Robust controller design of the integrated direct drive volume control architecture for steering systems

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ABSTRACT

Recently, much effort has been directed toward the large throttling loss and low efficiency of the valve control system widely applied in steering system of ships. This paper presents an Integrated Direct-Drive Volume Control (IDDVC) electro-hydraulic servo system with the advantages of high efficiency and energy conservation. Firstly, the simulation model of IDDVC is improved by software AMESim, including the nonlinear interaction of the motor-pump and the oil supply ignored by traditional transfer function model. Then, by establishing discrete state equations, a controller based on robust sliding control strategy has been designed to enhance the practicality and real-time performance. Finally, the accuracy of the model and the effectiveness of the controller are proved through the experiments which are conducted after constructing the IDDVC prototype.

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1. Introduction

The steering gear(SG) hydraulic system is one of the most important marine auxiliary systems which can control ship course and ensure navigation maneuverable. According to the traditional architecture, hydraulic servo valves are often chose as the control units which have fast response and high control precision. However, this system also exists several problems. Firstly, the throttling loss is large and the efficiency is low. This disadvantage is becoming much more serious under the background of energy shortages and environmental problems, recently [1–3]. Secondly, the requirement for the cleanliness of oil is high and its reliability is poor [4]. Besides, the cost of servo valve and integrated system is unaffordable [5–7]. Benefiting from the rapid development of converter technique, a new electro-hydraulic servo system, Direct Drive Volume Control(DDVC), has been investigated during the past 25 years [8,9]. Compared with the valve-controlled system, DDVC has the advantages of high efficiency, easy operation and low cost. Therefore, it provides a promising direction to realize innovations by applying DDVC into SG.

DDVC has the similar principle with the Electro-Hydraulic Actuator (EHA) [10]. Both modeling and control algorithm research are investigated in recent years [11]. For modeling, the models among most papers are idealistic after simplifying the dynamic process. Most documents focus on the positive and negative

discontinuity of the Coulomb friction for the actuators. Only a minority of the papers refer to the nonlinear interaction of the motor-pump and oil supply, and fewer documents establish simulation model to simulate this property [12]. Besides, as the modern control theory gradually replaces the classical one, it can be found that there are several researchers focusing on the control algorithm to improve the control performance from both of theory and practical aspects, including back-stepping [13], fuzzy neural networks [14–16], adaptive control or their combination [17,18]. Especially, sliding mode control method is preferred in practical system because it is inherently robust against the system uncertainty and the external disturbance, and has a good transient response [19]. Moreover, some fault diagnosis strategies are also considered for improving the control performance [20,21]. However, there are an ocean of researches aimed at DDVC continuous system which lacks the transformation to the discrete system and limits the application into practical engineering projects. Furthermore, although quite a few control algorithms possess good simulation performance, the calculation is tedious and it is hard to guarantee the real-time performance.

Therefore, it is meaningful for applying DDVC on SG hydraulic system to reduce the energy consumption and improve the control performance. However, the precondition is making a precise model for the DDVC system, especially investigating the control algorithm which has quick real-time and good practical performance. In this paper, an Integrated Direct-Drive Volume Control (IDDVC) electro-hydraulic servo system is presented for SG system application, and both of the controller simulation and experiment are proceeded to verify the effectiveness.

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2. Modeling analysis for the IDDVC

The equivalent schematic of the SG electro-hydraulic actuator based on IDDVC is shown in Fig. 1. It uses adjustable-speed servo motor to drive bi-direction hydraulic pump, thus replacing the servo valve and servo variable pump used in conventional electro-hydraulic system. The direction, speed and position of the hydraulic cylinder can be controlled by changing the direction, speed or running time of the servo motor. By limiting the torque of the servo motor, the maximum pressure of the hydraulic cylinder can be restricted, thus playing the role of the overload protection. It should be noticed that one single-rod double-acting symmetrical hydraulic cylinder is used in this setting. Fig. 2 shows two kinds of single-rod double-acting hydraulic cylinders, the right one has the equal acting area ($A_1 = A_2$) of the two chambers compared with the left one. Hence, it should reduce the control difficulty because the unbalanced flow rate effect is not needed to consider.

For the calculation and simulation of the IDDVC, the transfer function model for the power mechanism of the pump controlled cylinder is adopted. Although it is concise and efficient, quite a few dynamic details of the model are ignored. This paper, first of all, has built an IDDVC transfer function model. And then, after theoretical analysis, simulation model is built under AMESim environment in order to precisely describe the working process of the IDDVC. The model can reflect the nonlinearity of the system, which supports the controller design.

2.1. Transfer function model of the IDDVC

The model can be divided into synchronous motor subsystem and hydraulic subsystem. This is a traditional process and this part is omitted. The motor can be simplified as a proportional component after comparing the frequency response between the prototype and servo motor which are 2 Hz and 500 Hz respectively.

$$G_{sp} = \frac{W_p(s)}{U^*(s)} = K_{sp} \tag{1}$$

where K_{sp} is the driver gain of the servo motor, $K_{sp} = 10\pi$ rad/s.

The IDDVC can be simplified as the power mechanism of pump controlled cylinder. Then, the IDDVC actuator needs to overcome internal unbalanced force, friction, viscous force, preload and other forces, which has strong time-variant characteristic and uncertainty. Besides, in order to simulate the actual working condition, this project has designed a group of disc-springs as actuator load [22]. Therefore, considering the elastic load, the displacement of the power mechanism of pump-controlled cylinder is expressed as Eq. (2),
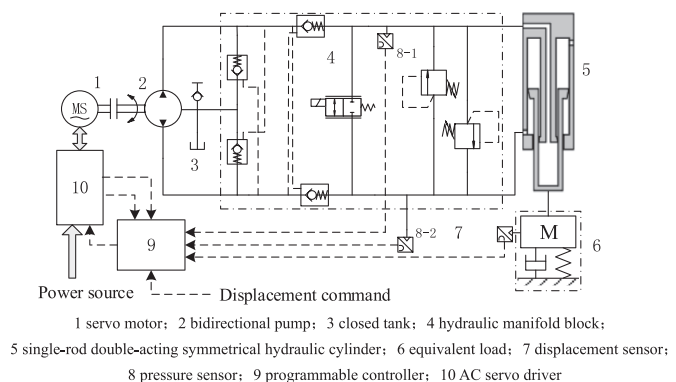
Power source - - - - - Displacement command

1 servo motor；2 bidirectional pump；3 closed tank；4 hydraulic manifold block；
5 single-rod double-acting symmetrical hydraulic cylinder；6 equivalent load；7 displacement sensor；
8 pressure sensor；9 programmable controller；10 AC servo driver

Fig. 1. Equivalent Schematic of the SG electro-hydraulic actuator based on IDDVC system.
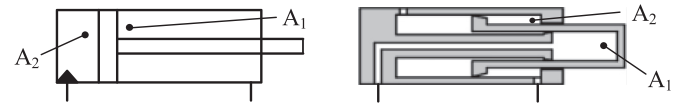
Fig. 2. Schematic of the two kinds of single-rod double-acting hydraulic cylinders.

$$X = \frac{\frac{D_p}{A}\omega_p - \frac{C_t}{A^2}\left(\frac{V_0}{\beta_e}s + 1\right)F_L}{\left(\frac{1}{\omega_r}s + 1\right)\left(\frac{s^2}{\omega_h^2} + \frac{2\xi_h}{\omega_h} + 1\right)} \tag{2}$$

where natural frequency ω_0 and ζ_0 are

$$\omega_0 = \omega_h\sqrt{1 + \frac{K}{K_h}}$$

$$\xi_0 = \frac{Ct}{2A}\left(1 + \frac{K}{K_h}\right)^{-\frac{3}{2}}\sqrt{\frac{\beta_e M}{V_0}} + \frac{B_c}{2A}\left(1 + \frac{K}{K_h}\right)^{-\frac{1}{2}}\sqrt{\frac{V_0}{\beta_e M}}$$

2.2. AMESim model of the IDDVC

A multitude of details are ignored according to the modeling method mentioned above. After theoretical analysis, a compound simulation model based on the AMESim (AMESim model for short) is presented. It has two sub-models including volume pump model and pilot operated check valve model. Moreover, it contains the modelling part for the closed volume cavity.

2.2.1. Volume pump model

In order to establish the pump model precisely, the leakage should be considered in detail. Firstly, it is a must to distinguish the internal leakage and external leakage for the pump. Then, the loss of flow Q_L can be divided into laminar flow loss $Q_{L\mu}$ and turbulence loss $Q_{L\rho}$ [23]. As the pump works in both directions, it can be assumed that the internal structure with in-out oil ports is symmetric. The leakage laminar flow factors and turbulent factors of the in-out ports are given as $C_{\mu i}$, $C_{\mu e}$, $C_{\rho i}$ and $C_{\rho e}$ respectively. The average of the pressure is used as reference pressure to amend the calculation leakage flow. Then the leakage of the three oil ports is calculated in Eq. (3). Since the gap leakage between the gear faces accounts for the vast majority of the whole leakage (about 75–80%), the out leakage laminar flow factor $C_{\mu e}$ plays a dominant role. Besides, the oil compressibility is amended and is converted into the flow under the 0 MPa reference pressure.

$$\begin{cases} Q_{L_{12}} = \left(C_{\mu i}\frac{\Delta p_{12}}{\mu} + C_{\rho i}\sqrt{\frac{|\Delta p_{12}|}{\rho(p_{12ref})}} \cdot \text{sgn}(\Delta p_{12})\right) \cdot \frac{\rho(p_{12ref})}{\rho_0} \\ Q_{L_{13}} = \left(C_{\mu e}\frac{\Delta p_{13}}{\mu} + C_{\rho e}\sqrt{\frac{|\Delta p_{13}|}{\rho(p_{13ref})}} \cdot \text{sgn}(\Delta p_{13})\right) \cdot \frac{\rho(p_{13ref})}{\rho_0} \\ Q_{L_{23}} = \left(C_{\mu e}\frac{\Delta p_{23}}{\mu} + C_{\rho e}\sqrt{\frac{|\Delta p_{23}|}{\rho(p_{23ref})}} \cdot \text{sgn}(\Delta p_{23})\right) \cdot \frac{\rho(p_{23ref})}{\rho_0} \end{cases} \tag{3}$$

where Q_{Lij} is the leakage flow between oil ports; $\Delta p_{ij} = p_i - p_j$ means the pressure difference between oil ports; p_{ijref} represents the reference pressure of Q_{Lij} and $\Delta p_{ij} = (p_i + p_j)/2$; ρ_0 denotes the oil density under 0 MPa reference pressure.

Besides, the flow loss ΔQ_{vf} should be considered when the pump rotates in high speed due to the insufficient filling of the oil absorption port. A critical speed of rotation ω_{fl} can be further assumed and the oil absorption is regarded as sufficient when the speed is below ω_{fl}. Also, ΔQ_{vf} is expressed by using the polynomial