Auto-Commissioning and Adaptive Tuning of Servo Control Parameters in an Electro-Hydraulic System Based on Physical Plant Model

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***Abstract*—In industrial servo control systems, the system performance is significantly affected by the physical parameter variance from different machine configuration and from the hydraulic loop non-linearity. Parameter detuning deteriorates the system performance and may even cause system instability issue, and parameter tuning requires much labor effort. In this paper, based on the physical plant model, an auto-commissioning and adaptive tuning method is proposed in a servo motor driven hydraulic control system. The hydraulic loop parameter which is time-variant is off-line or on-line estimated and then used for auto-commissioning and adaptive tuning of control parameters. The proposed method is verified with experiment to achieve high repeatability of dynamic and steady-state performance in different working conditions, and requires zero effort for parameter tuning.**

***Keywords*—Electro-hydraulic system; auto-commissioning; adaptive control; control system**

I. INTRODUCTION

In industrial servo control systems that require high dynamic performance, the knowledge of physical model plays a significant role in servo controller design. The servo control performance is significantly affected by the physical parameter estimation accuracy. The model parameters vary from machine to machine due to different physical configurations, and they vary among different working points due to hydraulic loop non-linearity, and they also vary after long-time operation due to physical heating and wearing etc. All these factors lead to model mismatch, or parameter detuning, which severely deteriorates both the dynamic and steady-state performance. Furthermore, parameter detuning may even cause system instability issue. Piecewise PID method is adopted traditionally for performance consistency at different working points, and thus the commissioning of control parameters demands tremendous human work through trial and error, and extra downtime will occur if the system instability happens[1],[2].

Mitigation of the above-mentioned problem calls the demand of auto-commissioning or adaptive control techniques. In this paper, an automatic process is proposed to achieve high repeatability of dynamic and steady-state performance. The electro-hydraulic system model is presented as a basis for the controller design. The servo controller is then derived with only one parameter varying in different working conditions. Based on the physical system model, an auto-commissioning and adaptive parameter tuning method is proposed after obtaining the knowledge of this key model parameter. The controller can then be commissioned and tuned to achieve the desired performance with high repeatability in different conditions. A servo permanent magnet synchronous machine (PMSM) driven hydraulic system was taken as an example and the experimental result shows that the proposed method exhibits reproducibility of system performance among different machine configurations and at different working points.

II. ELECTRO-HYDRAULIC SYSTEM MODELING

A servo hydraulic pump system is taken as an example, as illustrated in Fig.1. The system is composed of the variable speed drive, the servo motor (PMSM) and the hydraulic pump, etc. The hydraulics cylinder is selected as the load. In order to derive the auto-commissioning or adaptive estimation

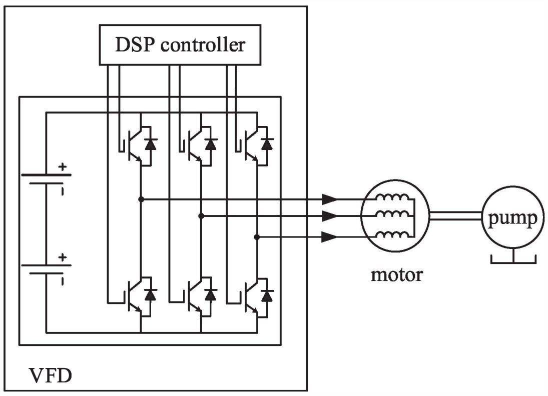

Fig. 1. Servo hydraulic pump system structure.

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method, the electro-hydraulic system plant model is developed.

To establish the mathematical model, assume the pump leakage to be laminar flow, ignore cylinder leakage, pipeline (including valve) pressure loss and dynamic process, and take no consideration of the pulsation of oil supply from pump. Based on these hypotheses, the flow equation can be derived as following.

$$Q_B = Q_R + Q_L = D_P n \tag{1}$$

Where Q_B is hydraulic pump theoretical output flow, Q_R is hydraulic pump practical output flow, Q_L is hydraulic pump practical total leak flow, D_P is pump's displacement, n is the speed of motor.

Flow continuity equation is

$$Q_B = A_T v + Q_L + \frac{V_t}{\beta}\frac{dP_R}{dt} \tag{2}$$

$$Q_L = C_{se} P_R = \frac{C_P}{\mu_T} P_R \tag{3}$$

Where A_T is the effective area with high pressure oil on the hydraulic cylinder piston, V_t is oil volume of pump output chamber and the part of hydraulic cylinder on the oil's side, β is elastic modulus of the oil, C_P is pump leak coefficient, P_R is pump outlet pressure, μ_T is dynamic viscosity of the oil, C_{se} is the total leakage coefficient of the pump, wherein comprises the hydraulic fluid viscosity term.

Establish cylinder dynamic equations taken piston rod as the object

$$A_T P_R = m_t \frac{dv}{dt} + Bv \tag{4}$$

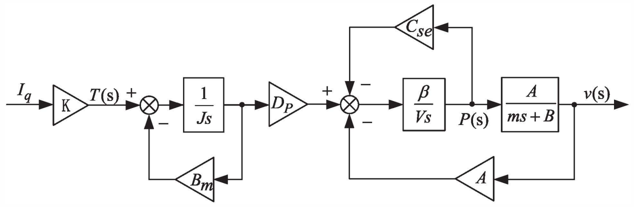

Fig. 2. Electro-hydraulic plant model for the servo hydraulic pump.

Where v is hydraulic cylinder piston velocity, m_t is total mass of load and piston, B is viscous friction coefficient of hydraulic cylinders. According to the formula above, establish hydraulic system block diagram shown in Fig.2.

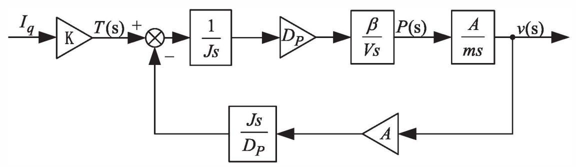

Fig. 3. Modified electro-hydraulic plant model.

Ignoring hydraulic pump leakage, hydraulic cylinder viscous friction and motor friction, the block diagram can be transferred into Fig.3.

Take the pressure signal as an output signal, for pressure is a major concern during pressure control mode of electro-hydraulic system. Combine parameters and a simplified block diagram is obtained as Fig.4. Define $K_T = \frac{A^2 J}{m D_P}, K_f = \frac{D_P \beta}{V}$.

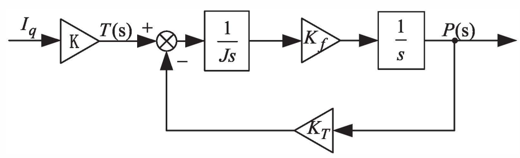

Fig. 4. Simplified electro-hydraulic control system.

As shown in the plant model in Fig.4, there are 3 parameters in the physical plant: the inertia J, the ratio of the load torque to the pressure at the pump outlet K_T, and the hydraulic loop characteristic parameter K_f. The inertia of the motor and pump doesn't vary among different working points or during long-time operation, and there is no significant variance of the inertia among products. For this reason, the inertia J is considered constant and can be measured accurately during the product development, and thus there is no need for on-line estimation. The parameter K_T, which is totally decided by the selected hydraulic pump, is in a similar case. And thus auto-commissioning should not consider the variance of those parameters. K_f represents the ratio of the derivative of the pressure (bar) to the shaft speed (rad/s), as given by (5), which is similar to the spring coefficient.

$$K_f = \frac{\dot{P}}{\omega} \tag{5}$$

K_f varies during operation and among different machine applications for three reasons. Firstly, for a whole hydraulic loop, consisting of oil pipes, hydraulic valve, hydro cylinder, oil tank, etc, K_f differs among machine applications because of different mechanical configurations. Secondly, K_f varies as the outlet pressure varies due to the non-linearity of the oil hose, i.e., K_f is not constant when operating at different pressure working points. Therefore, an auto-commissioning and auto-tuning method based on K_f estimation is required for the performance consistency of different machines and at different operation conditions.

III. Auto-Commissioning and Auto-Tuning Technique for System Controller Parameters

A. Parameter Estimation Method

To solve the problem of K_f variation as described above, an auto-commissioning method is proposed to measure the parameter K_f automatically before the system start-up and during machine operation, and auto-commissioning and auto-tuning of controller parameters can be derived based on the estimated information.

The servo pump system controller includes two functions: the speed control and the output pressure control, as shown in Fig. 5. In pressure control mode, the pressure holding precision and pressure rising time are critical to the system performance.

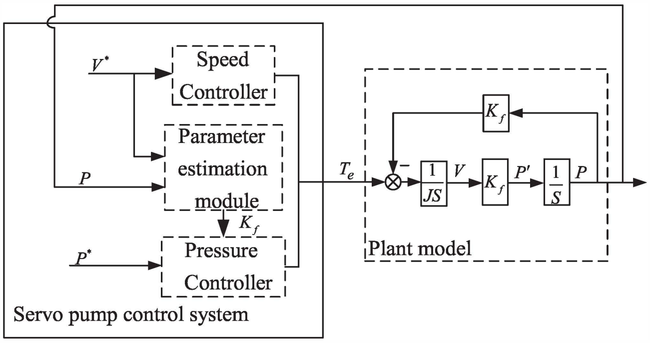

Fig. 5. Electro-hydraulic control system model for the servo hydraulic pump.

The proposed method is to apply a constant motor speed and calculate the derivative of the output pressure to acquire parameter K_f which is a function of pressure level. In practice, a constant speed command ω* is given, then the motor will drive the pump to spin with constant speed. And the pressure starts to rise up due to the "load integrator" modeled in Fig.5. The pressure signal is recorded and the waveform will be utilized for the following process. This process can be performed manually before system start-up, and can also be naturally a part of a machine duty cycle.

In order to reject the signal noise and pressure ripple produced by the pump, pressure signal needs to be filtered with a linear-phase filter to become a smooth and monotonic waveform. Then the derivative of the pressure can be calculated in the digital controller. In the experiment, considering the data processing capability and memory size of the digital controller, the pressure response curve can be approximated to a piecewise line.

Fig.6 is shown as an example. In this figure, the pressure curve is approximated to a three-segment line of which the knee point is at the pressure of 30bar, 40bar. The time instants are recorded when the pressure reaches certain points such as 20bar, 30bar, 40bar and 80bar. The knee points are chosen based on experience. After the time instants are known, the

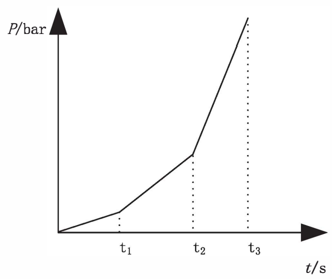

Fig. 6. Pressure rising curve under constant speed control condition.

three slope values, which represent the continuously changing slope of pressure in different pressure working points, can be calculated in the digital controller. And as shown in (6), the knowledge of the pressure-varying parameter K_f has been acquired automatically without the need of human intervention.

$$K_f(k) = \frac{\dot{P}(k)}{\omega} = \frac{P(k+1) - P(k)}{\omega\left[t(k+1) - t(k)\right]} \tag{6}$$

B. Control System Design

For the linear system shown in Fig.5, the traditional PID control scheme is the most widely and popularly adopted method in the industry. The system pressure level, which produces the motor load torque, can be regarded as the system disturbance to be rejected by feedback controller. Besides, the hydraulic plant is actually a varying load with a pressure-varying gain K_f, which results in time-variant property during different pressure working points. However, after we learn the K_f parameter in advance and identify the hydraulic system parameters at different working points with the process described in Part A, the traditional PID control performance can be improved with two methods: disturbance decoupling and load correction. Fig.7 shows the virtual system plant model after disturbance decoupling and load correction as well as the simplified equivalent plant model.

Firstly, the varying load can be corrected by a load correction item with the knowledge of K_f under different pressure. The varying parameter K_f can be cancelled out by the parameter $1/K_f$ in the controller. As a result, the "virtual system plant model" after the compensation becomes constant and invariant, which guarantees a consistent pressure response when aiming at different pressure levels. Thus the response

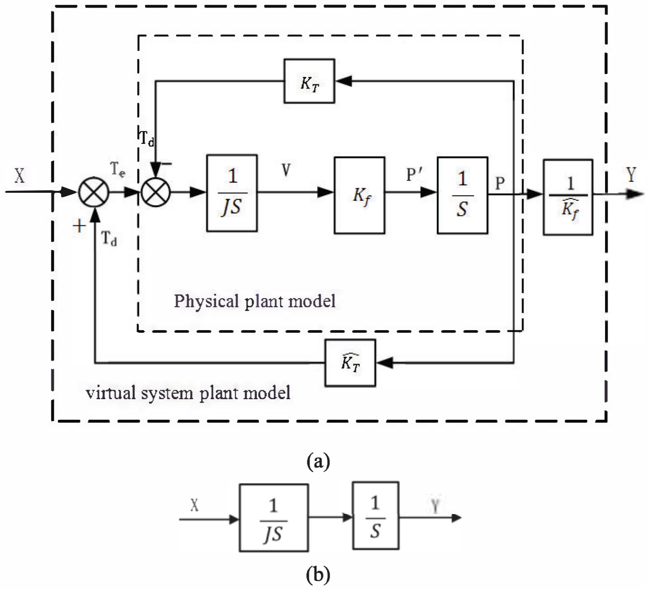

Fig. 7. (a) Virtual system plant model after disturbance decoupling and load correction; (b) Simplified equivalent plant model

consistency can be successfully achieved with the information of a precisely estimated parameter $1/K_f$. Fig.7 shows the virtual system plant model after disturbance decoupling and load correction as well as the simplified equivalent plant model, which makes great contribution to convenient and effective controller design.

IV. EXPERIMENTAL RESULTS AND ANALYSIS

Tests were performed in the laboratory testbed shown in Fig.1. The controller is TMS320F28335. The rated power, rated current and rated voltage of the PMSM are relatively 30kW, 53A and 380V. The switching frequency is 4kHz. The system is controlled to rotate at a constant speed, in which condition the pressure rises from zero to the protection value. The waveform during a ramp starting of pressure is shown in the following figure, and the calculated K_f at different time point and at different pressure is shown below. From Fig.8, we can obtain the key parameter of the model in different condition.

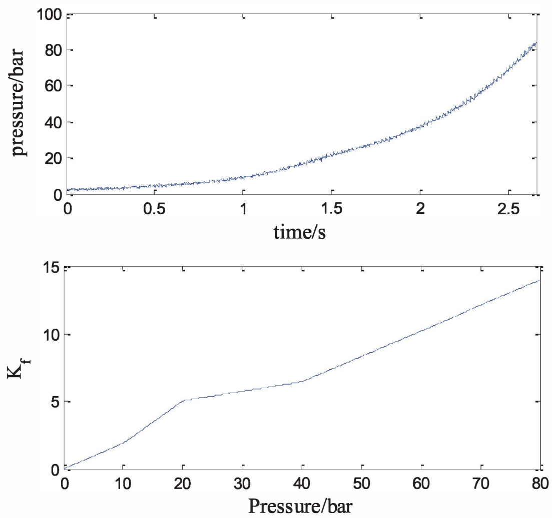

Fig. 8. Experimental result of auto-commissioning process (pressure signal and estimated K_f)

Having the full knowledge of the plant model, a PID controller was designed to perform the pressure control function in the electro-hydraulic system. We made compensation in the physical plant model to eliminate the effect of the parameter K_f as discussed in the last section. Fig.9 shows the outlet pressure control performance showing current, speed and pressure respectively from top to bottom. It can be seen that the pressure precisely track the pressure command. A zoom-in figure of the pressure response is shown in Fig.10. It shows that good dynamic performance is achieved with auto-commissioning method because precise control is performed at different working points throughout the pressure rising period. The auto-commissioning method is tested on another machine in order to illustrate the reproducibility..

For comparison, experiment is performed without the knowledge of K_f from auto-commissioning. A traditional PID control system is adopted. Fig.8 shows the pressure control results. It can be seen that both the dynamic and steady performance is worse than that with the knowledge of K_f from auto-commissioning due to the parameter detuning.

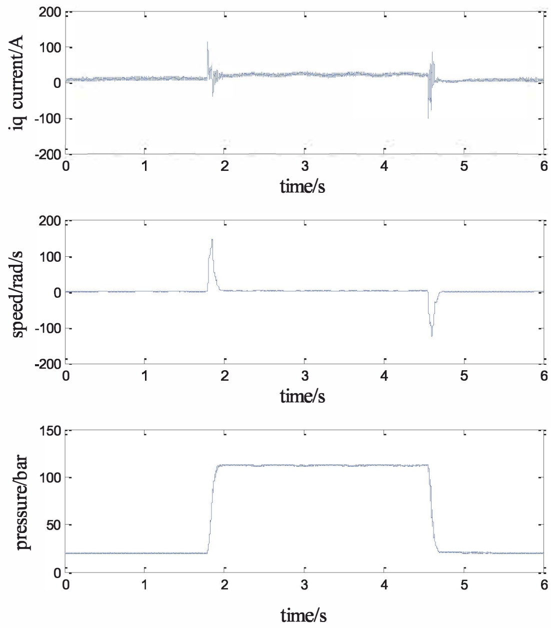

Fig. 9. Pressure control performance (current [A], speed [rad/s] and pressure [bar])

V. CONCLUSION AND DISCUSSION

In this paper, a servo hydraulic pump system is taken as an example to illustrate an auto-commissioning and adaptive tuning method. The simplified model of the electro-hydraulic

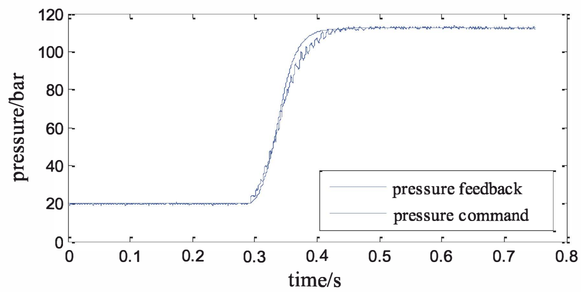

Fig.10. Pressure-tracking response after auto-commissioning (pressure signal from 20bar to 110bar)

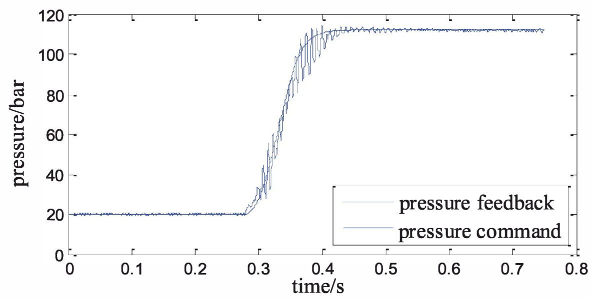

Fig.11. Pressure-tracking response without auto-commissioning (pressure signal from 20bar to 110bar)

system plant is established, then an auto-commissioning method is proposed based on the physical plant model. The hydraulic loop parameter which is time-variant is off-line or on-line estimated and then used for auto-commissioning and adaptive tuning of control parameters. The proposed method is verified with experiment to achieve high repeatability of dynamic and steady-state performance in different working conditions, and requires zero effort for parameter tuning.

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