Identification and Modeling of Electrohydraulic Force Control of the Material Test System (MTS)

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Abstract. In the heavy-duty material test device, an electrohydraulic force servo system is usually utilized to load the tested samples. The signal from the pressure sensor is compared with the instruction and the difference between them is then fed to a digital servo valve to form a closed loop control to the target force. The performance of the electrohydraulic force servo system is not only closely related to how accurate to feed the flow rate to the hydraulic cylinder, but also the stiffness of the system which is dominated by the compressibility of oil. Thus the clarification of the characteristic parameters becomes the key of the solution to optimal force control. To identify the electrohydraulic force servo system various step signals are input to excite the dynamic response of the system. From the relationship between the step magnitude and the force response, the system model and the key control parameters are determined. The electrohydraulic force servo system is identified as a first order system with time constant varied with the pressure and their relationship is also clarified by fitting. Based on the identification of the system optimal control parameters are finally obtained and force rate error is reduced to 0.2% from original 3%.

1. Introduction

The test to material’s mechanical characteristics is usually carried out in a material test device. For a heavy-duty loading force required, an electrohydraulic loading system is commonly used to superimpose force on the tested samples during the testing process. Being a key part of the MTS the electrohydraulic force servo system is required to be extremely accurate in the control of both loading force and force rate. Taking the hydraulic cement press as an example, the loading rate of the electrohydraulic loading subsystem should be at 2.4 KN/s with an error less than 5% and error rate for the constant force control should be maintained within a range 0.2% according to the GB7314-87 (also ISO) Standard for cement test. Therefore, an electrohydraulic force servo system is usually utilized for the purposes. The signal from the pressure sensor is compared with the instruction and the difference between them is then fed to a digital servo valve to form a closed loop control to the target force. The performance of the electrohydraulic force servo system is not only closely related to how accurate to feed the flow rate to the hydraulic cylinder, but also the stiffness of the system which is dominated by the compressibility of oil. Thus the modeling and identification of the system are virtually important for the fine force control of the system.

In this paper, the working principle of the electrohydraulic force control of MTS is first introduced. To identify the important control parameters and modeling the system an experiment is designed to obtain the relationship between input and the output. The force control is identified as a first order system with time constant varied with the pressure and their relationship is also clarified by fitting.
Based on the identification of the system optimal control parameters are finally obtained and force rate error is reduced to 0.2% for 3%.

2. Modeling of system

The electrohydraulic force servo system is shown in Figure 1. Since the cylinder is specially machined to get rid of the frictional force, the control of the load to the pressure is equivalent to the control of the force. A pressure sensor is used to measure the pressure (force), signal from which is compared with the instruction and the difference between them is then fed to a digital servo valve to form a closed loop control to the target force. A bypass valve is connected with the digital servo valve in parallel and its valve port opening is controlled by the pressure differential across the digital valve. Through the drainage of the bypass valve, the pressure drop across the digital valve is kept constant and the value is dependent on the pretension of the biased spring in the bypass.

Figure 1. Electrohydraulic force servo system.

By reference to the system shown in Figure 1, the transfer function of the pressure vs. the input of the displacement of the digital servo valve is acquired and shown in Figure 2, under assumptions below:

- the compressibility of oil in the sensitive chamber of the bypass valve is negligible.
- regardless of the pumping effect of spool of the bypass valve.

The system mainly consists of three control loops. Loop 1 demonstrates that the increasing pressure of the load $P_l$ will cause the reduction of the flow rate and the opening of the bypass valve. Both of them will result in the pressure rising of the pump and an effect of positive feed back to the
system. Whereas loop 2 demonstrates that the increasing of pump pressure will induce the gain of the flow rate through the digital servo valve and the opening of the bypass valve. Both of them cause the pressure of the pump and an effect of negative feedback to the system. Since the gain of loop 2 is always large than that of loop 1, the net effect of loop 1 and loop 2 tends to be a negative and the system is always maintained stability. Moreover, the input of the displacement of the digital servo valve sill superimposes a forward feed control the system as shown in loop 3, the dynamic response of the system is improved. In the design and the physical parameters selection of the system the gain of the digital servo valve is kept small enough and the dynamic response of the bypass valve is much fast than the pressure inside the cylinder, the model of the electrohydraulic force servo system can be further simplified as a first order system

$$W(s) = \frac{K_{a1}}{\tau s + 1}$$

Where $K_{a1}$ is the gross gain of system including the electronic part; $\tau$ is the time constant of the system

$$\tau = \frac{V_L}{E_h K_p}$$

Where $V_L$ is the volume of the cylinder; $E_h$ is the bulk modulus of oil; $K_p$ is the flow-pressure coefficient due to the leakage flow. Under the step input of various magnitudes the measured pressure response is presented in Figure 3.

Figure 3. Pressure responses under various step input.

3. Identification of system

In the equation (2), the volume of the cylinder $V_L$ is regarded as a constant since the deformation of the tested sample is small and negligible and the flow-pressure coefficient $K_p$ is independent of the pressure because of the laminar leakage. Thus the key to the identification of the constant of the system is to clarify the variation of the bulk modulus with pressure.

Oil used as the transmission media in a hydraulic system has a great opportunity to contact with air, such as on the free surface of oil tank, and inevitably some of initial air will be mixed into the oil. The mixture of air in oil generally exists in two forms, free air and dissolution. The molecule of dissolved air with those of oil to form air-in-oil emulsified liquid. While the free air drastically reduced the gross modulus of oil, there is no evidence that the dissolved air has a noticeable effect on the bulk modulus within the range applied in the hydraulic system. However, the dissolved air and
free is convertible from each other. According to the Henry’s Law, the relative volumetric of air dissolved in oil is proportional to absolute pressure at given temperature. As pumping circulation and pressure fluctuation in the hydraulic system, the free air will dissolved into or separate from oil. Therefore, relative volumetric amount of either free air or dissolved air is dependent the pressure, in addition to their initial amount. Based on above discussion the air content oil bulk modulus in a hydraulic cylinder can be expressed as

$$K_{of} = \left\{ \begin{align*} & K_e \left[ p_e XK_e \left( \frac{1}{p^2} - \frac{1}{p_c^2} \right) + 1 \right] & \text{for} \quad (p_c > p > p_e) \\ & K_e & \text{for} \quad (p \geq p_c) \end{align*} \right.$$  

(3)

Figure 4. Bulk modulus of oil content oil.

Figure 5. Theoretical and experimental results.

According to the model the mixed free air exists and affects the gross bulk modulus within the area below a critical pressure, while above this critical pressure the mixed free air will no longer influence the bulk modulus of oil because of complete dissolution as shown in Figure 4. The critical pressure is proportional to the square root of the amount of the initial mixed free air. The speed of the pressure has also a substantial effect on the critical pressure and thus on the bulk modulus. The critical pressure increases and tends to be an upper constant as pressure response speed goes up, while both decrease and tend to be a lower constant as response speed slows down. The model of the bulk modulus given in Equation (3) agrees well with experimental results as shown in Figure 5.

The typical loading process of the compression machine is shown in Figure 5. In this curve, the loading force rate is 2.4 kN/sec, an authorized rate for the cement sample test. From this Figure, it can be find that the loading process displays a linear manner, except at the starting area. Because of the large scale, the ripple is masked in the curve. For this reason, an analysis to the loading force rate is given in Figure 6. The force rate is defined as the force increment for each second. From this Figure, much insight can be accessed: The oscillatory rate of the loading curve is within an area 2.4(1 \pm 0.02). Within 4 seconds the loading force rate will become stable and starting area.

4. Experiment and conclusions

The identified model of the electrohydraulic force servo system is applied in the cement test device. The typical loading process of the compression machine is shown in Figure 6. In this curve, the loading force rate is 2.4 kN/sec, an authorized rate for the cement sample test. From this Figure, it can be found that the loading process displays a linear manner. Because of the large scale, the ripple is masked in the curve. For this reason, an analysis to the loading force rate is given in Figure 7. The force rate is defined as the force increment for each second. From this Figure, much insight can be
accessed: The oscillatory rate of the loading curve is within an area $2.4(1 \pm 0.02)$. Within 4 seconds the loading force rate will become stable and starting area.

According to the GB7314-87 (also ISO) Standard for cement test, the loading rate of the electrohydraulic loading subsystem should be at 2.4 kN/s with an error less than 7% and error rate for the constant force control should be maintained within a range 0.2%. With the application of the stage control these two specified errors are limited to 2% and 0.05% respectively and many other technical performances are also enhanced.

![Figure 6. Loading Curve.](image1)

![Figure 7. Loading Force Rate.](image2)

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