International Journal of Control and Automation
Vol. 9, No. 8 (2016), pp.31-40
http//dx.doi.org/10.14257/ijca.2016.9.8.04

Program Control with Motion Discipline Predicting of Electric Load System

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Abstract

Non-linear interference to load torque in the actuator load system, which is used to test the performance of the actuator. is a key factor in reducing the load accuracy of the load system. The disturbance torque error and load motor responding delay error are the main problem in control the Non-linear interference. To solve these problems, this paper proposes a program control method with motion discipline prediction of actuator. In the method, An actuator motion discipline predicting model is developed to obtain the accuracy disturbance torque and lower the disturbance torque error; A load torque command sending time model is established to obtain instruction set with timestamp and reduce the load motor responding delay error. The new method is compared with two traditional methods under certain frequency. A step motion experiment is carried out and the result indicates that the new method has obvious advantage in the field of load accuracy and bandwidth.

***Keywords**: Discipline prediction; Program load; Disturbance torque, Electric load system*

1. Introduction

Electric load simulator (ELS) is equipment which simulates load conditions of flight control surfaces under laboratory conditions and assesses actual performances of actuator system. During experiments, forced motion of load system generates a strong non-linear disturbance torque [1] that not only reduces the load accuracy of the loading system, but also easily generates system vibration, and even produces impact damage to connected tooling or tested actuator [2]. Therefore the study on disturbance torque suppression of ELS aims to improve load accuracy and load performance of the system.

The disturbance torque of ELS is induced by actuator motion. Zhang and Qi *et al*. [3,4] calculated the disturbing torque by system inertia and acceleration, but higher order differential of actuator angular makes difficult its practical application. Due to the nonlinear and uncertain disturbance torque, scholars put forward many control method such as adaptive control[5,6] , sliding mode variable structure control[7,14]. Currently intelligent control method was still in simulation phase, but a lot of theoretical researches on this issue had been developed[8]. For instance, Sebastian [9] adopted the processing ability of dynamic problems of diagonal recurrent neural network to gain correction values and real-time tuning of PID parameters, but there still existed convergence problems. Hua Qing [10] studied an compound control method combining iterative learning and PID which had a good inhibitory effect of the disturbance. Intelligent methods have been the endeavor of modern research [11]. Many researchers worked for pursuing more ideal methods of inhibiting disturbance torque in last decades [12]. Yuexuan Wang *et al*. [13] developed a simulation system to predict actuator position, with this system he avoid noise introduced by real-time position collecting and get more

ISSN: 2005-4297 IJCA

accurate disturbance torque. Jinying Zhang [14] identified position feedback signal of drive motor and constructed ideal speed compensation signal to inhibit disturbance torque.

This paper presents an program control method with actuator motion discipline prediction establishing the actuator motion simulation model to predicts the actuator motion discipline under the corresponding command, and building the load instruction sending time model to synchronous compensate disturbance torque.

2. ELS Disturbance Torque Analysis

As shown in Figure 1 the physical structure of ELS is mainly composed by four parts: load motor, measured actuator, torque sensor and encoder, where the load motor provide the main load torque for the system, and the torque sensor is used to measure actual load torque of measured actuator, and the encoder is used to measure angular displacement of measured actuator.

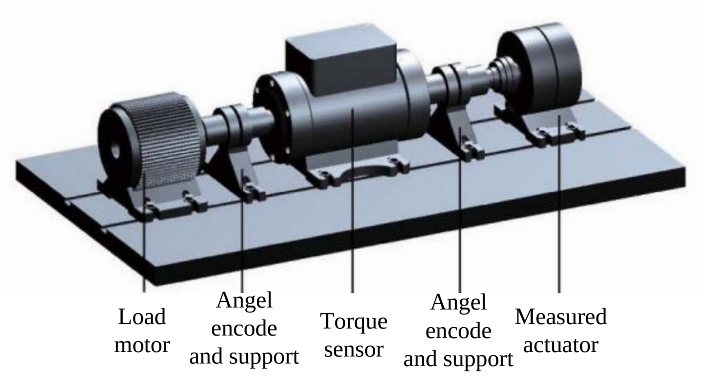


Figure 1. The Simplified Block Diagram of Actuator Electric Load System

The system control model is shown in Figure 2.

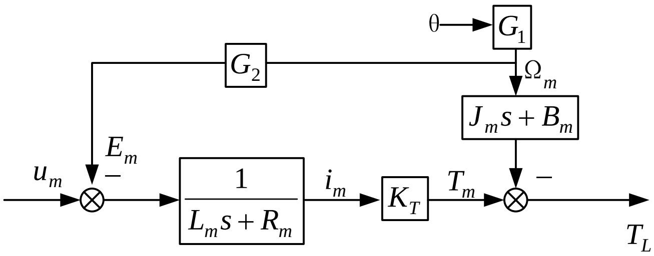


Figure 2. The ELS Control Diagram

In Figure 2, θ is the actual displacement collection amount of the actuator, and um is the voltage value corresponding to the required load torque of the actuator. When $u_m = 0$, it can be seen that as long as the actuator moves, the value of TL is bound to be nonzero and equals to the disturbance torque value.

3. Program Control with Motion Discipline Prediction

3.1. The Principle of Control Strategy

The diagram of program control mode with motion discipline prediction is shown in Figure 3. Firstly we send actuator motion command θ to actuator motion simulated system which outputs compensating torque command, then process this instruction by phase correction and actual load instruction of load motor is obtained. Therefore, the disturbance torque value and the phase correction time are the important factors on program control with motion discipline prediction. This paper will establish a actuator motion simulation system to simulate disturbance torque, and then adjusts the instruction sending time of load motor.

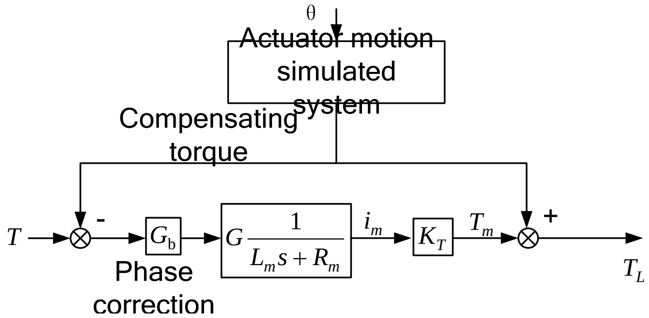


Figure 3. Program Control Strategy with Motion Discipline Prediction

3.2. Actuator Motion Simulated System

Taking electro-hydraulic proportional valve hydraulic actuator as the example, the transfer function of this actuator is described as Eq. (1):

$$Y(S) = \frac{U(S)K_qK_0/\left[A_1\left(\frac{S^2}{W_{SV}^2}+\frac{2\xi_{SV}}{W_{SV}}S+1\right)\right]-\frac{K_{ce}}{A_1^2}\left(1+\frac{V_t}{4\beta_e K_{ce}}S\right)F_L}{S\left(\frac{S^2}{W_h^2}+\frac{2\xi_h}{W_h}S+1\right)}\cdot K_f \quad (1)$$

Where Y is the actuator displacement, and U is the position command voltage, and K_q is the flow gain of electro-hydraulic proportional directional valve, and K_0 is the proportional amplification factor of the transduction of voltage signal into current signal, and K_f is the proportional amplification gain of the actuator position feedback, and A_1 is the effective piston area of rod chamber of actuator hydraulic cylinder, and W_{SV} is the natural frequency in electro-hydraulic proportional directional, and ξ_{SW} is the damping ratio of electro-hydraulic proportional directional valve, and K_{ce} is the total flow pressure coefficient, and ξ_h is the hydraulic damping ratio, and W_h is the hydraulic natural frequency, and β_e is the elastic modulus of effective volume, and V_t is the equivalent total volume, and F_L is the external load of the actuator and can be regarded as a constant. When the test occurs, U is the step voltage command, $U(S)=U/S$, the displacement function about time $y_{(t)}$ of the actuator can be calculated by Eq. (1).

It can be rewritten as:

$$Y(S) = U(S)\mathrm{G}_1(\mathrm{S}) + F_L\mathrm{G}_2(\mathrm{S}) \quad (2)$$

Where

$$\mathrm{G}_1(\mathrm{S}) = \frac{K_qK_0/\left[A_1\left(\frac{S^2}{W_{SV}^2}+\frac{2\xi_{SV}}{W_{SV}}S+1\right)\right]}{S\left(\frac{S^2}{W_h^2}+\frac{2\xi_h}{W_h}S+1\right)}\cdot K_f \quad (3)$$

$$\mathrm{G}_2(\mathrm{S}) = \frac{-\frac{K_{ce}}{A_1^2}\left(1+\frac{V_t}{4\beta_e K_{ce}}S\right)}{S\left(\frac{S^2}{W_h^2}+\frac{2\xi_h}{W_h}S+1\right)}\cdot K_f \quad (4)$$

Making U=0, we can recognize G1(S). Also ,making FL=0, we can recognize G2(S).

3.3. The Instruction Phase Correction of Load Motor

The determination of Δt_1 should comprehensively consider the response time of load system. It can be described as Eq. (5).

$$i_{(t)_j} - i_{j-1} = \frac{\left(u_j-u_{j-1}-E_{m_j}+E_{m_{j-1}}\right)}{R}\cdot\left(1-e^{-\frac{R}{L}t}\right) \quad (5)$$

Taking the *j*-th command as the study object t_{j-1} is the initial current, and u_{j-1} is the initial voltage, and E_{mj-1} is the initial back-EMF, and $i_{(t)j}$ denotes the current at the t moment of the *j*-th command starting from the *j*-th instruction. When $t=\Delta t_1$, in order to meet $K_T i(\Delta t_1)_j = T_{bj}$, that is

$$K_T i_{(\Delta t_1)_j} = K_T \cdot \left[\frac{\left(u_j - u_{j-1} - E_{m_j} + E_{m_{j-1}}\right)}{R} \cdot \left(1 - e^{-\frac{R}{L}\Delta t_1}\right) + i_{j-1} \right] = T - J \left.\frac{d_\theta^2}{d_t^2}\right|_{t=j\cdot\Delta t} \tag{6}$$

It can be obtained from the above equation that:

$$u_j = R \cdot \left[\left(\frac{T - J\left.\frac{d_\theta^2}{d_t^2}\right|_{t=j\cdot\Delta t}}{K_T} - i_{j-1} \right) \cdot \Big/ \left(1 - e^{-\frac{R}{L}\Delta t_1}\right) \right] + \left(E_{m_j} - E_{m_{j-1}} + u_{j-1}\right) \tag{7}$$

In Eq.(6), u_j is the function about Δt_1 where $0<u_j<u_{max}$ and $0<\Delta t_1<\Delta t$. u_{max} is the allowed maximum output voltage of controller. When j = 1,2,3 ... n, a curve cluster can be formed by n curves about u_j and Δt_1. As shown in Figure 4, select any point in time between 0 and Δt, draw a straight line parallel to the u axis. If there are points of intersection between the straight line and all the curves within the rectangle formed by both axes and the line of u_{max} and Δt, it illustrates that the point meets the requirements of Δt_1, and the value of u corresponding to the intersection of curve is the torque command value of motor at the time. And n command values form the load command array of motor.

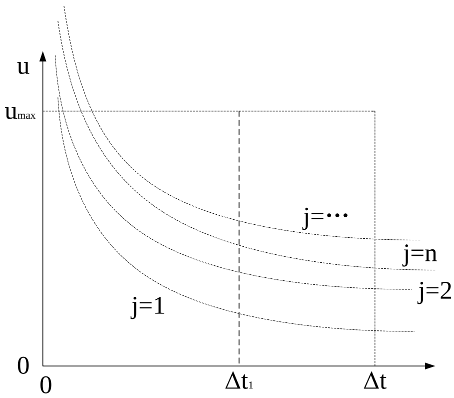


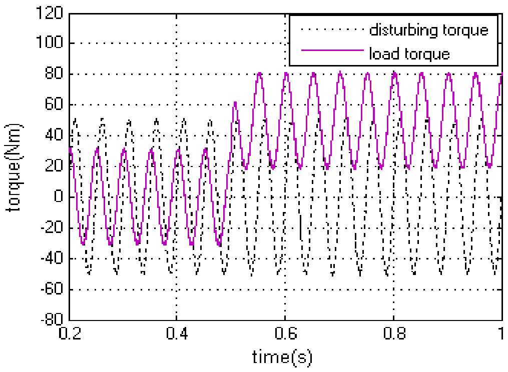
Figure 4 Command Array Generation oof Load Motor

4. Simulation Analysis

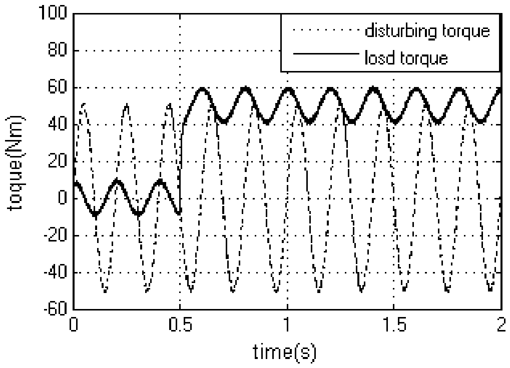
The simulation model developed in Matlab will be processed respectively torque feedback control, velocity feed-forward control and the proposed program control with motion discipline prediction. Corresponding control results of the above three methods will be compared, and further the influence of disturbance torque and phase-corrected time in the program control with motion discipline prediction to loading result will be analyzed.

4.1. The Comparison Analysis of Control Results

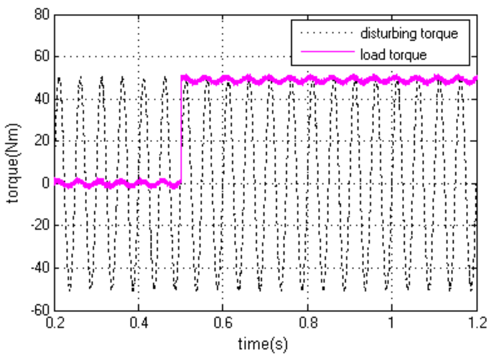
Respectively load an external noisy disturbance torque with amplitude of 50Nm at a frequency 20 Hz and 5Hz Three different control schemes with a command of 50Nm are used at 0.5 ms moment under two frequencies and control results are shown in Figure 8. Figure 5-a) describes the result of velocity feed-forward control under 20 HZ disturbance torque. As is seen, load torque is also a sine wave which similar to disturbance torque and its amplitude less than the one of disturbance torque when the stable stage reaches. The effect of disturbance torque is weakened by 40%. When the frequency of disturbance torque is dropped to 5 HZ with others keep unchanged, its control result is shown in Figure 5-b) and the effect of disturbance torque is weakened by 80%, which demonstrates that the fluctuation of load torque at the stabilization stage is decreased contrasting with Figure 8-a). It illustrates that the lower the frequency of disturbance torque is, the effect of velocity feed-forward control is better. Figure 5-c) shows the control result of closed-loop torque feedback control, As is seen, load torque still exists small oscillations, but has a relative great effect compared with velocity feed-forward control. The disturbance torque is weakened by 90% .Similarly when the frequency of disturbance torque is dropped to 5 HZ with others keep unchanged, its control result is shown in Figure 5-d) where the oscillation is lowered, but has some output deviation yet. In practice the high frequency component of actuator motion is generally much higher than 20Hz, and the system using torque feedback control method is easy to generate shock. Figure 5-e) is the result of the proposed control method which demonstrates an obvious suppression effect of disturbance torque and the effect of disturbance torque is weakened by 98%. The control result does not change when the frequency of disturbance torque is dropped to 5 HZ as shown in Figure 5-f). Through the aforementioned comparative analysis, the proposed method outperforms torque feedback control and velocity feed-forward control for disturbance torque suppression and is more suitable for a wider system frequency band.



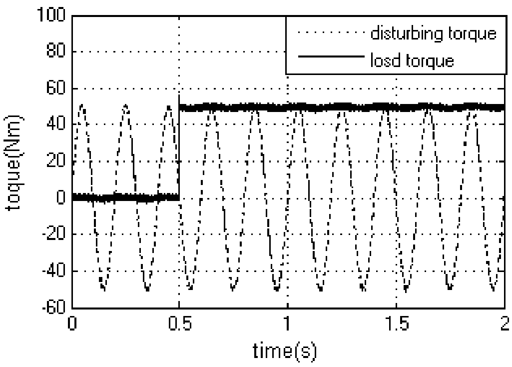
A) Torque Feed-Forward Control With 20HZ Disturbance Torque



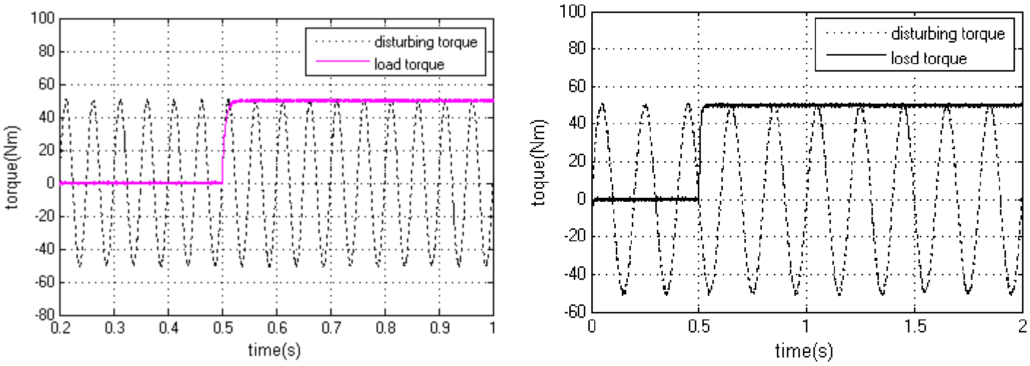
B) Torque Feed-Forward Control With 5HZ Disturbance Torque



C) Torque Feedback Control with 20HZ Disturbance Torque



D) Torque Feedback Control with 5HZ Disturbance Torque



e) Program Control Under 20HZ Disturbance Torque

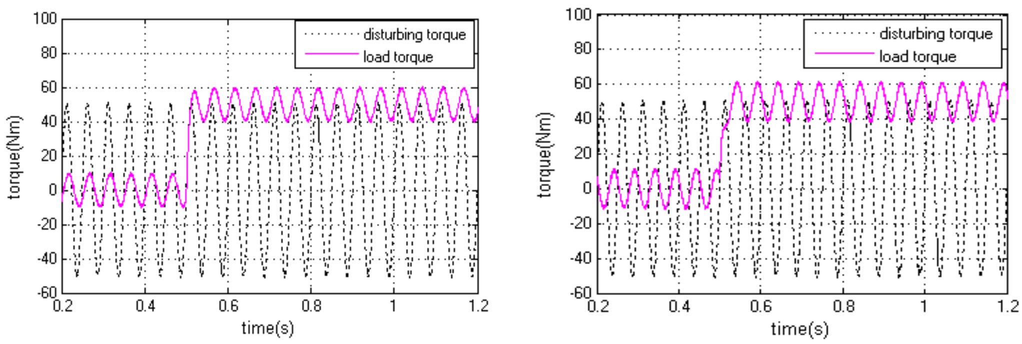
F) Program Control Under 5HZ Disturbance Torque

Figure 5. 50Nm Constant Torque Output of System with Disturbance Torque of 50Nm, 20Hz Or 5HZ

4.2. The Influence of Amplification Factor and Correction Time to Control Result

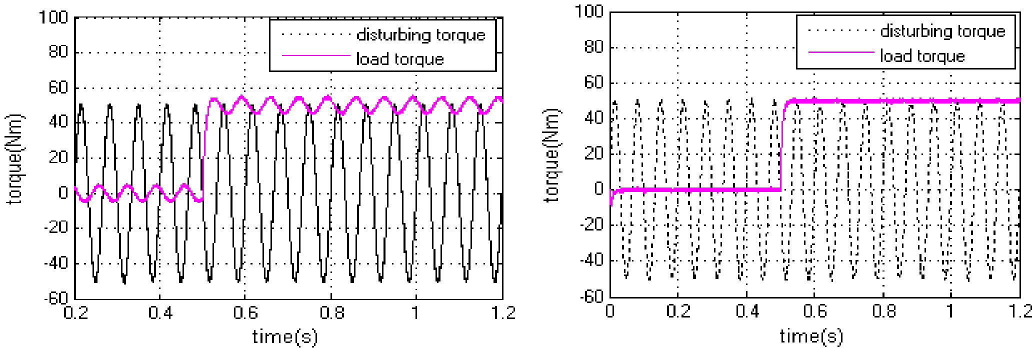
The higher the frequency of the load motor input command, the more obvious the decay of output. Therefore, the disturbance torque must be preceded by amplification calculation. The influence of the amplification factor to the control result is simulated as shown in Figure 6. The amplification factor is 1.2318 and the correction time is 4.96ms in Figure 5-d). When the amplification factor is 1 and 1.5, the corresponding results are shown in Figure 6-a) and Figure 6-b). It means that as the ideal amplification factor is a center, the larger the deviation value, the worse the control result. Figure 6-c) shows that the control result is still not ideal. When the amplification coefficient and correction time are not modified, namely that the amplification factor is 1.2318 and the correction time is 4.96 ms, and the disturbance signal is 15Hz and 50Nm. Through calculation, the amplification factor changes to 1.1364, while the correction time is 5.21ms, the control result is shown in Figure 6-d). Thus, the mentioned previously analysis demonstrates that when the frequencies of disturbance signals are different, corresponding amplification factors and correction times are different, and the amplification factor is related to the frequency amplitude of the disturbance signal.

In summary, the program control method with motion discipline prediction is to disperse the overall control process into an array of commands allocated a respective amplification factor and correction time. Moreover the size and frequency of disturbance torque is changing within the whole step motion process of actuator. According to the analysis in subsection 3.2, the longest correction time point can be selected as the desirable correction time of all discrete points, and the amplification factor changes with the nodes.



a) The Amplification Factor Is 1(20Hz) Is

B) The Amplification Factor 1.5(20Hz)



C) Amplification Factor Is 1.2318 and Correction Time Is 4.964ms (15Hz)

D) Amplification Factor Is 1.1364 and Correction Time Is 5.21ms (15Hz)

Figure 6. The Influence Analysis of Amplification Factors and Correction Times to Control Results

5. Experimental Verification

This paper has designed a load system using the proposed method for a research institute in Guizhou province. The initial design of test bench use of the torque feedback control, and its control result is shown in Figure 7. As is seen, dramatic torque fluctuations have occurred when actuator starts moving. The re-designed scheme adapts the velocity feed-forward control The 300Nm constant torque loading in step motion of actuator servos is shown in Figure 8 where the torque quickly changes from 0Nm to 300Nm when the actuator is starting, and then rapidly reduces to 0Nm at the moment of reaching the step position of actuator. The load torque changes in step process of actuator and not meet the load requirement. Moreover due to the extremely quick speed change of the step motion of actuator, rapidly changing torque may cause system impacts, even bring out the tooling breakage that once appeared in experiment.

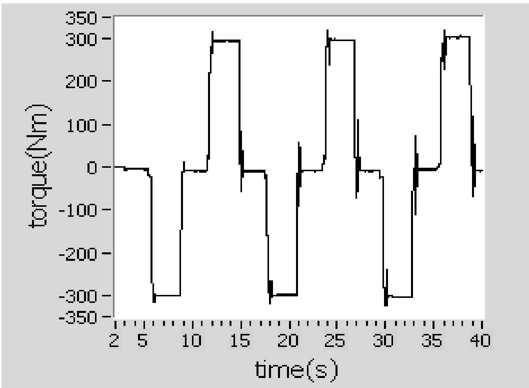


Figure 7. 300Nm Constant Torque Loading Used Torque Feedback Control

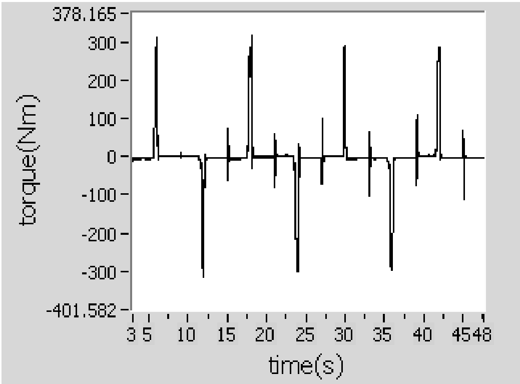
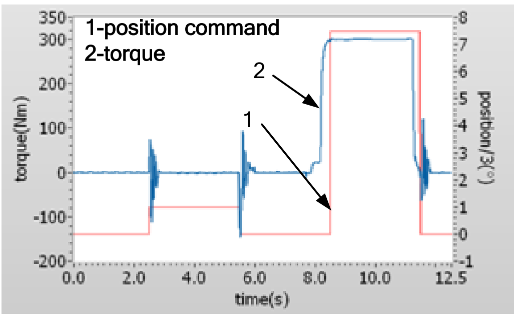


Figure 8. 300Nm Constant Torque Loadingused Speed Feed-Forward Control

Finally we use the open-loop compensation control to separately load the constant 300Nm and 500Nm to the system as shown in Figure 9 and Figure 10 where the curve 1 denotes the command curve of actuator step motion, and the curve 2 is the load torque. Because the load platform and actuator system have been designed separately are not controlled by the same PC in design, therefore a time synchronization link needs to be added. The actuator controller sends a step command per 3s in loading process. As shown in the both figures, the controller sends the first rise step at 2.5s without any control of load system, and takes this moment as the starting point. As is seen, the load torque occurs fiercely fluctuations without any compensation At the time of 5.5s, the controller sends the first drop step, and the load system similarly has not any control and corrects for the time. When the time synchronization has been realized, the load system starts to process the load program at 8.5s as the t_0 point. Because the torque to be loaded is large and the number of connecting portions is much more due to the linear motion of the actuator converting into the rotary one through the linkage, a small 20Nm load torque is firstly added before the program control to eliminate motion gap and then load 500Nm in order to avoid the impact. As is seen from the figures, the load torque generates non-fluctuation at 8.5s when the system presents the compensation in the same step motion. That means the system effectively suppresses disturbance torque. When the actuator arrives at a specified position, maintain the static 500Nm load to test whether the actuator maintains a stable in the effect of 500Nm. Before the arrival of the second falling edge of the actuator, unload advances 0.5s and then the actuator returns to zero point by a step movement. Thus, the aforementioned analysis demonstrates the proposed program control with motion discipline prediction in the load system can effectively suppress the disturbance torque.



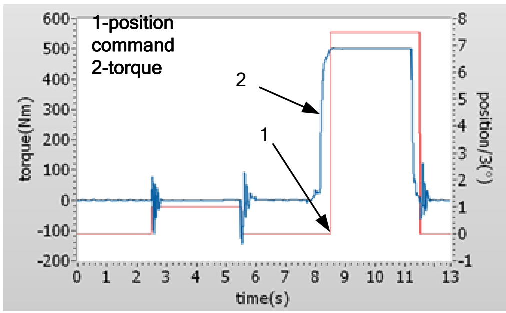


Figure 9. 300Nm Constant Torque Loading **Figure 10. 500Nm Constant Torque Loading**

6. Conclusions

This paper presented a new program control method with motion discipline prediction to inhibit the disturbance torque. Based on the analysis of the origin of the disturbance torque, A actuator motion discipline predicting model has been established to get the disturbance torque value. Then a load torque sending time model is built to weaken the influence of the responding delay of the load motor. Finally, through the simulation and experiment, it has been validated that the proposed method has advantage as follows:

(1).The new method proposed in this paper can eliminate the disturbance torque by 98%. Compared to the traditional method, it has a more obvious inhibit impact.

(2).The new method proposed in this paper can work in a wider bandwidth with an almost unchanged accuracy.

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