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Control via Input Shaping of a Pneumatic Crane System

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Abstract— **The application of an input shaped controller to a pneumatic crane system is the subject of this paper. Precise positioning of the crane and the suppression of the load-swing during and after the system's movement dictate the need for careful tuning of the shaper's parameters. The shaped system's performance is contrasted in experimental studies with that of a classical On/Off controlled case.**

I. INTRODUCTION

Fast and swing-free transfer of a suspended load in crane systems is a difficult procedure in the presence of system uncertainties and disturbances. The system's performance deteriorates from the underdamped nature of the pendulum-like swinging motion of the payload. The natural frequency of this motion depends on the load mass and the length of the suspending "rope".

In order to achieve fast and precise payload positioning several researches have developed various control techniques for crane systems.

Fang, Dixon, Dawson and Zegeroglou (2001) proposed a simple PD controller to asymptotically regulate a crane system and two nonlinear controllers increasing the coupling between the planar gantry position and the payload angle. A simple output feedback PD controller is also proposed by Kiss, Levine and Mullhaupt (2000) for nonlinear cranes. Yu, Lewis and Huang (1995) utilised a time-scale separation approach to control a two-degree-of-freedom overhead crane system where a linearized model has been introduced to facilitate the construction of the error system. Yoshida and Kawabe (1992) proposed a saturating control law based on a guaranteed cost control method for a linearized version of the two-degree-of-freedom crane system dynamics.

In most of these research approaches, the primary guaranteed performance is related to achieving zero swing at the end of the transport. This fundamental consideration is pointed out in this paper where an effective approach to obtain precise final positioning and the swing suppression throughout the load movement is presented, based on the input shaping technique (Kapila, Tzes and Yan 2000).

The modelled crane-pendulum system consists of a double acting pneumatic long cylinder with a carrier bracket playing the role of the trolley from which the pendulum-load is suspended.

The input shaping technique is a simple and effective method for reducing the residual vibrations of linear systems and is well known in the area of flexible-link manipulator control (Tzes, Englehart and Yurkovich 1989) where several researchers have examined the reduction of oscillations via the usage of this input preshaping. An adaptive version of this precompensator has been implemented by Tzes and Yurkovich (1993) by combining a frequency domain identification scheme resulting in the generation of a sequence of impulses at prespecified time instants that produced a vibration free output. Rhim, Hu, Sadegh and Book (2001) introduced a multirate repetitive learning controller in conjunction with a command shaping method for discrete time joint control of a single flexible link manipulator containing also nonlinearities. The input shaping for vibration-free positioning of flexible systems has been examined by M. Sahinkaya (2001), where a continuous and differentiable function is introduced to define the desired motion and then the input is shaped by inverse dynamic analysis, while other researchers (see for example Petropoulos, *et al*, 2004) have reported various approaches indicating that the input shaping method for controlling vibration in load positioning systems remains an active research area.

This paper is organized in the following manner. In Section II the mathematical system's model is constructed, while the design of the Input Shaping precompensator is analyzed in Section III. The utilized experimental setup is presented in Section IV. In Section V the parameter identification and the controller design methodology is given while primitive experimental results are presented. Finally, conclusions are drawn in the last Section VI.

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II. SYSTEM MODEL

A. The Pneumatic System Model

Consider a pneumatic cylinder-valve system of the type shown in Fig.1, where the carrier bracket with mass M plays the role of the crane trolley from which the load, with mass m, has been suspended through a weightless "rope" of constant length ℓ. The moving pendulum's dynamic equations are:

$$(M+m)\ddot{x}+B_L\dot{x}+m\ell(\cos(\vartheta)\ddot{\vartheta}-\sin(\vartheta)\dot{\vartheta}^2)+F_f=F_L \quad (1)$$

$$m\ell^2\ddot{\vartheta}+B_r\dot{\vartheta}+mg\ell\sin(\vartheta)+m\ell\cos(\vartheta)\ddot{x}=0$$

where F_L is the applied force on the carrier bracket in both directions : $F_L=(P_1-P_2)A$, M is the total mass of the carrier bracket, m the pendulum's mass, x the displacement of the carrier, B_L the viscous friction coefficient of the cylinder, ℓ is the length of the pendulum, F_f the Coulomb friction force, F_L the actuator force applied on the carrier bracket, ϑ is the angular position of the pendulum, B_r the joint viscous friction coefficient, P_1 and P_2 the absolute pressures inside the cylinder champers, with A the energetic area of the cylinder's diaphragm.

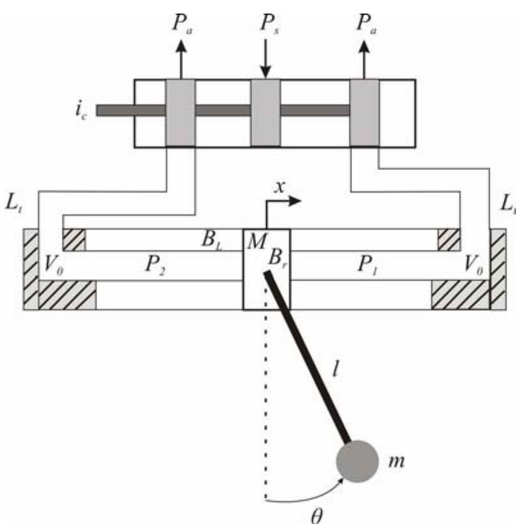

Fig.1. Cylinder-valve system

The electro-pneumatic valve is a significant component of the pneumatic system since its rapid response, affects the kinematic behavior of the carrier and the position accuracy. An equation for the spool motion has been derived by Richer and Hurmuzlu (2000). In our experiments, due to the on-off nature of the "fast" values, the valve dynamics can be neglected.

In order to complete the mathematical model of the pneumatic equipment we also need to express the pressure in two champers of the cylinder as a function of the in-out air flow and the carrier bracket position.

Liu and Bobrow (1988), Ben-Dov and Salcudeam (1995) and Scavarda (1996) considered the procedure of pressing and exhausting the two cylinder champers as an adiabatic. Ibrahim and Otis (1992) pointed out that the temperature inside the cylinder champers is varied between the adiabatic and isothermal boundary curves.

Assuming that the air pressure and temperature of the cylinder champers are uniform, the diferential equations for the two pressures are,

$$\dot{P}_1=\frac{C_f R\sqrt{T}}{V_0+A\left(\frac{1}{2}L+x\right)}[a_{in}A_{v1in}P_s\dot{m}_r(P_s,P_1)-a_{ex}A_{v1ex}P_1\dot{m}_r(P_{1,}P_a)]-a\frac{P_1A}{V_0+A(\frac{1}{2}L+x)}\dot{x} \quad (2)$$

$$\dot{P}_2=\frac{C_f R\sqrt{T}}{V_0+A\left(\frac{1}{2}L-x\right)}[a_{in}A_{v2in}P_s\dot{m}_r(P_s,P_2)-a_{ex}A_{v2ex}P_2\dot{m}_r(P_{2,}P_a)]+a\frac{P_2A}{V_0+A(\frac{1}{2}L-x)}\dot{x}$$

where C_f a non-dimensional valve discharge coefficient, R is the ideal gas constant, T is the air temperature inside the chambers, L is the useful piston stroke, a_{in} and a_{ex} are the heat transfer coefficients for the charging and discharging processes respectively, V_o the dead volume at the end of the stroke, A the effective piston area where the expression for $\dot{m}_r(P_u,P_d)$ is derived by Richer and Hurmuzlu (2000). Note that since we use on-off valves, the terms A_{vin}, A_{viex}, i=1,2 in Eq. 2 can take only one of the distinct values {0 or πR^2_v}, where R_v is the radius of the valve effective area.

B. Control problem statement

The goal is to transfer the suspended load as faster as possible from an initial position x_i to a final position x_f, moving along an one-dimension trajectory and keeping the load swing angle ϑ as small as possible both during transfer and after the desired location is reached. Fig. 2 shows the schematic diagram of the pneumatic control system with the measuring and computing components.

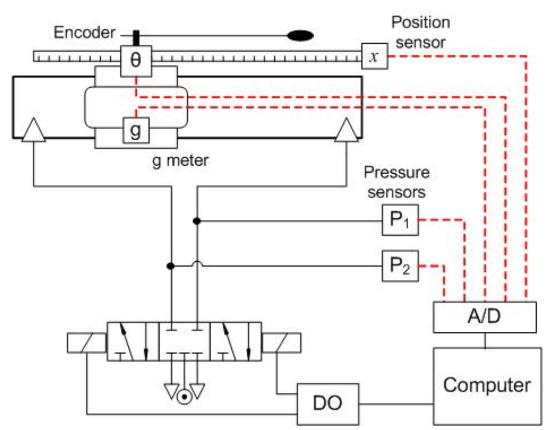

Fig.2. Long pneumatic cylinder with pendulum system

According to the assumption that the dynamic behavior of the pneumatic cylinder and valve components is faster compared to that of the moved pendulum-load, the applied differential pressure $P_1 - P_2$ creates a force F_L based on the simple analog relation $F_L(t) = \delta(P_1(t) - P_2(t))$.

Due to our consideration of small rotations of the pendulum around the equilibrium point $\vartheta \cong 0$, this leads to a linearized system which has low damped eigenvalues

III. Input Shaping Precompensator Design

The application of an input precompensation scheme for vibration suppression in load positioning systems corresponds to a feed-forward term that convolves in real time the desired reference input with a sequence of impulses, as symbolically illustrated in Fig.3, and produces a vibration-free output. The input shaping technique comes from control theory of second-order dynamic systems,

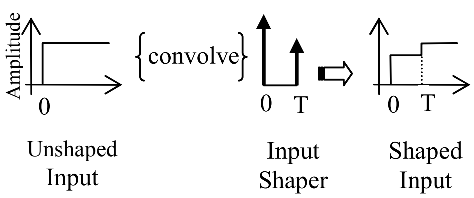

Fig.3. Schematic of input shaping process

whose natural frequencies and damping ratio are used to determine the proper input shaper parameters. The natural frequencies ω_i and damping ratio ζ_i of the system $G(s)$ are first identified along the desired trajectory. Subsequently, these data are used to compute an appropriate input shaper which is simply a sequence of impulses. These impulses are then convolved with the desired input to yield the shaped input. The magnitudes $A_j, j=1,\dots,N_1$ of the impulses and the time instants t_j of their application can be computed from the following expressions

$$A_j = \frac{\binom{N-1}{j-1} K^{j-1}}{\sum_{i=0}^{N-1} \binom{N-1}{j-1} K^{j-1}}, \quad t_j = (j-1)\frac{\pi}{\omega_1\sqrt{1-\zeta_1^2}}, \quad K = e^{-\zeta_1\pi/\sqrt{1-\zeta_1^2}} \tag{3}$$

If the magnitudes and the time instants of these impulses are suitably selected then the system response will be free of vibrations. In other words, the command to the system is filtered by an input shaper, dependent on the linearized parameters of the system. The robustness of such a controlled system against variations of system's parameters may be elementary but can be improved by convolving the input with a longer sequence of impulses.

In our case study, due to the utilized on-off pneumatic valves, a Unity Magnitude Shaper (UMS), is appropriate. Since a closed form expression for the time instances of UM-Shapers has not yet been extracted, an approximate expression for highly underdamped systems has been derived by Pao and Singhose (1996) computing the impulse train time intervals with respect to ζ_1 and ω_1 for the design of Zero Vibration (ZV), Zero Vibration and Derivative (ZVD) and Extra Insensitive (EI) Shapers.

IV. Pneumatic Crane Experimental Setup

In pneumatics, valves have a significant influence on the behaviour of open loop and closed loop systems. The nonlinearities of these valves influence not only the steady state response of the system, but have profound impact on the transient response. The pneumatic experimental system consists of a pneumatic cylinder 1.2 m long, two controlled solenoid electro-valves, a PC-based measurement system comprised of four sensors, and a source of compressed air, as shown in Fig.4.

The cylinder of Joucomatic Ltd. is a double-acting one with a carrier bracket on which a pendulum has been suspended. Each direction of motion is selected via appropriate actuation of the two 5/3 way electro-valves, which convert the electrical signal to on-off air flow. The maximum speed of 4m/sec can be achieved by the use of the cylinder chosen, when fed with maximum pressure of 8 bars. The active displacement of the cylinder's carrier bracket (stroke) is 1.0 m and its bore 25mm. It should be noted that the lag induced to the system dynamics due to the 0.5m tubes connecting the valves with the cylinder (approximately 1-2msec) has been neglected.

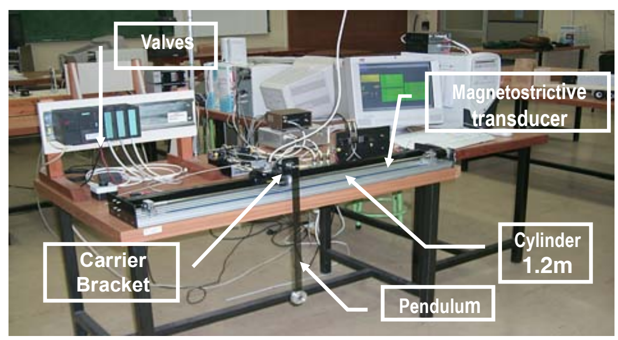

Fig.4. An overall view of the experimental set-up.

The instrument for carrier bracket position measuring is the Schaevitz MagneRule Plus magnetostrictive linear displacement transducer. The measurement system also includes an accelerometer located on the carrier bracket, an encoder located on the pendulum joint for measuring the swing angle and two pressure sensors. The encoder measures movements with a resolution of 500 pulses per rotation, resulting in measuring a deviation of the load with accuracy equal to 0.0125rads. The data acquisition relies on hardware (2xPCI 6024E DAQ) from National Instruments (NI) and the utilized software platform was NI's LabView.

V. Parameter Identification and Controller Design

Prior to the controller design the system's parameters are identified through experimental measurements. The oscillation period of the pendulum was measured for various initial angles, resulting in a constant value for small initial angles and a nonlinear function for large ones. The position, velocity and acceleration of the carrier bracket and the pressures in the cylinder champers are measured during a pulse response in order to estimate the system's behavior from a qualitative point of view. Subsequently, the dc-gain of the system is determined by measuring the final position of the crane trolley as a function of the initial position and the excitation duration. The measurements confirmed the strong nonlinear nature of the system.

A. On-off controller

The simplest controller is the primitive on-off controller shown in Fig. 5. The crane trolley reaches three different desired positions and swings around them, while the pendulum's response shows intense undesirable swings which at the worst case reach forty degrees as shown in Fig. 6.

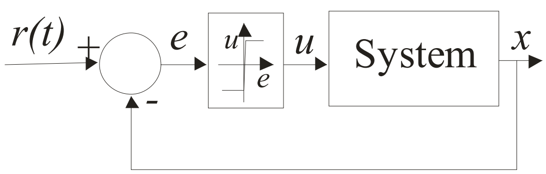

Fig.5. On-off controller

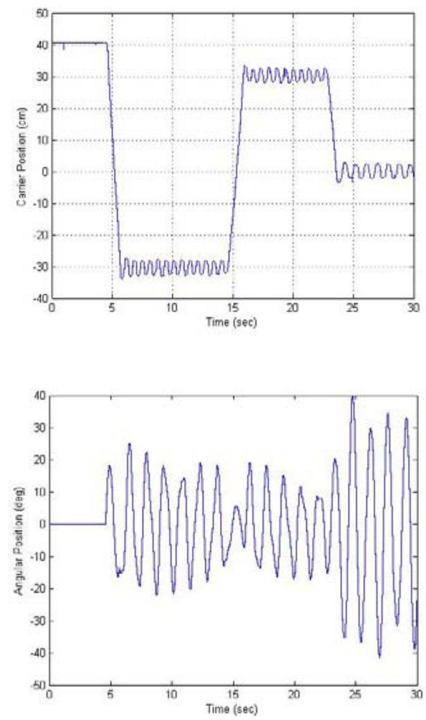

Fig.6. Responses of the carrier position and the pendulum's angle based on on-off control.

The carrier's oscillation is due to the delayed response of the carrier bracket in regard to the excitation time instant.

B. Input Shaped Controller

The second controller that has been experimentally tested is the UMS. The control scheme is shown in Fig.7. The reference signal (carrier desired position) and the initial position of the carrier are used in order to determine the duration that the valve will be "on". Then this signal is convolved in real-time with the impulse sequence in order to provide the input to the system.

The time intervals of the shaper precompensator are derived according to the UM-Shaping technique described in Section III. The mode we want to suppress is the oscillation of the pendulum, thus the utilized natural frequency and damping factor are those measured by the free response of the pendulum.

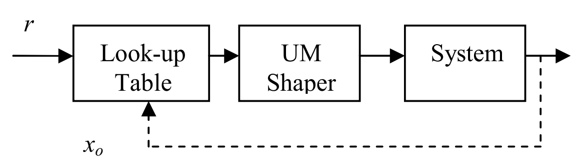

Fig. 7. Input Shaping of the pneumatic crane–pendulum system.

Since the pneumatic system is from its nature highly nonlinear we need to increase the robustness of the Input Shaper against variations of ζ_1 and ω_1. We experimented with two different pendulum masses (100gr and 1kgr) and two kinds of shapers (ZV and ZVD).

In Figures 8 and 9 we present the system's response (carrier position and pendulum's angle) when implementing ZV and ZVD Unity Magnitude Shapers (Input Shaping Control-ISC) respectively for a load mass of $m = 100gr$, while Figs. 10 and 11 are for a load mass $m = 1kgr$.

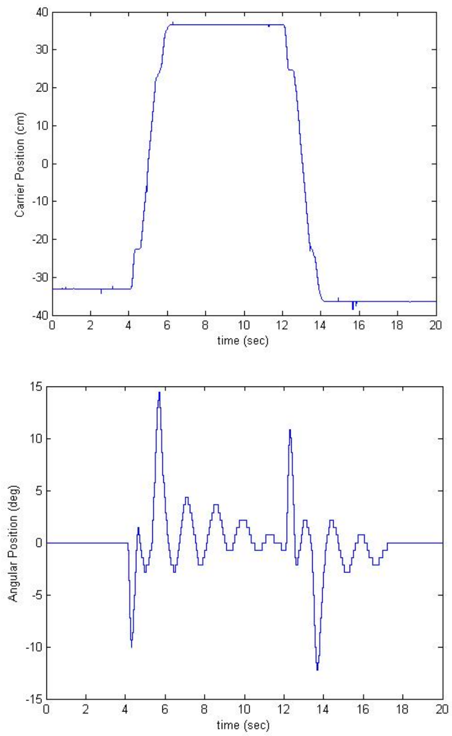

Fig.8. Responses of the trolley position and the pendulum's angle based on ZV ISC (m = 100gr)

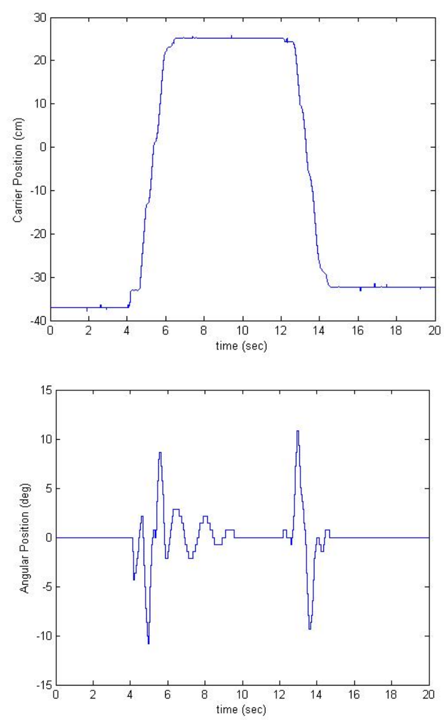

Fig.9. Responses of the trolley position and pendulum's angle based on ZVD ISC (m = 100gr)

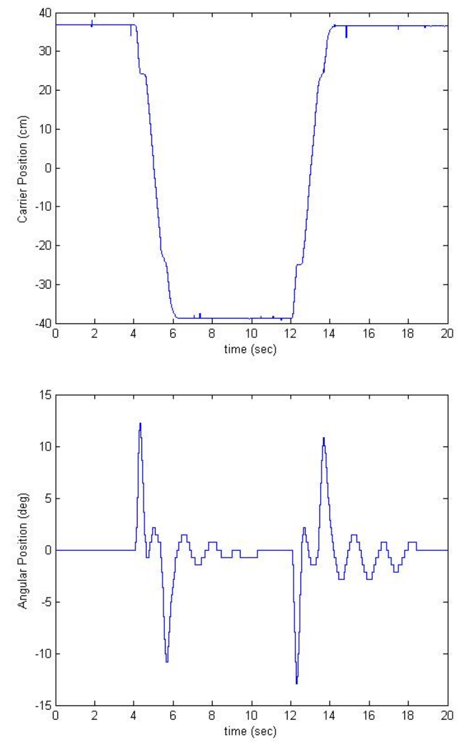

Fig.10. Responses of the trolley position and pendulum's angle based on ZV ISC (m = 1kgr)

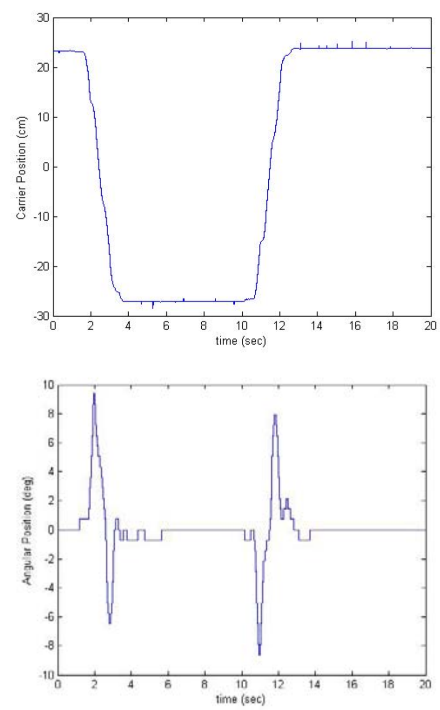

Fig.11. Responses of the trolley position and pendulum's angle based on ZVD ISC (m = 1kgr)

It is apparent from the previous results that not only exact positioning of the carrier is achieved, but the pendulum's oscillation is almost eliminated just a few seconds after the command was given. Note that the peak amplitude of the pendulum's response is reduced with the ZVD than with the ZV Shapers since the first is more robust to modelling inaccuracies, which exist in our case due to the nonlinearities of the cylinder's dynamics.

VI. Conclusions

In this paper the control problem of the position and the swing of a pneumatic pendulum system based on the Input Shaping technique was investigated. The input shaped system was contrasted in experimental studies with an on-off controlled system. The obtained experimental results prove the efficacy of the suggested scheme.

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