Study on Ultimate Performance of Light-duty Electro-Hydraulic Torque-Load Simulator

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Abstract—**In order to improve the ultimate performance of Light-Duty Load Simulator, a design with electro-hydraulic loading scheme is discussed in this paper. Some key points of the mechanism of Light-Duty Electro-Hydraulic Load Simulator (HLS) are researched and revealed. Based on analysis of mathematical model and control strategy, a compound controller including disturbance observer, fuzzy-PID and velocity synchronizing decoupling compensation is designed. Results of experiments show that the compound controller improves the ultimate performance of system. In application to a special aircraft actuator test, the HLS is demanded to track the reference signal with small amplitude and high frequency for emulating the state of spring board loading system. For the special test, a new compensation network is created. Results of simulation show its effects on reducing the tracing error.**

Keywords—hydraulic load simulator, light-duty, ultimate performance, compound controll, high-frequency actuator test, motion disturbance

I. INTRODUCTION

The Electro-hydraulic Load Simulator (HLS) is a typical force servo control system with motion disturbance, which exerts loads while moving with actuator. It is widely used in actuator test or flight control simulation by tracking air-dynamic load precisely.

With the rapid improvement of aircraft cruise velocity, overload and dynamic performance in recent years, it has been critical issues that how to make loading system with higher ultimate performance. It brings a new challenge for the study on Electro-Hydraulic Load Simulator. Moreover, with the urgent demand for accurate attack, many high maneuverable and precise aircrafts such as air-to-air missile, anti-tank missile and tactical pilot less aircraft put forward new target to the ground simulation on actuators and ultimate performance of the testing system. All of these challenge the small-torque electro- hydraulic load simulators.

Up to now, the electro-hydraulic load simulator with maximum torque low than 20Nm is not prevalent in China. Research [1] shows that the minimal electro-hydraulic load simulator S-105-0.5 made by ACUTRONIC can output 57Nm torque. Beijing University of Aeronautics and Astronautics (BUAA) have studied on small-torque electro- motor load simulator for many years, and research [2] presents a simulator with 20-100Nm output torque. Research [3] shows that the electro-motor load simulator developed by BUAA can output torque no more than 10Nm. However, compared with electro-hydraulic simulators, small-torque electric-motor load simulator has not been used widely for larger inertia and narrow frequency band.

Based on analysis above, BUAA has successfully developed a HLS with 20Nm output torque, and is studying on simulators with 6Nm output torque now.

II. KEY POINTS IN RESEARCH ON LIGHT-DUTY HLS

According to actual demand of the loaded objects, ultimate performance of the researched little-torque HLS can be divided into four aspects as follows.

A. *Minimal loading torque (No more than 20Nm);*

Minimal loading torque challenges designing for hydraulic load actuator system.

B. *Demand for minimal inertia load;*

It is a complex issue to select and design the torque sensors, angle sensors and the rotor in hydraulic motor.

C. *High precision of dynamic tracing of torque table;*

It presents high demand for the dual ten frequency band (magnitude 10% & phase10deg) which can reach 20Hz. And it requests to widen the flat part in system frequency character curve as more as possible. Furthermore, it also requests strong capability to eliminate the motion disturbance.

D. *Frequency characteristic testing to the actuator with small amplitude and high frequency。*

It is demanded to apply active loading carried out by loading simulator other than passive loading implemented by spring board to measure the actuator's frequency characteristic. It requests that loading simulator has both high frequency band without disturbance and good performance tracing small amplitude and high frequency load.

High loading frequency band without disturbance demands that the system has enough mechanical loading stiffness. In this case, optimal generalized coupled stiffness described in research [4] will not be applicable.

III. FRAMEWORK DESIGNING FOR SYSTEM

The whole framework and principle of system uses the one described in research [5], which structure is depicted in Fig 1. As shown in Figure 1, torque sensor, hydraulic loading motor

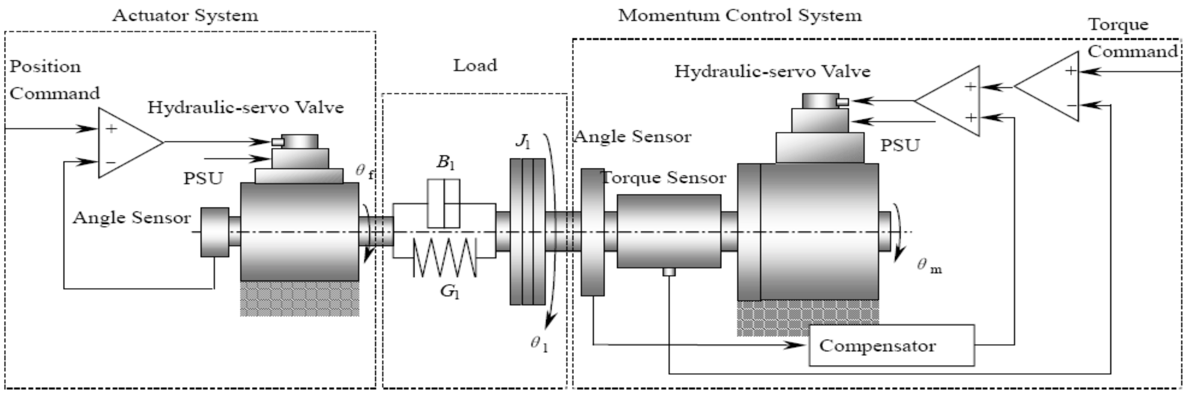


Figure 1. The conventional architecture of the load simulator

and loaded actuator are rigidly connected with the same axis.

Aimed at the key points listed in section II, design scheme in this paper adopts a new type small-torque hydraulic motor with limited rolling angles, and the special small-torque sensor. At the same time, scheme in this paper also select high precise angle position sensor and servo-valve with high performance. Designing principles adopted in this paper are involved as small torque, low inertia and high stiffness.

A. Hydraulic actuating mechanism

Based on the demand for 20Nm maximum loading torque, this paper designs a kind of smart hydraulic motor with limited rolling angles shown in Figure 2.

Figure 2. .20Nm loading hydraulic mortar

The smart hydraulic motor mainly has three characters listed as follows.

- The motor has small torque and small displacement which brings difficulties in designing process. Compact structure can be achieved by optimizing design parameters, reducing rotor inertia and adopting integrative vane manufacture technique.
- Little friction can be obtained by using Static Pressure Supporting technique combined with special sliding bearing.
- Internal leakage control. Due to the small volume of motor, the seal area causing leakage is relative big. Therefore, it is more difficult to suppress the leakage in small motor than in big one. The internal leakage can be limit to 10% successfully by the precise manufacture and surface spray on friction pairs.

According to the characters above, the motor can get working parameters as follows. Displacement is 1.54cm3/rad, and max output torque is 20Nm in case of 600deg/s max speed. The max load flow can reach 1.1Lpm.

B. Sensors and loading channel shafting

Designing principles of torque sensors are limited by three conditions listed as follows.

- System's stiffness is assured as high as possible if the inertia is permit. And it can improve the system loading frequency width without disturbance.
- Dimension and stiffness of spring parts in system must consider sensitivity effect of the strain foil.
- Solving spaces non-linear by adopting special non-gap friction shaft coupling.

Based on the restrictions above, designing scheme in this paper adopts following techniques.

- To satisfy rigorous frequency width and inertia demand, this paper combined torque sensors with loading shaft as a part which is machined integrally and is calibrated after attaching strain foil.
- Select angle encoder with super low inertia.
- Adopt Backlash-free shaft/hub connections.
- A cantilever style is adopted. It is in order to reduce the friction and inertia that flange of motor is the only one supporting and assemble point.

C. Servo Valve

This paper adopts two stage flow control servo valve with mechanical feedback pilot stage MOOG-G761-3001 made by MOOG corporation [6], which frequency band can achieve more than 200Hz. This design can make system be free of servo valve's effect.

Based on the designing technique above, the structure of loading channel is described as Figure 3.

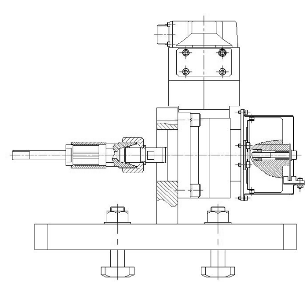


Figure 3. The assembly of load channel.

IV. MATHMATICAL MODEL OF SYSTEM

As shown in section III, load simulator is used to measure actuator's frequency characteristic and trace loads with small amplitude and high frequency, so mathematical model of servo valve can be calculated with second-order model as follows:

$$G_{sv}(S)=\frac{x_V}{I_{sv}}=\frac{K_s\omega_{sv}^2}{S^2+2\xi_{sv}\omega_{sv}S+\omega_{sv}^2} \tag{1}$$

Equation of torque sensor is as follows.

$$U_f=K_F\cdot M \tag{2}$$

$$M=G_\varepsilon(\theta_m-\theta_l) \tag{3}$$

Equation of the current amplifier of the servo valve drive is as follows.

$$I_{sv}=K_{V/I}\cdot U_c \tag{4}$$

Equation of the flow into the motor is as follows.

$$Q_f=D_m\frac{d\theta_m}{dt}+\frac{V_m}{4E_y}\frac{dp_f}{dt}+C_{sl}p_f \tag{5}$$

Equation of the rotor of hydraulic motor is as follows.

$$D_m p_f=J_m\frac{d^2\theta_m}{dt^2}+B_m\frac{d\theta_m}{dt}+G_\varepsilon(\theta_m-\theta_l) \tag{6}$$

Equation of the load is as follows.

$$G_\varepsilon(\theta_m-\theta_l)=J_l\frac{d^2\theta_l}{dt}+B_l\frac{d\theta_l}{dt}+G_l(\theta_l-\theta_f) \tag{7}$$

Where:

$K_{tm}=K_c+C_{sl}$, The total stiffness coefficient;
Q_f, The flow rate of load (m^3/s);
p_f, The pressure of load (N/m^2);
E_y, The Volume Bulk Module (N/m^2);
K_Q, The flow rate gain of servo valve (m^2/s);
D_m, The displacement of motor (m^3/rad);
C_{sl}, The leakage coefficient of motor ($m^5/N\cdot s$);
J_l, The load initial($Kg\cdot m^2$);
G_ε, The stiffness of torque sensor ($N\cdot m/rad$);
B_l, The damping of the load ($N\cdot m\cdot s/rad$);
M, The output torque ($N\cdot m$);
θ_l, The angle displacement of load (rad);
θ_f, The angle of actuator output shaft (rad);
U_f, The output voltage of torque sensor (V);
U_c, The output voltage of controller (V);
G_l, The rotation stiffness of load ($N\cdot m/rad$);
$G_{sv}(S)$, The transfer function of servo valve;
I_{sv}, The input current of servo valve (A);
K_{sv}, The gain of servo valve (m/A);
ξ_{sv}, The damping of servo valve (rad);
ω_{sv}, The cut-off frequency of servo valve (rad/s);
$K_{V/I}$, The gain of current amplifier (A/V);
K_c, The whole factor of flow rate to pressure ($m^5/N\cdot s$);
x_V, The displacement of servo valve spool displacement (m);
V_m, The total control oil volume of hydraulic motor (m^3);
J_m, The rotor inertial of the hydraulic mortar($Kg\cdot m^2$);
B_m, The damping of the hydraulic mortar($N\cdot m\cdot s/rad$);
θ_m, The swiveled angle of hydraulic motor output shaft(rad);
K_F, The transfer coefficient of torque sensor ($V/N\cdot m$);

The accurate mathematical model of hydraulic load simulator is shown in Fig. 4.

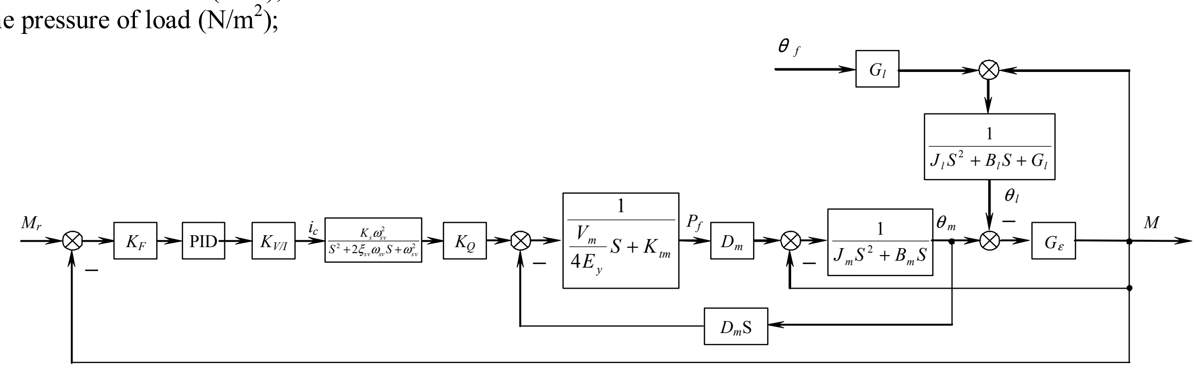


Figure 4. The accurate model of hydraulic load simulator.

V. CONTROL METHOD OF TORQUE CLOSED LOOP SYSTEM

Aimed at small-torque load simulator being sensitive to friction, this paper designs a parallel controller combined fuzzy-PID and common-PID arithmetic methods as shown in Figure 5.

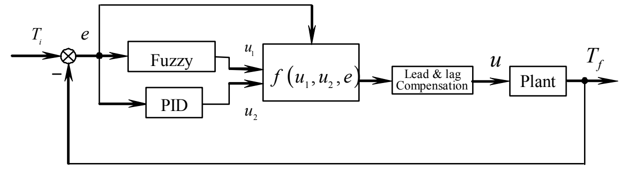


Figure 5. The complex controller of HLS's torque close-loop.

In Figure 5, system adopts exponential switch instead of traditional simple addition. It does not need to set the threshold when use exponential switch method, the outputs of fuzzy-PID and common-PID are mixed in the range from zero to infinity.

The switch speed is determined by "a". The switch is more acute when "a" is larger [7].

$$u=f(u_1,u_2,e)=u_1\left(1-\exp(-a|e|)\right)+u_2\exp(-a|e|) \tag{8}$$

Based on MOOG servo valve's high performance more than 200Hz, the ultimate frequency characteristic is free of restriction of the valves in system. In this case, stiffness and vibration modal of loading mechanism become the main factors restricting ultimate performance of system. In order to let system character overcome load natural frequency, before controller output reaches servo valve, the lead & lag compensation item is introduced into system to restrain frequency overshoot and broaden system loading frequency band without disturbance.

VI. COMPENSATION FOR MOTION DISTURBANCE

According to research [8], motion disturbance relates with actuator's velocity and accelerator, and velocity synchronizing control is still a powerful method to eliminate extraneous torque. To improve the robust of against outer disturbance and non-linear friction system, the Dynamic Robust Compensation [9] is designed through robust compensator introduced into the velocity synchronizing control [8] network, which is shown as figure 6. The velocity synchronizing control is used to suppress strong disturbance, while the robust compensator is adopted to reduce the tracing error created by nonlinearity, such as load actuator friction.

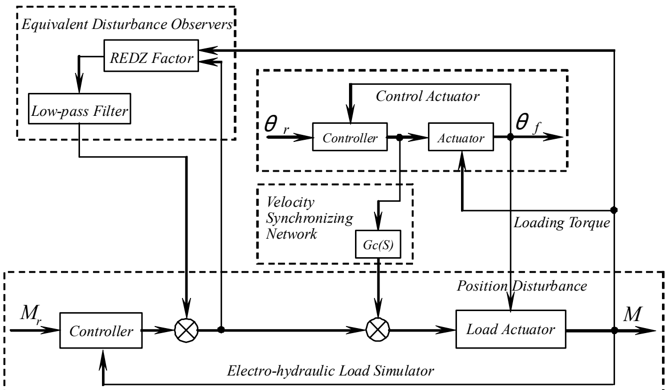


Figure 6. The Dynamic Robust Compensation(DRC) scheme.

VII. FREQUENCY CHARACTERISTIC TESTING OF ACTUATOR WITH SMALL AMPLITUDE AND HIGH FREQUENCY

Up to now, standard loads for actuator frequency testing are still implemented by spring boards. However, with wide use of load simulators, substituting active loading with load simulator for passive loading with spring board becomes more and more urgent.

In this case, simulator operates in form of grads tracking. Torque input and moving disturbance are all sweep sine signal with small amplitude and high frequency.

A high band width without disturbance of load simulator is requested. Furthermore, the dual ten frequency characteristic is asked wide enough.

Usually, the -3dB frequency band ω_{nc}(Hz) of small-torque actuator is from 30Hz to 50Hz. During the frequency test, the stimulus of angle actuator is sine sweep signal from 0.1Hz to 50Hz which the amplitude is no more than 1 degree. The spring board is fixed to a certain point in order to apply required linear load on actuator.

For instance, the stimulus A_{in} of actuator is a small amplitude sine wave with a amplitude A(deg) and frequency ω_{nc}(Hz). The output of actuator is about 0.707A (deg) sine wave with about 60(deg) phase lag. The load ratio is K. In order to emulate the spring board, the real torque stimulus must be:

$$T_{in} = KA_{out} \tag{9}$$

Because the frequency of stimulus is rather high, the amplitude change and phase lag of load simulator are big. It is so difficult to compensate the error and the system may be not stable.

The following schemes are proposed:

- Product of actuator stimulus A_{in} and load ration K is used as a nominal input load signal $\hat{T}_{in} = KA_{out}$ to excite the HLS.
- Zero-phase error feed forward control [10][11] [12] is adopted to reduce the phase lag of $\hat{T}_{in}$.
- Velocity synchronizing control is used to suppress the motion disturbance.
- Product of position sensor feedback A_{out} with high accuracy and load ratio K is the real load table $T_c = KA_{out}$ as a compensation signal. The difference of T_c and real torque feedback is used to multiply with a constant gain K_c. Then it is added to the output of controller.

The structure of compensation is depicted in Fig 7.

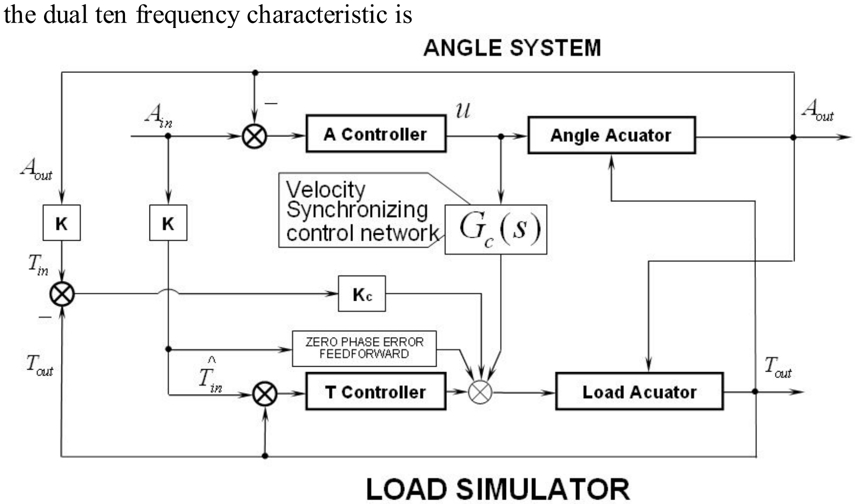


Figure 7. The scheme of the compensation in special aircraft actuator's frequency characteristic test by HLS.

VIII. Simulation and Experiments

This paper studied the 20Nm HLS (as shown in Figure 8) developed by BUAA.

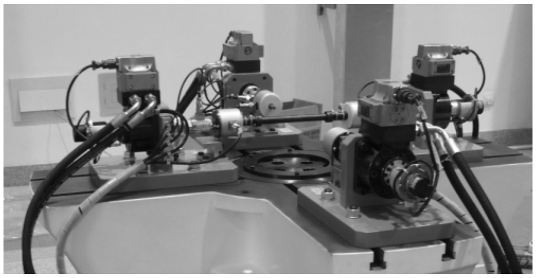


Figure 8. The test rig of the actuator and its loading system.

A. *Experiments on operating states of the Fuzz controller*

By inputting step signals to actuator system, torque orders and feedback curves can be obtained as shown in Figure 8. and parameters self-adjusting curve of the fuzz controller is shown in Figure 9.

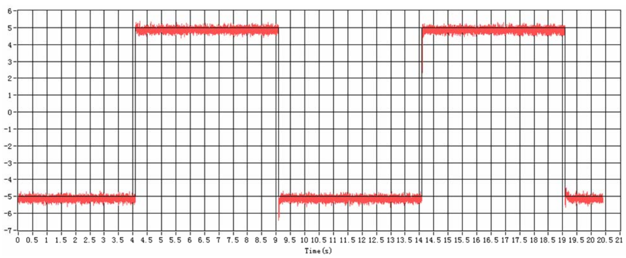


Figure 9. The step response of HLS by Fuzzy-PID controller test.

As shown in Figure 9, system raising time to response step signal is 5 mm, and in this case, system has no overshoot.

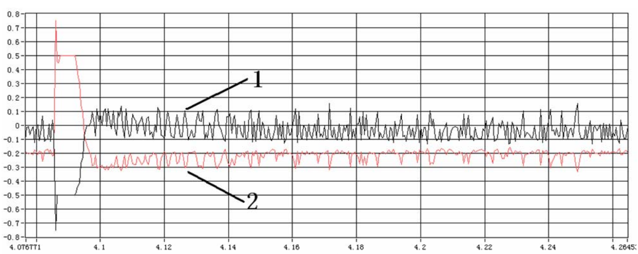
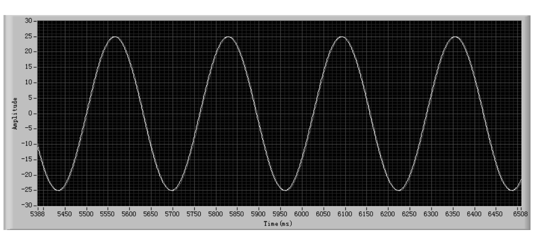


Figure 10. The self-adjusting of parameters in Fuzzy-PID controller test.

The curve 1 in Figure 10 describes the varying process of ΔKp in Fuzz PID controller vs. time. The curve 2 describes varying process of ΔKi in Fuzz PID controller vs. time.

B. *Experiment on Eliminating extraneous torque*

When the stimulus A_{in} of actuator is 25deg-3.8Hz sine wave (with max speed of system), by inputting 0Nm signals to HLS system, torque orders and feedback curves can be obtained as shown in the upper Figure 11.



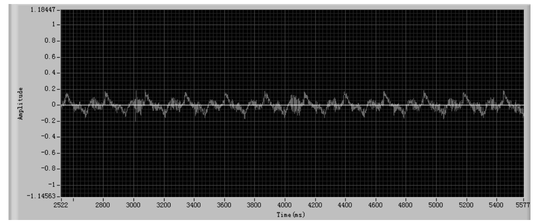


Figure 11. The test to eliminate the extraneous torque.

In the lower curve of figure 10 the tracing error describes the very small extraneous torque with the p-p value of 0.2Nm.

C. *Frequence response experiment of HLS without disturbance*

The simulator's frequency characteristic without motion disturbance can be measured as shown in Figure 12.

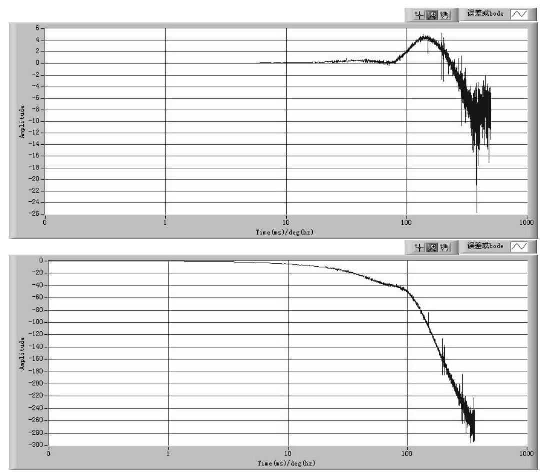


Figure 12. The frequency response of HLS without disturbance.

In the upper curve of Figure 12, magnitude-frequency characteristics of -3dB is 260Hz. In the lower curve, phase-frequency characteristics of -90deg is 134Hz. The dual ten frequency band (magnitude 10% & phase10deg) is 20Hz.

D. *Simulation on frequency characteristic test of actuator with small amplitude and high frequency*

During the simulation, the stimulus of actuator is 1deg-40Hz sine wave, and load ration K is 10Nm/deg. Angle orders and feedback curves can be obtained as shown in Figure 13. The lag of phase is big.

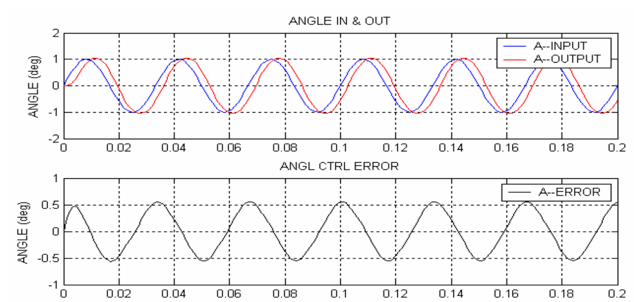


Figure 13. The simulation of angle actuator system.

Product of position sensor feedback A_{out} with high accuracy and load ratio K is used as torque stimulus $T_{in} = KA_{out}$. The simulation of HLS curve can be obtained as shown in Figure

14. In this case of high frequency stimulus, the lag of phase is big as angle actuator, which leads to huge load tracing error.

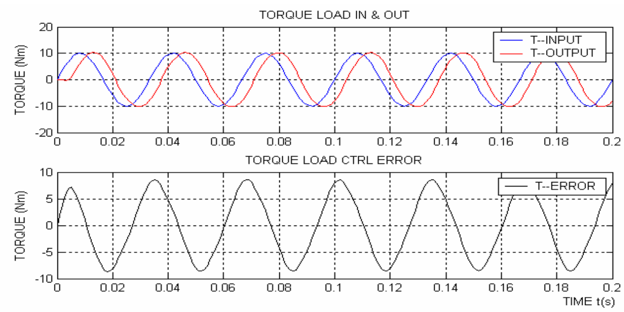


Figure 14. The simulation of HLS with real stimulus T_{in}.

Product of actuator stimulus A_{in} and load ration K is used as a nominal input load $\hat{T}_{in} = KA_{out}$ to excite the HLS, while the Zero-phase error feed forward is adopted. The simulation of HLS curve can be obtained as shown in Figure 15. The lag of phase is smaller, but it cannot be used in the real system because this nominal input is not compared with real load table.

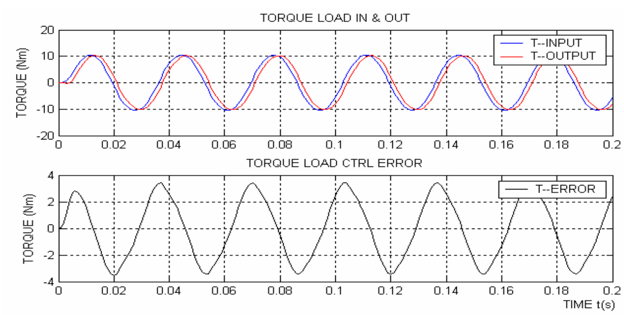


Figure 15. The simulation of HLS with nominal stimulus $\hat{T}_{in}$.

The structure of compound compensation depicted in Fig 7 is added to the system when the gain factor K_c has been adjusted well. Results of simulation show that the special designed compensation network is effective on reducing the tracing error and emulating the situation as spring board as shown in Figure 16.

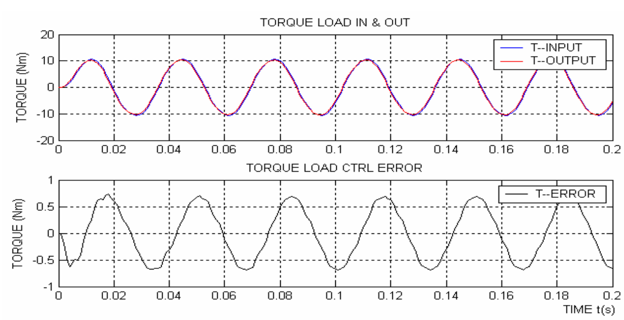


Figure 16. The simulation of HLS with the special compensation

IX. CONCLUSION

This paper summarizes the critical issues on Light-Duty HLS, and clearly discloses factors influencing its ultimate performance. By deeply analyzed from system framework and control methods, some conclusions can be obtained as follows.

According to demanding for high-performance actuator loading test, electro-hydraulic servo system can be successfully used to Light-Duty load simulator, which has higher performance than electro-motor load simulator.

Mechanism of load actuator designed in this paper can improve shafting stiffness in case of limited inertia, and it can broaden the frequency band without disturbance.

Results of Experiment show that a compound controller including disturbance observer, fuzzy-PID and velocity synchronizing decoupling compensation is efficiently improves system ultimate performance.

Based on demanding for frequency characteristic test of actuator with small amplitude and high frequency by HLS, this paper presents a new compensation network structure. Results of simulation show that the special designed compensator is effective on reducing the tracing error and emulating the state of spring board. Its experimental result still need to be proved.

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