Dynamic surface control associates adapting external torque algorithm for electro-hydraulic velocity servo system

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Abstract: The nonlinear model of the electro-valve driven hydraulic system is presented. Based on the mathematical model, a speed control of an electro-hydraulic system is investigated in the paper based on Dynamic Surface Control (DSC). DSC is a systematic controller calculated by combining backstepping technique and sliding mode control. In order to regulate the system's state to the desired speed under the changes of external torque, an adaptive control is proposed in this paper. The system stability is proven, and a number of simulations are given to illustrate the effectiveness of the controller.

Keywords: Hydraulic system, Dynamic surface control, Adative control, Backstepping-sliding mode control, Lyapunov's stability

I. INTRODUCTION

Hydraulic systems are broadly used in different applications because of the capacity in handling large inertia and loads. Typically like processing of plastics, industrial robots, lunch vehicles, flight simulators, floating cranes, and various military applications [1]. Depending on desired control objectives, the researches about this system are divided into numerous feature like position, velocity or torque system. Many researches focus on position and tracking problems of the hydraulic systems in the quest of seeking appropriate control schemes for better control performance [2]-[4]. With an idea of exploiting simplicity of pulse-width-modulated on/off valve accompanied with a feedback linearization control to stabilize the supply pressure, the authors successfully construct a nonlinear position control for a hydraulic system [5]. This paper considers about velocity control schemes for hydraulically driven process. For successful closed loop velocity control, development of suitable controller which could reflect such characteristics is very significant, although the dynamics of electro hydraulic servo system is highly nonlinear [6]. Although the using the feedback linearization controller successfully achieves the desired objectives theoretically, another controller is designed using the backstepping approach. It avoids adverse effects of the uncertainties or non-modelled dynamics. The authors Krstic et al. in their book [7] introduced the fundamental concept of backstepping method. The backstepping control method is also presented in [8-10] where this technique is explained in detail for regulating and tracking problems. Dynamic surface control (DSC) is a outstanding schemes that developed from backstepping technique and multi sliding mode control. In the book [11], J. Karl Hedrick et al. introduced basically and using DSC for uncertain nonlinear systems in generally. This method was also applied in numerous system as electrical motors [12], power system [13]. This paper proposes a new controller for electro-hydraulic velocity servo system base on dynamic surface control. To increase the performance of system with external torque, an adaptive algorithms was added to adjust this parameters. The efficient designed controller is validated by the appropriate simulation of nonlinear mathematical model of the system.

II. SYSTEM DESCRIPTION AND DYNAMIC MODEL

Two stages servo valve consists of three main parts: the electrical torque motor, the hydraulic amplifier, and the valve spool assembly. The dynamics of the valvespool with no noticeable decline in accuracy in a widerange of frequencies can be described through the firstorder transfer function between the valve opening area \( A_v \) and control input \( u \).

\[
\tau_v \dot{A}_v + A_v = K_{sv} u
\]  \hspace{1cm} (1)

where \( K_{sv} \) is the servo valve gain and \( \tau_v \) is the servo valvetime-constant. The constants mentioned can be determined for by certain tests. Due to the fact that the input of the valve is an electric current but the interface card output is in the form of an electric voltage, it is common to use a current to voltage converter.
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For an ideal critical center, the servo valve with a matched and symmetric orifice the input/output flow rate from the servo valve through the orifices (assuming negligible leakage) can be expressed in the following form:

\[ Q_l = C_d A_m \sqrt{\frac{P_r - p_r \cdot \text{sign}(A_m)}{\rho}} \]  \hspace{1cm} (2)

where \( p_r = p_{c_1} - p_{c_2} \) is a load pressure or pressure difference between both chambers, \( p_s = p_{c_1} + p_{c_2} \) is the supply pressure and \( Q_l \) is the load flow. Assuming no external leakage, \( Q_l \) can be considered as the average flow in each path \( \bar{Q_l} = \frac{Q_{c_1} + Q_{c_2}}{2} \), \( Q_{c_1} \) and \( Q_{c_2} \) are flow rates to and from the servo valve. Adding a low-pass filter to this filter as:

\[ V_0 \frac{2\beta}{\rho} \bar{p}_l = C_d A_m \sqrt{\frac{p_r - p_s \cdot \text{sign}(A_m)}{\rho}} - D_m \dot{\theta} - C_L \bar{p}_l \]  \hspace{1cm} (3)

where \( \beta \) and \( V_0 \) are, respectively, the fluid bulk modulus and the oil under compression in one chamber of the actuator. \( D_m \) and \( C_L \) represent the actuator volumetric displacement and total leakage coefficient, respectively. By applying Newton’s second law for the rotary motion of a hydraulic actuator and neglecting the Coulomb’s frictional torque:

\[ J_l \ddot{\theta} = D_m \bar{p}_l - B \dot{\theta} + T_i \]  \hspace{1cm} (4)

Combining (1), (2), (3) and (4), the third-order nonlinear system that describe the system dynamics can be derived as:

\[ \ddot{\theta} = -a_1 \dot{\theta} + a_2 \bar{p}_l + a_3 \]  \hspace{1cm} (5)

\[ \ddot{p}_l = -a_4 \dot{\theta} - a_5 \bar{p}_l + a_6 \left( \frac{p_r - \bar{p}_l}{V_0} \right) A_m \]  \hspace{1cm} (6)

\[ \dot{A}_m = -a_7 A_m + a_8 u \]  \hspace{1cm} (7)

where \( a_1 = \frac{B}{J_l} \), \( a_2 = \frac{D_m}{J_l} \), \( a_3 = \frac{T_i}{J_l} \), \( a_4 = \frac{2\beta D_m}{V_0} \), \( a_5 = \frac{2\beta}{V_0 C_L} \), \( a_6 = \frac{2\beta}{V_0 \sqrt{\rho}} C_d \), \( a_7 = \frac{1}{\tau_m} \) and \( a_8 = \frac{K_m}{\tau_m} \).

### III. SYSTEM DESCRIPTION AND DYNAMIC MODEL

#### A. Dynamic surface control

Dynamic Surface Control (DSC) is constructed based on integrator backstepping (IB) technique and multiple sliding mode (MSS) method. So that, this controller is inherited all the advantages of both above mechanisms. Beside, with an important addition in the design: the low-pass filter; this controller also brought significant effect in diminishing error in calculating and minimizing the amount of computation by avoiding the “explosion of terms” phenomenon. Let consider the following to express the DSC approach for the nonlinear system:

\[
\begin{align*}
\dot{x}_1 &= x_2 + f(x_1) \\
\dot{x}_2 &= u
\end{align*}
\]  \hspace{1cm} (8)

Where the function \( f(x) \) is non-Lipschitz nonlinearity and completely known.

Defining the first error valuable: \( S_1 = x_1 - x_{1r} \) \hspace{1cm} (9)

Firstly, we determind the virtual control signal \( \overline{x}_{2r} \) so that if \( x_2 \) tracks this value, \( x_1 \) will approch the reference. Proposing Lyapunov candidate function for \( S_1 \) as:

\[ V_1 = \frac{1}{2} S_1^2 \]  \hspace{1cm} (10)

Differentiating (10) gives:

\[ \dot{V}_1 = S_1 \dot{S}_1 = S_1 \left( \dot{x}_2 + f(x) - \dot{x}_{1r} \right) \]  \hspace{1cm} (11)

Choosing \( \overline{x}_{2r} = -f(x) + \dot{x}_{1r} - k_1 S_1 \), where \( k_1 \) is a positive gain, thus \( V_1 = -k_1 S_1^2 \leq 0 \) or \( x_1 \) will be driven to \( x_{1r} \) by \( x_{2r} \).

Signal \( \overline{x}_{2r} \) determined above is a virtual signal. At this step, a low pass filter is added, \( x_{2r} \) track to \( \overline{x}_{2r} \) through this filter as:

\[
\begin{align*}
\tau \ddot{x}_{2r} + x_{2r} &= \overline{x}_{2r} \\
x_{2r}(0) &= \overline{x}_{2r}(0)
\end{align*}
\]  \hspace{1cm} (12)
Defining sliding surface:
\[ S_2 = x_2 - x_{2r} \]  
(13)

The control signal \( u \) will drive \( S_2 \) to zero, thus, \( u \) is chosen as:
\[ u = \dot{x}_{2r} - k_2 S_2 = \frac{x_{2r} - x_{2r}}{e} - k_2 S_2 \]  
(14)

The stability of system will be consider by Lyapunov cadidate function following:
\[ V = \frac{S_1^2 + S_2^2}{2} \]  
(15)

We will discuss specifically about the stability of DSC for roll-to-roll system in the next section.

B. Dynamic surface control for electro-hydraulic velocity system

The controller aims to keep the velocity at desired values, web speed is controlled through tracking angular velocity at references. This section will express the DSC controller formular for the electro-hydraulic system, then we will demonstrate the stability of system with that controller. The outstanding characteristics will be also discussed in this section and illustrated through simulation results in the final section. Firstly, define some error variables that are considered for design controller as following:
\[ x_1 = \dot{\Theta} - \dot{\Theta}_d \]  
(16)
\[ x_2 = p_e - \alpha_1 \]  
(17)
\[ x_3 = A_p - \alpha_2 \]  
(18)

Where \( \dot{\Theta}_d \) is desired desired rewind velocity. \( \alpha_1, \alpha_2 \) are the control virtual signals. The conventional ideal is using virtual control signals \( \alpha_{id}, \alpha_{2d} \) generated through backstepping technique in order to \( x_1 \to 0 \) and \( x_2 \to 0 \), respectively. But in this research, instead of using directly virtual signal \( \alpha_{id}, \alpha_{2d} \) we use \( \alpha_1, \alpha_2 \) that are filtered signal after let \( \alpha_{id}, \alpha_{2d} \) through low-pass filter that was defined as:
\[
\begin{align*}
\alpha \dot{\alpha} + \alpha &= \alpha_d \\
\alpha(0) &= \alpha_d(0)
\end{align*}
\]  
(19)

This addition helping decrease significantly amount of computation from caculating derivative of virtual control, that generate “exploation of term” phenomenon. Rewrite the mathemtical system by these error variables as:
\[
\begin{align*}
\dot{x}_1 &= -a_1 \dot{\Theta} + a_1 p_e + a_1 - \dot{\Theta}_d \\
\dot{x}_2 &= -a_2 \dot{\Theta} - a_2 p_e + a_2 \left( \sqrt{p_i - p_e} \right) A_p - \dot{\alpha}_1 \\
\dot{x}_3 &= -a_3 A_p + a_3 u - \dot{\alpha}_2
\end{align*}
\]  
(20) (21) (22)

Choose the first Lyapunov candidate funtion as:
\[ V_1 = \frac{1}{2} x_1^2 \]  
(23)

Take time derivative (23) and using (20), we obtain:
\[ V_1' = x_1 \dot{x}_1 = x_1 \left( -a_1 \dot{\Theta} + a_1 p_e + a_1 - \dot{\Theta}_d \right) \]  
(24)

From equation (24), the first control virtual signal can be chosen as:
\[ \alpha_{id} = \frac{1}{a_2} \left( a_1 \dot{\Theta} - a_1 + \dot{\Theta}_d - k_1 \text{sign} (x_1) \right) \]  
(25)

Where \( k_1 \) is a positive gain. Replace the signal (25) to (24), we obtain:
\[ V_1' = -k_1 x_1 \text{sign} (x_1) \leq 0 \]  
(26)

The derivative of Lyapunov funtion is negative definite as above demonstrate, signal \( x_1 \) will approch to zero as Lyapunov stability theory.

Next, we will determine the second control virtual signal. The desired value of pressure difference between both chambers is \( \alpha_1 \) which is filtered signal after let \( \alpha_{id} \) through the low-pass filter. Choose the second Lyapunov candidate funtion as:
\[ V_2 = \frac{1}{2} x_2^2 \]  
(27)

Take time derivative(27)and using(21), we obtain:
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\[ \dot{V}_1 = x_1 \dot{x}_1 = x_1 \left( -a_0 \dot{\Theta} - a_1 p_L + a_2 \left( \sqrt{P_e - p_L} \right) A_n + \dot{\alpha}_1 \right) \]  

(28)

From (28), choose the desired value of \( A_n \) as:

\[ A_{\text{end}} = a_{2d} = \frac{a_0 \dot{\Theta} + a_1 p_L + \dot{\alpha}_1 - k_2 \text{sign} \left( x_1 \right)}{a_2 \left( \sqrt{P_e - p_L} \right)} \]  

(29)

Where \( k_2 \) is a positive gain. Replace the fomular above to (28), we obtain:

\[ \dot{V}_1 = -k_2 x_1 \text{sign} \left( x_1 \right) \leq 0 \]  

(30)

The derivative of the second Lyapunov function is negative definite as demonstrate, signal \( x_1 \) will approach to zero as Lyapunov stability theory.

The formular (29) is the second control signal, after letting this signal through the low-pass filter, we obtain the desired value of valve opening area \( \alpha_1 \). Choose the third Lyapunov candidate function as:

\[ \dot{V}_3 = \frac{1}{2} x_3^2 \]  

(31)

Take time derivative (31) and using (22), we obtain:

\[ \dot{V}_3 = x_3 \dot{x}_3 = x_3 \left( -a_1 A_n + a_2 u - \dot{\alpha}_2 \right) \]  

(32)

Choose the control signal as:

\[ u = \frac{1}{a_2} \left( a_1 A_n + \dot{\alpha}_2 - k_3 \text{sign} \left( x_3 \right) \right) \]  

(33)

Where \( k_3 \) is a positive gain.

With the control signal as chosen, the derivative of third Lyapunov candidate function can be rewritten as:

\[ \dot{V}_3 = -k_3 x_3 \text{sign} \left( x_3 \right) \leq 0 \]  

(34)

The derivative of Lyapunov function is negative definite as above demonstrate, signal \( x_3 \) will approach to zero as Lyapunov stability theory.

To demonstrate the stability of all states, we propose the total Lyapunov candidate function as:

\[ V = \frac{1}{2} x_1^2 + \frac{1}{2} x_2^2 + \frac{1}{2} x_3^2 \]  

(35)

Take time derivative of (35) and use the control signals (25), (29), (33), we obtain:

\[ \dot{V} = -k_1 x_1 \text{sign} \left( x_1 \right) - k_2 x_2 \text{sign} \left( x_2 \right) - k_3 x_3 \text{sign} \left( x_3 \right) \]  

(36)

From above fomular, we can infer that derivative of total Lyapunov candidate function is negative definite. The system will be stable as Lyapunov stability theory.

C. Adaptive external torque for electro-hydraulic velocity system

In fact, when the system is operating, the external torque is an uncertainty. Although the DSC controller proposed in previous section has adjustable disturbances function, is is necessary to associate with a adaptive external torque algorithm if this parameter is changed significantly. Base on the previous designed controller, we can rewrite the virtual control signal (25) as:

\[ \dot{\alpha}_{\text{ad}} = \frac{1}{a_3} \left( a_1 \dot{\Theta} - \frac{\dot{\Theta}}{J} \right) \]  

(37)

Where \( \ddot{T}_L \) is the adapted value of the torque. Propose the Lyapunov candidate function as:

\[ V = \frac{1}{2} x_1^2 + \frac{1}{2} x_2^2 + \frac{1}{2} x_3^2 + \frac{1}{2} \left( \ddot{T}_L - T_L \right) \]  

(38)

Take time derivative of (38) and use the control signals (29), (33), (37) we obtain:

\[ \dot{V} = -k_1 x_1 \text{sign} \left( x_1 \right) - k_2 x_2 \text{sign} \left( x_2 \right) - k_3 x_3 \text{sign} \left( x_3 \right) + \left( \dddot{T}_L - T_L \right) \left( \dot{T}_L - T_L \right) \]  

(39)

From (39), to guarantee that the derivative of Lyapunov candidate function is negative definite, we choose the adapted law as:

\[ \ddot{T}_{L} = \lambda \frac{x_1}{J} \]  

(40)
The next section will express some simulation results with proposed algorithm with the uncertain torque element considered.

**IV. SIMULATION RESULT**

In this section, the performance of the proposed controller will be evaluated through a numerical simulations. In order to test the quality of the designed controller, there will be three cases and the simulation results of the presented controller is compared to the reference. To evaluated the outstanding efficient, we will also compare with performances of PID controller. The parameters of system is shown in table below:

<table>
<thead>
<tr>
<th>Physical parameters</th>
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<tbody>
<tr>
<td>$J_s = 3.4 \times 10^{-3} \text{kgm}^2$</td>
</tr>
<tr>
<td>$B = 1.25 \times 10^{-6} \text{Nms/rad}$</td>
</tr>
<tr>
<td>$\beta = 0.35 \times 10^9 \text{Pa}$</td>
</tr>
<tr>
<td>$C_g = 0.65$</td>
</tr>
<tr>
<td>$K_m = 4.23 \times 10^{-7} \text{m}^2 / \text{V}$</td>
</tr>
</tbody>
</table>

In the first case, the reference is 40 (rd/s) and the system will be operate from initial state. The figure 1 is shown the response of proposed controller in this case. The figure 2 is expressed the performance when the desired value changes suddenly. In last case, an unknown torque will be added to system when it’s operating, the details will be shown in figure 3.

**Figure 1:** The velocity responses with constant reference

**Figure 2:** The velocity responses with suddenly changed reference
As the simulation results, the responses of velocity and the are shown in fig.1-3 coresponding to three cases. It can be seen that the velocity responses can track the reference and converge to the desired value in a short time in all case but the response of new controller is faster than PID algorithm. The overshoot is significantly different in the performance of the controller between the proposed controller and conventional one in all case. Once the external torque is suddenly added, both the adaptive DSC and PID controller ensure the settling time is small, but PID’s overshoot is bigger than DSC. The simulations showed that the outstanding efficient of new algorithm compare with conventional controller. Especially when unknown disturbance.

IV. CONCLUSION

In the paper, an adaptive dynamic surface control for electro-hydraulic system is introduced. Hydraulic nonlinearities are taken care of using the systematic approach. Uncertain external torques handled thanks to the adaptive algorithm that was associated with DSC controller. The system is proven to be stable in Lyapunov’s sense. Several numerical studies are included to show the effectiveness of the proposed control.

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