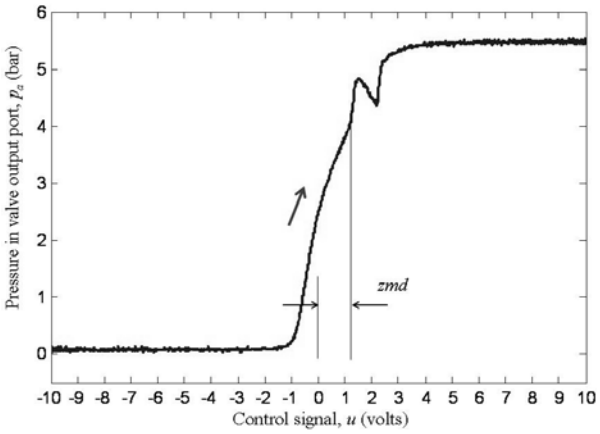

Fig. 5: *Pressure behavior p_a in valve's output port and the evidence of the right dead zone*

In the second step, the p_b pressure curve (Fig. 6) is analysed in the valve gap for the u control signal range from 10 to -10 volts. The same thought described in the previous paragraph is used when the control signal crosses the left dead zone value (left break-point, *zme*), when also there is a similar sudden pressure's variation in u = -0.8 V as it is shown in Fig. 6.

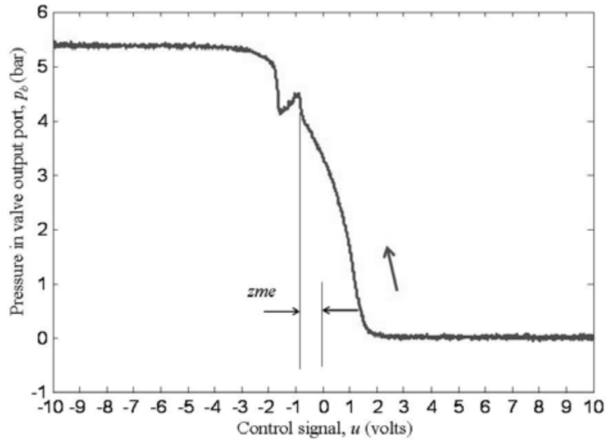

Fig. 6: *Pressure behavior p_b in valve's output port and the evidence of the left dead zone value*

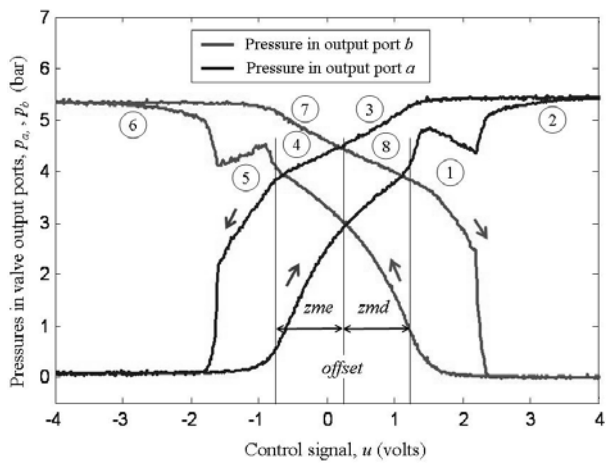

Fig. 7: *Graphical representation of the dead zone values (break-points) and the enumeration of the pressure behavior branches*

In the third step, the analysis of pressure's behaviour in actuator system is carried out. Both pressure behaviors, as functions of the control signal, are plotted in an only figure (Fig. 7), with u ranging from - 4 V to 4 V. In this amplified graphical representation, the constant *offset* value was obtained in the way to centralize the valve spool and have equal values to spool overlap in both right and left sides. This *offset* = 0.2 V can be added to control signal value and permits of the identical dead zone values ($zmd = zme = 1.0$ V). The *offset* value can be important in some control applications as used in Ning and Bone (2005).

For the best comprehension of Fig. 7, main pressure behaviour branches are enumerated and commented in Tab. 2. This table describes the behaviour of the pneumatic system's elements during experimental tests.

The case study presented in section 5 depicts the efficiency of this method through a pneumatic actuator application with dead zone compensation in proportional controller for trajectory tracking.

5 Study Case: Pneumatic Actuator Application

The dead zone compensation strategy was implemented on the pneumatic actuator shown in Fig. 8 using dead zone parameters identified in section 4.2. The detailed experimental setup is the same that was described in section 4.1.

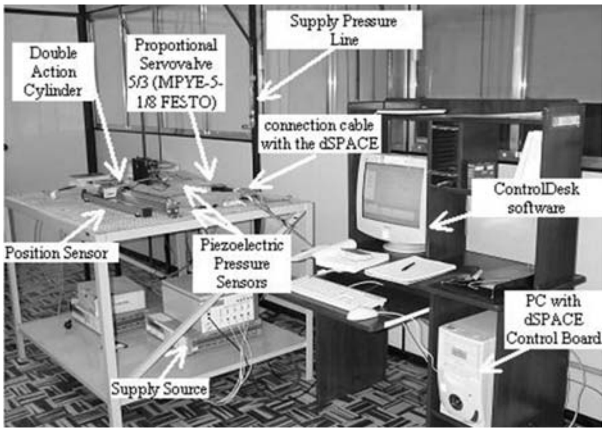

Fig. 8: *Experimental Pneumatic Actuator*

In this experimental work, a classic proportional controller (P) was implemented without and with dead zone compensation in pneumatic servovalve for comparison. According to Andrighetto et al. (2003), the use of P controllers in servo pneumatic systems is associated with poor performance, with oscillation and great errors. This controller was chosen because it is easy to implement, has only one parameter to adjust (K_P gain) and already there are results presented in previous works (Andrighetto et al., 2003). Also, the results of dead zone compensation are easier to see with P controller. The control signal u generated by a P controller and applied to servovalve is given by Eq. 3.

$$u = K_{\mathrm{P}}\left(y(t) - y_{\mathrm{d}}(t)\right) \tag{3}$$

where K_{P} is Proportional gain, $y(t)$ is position of pneumatic actuator and $y_{\mathrm{d}}(t)$ is desired position of pneumatic actuator.

The dead zone compensation is carried out through the addition in controller output of an inverse of dead zone function to cancel the dead zone effect in the system. According to Valdiero (2005), the perfect compensation of dead zone is difficult, but its effects of performance degradation can be minimized by an estimate of dead zone parameters and an increased softness in the range near to zero position. The inverse function used to compensate dead zone is described by Eq. 4. Figure 9 depicts the graphical representation of the smoothed dead zone inverse.

$$u_{\mathrm{czm}} = \begin{cases} \dfrac{u_{\mathrm{d}}(t)}{md} + zmd & \text{if} \quad u_{\mathrm{d}}(t) \ge lc \\ \dfrac{u_{\mathrm{d}}(t)}{me} + zmd & \text{if} \quad u_{\mathrm{d}}(t) \le -\, lc \\ \left(\dfrac{zmd + lc/md}{lc}\right) u_{\mathrm{d}}(t) & \text{if} \quad 0 \le u_{\mathrm{d}}(t) \le lc \\ \left(\dfrac{zme + lc/me}{lc}\right) u_{\mathrm{d}}(t) & \text{if} \quad -lc \le u_{\mathrm{d}}(t) \le 0 \end{cases} \tag{4}$$

where $u_{\mathrm{d}}(t)$ = desired control signal input, without considering dead zone; lc is smoothness width used in compensation; and u_{czm} is compensated output signal.

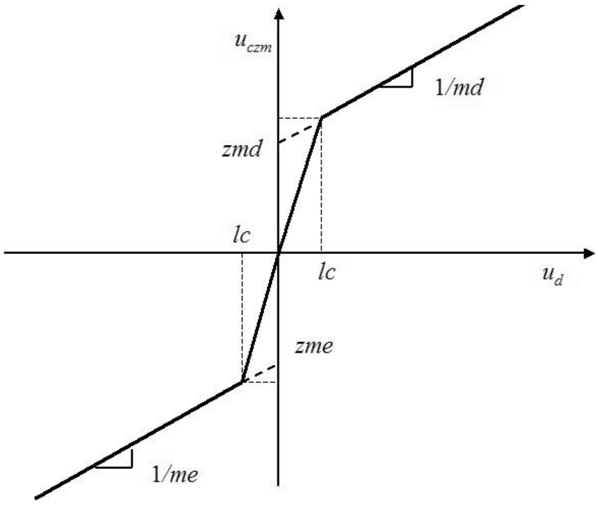

Fig. 9: *Graphical representation of the smoothed dead zone inverse function*

A control structure block diagram is shown in Fig. 10. The dead zone compensation is located in controller output.

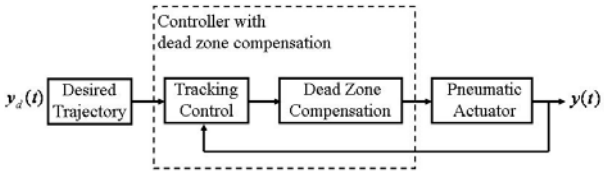

Fig. 10: *Control structure block diagram with dead zone compensation in controller output*

In experimental tests, $K_{\mathrm{P}} = 20$ was used. This value conducts to a moderate position error without system oscillations. For dead zone compensation, $md = me = 1$ and $lc = 0.05$ were used, that are in Eq. 4 and are identified as shown in section 4. The lc value represents a compromised decision between control signal quality and effective dead zone compensation. For example, if *lc* is too large, dead zone compensation is poor. If *lc* is too small, oscillations in control signal can occur near of origin.

Experimental tests were carried out with a sinusoidal reference tracking signal of period 10 s and amplitude 0.20 m. Results obtained from dead zone compensation to trajectory tracking tests to sinusoidal trajectory are depicted in Fig. 11. For comparison, Fig. 11 also depicts results of this trajectory without dead zone compensation. The analysis of these performances permits to see that dead zone non-linearity causes highly expressive limitations in the performance of controller, with a large position lag to tested trajectory (curve without compensation compared with desired). After compensation, this lag is minimized and executed trajectory is very close to desired trajectory. This successful dead zone compensation was possible because dead zone was adequately identified in a simple way by proposed methodology in this paper.

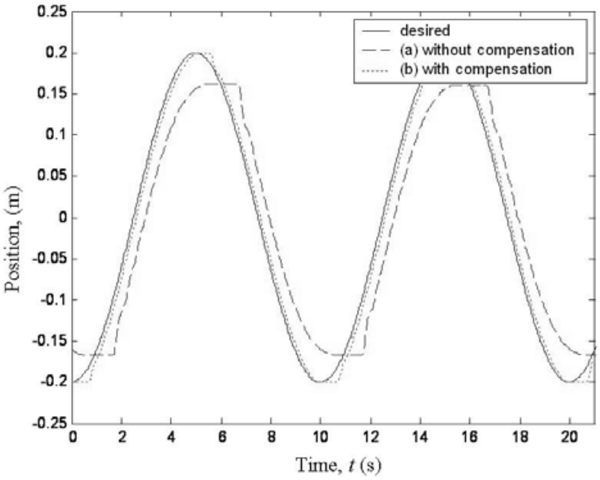

Fig. 11: *Experimental results to sinusoidal trajectory tracking: (a) without and (b) with dead zone compensation*

5 Conclusions

This paper presents a new methodology that addresses the experimental dead zone identification, based on the pressure dynamic behaviour in the valve gaps. Dead zone is a typical nonlinearity in proportional directional pneumatic valves and it is treated as imperfection of mechanical components.

The dead zone analytical model is characterized by set of parameters and the main aspect considered is its identification. The results of this paper show that is possible to obtain the parameters to dead zone model, in a simpler and easier way, based on observing the dynamic behaviour of the pressure in the valve gaps. These results were confirmed experimentally by a study case with pneumatic actuator.

This methodology is cheaper than the conventional ones because it needs only pressure transducers. With this paper, the authors intend to contribute in the study and research of advances in pneumatic servo position control to open the doors to new industrial applications for these systems.

Nomenclature

lc	smoothness width	[V]
md	right slope of output	[1]
me	left slope of output	[1]
offset	control signal to centralize valve spool	[V]
p_a	pressure in valve output port a	[bar]
p_b	pressure in valve output port b	[bar]
u	control signal	[V]
u_{czm}	compensated output signal	[V]
u_d	desired control signal input	[V]
u_{zm}	output value after dead zone	[V]
y	position of pneumatic actuator	[m]
zmd	right dead zone value	[V]
zme	left dead zone value	[V]

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