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Auto-Selective Three Term Control of Position and Compliance of a Pneumatic Actuator

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***Abstract*—**Due to their high power-to-weight ratio and low cost, pneumatic actuators are attractive for robotics and automation applications; however, achieving fast and accurate control of their position have been known as a complex control problem. The paper presents a methodology for obtaining controllers that achieve high position accuracy and preserve the closed-loop characteristics over a broad operating range. Experimentation with a number of conventional (or "classical") three-term controllers shows that, as repeated operations accumulate, the characteristics of the pneumatic actuator change requiring frequent re-tuning of the controller parameters (PID gains). Furthermore, three-term controllers are found to perform poorly in recovering the closed-loop system after the application of load or other external disturbances. The key reason for these problems lies in the non-linear exchange of energy inside the cylinder relating, in particular, to the complex friction forces that develop on the piston-wall interface. In order to overcome this problem but still remain within the boundaries of classical control methods, we designed an auto selective classicaql controller so that the system performance would benefit from all three control gains (KP, Kd, Ki) according to system requirements and the characteristics of each type of controller. This challenging experimentation took place for consistent performance in the face of modelling imprecision and disturbances. In the work presented, a selective PID controller is presented for an experimental rig comprising an air cylinder driven by a variable-opening pneumatic valve and equipped with position and pressure sensors. The paper reports on tests carried out to investigate the capability of this specific controller to achieve consistent control performance under, repeated operations and other changes in operating conditions.

***Keywords*—**Classical selective controller, long-term experimentation, pneumatic actuator, position accuracy.

I. INTRODUCTION

SERVO pneumatic systems play an indispensable role in industrial applications thanks to their variety of advantages like: simple operation, clean, low cost, high speed and easy maintenance. The dynamics of these systems are highly nonlinear and their models inevitable contain parametric uncertainties and unmodeled dynamics. The pneumatic servo system is a very nonlinear time-variant control system because of the compressibility of air, the friction force between the piston and the cylinder, energy and thermal effects inside the cylinder, the flow rate through the servo valve, etc. In recent articles [1], [2], [3] it is demonstrated that the application of nonlinear robust control techniques is a necessity for successful operation of pneumatic systems. Although PID (Proportional, Integral, Derivative) controller is still the most widely used approach due to its ease of implementation, the need for overcoming highly nonlinear phenomena turns away the use of classical PID controllers nowadays, see [4]. Therefore modern control techniques were designed and tested in pneumatic actuators in order to improve the performance of such systems considering position accuracy and repeatability as the two main performance characteristics. Fuzzy logic control, neural networks method, adaptive control, self-tuning or gain scheduling, the so-called “Soft Computing” control techniques, are approaches that many researchers such us [4], [5], [6], [7], [8] have implement in the recent past in an attempt to build a robust controllable pneumatic system. Among these techniques, much previous work from the above authors approved to be very confident in solving position control problem of a pneumatic servo system and all non-classical control methods have attracted considerable attention because they provide a systematic approach to the problem of maintaining stability and consistent performance in the face of modeling imprecision and disturbances. In this paper, our scope is to investigate and afterwards demonstrate how the all time classic PID controller can be improved with a specific technique of auto-selective use of the control terms and achieve the ultimate system results in the position-tracking problem of a servo pneumatic system applying a traditional PD controller in the first step of the controller design. The use of the I-term control will be selective if the system performance requires although it will be included in the control algorithm from the beginning.

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II. EXPERIMENTAL RIG AND DYNAMIC ANALYSIS

The servo pneumatic system, which is the subject of this study, is composed of a double acting cylinder (FESTO DSW-32-80-A), a proportional servo valve (FESTO MPYE-5-1/8-), a linear variable differential transducer (RDP ACT1000C), the Siemens C164CI microcontroller development kit and finally a COMPAQ Presario PC to edit and download the code to the

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microcontroller. An overall layout of the system is provided in Fig. 1.

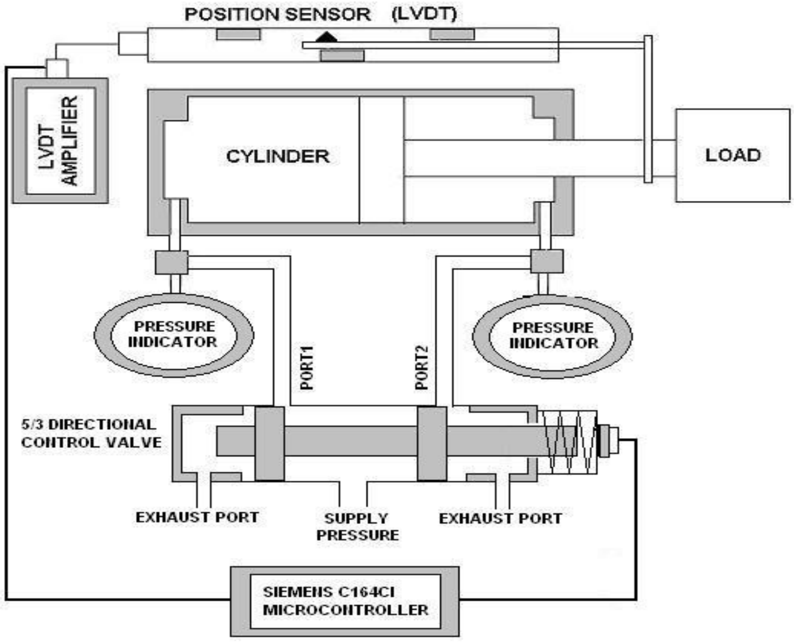

Fig. 1 The main layout of the system

The dynamic analysis of such a system can be easily found in many previous research projects like [9]; [10]. For illustration purposes we will provide at this point some mathematical equations, which will be included in the control algorithm. The interesting part of this analysis is focused on the servo valve dynamics, regarding air mass flow rate (Q):

$$Q = \begin{cases} C_f AiC_1 P_1 / \sqrt{T}, \\ if\ P_2 > P_1 P_{cr} \\ C_f AiC_2 P_2 / (\sqrt{T}) (\frac{P_2}{P_1})^{(\frac{1}{K})} \sqrt{1 - (\frac{P_2}{P_1})^{(\frac{K-1}{K})}}, \\ if\ P_2 \le P_1 P_{cr} \end{cases} \quad (1)$$

Where,
Cf, C_1, C_2, K, are constant coefficients, Ai is the valve orifice area, P_1 and P_2 are the air pressures entering the two cylinder chambers and P_{cr}=0.528 is the critical pressure value separating the supersonic from the subsonic circumstances of the system operation.
According to the same authors the pressure derivative build-up equations inside each of the two cylinder chambers is shown below:

$$\overset{o}{P}_{1,2} = [RTQ_{1,2} - \alpha P_{1,2} A_{1,2} \overset{o}{x}] / A_{1,2}(L - x) \quad (2)$$

Where

$\overset{o}{x}$ the piston velocity, x is the piston position, T is the temperature, R is the ideal gas constant, $A_{1,2}$ is the piston's areas from both sides, L is the piston stroke, a is a coefficient depending on the heat transfer and their values can be easily calculated and for simplicity reasons reporting them is neglected here. The equation of the piston motion is derived from the second Newton's Law:

$$P_1 A_1 - P_2 A_2 - Mg - Ff - bx = M \overset{oo}{x} \quad (3)$$

Where,

$\overset{oo}{x}$ the piston acceleration, M the mass of the load, g the gravitational force (the cylinder is placed vertically), Ff the Lugre friction force, P_1, P_2 the chambers pressures, b is the spring constant and A_1, A_2 the piston area in each chamber respectively.

III. Internal Friction Modelling

Friction is perhaps the most important nonlinearity that is found in any mechanical system with moving parts. For the system considered in this research project, friction, which arises in the contacts of the piston with the cylinder walls as well as the linear slide-way and other minor rubbing elements, has a direct impact on the dynamics of the system in all regimes of operation. All classical friction models do not consider the presliding region; the system does not move as long as the applied force is smaller than the maximum static force so, in order accurately to design compensation, friction has to be identified in both the sliding and presliding regions. To construct a general friction model from physical first principles is simply not possible. What researchers look for instead is a general friction model for control applications, including friction phenomena observed in those systems. This task is by no means a simple one since no universal friction model exists, on the one hand, and the practical measurement of friction is not straightforward on the other. A good model structure for the identification of friction would ease the desired task. The LuGre model is a dynamic friction model introduced in [12]. The same researchers developed extensive analysis of the model and its application two years later [11]. The model is related to the bristle interpretation of friction. The deformation of the bristles is presented in the next layout:

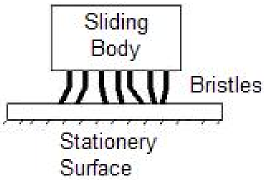

Fig. 2 Bristle deformation of LuGre model

Friction is modelled as the average deflection force of elastic springs. When a tangential force is applied the bristles will deflect like springs. If the deflection is sufficiently large the bristles start to slip. The velocity determines the average bristle deflection for a steady state motion. It is lower at low velocities, which implies that the steady state deflection decreases with increasing velocity. The LuGre friction model is illustrated in the following Fig. 3.

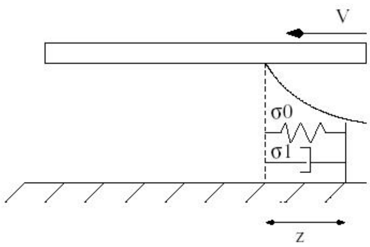

Fig. 3 The LuGre friction model

This models the phenomenon that the surfaces are pushed apart by the lubricant, which is between the cylinder body and the piston rubber. The model is widely used in modern pneumatic research projects and it is considered the most appropriate for control tasks, like reference [13]. The LuGre friction model has the general form:

$$F_f = s_0 z + s_1 \frac{dz}{dt} + s_2 v \tag{4}$$

Where z is friction internal state that describes the average elastic deflection of the contact surfaces during the stiction phases, the parameter σ_0 is the stiffness coefficient of the microscopic deformations z of the bristles during the presliding displacement, σ_1 is a damping coefficient associated with dz/dt and σ_2 represents the viscous friction, whereas v is the current velocity of the piston.The dynamic of the internal state z is expressed by:

$$\frac{dz}{dt} = v - \frac{s_0 \left|v\right|}{g(v)} z \tag{5}$$

Where g(v) is a positive function that describes part of the steady state characteristics of the model for constant velocity motions and it is given by:

$$g(v) = Fst + (Fv - Fst)e^{-(v)^2} \tag{6}$$

Where Fs, Fv are the static and viscous friction. From the control point of view the simplest yet most effective approach to counteract the friction phenomenon appears to be the use of sliding mode control; in fact, one of the main characteristics of this control technique is its robustness against bounded uncertainties and disturbances. Friction is regarded as a bounded disturbance of unpredictable sign and therefore counteracted by choosing suitable control amplitude. The LuGre friction model turned out to be robust under the influence of all the parameters of the control methods applied to the simulation model.

IV. Design of the Controller

Although the design of the classical three-term control for nonlinear multivariable systems has been extensively studied in many books and papers, the design procedures for such high order nonlinear systems, like the pneumatic systems, may be complicated and vary from case to case. A simple approach to robust control, and the main topic of this chapter, is the implementation in the system of a classical controller which would automatically choose whether or not all terms of control P, I, D are appropriate for the application. During experimentation we managed to witness that the system operates satisfactory with the use of PD control and that the existing steady state position error was the main unwanted characteristic of the system behavior. The solution to that was to address the I-term control in the controller so that the steady state error would be eliminated. The need for retuning three rather than two control gains on one hand and the fact that the overall system behavior became oscillatory under the influence of the I-term, turned us to implement a clever idea of using the I-term only for beneficiary results. The new technique, which was addressed in the system, is based on “switching” the Integration term ‘on’ and ‘off’ according the value off the steady state error. Although the integration term is limiting the steady state error to tiny values close to zero over short periods, it is also producing the unacceptable system response over long time periods of operation. This technique introduces another secondary ‘zone’ of the steady

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state error which is placed in the middle of the primary 'zone' of 4mm (+/- 2mm) when the I-term switches on, discussed earlier in this section. The secondary zone of values is, like the primary zone, split into two parts, 0.5mm above and 0.5mm below the demand piston position respectively. The idea of the new control algorithm is that the I-term is switched on when the error is within the primary zone (2mm above the demand position and 2mm below it) but when the error values are very small, i.e., within the secondary zone of values, it switches off. When the system performs without the influence of the I-term, the behavior is not oscillatory, and therefore if the steady state error remains always in the secondary zone, the response is considered to be acceptable. The new algorithm is based on the generic PID algorithm with this slight modification allowing the existence of the secondary zone and in fact this method provides the privilege to the system of 'deciding', according to the value of steady state, error whether to perform with or without the integration influence.

The operation of the system required the implementation of manual retuning, but the undesired oscillations of the system were eliminated. In Fig. 3 the system response when this method is applied to it is shown. It is useful to explain at this point some multiple critical comments, which can be recorded from the plot below. In order to estimate the steady state error, the position axis of the plot is extremely focused around the desired position value. The system responses, as well as the demand position signal, therefore appear noisy. There are two different system responses, test1 (red line) and test2 (blue line), which are the average curves of ten different experiments each, with the same control gain values and the desired target position is set to the ¾ of the piston stroke. A manual retuning method was followed and the ultimate gain values are provided in the below table:

Test Number	Proportional Gain (Kp)	Integral Gain (Ki)	Derivative Gain (Kd)
Test 1 (red line)	1.5	0.002	2
Test 2 (blue line)	1.3	0.002	2

The system with the new method of the secondary zone of error values performs rather well, there are no oscillations during long time operations (50 sec) as there were before, with the simple PID control and the critical factor of this research project, the position accuracy, is minimized in values between 0.16mm and 0.2mm of the demand position. The piston position percentage accuracy is calculated as the 0.28% of the overall stroke of the piston, a value that is excellent considering the nature of the system. After all, this new method of control improved the overall system response and the time spent on retuning the re-designed algorithm was worth it for the aim of this project. The experimentation to back-up our results was designed as follows. During 8 hours continuous operations of the system all responses of the PD, PID, Auto-selective I-Term, were recorded. Then, an average response of each one of the techniques was plotted and we provide the outcome of it in Fig. 5.

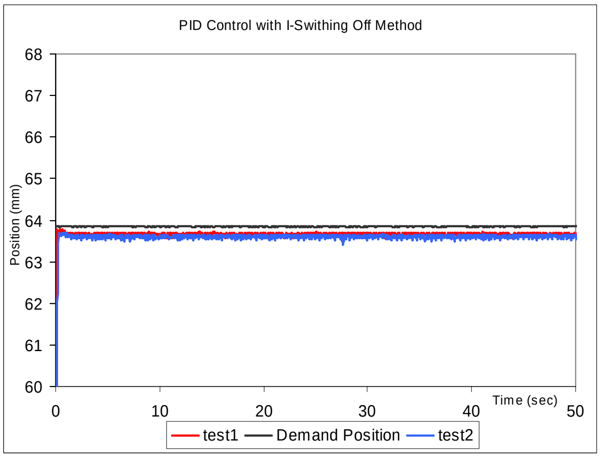

Fig. 4 The Performance of the system using the "selective" I-Term controller

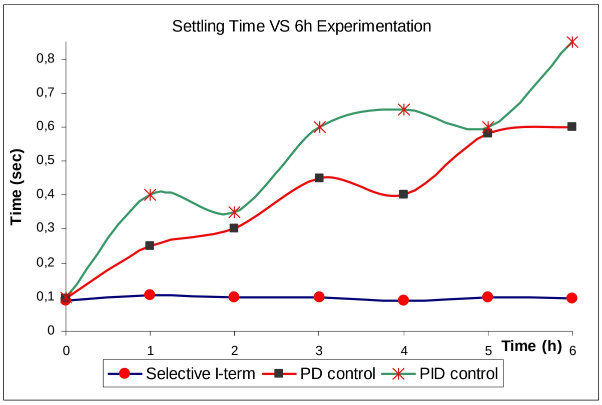

Fig. 5 The settling time of the system using PD, PID, Auto-selective I-Term

V. Conclusion and Further Work

The reason for testing this kind of controller, even though the PD controller's results exceeded expectation, was to use the I-term optionally in order to try to reduce (or eliminate) the steady state error and afterwards compare all the classical control techniques together. In order to avoid changes to system rise time due to the I-term, it was decided to use the integral term only when the system's position was within some small limitations around the demand position. The increased problem of estimating the PID controller gains values rather than the PD controller is time consuming, and the actual values could be estimated only by experimentation. This is why the auto-selection of I-term experiment was designed, which was based on the assumption that, when the system performed within a smaller secondary limitations zone of error values, the I-term should switch off automatically and therefore the controller should revert to PD operation. With this technique the oscillatory response of the system was eliminated, the selective existence of the I-term minimized the steady state error and the final position percentage of 0.28% is considered to be excellent for such a complex mechanical system.

Based on the laboratory measurements we were able to conclude that the controller is suitable and effective for positioning experiments with our rig. As research continues the improvement of the controller will continue also and tests in simulation, using the already built simulation model, as well as in the real experimental rig will take place. Furthermore we are interested in trying out several different tests with this promising control technique and try to prove its robustness under any operation conditions, like compliant load. In addition to this, the simulation model will be finalised in order to provide us a comparative and predictive tool of the real pneumatic system.

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