PNEUMATIC POSITIONING SYSTEM CONTROLLED BY ON-OF VALVES

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ABSTRACT: *The pneumatic cylinders are widely used in industrial applications for many automation purposes thanks to their advantages. The design of a stable robust position controller for a pneumatic servo-system is difficult. In most cases, applications of pneumatic actuators use servovalves. In the past few years there has been a wide interest in the use of cheap high speed solenoid valves. The main contribution of this paper is instead of using a servovalve, two solenoid valves are applied in the positioning system. The sliding mode controller is implemented on a DSP. The main challenge is the higher accuracy than 10 μm.*

Key Words: *Pneumatic/Positioning/Sliding mode*

1. INTRODUCTION

As an important driver element, the pneumatic cylinder is widely used in industrial applications for many automation purposes thanks to their variety of advantages, such as: simple, clean, low cost, high speed, high power to weight ratio, easy maintenance and inherent compliance. Traditionally, they are used for motion between two hard stop. The design of a stable robust position controller for a pneumatic system is difficult since it is a very nonlinear time-variant controlled plant because of the compressibility of air, the friction force between the piston and the cylinder, air mass flow rate through the servo-valve, etc.

The early applications based on linear PID controllers proposed by Burrows and Web, 1966; Vaughan, 1965 had limited operation area. Fok and Ong, 1999 [1] reached precision of ± 0.3 mm. Fujiwara et al., 1995; Matsukuma et al., 1997 proposed artificial neural network and Jeon et al., 1998 proposed genetic algorithm to tune the PID controller. The precision was ± 0.1 mm in the best case. The best precision (0.01 mm) was reached by Wikander, 1988 [2]. Sliding mode control was proposed by Noritsugu and Wada, 1989; Tang and Walker, 1995 [3]; Surgenor and Vaughan, 1997 [4]; Paul et al., 1994 [5]; Song and Ishida, 1997 [6] but the accuracy was limited.

The pneumatic valve is the key element in the system. There are two types of valves used in the pneumatic positioning, servovalves and on-off valves. With conventional on-off valves accurate position control is difficult to achieve because of the limitation of the valve response time. In the past few years there has been a wide interest in the use of cheap high speed solenoid valves [7]. The most of applications are on pulse with modulation (PWM). By the advent of DSPs with high computation power, the precise and robust control of pneumatic actuators has become possible.

Sliding mode control was introduced in the late 1970's [8] as a control design approach for the control of robotic manipulators. Among experimental studies, a few succeeded in showing closed-loop system behaviour which was predicted by the theory [9].

Another solution is to employ the advanced nonlinear control strategies developed in recent years (soft computing) [10].

2. DESIGN OF A SLIDING MODE CONTROLLER

In order to design a robust controller and predict the control performance for the pneumatic test rig, a theoretical and practical modelling of the rig is needed. The dynamic of the piston is modelled by the mass "m", the damping "d" and the spring "k". The friction force is denoted by "F_f". The piston can be moved by the pressure difference between the two sides of the piston. The pressures p_a and p_b can be influenced by the input and output air flow rates, which can be controlled by the input and output valves. The system, which can be described by a second ordered nonlinear motion equation:

$$m\ddot{x} = p_a(u)A_a - p_b(u)A_b - d\dot{x} - kx - F_f \tag{1}$$

where x is the position, u is the control signal. The dynamics of the valves are ignored. The other parameters and variables T, V, A, Q and c are the temperature, volume, area, heat energy and specific heat respectively. The subscription refers to the location of actual variable. The calculation of p_a and p_b is based on two main laws, balance of the input, output and inner energies and balance of the input, output and inner masses.

The design of a sliding mode controller consists of three main steps. One is the design of the sliding surface, the second step is the design of the control which holds the system trajectory on the sliding surface, and the third and key step is the chattering-free implementation. The purpose of the switching control law is to force the nonlinear plant's state trajectory to this surface and keep on it. When the plant state trajectory is „above" the surface, a feedback path has one gain and a different gain if the trajectory drops „below" the surface.

Consider a single-input, single-output second-order nonlinear dynamic system:

$$\ddot{x} = f(x,\dot{x},u) \tag{2}$$

Where x is the output signal (position) of the controlled plant, u is the control signal. If x_d denotes the desired value, then the error between the reference and system states may be defined as $e=x_d - x$.

2.1. Sliding surface design

Classically, a scalar variable s is calculated as a linear combination of the error and its derivative.

$$s = e + \lambda \cdot \dot{e} \tag{3}$$

Let $s(\dot{e},e) = 0$ define the „sliding surface" in the space of the error state. The purpose of sliding mode control law is to force the state trajectory of the error to approach the sliding surface and then move along the sliding surface to the origin (Fig.1.).

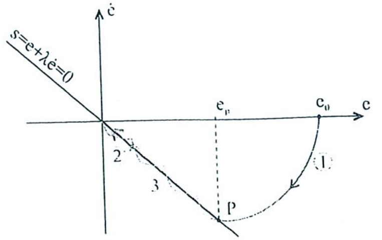

Fig.1. Sliding motion in the state space

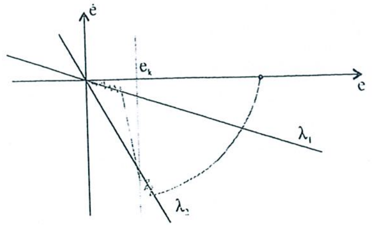

Fig.2. Two sliding lines in the state space

The process of sliding mode control can be divided into two phases, that is, the approaching phase with $s(\dot{e},e)\neq 0$ and the sliding phase with $s(\dot{e},e)=0$. If the system is in sliding mode the error is decreasing exponentially, where λ is a time constant type parameter. If λ is big than the system response is slow but accurate. If it is small than the system response is fast but the system might chatter (see the experimental results). The proposed solution is the application of two sliding lines (see Fig. 2.).

The steep sliding line described by λ_2 ensures the fast response. When the trajectory get close to the origin, the system change over to switching line of λ_1, to avoid the chattering.

2.2. Selection of the control law

In order to guarantee that the trajectory of the error vector e will translate from approaching phase to sliding phase, the control strategy must satisfy the sliding condition

$$s(\dot{e},e)\cdot\dot{s}(\ddot{e},\dot{e})<0\,. \tag{4}$$

This means that e will always go toward the sliding surface. A proper control should be selected to satisfy the condition (4) in any time instant. The simplest control law that might lead to sliding mode is the relay. $u=\delta\cdot sign(s)$

2.3. Chattering free implementation

Chattering is the main problem of sliding mode control and chattering free implementation is the key step in design of a sliding mode controller. A quite general solution is that the relay (which changes its output value suddenly) is replaced by a saturation function. There is a boundary layer around the sliding surface where the control signal is changing continuously. If the system trajectory is close to the sliding surface and the control signal is small, than the system might stick before the goal.

To avoid it a modified saturation function shown in Fig. 3. is proposed. When the limitation of the position is satisfied, all high-speed on-off solenoid valves are ON to stop the overshoot. The control will be finished when $|e_s|$ is smaller than e.

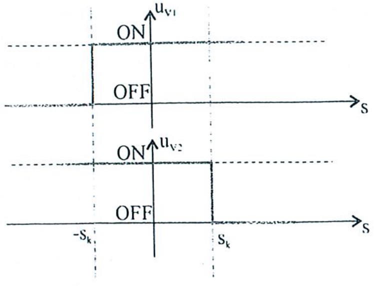

Fig.3. Modified saturation function

3. THE SERVOPNEUMATIC POSITIONING SYSTEM

The system is shown in Fig.4. and Fig.5. It consists of a double-acting pneumatic rodless cylinder (MECMAN 170 type) with bore of 32 mm, and a stroke of 500 mm, controlled by tree-way solenoid valves (Bosch Rexroth 579, FESTO MHE2-MSH-3/2O-QS-4-K fast switching types). A linear encoder (LINIMIK MSA 320 type and BTL5-S101 type Micropulse Linear Transducer with 1 μm resolution from Balluff) gives the position. Velocity and acceleration are obtained by numerical derivation. Pressure sensors (Motorola MPX5999D) are set in each chamber. The controller is implemented in a DSP board of „eZdspTM for TMS320LF2407" from Spectrum Digital.

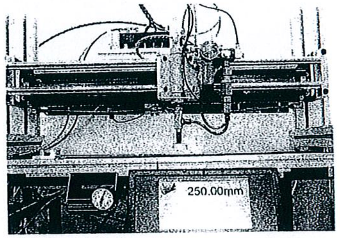

Fig.4. The experimental positioning system

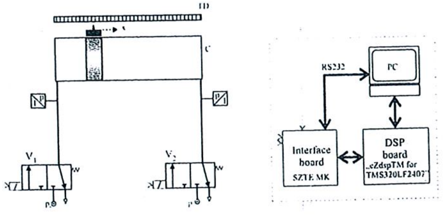

Fig.5. Configuration of pneumatic positioning system

The system pressure is set to be 6 bar, the sampling time is 2 ms. In order to analyze the positioning methods a real-time data acquisition program was designed for a PC to capture the system output data through the communication interface between the PC and the DSP controller.

The control program is in the DSP program memory. So the DSP controller can operate independently. The DSP Starter Kit (DSK) enables the user to connect the DSP to the parallel port on a PC and download code using a DOS interface. The control algorithm is written in "C" language, and compiled into assembly language and downloaded into the DSP board.

4. EXPERIMENTAL RESULT

The conventional (Bosch Rexroth 579), single stage solenoid operated on–off valves are very bulky and their dynamic performances are low. With these valves fine motion control is difficult to achieve because of the limitation of the valve response time. With on-off control the system will never reach a steady state value.

The actual position will tend to oscillate around the desired position. (see in Fig.6. and Fig.7.). The second measurement is a positioning with high-speed on-off solenoid valves. The time functions of the position, speed and control signal is shown Fig.8. and Fig.9. The position error of the DSP based relay type sliding mode control is within ±0.01 mm.

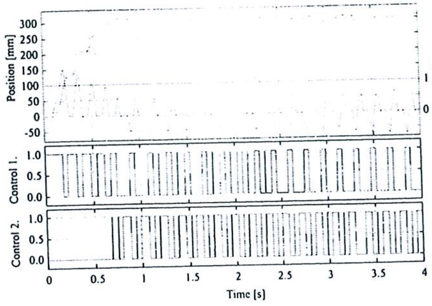

Fig.6. Time functions position and control with conventional on-off valves

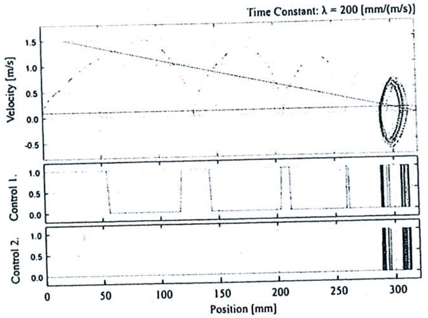

Fig.7. Phase plane trajectory with conventional on-off valves

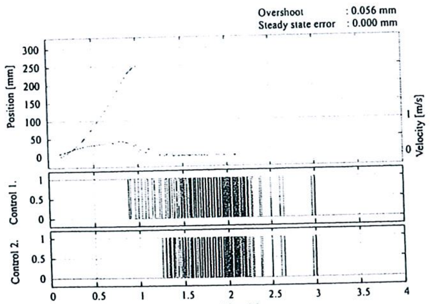

Fig.8. Time functions position and control with fast switching on-off valves

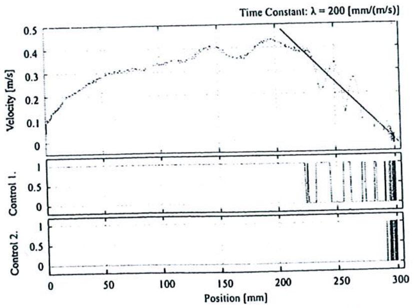

Fig.9. Phase plane trajectory with fast switching on-off valves

5. CONCLUSIONS AND FUTURE WORK

Based on the laboratory measurements we can conclude that the pneumatic servo-systems can be used for precise robust position control, not only movement between two hard stops. The sliding mode control is a promising tool for controlling such systems. The proposed double line and modified saturation function can eliminate the chattering, which is the main problem in case of sliding mode control.

A video can be downloaded from the official web page [11] of International Student Experimental Hands-on Project Competition via Internet on Intelligent Mechatronics and Automation.

Further works we have done with applying the input shaping method. Once the system has reached the setpoint, the residual oscillation will degrade positioning accuracy and may cause a delay in task completion. Input Shaping is a feedforward control technique for reducing vibrations in computer controlled machines. The method works by creating a command signal that cancels its own vibration. That is, vibration caused by the first part of the command signal is canceled by vibration caused by the second part of the command. Input shaping is a command generation technique that is used to reduce command-induced vibration (as opposed to disturbance-induced vibration) [12]. Input shaping is implemented by convolving a sequence of impulses, called an input shaper.

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FLOUR QUALITY AND WHEAT KERNEL HARDNESS CONNECTION

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SUMMARY

Wheat kernel hardness is a very important inheritable parameter; it determines quality, flour yield, flour particle-size, water absorption and other quality characteristics. Besides, kernel hardness has great effect on the resulting flour's baking properties, too (Békési, 2001). The relationship between wheat protein content and kernel texture is usually positive and kernel texture (hardness) influences the grinding energy. Hard Wheat grains require more grinding energy (e_g) than Soft one (Véha, Gyimes 1999).
The aim of our research was to determine the possible relationship between kernel hardness and various other parameters of the flour (dough visco-elastic characteristics, wet gluten, water absorption, flour recovery, alveograph-traits).
We used Perten SKCS 4100 to determine the kernel hardness, while the Perten 3303 mill was used to establish the grinding energy (e_g). Registered and widely used Hungarian wheat varieties (7 of HRWW and 4 of SRWW) were examined. Twin correlations were used to determine the relationship among the various traits. According to the results, there is a very strong correlation between the e_g and the kernel hardness ($r=0.991$). The correlations between hardness index and the examined flour parameters were also significant. We found strong correlation between the e_g and water absorption of the flour. The associations found in this study will help to better understand the technological aspects of wheat and flour quality as well as provide useful information the breeders to develop new, high quality hard or soft wheat varieties.

1. INTRODUCTION

The kernel hardness has great effect on the baking properties of the resulting flour. Flour, which is made from hard wheat generally have a medium to high protein content and stronger gluten than that, which is made from soft one. The Hardness-locus on chromosome 5D is the main determinant of grain texture in bread wheat. Puroindoline-a (pin-a), puroindoline-b (pin-b) and Grain Softness Protein (GSP) genes are tightly linked at this locus and their products are the predominant components of friabilin, a 15 kD endosperm protein complex. The friabilin protein complex determines the kernel hardness. Generally, when the amount of the friabilin is high, the kernel hardness is soft and reverse (Ácsné, 2001). We can sort the kernel hardness in these two groups. Hardness in wheat is largely controlled by genetic factors but it can be affected by the environment, for example the weather (Gyimes, 2004). The transgenic expression of wild type Pin-a sequence in the Pin-a null genotype gave soft grain with the characteristics of soft wheat including stronger starch bound friabilin.