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Design and experimental evaluation of an electrohydraulic vibration shaker

Thanh Danh Le[1*] and Van Nguyen Ho[2]

[1] Industrial University of Ho Chi Minh City, 12 Nguyen Van Bao Street-Go Vap District-Ho Chi Minh City, Viet Nam.
[2]Vocational college of Tay Ninh Province, 19 Vo Thi Sau Street – Tay Ninh City, Tay Ninh Province, Viet Nam

[*]*Corresponding author E-amil: lethanhdanh@iuh.edu.vn*

ABSTRACT: The main objective of this paper is to experimentally evaluate of a low frequency vibration exciter featured open-loop hydraulic transmission which can produce a harmonic signal having frequency smaller than 5Hz through controlling the proportional valve. The first work is that the dynamic model of the system is built based on the basic principle of the hydraulic power. Due to the nonlinear dynamic characteristic and unknown system parameters, it is difficult to design a model-based controller. Hence, the next work is that an intelligent controller which is a combination between the sliding mode and fuzzy logic control algorithm is introduced. The proposed controller's feature is that the fuzzy rules are designed according to the Lyapunov stable criterion and the chattering phenomenon from the conventional sliding mode control algorithm is removed. Then, the experimental apparatus is set up to evaluate the control performance of the shaker. The result confirmed that the control performance of the shaker tracks well the desirable trajectory.

KEYWORDS: Shaker, Hydraulic transmission, Sliding mode control, Fuzzy logic control

INTRODUCTION

In general, vibration shaker plays an important role in a vibration testing system because of the generation of desirable vibration motion [1]. Hence, the vibration exciter has been studied and used widely in many fields such as vibration experimentation, earthquake's impact evaluation, structural testing, dynamic testing, etc. A group researchers designed and experimented a mechanical vibration exciter in which the unbalanced mass is used to generate uniaxial vibrations [2]. Another mechanical vibration shaker using eccentricity mechanism had been designed and tested, showing that the vibration testing can be realized with reasonable results [3]. An employed a cam and follower mechanism for constructing a mechanical exciter which can produce the displacement through a given range of frequency [4]. Besides, electromagnetic exciters have been studied and designed obtaining the resonant frequency of 80Hz and the excitation force of 420 [5]. Another group researchers built successfully the mathematical model of an electromagnetic vibratory exciter, which describes the mechanical movement and electrical behavior [6]. Some researchers carried out the simulation and experiment of a high damped electrodynamic shaker which has been modeled as a two degrees of freedom mechanical lumped element system coupled to an electrodynamic transducer [7].

In addition, hydraulic or pneumatic transmission systems have been studied to generate derivable vibration signal. As well known, the hydraulic exciter can offer the vibration motion having larger force and displacement than mechanical or electromagnetic exciters [8,9]. By using the rotation type directional control valve, a hydraulic vibration exciter was studied [10]. In this structure, rotational movement of the vale is performed by the hydraulic motor. Up to now, the hydraulic exciter has been developed as the commercial products in [http://www.econ-group.com/product/?id=49].

However, the vibration exciter featured with hydrostatic transmission system inherits strongly the nonlinear and time-varying dynamics, indicating that it can cause difficulties for applying the simple control algorithms such as proportional-integral-derivative (PID). Hence, modern control algorithms were proposed for monitoring and controlling model having dynamic variation. Choi designed a robust tracking controller for the hydraulic system in which an update law for the controller is realized based on the reconstruction error [11]. The adaptive robust control algorithm guaranteeing stability and high performance of the hydraulic system in spite of the nonlinearity and uncertainties was introduced [12]. A group researchers proposed a robust decentralized controller for SISI and MIMO systems [13]. In addition, the sliding mode control algorithm is one of the useful approaches for solving the nonlinear systems [14]. But the drawback of the conventional sliding mode control method is to need an accurate dynamic model of the system [15]. In order to solve these disadvantages, some control strategies have been proposed, for example, an adaptive time-varying sliding controller for hydraulic servo system [16]. A researcher used self-tuning control for a low friction pneumatic actuator under the influence of gravity [17]. Guan et al. proposed and experimented successfully an adaptive sliding mode controller for electro-hydraulic system [18]. Acarman et al. proposed a feedback-linearization control strategy with consideration of various status of the chamber pressure in the system model [19]. In addition, Fuzzy control technique is also considered as a good tool for the nonlinear structures without model-based requirement such as Earth mitigation structure with MR damper studied [20, 21]. However, the design of a traditional fuzzy controller depends on the expert or experience of an operator. That is one of the difficulties for designing a fuzzy logic controller. Therefore, hybrid controllers which combine control algorithms such as fuzzy logic, sliding mode, neural network…, have been currently proposed by many researchers [22-24].

In this study, the control performance of a low frequency vibration shaker featuring the hydrostatic transmission will be evaluated experimentally. Due to nonlinear dynamic and imprecise dynamic model, the intelligent controller based on the sliding mode and fuzzy logic control algorithm will be studied for the tracking the desirable trajectory. The fuzzy law is designed according to Lyapunov critical theories. The remainder of this paper is organized as follows Dynamic model of the vibration shaker is presented in section 2. The controller is designed in section 3. The experiment is illustrated in section 4. Some conclusions are drawn in section 5.

DYNAMIC EQUATION

3D physical model of the vibration shaker is presented in Fig. 1 in which the load plate is supported by a hydraulic cylinder meanwhile the base is fixed. The cylinder is only moved in the vertical direction through four guidable rods. In order to eliminate sliding friction, four linear bearings are positioned between the guidable rod and base. The linear movement of the hydraulic cylinder tracking the desirable trajectory is carried out by controlling the proportional valve with the hydraulic system presented in Fig. 2. Besides, a position sensor is positioned at the base to measure the displacement of the load plate compared with the base. A PC is used to system monitoring, data acquirement…

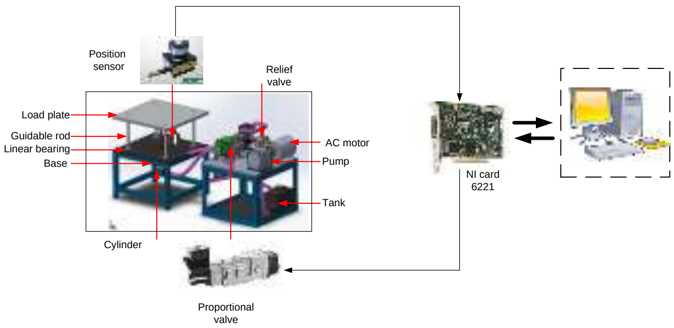

Figure 1. 3D model of the vibration shaker

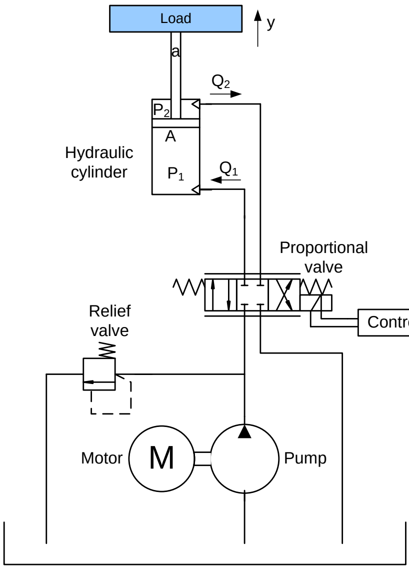

Figure 2. Hydraulic circuit of vibration shaker

By applying the Second Newton's Law, motion equation of the load plate is expressed as below:

$$P_1A - P_2(A-a) - Mg - F_f = M\ddot{y} \tag{1}$$

where A and a are the effectiveness areas of the piston and rod, respectively, M is the weight of the load, F_f is the friction force, y is the displacement of the load and g is the gravity acceleration. The pressure in the working chamber 1 (P_1) and 2 (P_2) is determined as following:

$$\begin{aligned}\dot{P}_1 &= \frac{\beta}{V_{o1} + Ay}\left(Q_{L1} - A\dot{y} - Q_{Lk}\sin g(P_1 - P_2)\right)\\ \dot{P}_2 &= \frac{\beta}{V_{o2} - (A-a)y}\left((A-a)\dot{y} + Q_{Lk}\sin g(P_1 - P_2) - Q_{L2}\right)\end{aligned} \tag{2}$$

with Q_{L1} is the flow rate entering chamber 1 meanwhile Q_{L2} is the flow rate leaving the chamber 2. V_{o1} and V_{o2} are the initial volumes of chamber 1 and 2, respectively. Q_{lk} is the internal leakage flow rate and β is the bulk modulus of the oil.

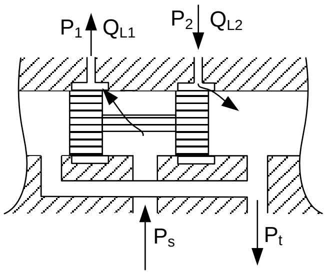

Figure 3. Schematic diagram of the valve

As shown in Fig. 3, where P_s is the supply pressure, P_t is the return pressure and the internal leakage is ignored, the flow rate Q_{L1} and Q_{L2} are defined as following:

$$Q_{L1} = \frac{1+sign(u)}{2}\left(C_d A_{ov}\sqrt{\frac{2(P_s - P_1)}{\rho}}\right) + \frac{1-sign(u)}{2}\left(-C_d A_{ov}\sqrt{\frac{2P_1}{\rho}}\right)$$
$$Q_{L2} = \frac{1+sign(u)}{2}\left(C_d A_{ov}\sqrt{\frac{2P_2}{\rho}}\right) + \frac{1-sign(u)}{2}\left(-C_d A_{ov}\sqrt{\frac{2(P_s - P_2)}{\rho}}\right) \tag{3}$$

in which $A_{vo} = k|u|$ based on zero-lap value assumption, *k* is the proportional gain, ρ is oil density, C_d is the discharge coefficient. u is the current supplying to the coil of the valve.

By introducing $x_1 = y;\ x_2 = \dot{y};\ x_3 = P_1;\ x_4 = P_2$ the dynamic equation of the system is expressed in the form of the state variable space as following:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = \dfrac{A}{M}x_3 - \dfrac{(A-a)}{M}x_4 - \dfrac{F_f}{M} - g \\ \dot{x}_3 = \dfrac{\beta}{V_{o1} + Ax_1}\left(Q_{L1} - Ax_2 - Q_{LK}sign(P_1 - P_2)\right) \\ \dot{x}_4 = \dfrac{\beta}{V_{o1} - (A-a)x_1}\left((A-a)x_2 + Q_{LK}sign(P_1 - P_2) - Q_{L2}\right) \end{cases} \tag{4}$$

The analysis indicates that the state variables of the system can be regulated through changing the opening area of the valve which is controlled by supplying the voltage to the coil. Next section will suggest an intelligent controller for the position tracking control of the cylinder according to the desirable trajectories.

FUZZY-SLIDING MODE CONTROLER

The sliding mode surface is defined as following:

$$s = \ddot{e} + 2\lambda\dot{e} + \lambda^2 e \tag{5}$$

where, λ is the sliding coefficient, $e = y_{ref} - y$ is the position error between the desirable (y_{ref}) and actual trajectory (y).

Taking derivative of the sliding surface obtained as following:

$$\dot{s} = \dddot{e} + 2\lambda\ddot{e} + \lambda^2\dot{e} = f\left(x_1, x_2, x_3, x_4\right) - G\left(x_1, x_3, x_4\right)u \tag{6}$$

By substituting Eq. (3) and (4) into Eq. (5), we have:

$$f\left(x_1,x_2,x_3,x_4\right)=\ddot{y}_{ref}+\frac{A}{M}\frac{\beta}{V_{o1}+Ax_1}\left(Ax_2-Q_{LK}sign(x_3-x_4)\right)$$
$$+\frac{A}{M}\frac{\beta}{V_{o2}-(A-a)x_1}\left(Ax_2+Q_{LK}sign(x_3-x_4)\right)+\frac{\dot{F}_f}{M}$$
$$G\left(x_1,x_3,x_4\right)=kC_d\frac{A}{M}\frac{\beta}{V_{o1}+Ax_1}\left(\begin{array}{l}\frac{1+sign(u)}{2}\sqrt{\frac{2(P_s-x_3)}{\rho}}\\+\frac{1-sign(u)}{2}\sqrt{\frac{2x_3}{\rho}}\end{array}\right)$$
$$+kC_d\frac{A}{M}\frac{\beta}{V_{o2}-(A-a)x_1}\left(\begin{array}{l}\frac{1+sign(u)}{2}\sqrt{\frac{2x_4}{\rho}}\\+\frac{1-sign(u)}{2}\sqrt{\frac{2(P_s-x_4)}{\rho}}\end{array}\right)$$

The sliding surface and time derivative are considered as the input signals of the fuzzy logic controller having the block diagram as shown in Fig. 4 in which k_1, k_2 and k_3 are the scaling factors while u is the control signal (voltage).

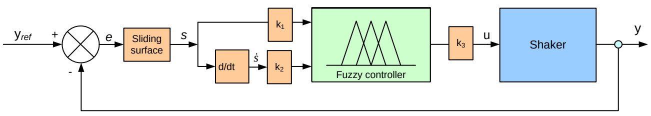

Figure 4. The overall controlling system

The linguistic variables of the input s and $\dot{s}$ and the output (u) signals are defined as following:

Big negative -"BN"; Middle negative - "MN"; Small negative - "SN"; Zero – "Z"; Small positive - "SP"; Middle positive - "MP"; Big positive - "BP"

The triangle type membership function is exploited to describe all of linguistic variables as shown in Fig. 5. The rules can be composed as follows:

Rule i: if s A_i and $\dot{s}$ B_i then u is C_i $\quad i$=1,2,3,…,n

with n is the number of the fuzzy rules; A, B, C are the linguistic variables of the input and output.

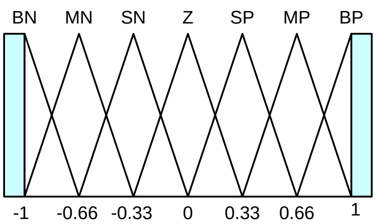

Figure 5. The membership function for input and output signals

Based on the Lyapunov stability theory, the fuzzy laws are built to satisfy the following condition:

$$\dot{V}=s\dot{s}<0 \quad \text{with} \quad V=1/2s^2 \tag{7}$$

As shown in Eq.(6), $\dot{s}$ will increase according to the decrease of the control signal (u) and reversely, growing the control signal will result in an deduction in the time derivative of $\dot{s}$ due to $G\left(x_1,x_3,x_4\right)>0$, obtaining the fuzzy laws listed in table 1.

Table 1: Fuzzy law table

$\dot{s}$ \ s	BN	MN	SN	Z	SP	MP	BP
BN	BN	BN	BN	BN	MN	SN	Z
MN	BN	BN	BN	MN	SN	Z	SP
SN	BN	BN	MN	SN	Z	SP	MP
Z	BN	MN	SN	Z	SP	MP	BP
SP	MN	SN	Z	SP	MP	BP	BP
MP	SN	Z	SP	MP	BP	BP	BP
BP	Z	SP	MP	BP	BP	BP	BP

By applying max-min aggregation method and centroid defuzzication method, the fuzzy reasoning result of the output is expressed as following:

$$u = k_3 \frac{\sum_{k=1}^{49} \mu_k R_k}{\sum_{k=1}^{49} \mu_k} \tag{3}$$

in which μ_k and R_k are the height and the weight of the control output obtained from the kth rule, respectively.

EXPERIMENTAL EVALUATE

The experimental apparatus is shown in Fig. 6. The proportional hydraulic valve manufactured by Yuken with mode EHDFG-04 is used to control the position of the cylinder. The position of the cylinder is measured by a linear sensor with mode RLP50S 1000B. In addition, A NI card 6221 with 68 pins is used as an A/D converter for communicating between the peripheral devices. A computer is employed to acquire the data and generate the output signal to control the proportional valve. The schematic diagram of implementation was depicted in Fig. 2.

Figure 6. Photograph of the hydraulic vibration shaker

The experimental results presented in Fig. 7 in which the desirable trajectory is denoted by the dashed line while the tracking control response is exhibited by the solid line. The sampling time was set at the value of 0.01s for all experiments. In this investigation, the desirable trajectory is sinusoidal signals having the amplitude of 8, 7, 5 and the frequency of 2, 3, 4, 5 Hz. It can be observed that at the beginning time, the position tacking responses cannot track well the references. It is evident because at the beginning, the initial states of the system and the reference are almost different. However, after that approximately 1.3s, the control response of the system is good agreement with the desirable trajectories with an acceptable accuracy. The maximum tracking error of the system is around 0.75mm. This experiment confirms the effectiveness of the low frequency vibration shaker controlled by the proposed controller.

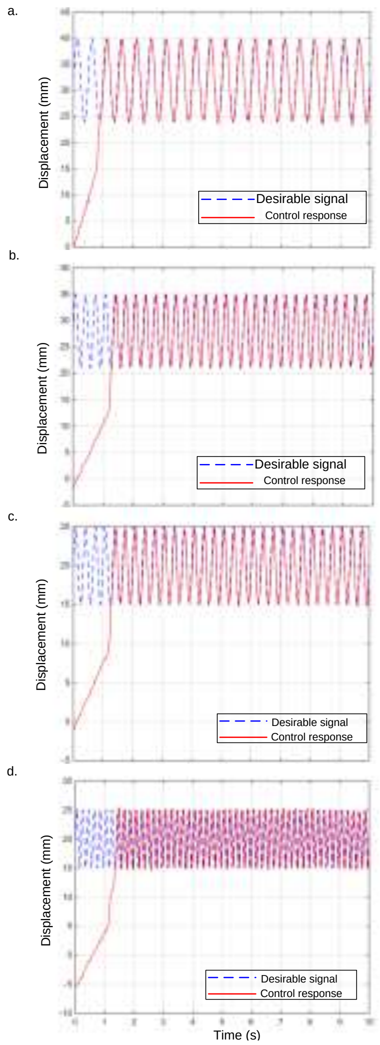

Figure 7. Position tracking performance of the vibration shaker for the sinusoidal signals having the amplitude and frequency: 8mm, 2Hz (a); 7mm, 3Hz (b); 5mm, 4Hz (c) and 5mm, 5Hz (d).

CONCLUSION

Based on basic principles of the hydraulic transmission, a dynamic model of the vibration shaker featuring the hydrostatic transmission was described in the form of the state variable space. From this dynamic model, an intelligent controller for controlling the proportional hydraulic valve to generate wanted displacement of the load plate including the amplitude and frequency was proposed successfully. Control law was established based on the Lyapunov stability theory. Finally, an experimental apparatus was built to evaluate the capacity of the vibrated generation of the system in low frequency areas. The results indicated that the system can generate the low frequency vibration signals with acceptable accurate. The time achieving the steady-state is about 1.3s.

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