ISA Transactions ∎ (∎∎∎∎) ∎∎∎–∎∎∎

Contents lists available at ScienceDirect

ISA Transactions

journal homepage: www.elsevier.com/locate/isatrans

Research article

Reduced-order model based active disturbance rejection control of hydraulic servo system with singular value perturbation theory

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ARTICLE INFO

Article history:
Received 21 April 2016
Received in revised form
13 September 2016
Accepted 5 January 2017

Keywords:
Active disturbance rejection control (ADRC)
Singular value perturbation theory
Hydraulic servomechanism
Reduced-order model
Mismatched uncertainties input

ABSTRACT

Hydraulic servomechanism is the typical mechanical/hydraulic double-dynamics coupling system with the high stiffness control and mismatched uncertainties input problems, which hinder direct applications of many advanced control approaches in the hydraulic servo fields. In this paper, by introducing the singular value perturbation theory, the original double-dynamics coupling model of the hydraulic servomechanism was reduced to a integral chain system. So that, the popular ADRC (active disturbance rejection control) technology could be directly applied to the reduced system. In addition, the high stiffness control and mismatched uncertainties input problems are avoided. The validity of the simplified model is analyzed and proven theoretically. The standard linear ADRC algorithm is then developed based on the obtained reduced-order model. Extensive comparative co-simulations and experiments are carried out to illustrate the effectiveness of the proposed method.

1. Introduction

Owning to the merits of small size-to-power ratios, hydraulic system is capable of handling large inertia and heavy loads with high accuracy and rapid response [1]. So far, they have been widely employed in many industrial applications. For example, they are important in the industrial sites such as the pitch system of turbine [2], robotic arm [3], mobile vehicle [4], mining [5], shaking-table [6] and flight testing fields [7], etc. In view of these facts, how to achieve high performance control of hydraulic servomechanism has attracted great attention of researches in both academia and industries over the last decade.

As a result of these problems, many experts and scholars have been focusing on the high-precision servo control of hydraulic system in the last dozen years. Up to date, there have been a number of elegant approaches contributing to this fields. The feedback linearization technology is an effective tools to address the nonlinearity of hydraulic system, and this approach have been well-studied in many literatures, for instance [8–11], etc. To cope with the parametric uncertainties problem, the adaptive control approach was introduced to the hydraulic systems, for example, as reported by [12–14]. In addition to the problems of nonlinearities and parametric uncertainties, the un-modeled dynamics and random external disturbances are also the main obstacles that need to be overcome. To solve these issues, the adaptive robust control (ARC) was developed by Yao [15]. Since the ARC method combines the merits of adaptive control and deterministic robust technology, it is able to handle the problems of parametric uncertainties, un-model dynamics, and external disturbances comprehensively. So far, ARC approach and its derivatives have been widely applied to several hydraulic systems, such as in [16–19]. Although these control technologies are effective on improving control performances, they all rely on the mathematical model of controlled plant to some extent, or the knowledge of uncertainties/disturbances bonds in advance.

Aiming to overcome the weakness of model dependence, Han [20] originally proposed the ADRC (active disturbance rejection control) technology in the late 1990s. The essence of ADRC approach is that all the modeling/parametric uncertainties and unknown external disturbances, no matter they arise from the plant itself or external environments, are lumped together as the so called "generalized disturbances". Good tracking precision can be obtained if the "generalized disturbances" could be estimated and compensated on-line in real-time. Its simplicity in engineering implementation and superior performance in dealing with a vast range of uncertainties have attracted several industrial companies, such as Parker Hannifin Extrusion Plant and Texas Instrument [21]. Besides that, the ADRC

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http://dx.doi.org/10.1016/j.isatra.2017.01.009
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technology has been successfully applied in many industrial control systems, e.g., the flywheel energy storage system [22], piezoelectric-actuator active-vibration system [23], pneumatic servo system [24], Nano-positioning Piezostage system [25], electromagnetic servo system [26], temperature regulation system [27], air–fuel ratio control system [28], two-mass drive system [29], etc.

Despite these numerous ADRC-based applications, there is few reports on the ADRC approach for hydraulic system. The primarily reason restricting the application of ADRC technology to hydraulic systems comes from the fact that the standard extended state observer (ESO) based ADRC approach is only applicable to the integral chain systems with matched uncertainties system [30]. Whereas the hydraulic systems is a typical mechanical/hydraulic double-dynamics coupling system with mismatched uncertainties input. Recently, Gao et al. [31] applied the ADRC technology to the hydraulic servomechanism with identification model of the controlled plant. In this study, extensive off-line model-distinguish experiments have to be performed to obtain the suitable model applicable to the technology in advance.

Although the idea of applying the ADRC approach to hydraulic servomechanism itself is not entirely new, given the important position of hydraulic system in industry fields, how to apply the ADRC technology to hydraulic servomechanism in a more direct and convenient way requires further investigation. In addition, high stiffness control and mismatched uncertainties input problems are still open problems in the hydraulic servo fields.

In this study, we develop an ADRC algorithm for the hydraulic valve-controlled position tracking (HVCPT) system without resorting to the identification model. The main contributions of this paper are as follows. We develop a practical framework to apply the standard ADRC technology to hydraulic systems. Meanwhile, this paper provides a solution to the problems of high-stiffness control and mismatched uncertainties existing in all the hydraulic servomechanism. Moreover, the stability proof of the internal states (i.e., the chambers' pressures of hydraulic actuator) is achieved. Basically, the aims of this paper are twofold: to achieve the high-performance tracking control for the HVCPT system by employing ADRC approach and to explore a simple implementation process of the advanced control technologies for the hydraulic servo systems.

To this end, the original model of the studied HVCPT system is built and analyzed firstly. Then, we obtain the reduced-order model by using singular value perturbation theory. Finally, the linear ADRC algorithm is directly carried out based on the simplified model. It is shown that, by means of singular value perturbation theory, the original mechanical-hydraulic dynamics coupling model of the HVCPT system can be regarded as the standard two-order integral chain system with matched uncertainties input. Therefore, the standard ADRC technology can be directly applied.

The remainder of the paper is organized as follows. Section 2 develops the full-order model of the HVCPT system. In Section 3, the original model is simplified. The justification of the model simplification process is analyzed with the singular value perturbation theory. Then, the standard linear ADRC control law is developed based on the obtained reduced-order model in Section 4. Section 5 gives the simulation and experiment results. Finally, conclusions are drawn in Section 6.

2. System architecture and full-order model

The schematic structure of the studied HVCPT system is illustrated in Fig. 1. The dynamics of inertia load can be described by

$$m\ddot{x}_p = A_1P_1 - A_2P_2 - b\dot{x}_p - f_c(x, \dot{x}, t) - f_d(t). \tag{1}$$

where m is load mass; x_p is load position; $\dot{x}_p$ is load velocity; P_1 and P_2 denote pressures inside the two chambers of cylinder; A_1 and A_2 are the ram areas of two chambers; b denotes the viscous friction coefficient, and $f_c(x,\dot{x},t)$ represents the un-modeled friction dynamics except the viscous friction force; $f_d(t)$ stands for the unknown external load and uncertainties of the mechanical dynamics.

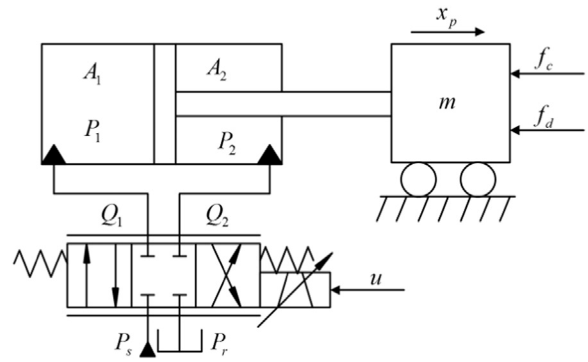

Fig. 1. Architecture of the discussed system.

Neglecting the external leakage, the actuator's flow continuity equation can be established as

$$\begin{aligned} \beta_e^{-1}\dot{P}_1 &= V_1^{-1}(t)\left[Q_1 - A_1\dot{x}_p - C_i(P_1 - P_2)\right], \\ \beta_e^{-1}\dot{P}_2 &= V_2^{-1}(t)\left[A_2\dot{x}_p - Q_2 + C_i(P_1 - P_2)\right]. \end{aligned} \tag{2}$$

where $V_1(t) = V_{10}+A_1x_p$ is the total control volume of the piston chamber, and $V_2(t) = V_{20}-A_2x_p$ is the total control volume of the rod chamber; V_{10} and V_{20} are the initial control volumes of the two chambers, respectively; β_e denotes the oil effective bulk modulus; Q_1 and Q_2 are the supplied flow rate into the piston chamber and return flow rate from the rod chamber, respectively; C_i is the internal leakage coefficient of the actuator.

Define function $S(*)$ as $S(*)=\begin{cases}1, & *\geq 0\\ 0, & *<0\end{cases}$. Then, the flow rate modulated by servo valve can be described by

$$\begin{aligned} Q_1 &= \alpha x_v\left[S(x_v)\sqrt{P_s - P_1} + S(-x_v)\sqrt{P_1 - P_r}\right], \\ Q_2 &= \alpha x_v\left[S(x_v)\sqrt{P_2 - P_r} + S(-x_v)\sqrt{P_s - P_2}\right]. \end{aligned} \tag{3}$$

where Q_1 and Q_2 are the supplied flow rate into piston chamber, and the return flow rate from rod chamber; $\alpha = C_d\omega\sqrt{2/\rho}$; C_d and ω are the flow coefficient and area gradient of servo valve; ρ is oil density, and x_v is the spool displacement; P_s and P_r are the supply pressure and return pressure, respectively.

Considering the response dynamics of servo valve (Moog D634-513) used in the study is much higher than actuator's operation frequency band, the linear model of valve is used

$$x_v = k_{xv}u. \tag{4}$$

where k_{xv} is valve gain and u is control output.

For simplicity, define $\gamma = \alpha\cdot k_{xv}$, $R_1=S(u)\cdot\sqrt{P_s - P_1}+S(-u)\cdot\sqrt{P_1 - P_r}$, $R_2=S(u)\cdot\sqrt{P_2 - P_r}+S(-u)\cdot\sqrt{P_s - P_2}$. Combining (4), we have

$$\begin{aligned} Q_1 &= \gamma\cdot R_1\cdot u, \\ Q_2 &= \gamma\cdot R_2\cdot u. \end{aligned} \tag{5}$$

From (1) to (5), the considered system can be described by (6)

$$\begin{cases} \ddot{x}_p = \left[A_1P_1 - A_2P_2 - b\dot{x}_p - f_c(x_p, \dot{x}_p, t) - f_d(t)\right]/m \\ \beta_e^{-1}\dot{P}_1 = \left[\gamma R_1 u - A_1\dot{x}_p - C_i(P_1 - P_2)\right]/V_1 \\ \beta_e^{-1}\dot{P}_2 = \left[A_2\dot{x}_p - \gamma R_2 u + C_i(P_1 - P_2)\right]/V_2 \end{cases} \tag{6}$$

For system (6), our goal is to generate suitable control output u such that the inertia load position x_p can track the specified motion

Please cite this article as: Wang C, et al. Reduced-order model based active disturbance rejection control of hydraulic servo system with singular value perturbation theory. ISA Transactions (2017), http://dx.doi.org/10.1016/j.isatra.2017.01.009