The Velocity Synchronizing Control on the Electro-Hydraulic Load Simulator

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Abstract: This paper builds up an accurate nonlinear mathematical model of an electrohydraulic force/torque servo control system, and provides a thorough theoretical analysis on the feedforward compensation for extraneous force/torque, whose limitation is analyzed and revealed. The nonlinear factors and the servo valve dynamics have much influence on the system characteristics. Subsequently a velocity synchronizing compensation method by using the control signal of the control actuator is proposed, which can reduce the lagging effects for the better performance. For the reason of similarity between the model of control actuator and that of the load simulator, the proposed method performs well against the influence of nonlinear factors. The simulations and the experiments confirm that this control scheme results in a quick response, robustness, and excellent ability against disturbance.

Key words: load simulator; synchronizing control; extraneous force/torque

Hydraulic systems are important actuators in modern industry, due to their high power/mass ratio, fast response and high stiffness. They are widely used both in position control and in force control. In particular, force tracking is important for some applications, such as vibration isolation and automotive active suspension. Hydraulic systems are complex and pose nonlinearities, which makes the modeling and design of feedback controllers challenging. The nonlinearities mainly remain in servo valve flow-pressure characteristics, orifice area openings, variations of fluid volume under compression and in part, to cavitations and seal friction. Many control methods are attempted in the force control, which is a difficult problem. Conventional PID controllers do not yield reasonable performance over a wide range of operating conditions (Alleyne, Liu & Wright, 1998; Niksefat and Sepehri, 1999). Several researchers have, therefore, considered the use of adaptive and sliding mode control techniques. Variable structure control is first used in a single-rod hydraulic cylinder by Chen, Lee and Tseng (1990). Robust control algorithms have also been studied to design hydraulic force control. H-infinity linear design approach is used in the force control by Laval, M’Sirdi and Cadiou (1996). Quantitative feedback theory (QFT) is another robust control, which is first used to hydraulic system by Niksefat and Sepehri (2000).
A load simulator, which exerts the force to the control actuator in real-time to ascertain the performance of the actuator in ground, is regarded as a hydraulic system in the force control. But for the load simulator designers, a new difficulty appears besides the general problems in the force control. They have to share part of their energies to handle with the extraneous force/torque, which comes from the disturbance of the control actuator. The architecture of the load simulator is shown in Fig. 1. The actuator movement is controlled by the actuator reference input and the force/torque is controlled by the load feedback separately. As the shaft of the loading motor is connected to that of the control actuator rigidly, the movement of actuator will inevitably educe the extra flow in the cylinder of the motor, which will generate the extra pressure, and leads the extraneous torque to build up. This extraneous torque is a significant disturbance especially when the actuator moves in high frequency, which can be obtained from the analysis of the system model in section 1.

For the load simulator, the extraneous force/torque is a main element that weakens the robustness and produces a great tracking error for the system. So, the eliminating measure is necessary and is a major argument in this field. To eliminate this error many methods are adopted. All these methods focus on how to make the loading motor move together with the actuator and how to make the system have a thick skin against the disturbance. It can be categorized into hardware eliminating and software eliminating. In the hardware eliminating, some measurements are introduced such as adding interconnected pore, oil container and buffer spring adjustments etc. They are aimed at releasing the extra flow in the cylinder of the motor. But for the hardware eliminating, it has some limitation and costs high. Another method is to improve the software, which focuses on decreasing the effects of the disturbance. Researchers are satisfied with this method, considering it as low-cost and convenient in use. Decoupling control does not get an approving result because it is difficult to carry out in reality. The feedforward compensation is a method that gets the most satisfying result in all the methods discussed above. It means to set the transfer function between output and the position disturbance to zero through adding new block into it. This eliminates the extraneous torque to a certain extent. But for the uncertainty and nonlinearity of the servo valve, the perfect results cannot be obtained yet.

According to this problem, a more effective method named velocity-synchronizing control is put forward in this paper. It depends upon the similar model between the load simulator and the control actuator. To educe this new method, some anterior results are also introduced in this paper. It builds up a more accurate nonlinear model on the load simulator, which further discloses the limitations of the feedforward compensation method. The nonlinear and dynamic characteristics of the servo valve are of a great influence on the loading system. So, the velocity synchronizing compensation method is proposed and is verified by the experiments.

1 The Nonlinear Mathematical Model of the Load Simulator

The structure of electro-hydraulic load simulator is shown in Fig. 1.

The mathematical model of the load simulator consists of three parts: servo valve, hydraulic motor and loading mechanism.

1.1 Servo valve

The model of the servo valve is composed of the transfer functions from electric current to spool position and from spool position to flow output.

The model described in Eq. (1) is of the servo valve, whose parameters are obtained through identification.

\[ G(s) = \frac{X_v}{I} = \frac{K_{X_v} s^2 + 2\xi_{v} s + \omega^2}{s^2 + 2\xi_{v} s + \omega^2} \quad (1) \]

where \( I \) is current input; \( X_v \) is displacement of the spool; \( K_{X_v} \) is spool displacement coefficient; \( \omega \) means the nature frequency of valve; \( \xi_{v} \) is damping coefficient.
Eq. (2) is the orifice equation of the servo valve, where the leakage is neglected, which is considered in the model of the hydraulic motor.

$$Q_f = C_q w X_v \sqrt{\frac{1}{\rho} (p_s - |p_f|)}$$

(2)

where $Q_f$ is load flow; $X_v$ is displacement of the spool; $p_f$ is load pressure; $p_s$ is power pressure supply; $w$ is area gradient; $C_q$ is flow coefficient of orifice; $\rho$ is oil density.

1.2 Hydraulic motor

Equation of flow into the motor is as follows,

$$Q_I = D_m \frac{d\theta_m}{dt} + \frac{V_m}{E_o} \frac{dp_f}{dt} + C_{sl} p_f$$

(3)

where $D_m$ is delivery capacity per radian; $V_m$ is total oil volume of motor chamber and the piping connected; $C_{sl}$ is coefficient of leakage; $E_o$ is bulk modulus of elasticity for oil; $\theta_m$ is angular displacement of the motor’s rotor.

The friction model of the motor includes viscous friction, Coulomb friction and static friction etc. The viscous friction can be put into the linear model of the system directly. So the friction torque here only represents the Coulomb friction and the static friction. The load equations can be described as

$$D_m \frac{d^2 \theta_m}{dt^2} + B \frac{d\theta_m}{dt} + G_s (\theta_m - \theta_l) + M_f$$

(4)

$$M = J_m \frac{d^2 \theta_m}{dt^2} + B_1 \frac{d\theta_l}{dt} + G_l (\theta_m - \theta_l)$$

(5)

where $J_m$ is rotary inertia of motor; $B$ is damping coefficient of motor’s rotor; $G_s$ is torsion stiffness between $\theta_m$ and $\theta_l$; $G_l$ is equivalent torsion stiffness of the load; $M_f$ is friction torque; $M$ is torque output; $\theta_l$ is equivalent angular displacement of the control actuator output. The accurate nonlinear mathematical model of electro-hydraulic force/torque servo system is shown in Fig. 2.

2 The Limitation of Feedforward Compensation for Extraneous Force Eliminating

From Ref. [3], extraneous momentum ($M_0$) is defined as the momentum output ($M$) under the condition that the system input ($M_0$) is set to zero. It can be described as

$$M_f = M | M_0 = 0$$

(6)

According to the diagram of the load simulator in Fig. 2, when electric current input is set to zero,
the system output is not equal to zero for the reason of the existence of $\theta_0$, which is the so-called extraneous torque.

![Fig. 3](image_url)

**Fig. 3** Experimental extraneous torque under the actuator displacement triangular command

Fig. 3 indicates the experimental result of the extraneous momentum when the loading motor moves with the actuator by triangle displacement commands. For further explanation, when the displacement locus is a triangle wave, the velocity of actuator is along a square wave, as the extraneous torque is shown in the results obtained in Fig. 3.

Conventionally, the feedforward compensation method can be adopted to reduce the effects of the disturbance, which rebuilds the transfer function from disturbance to output with the feedforward control, as shown in Fig. 4. In this paper, the feedforward compensation is carried out with the feedforward control by the actuator velocity.

![Fig. 4](image_url)

**Fig. 4** The architecture of the feedforward compensation method

From this theory, if the feedforward block $G_3$ for $\theta_0$ satisfies the equation $G_2 = G_3 G_1$, the disturbance of the actuator velocity will be eliminated completely. In fact, there is a nonlinear factor (orifice flow) in the block $G_1$, so it is difficult to design and realize block $G_3$. On the other hand, the actuator velocity signal is obtained from velocity sensor or displacement derivative circuit, which has the phase lag and high-frequency noise. It weakens the compensation effect. With the accurate nonlinear model built up in this paper, some simulation on the feedforward compensation is carried out, as the results shown in Fig. 5 (a) and (b) show the remained extraneous torques as the actuator moves along a sinusoidal wave with different frequency.

![Fig. 5](image_url)

**Fig. 5** Extraneous torque with feedforward compensation

(1. without compensation; 2. with compensation)

In Fig. 5, 80% ~ 85% extraneous torque is reduced, which is similar to the experimental result. The limitation of the feedforward compensation method is originated from the time delay for compensation blocks and some nonlinear factors. Next section puts forward a new method to overcome these shortcomings.

3 The Velocity Synchronizing Control with the Control Signal in Control Actuator Valve

According to the mathematical model, extraneous torque is closely related to the angular velocity of the actuator. If both the loading motor and the actuator move at the same velocity, there will be no extraneous torque at all. The expected load-
ing torque can be realized by the torque feedback control. In order to get a high performance, the velocity synchronization of two output shaft between the actuator and the load simulator is required. But for the reason of the time delay and nonlinearity, the expected result is difficult to obtain with the direct velocity feedforward. In feedforward compensation method, making use of the angular velocity means to get the information of the actuator movement. Then introduce angular velocity signal to the control loading motor, which makes it move with the actuator. Whereas the lag of the feedforward compensation blocks and some nonlinear factors are unavoidable, the movement information of the actuator obtained in advance will be beneficial for the compensation.

According to the hydraulic servo control principle, the control actuator moving speed is proportional to the servo valve spool displacement under a fixed load, which is also proportional to the input electric current. If the current signal can be introduced to the control loading motor in real time, due to similarity of the mathematical models between the actuator and motor, it is easy to ensure the velocity synchronization between the actuator and the loading motor. Consequently this scheme leads to a reduced extraneous momentum. Normally, the extraneous torque is related not only to the velocity of the actuator but also to the acceleration of the actuator. So further to add the differential of this signal to compensation block can lead to the better results. The new control scheme is illustrated in Fig. 6.

![Diagram of velocity synchronizing control scheme](image)

Fig. 6 The velocity synchronizing control scheme

Considering the gain difference in transfer function between the two systems, the adjusting block in the compensation loop is added. Since it is already declared that the acceleration is also a feature that affects the extraneous torque, especially in high frequency, the compensating block is chosen as Eq. (7).

$$G_e(s) = \frac{K_{com}(T_{com}s + 1)}{T_1s + 1}$$  \hspace{1cm} (7)

where $K_{com}$ is gain of the compensating block; $T_1$ is time constant of the filter, and it is chosen to make the filter bandwidth be five times to bandwidth of the control actuator; $T_{com}$ is differentiating time constant of the compensating block.

The simulation result with this method is shown in Fig. 7, where the simulation parameters and conditions are the same as shown in Fig. 5. About 95% or above of the extraneous torque is eliminated both in low frequency and in high frequency, so it can be concluded that this control scheme is superior to that of the conventional feedforward compensation method. So both the availability and the validity of this principle are cor.
firmed in the theory explained.

4 Experiments on the Velocity Synchronizing Control

The experiments are carried out on the test rig shown in Fig. 8. The closed-loop control method is an improved PID control method and extraneous torque compensation is based on velocity synchronizing scheme shown in Fig. 6. The extraneous torque under various frequencies is tested. The maximum angular velocity of the actuator is $220\,\text{deg/s}$. Fig. 9 gives the extraneous torque compensation results.

In Fig. 9 (a), (b), (c) and (d), the first curve indicates extraneous torque without compensation, the second one is extraneous torque with control signal of servo valve on compensation, and the third one is extraneous torque through adding acceleration information on the compensation. It is shown that applying the control signal of the servo valve and its differential on compensation, the extraneous torque is eliminated greatly, especially under higher frequency. Only 10% or less extraneous torque remains, which confirms the availability and the validity of the principle in practice.

In other experiments, the simulator is tested to track sinuous load under various frequencies. Positive and negative loading gradients are tested under each frequency to ensure the effectiveness, which means that the torque command is proportional to the actuator displacement, either by positive or by reverse direction. The maximum angular speed of the actuator is $220\,\text{deg/s}$. Experimental results are shown in Fig. 10-Fig. 13. The accuracy is enough to trace a load spectrum in reality, as it is shown in Fig. 14; the load tracing error is not greater than 4%.

Through a lot of the experiments and applications in practice, the velocity synchronizing control
method is proved effective and practicable. Although in this paper only PID improved method is adopted, in application it can be combined with any advance control method such as neural network controller, see Refs. [7-9].

5 Conclusions
An accurate nonlinear mathematical model of an electro-hydraulic force/torque servo system is
Fig. 14 Loading spectrum tracing

built up in this paper, and the essential reason for extraneous torque is discussed. The limitations of the feedforward compensation method are analyzed. A novel compensation scheme, velocity synchronizing control method is proposed, in which the control signal of servo valve in actuator system is introduced. The simulation and the experiments prove that the control method is effective. It can be concluded:

(1) The accurate nonlinear model presented in this paper reflects the essence of the load simulator in reality.

(2) The limitation of the feedforward compensation method in eliminating extraneous torque lies in the nonlinear and dynamic characteristics of the system.

(3) In velocity synchronizing control, making use of control signal of the servo valve in actuator system results greatly in eliminating extraneous torque.

References


Biographies:

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