The Application of A New Disturbance Observer in Variable-load Electro-Optical Stabilized Platforms

Zhi-qiang LI¹, Chao SONG², Feng-lei XU¹, Jie HUANG¹ and Xiao-fei LUAN¹

¹ Naval Aviation University Qingdao Branch, Qingdao 266000, China; ² Naval Aviation University Coastal Defense College, Yantai 264001, China

Abstract. Aiming at controlling difficulty of variable load Electro-Optical Stabilized Platforms (EOSPs), a control algorithm of perturbation rejecter + Disturbance Observer (DOB) is proposed by using the disturbance suppressor to compensate the disturbance observer due to model mismatch, to improve the robust performance of whole control system and the performance of disturbance suppression. The above control methods are validated in dual-rate loop structure under different friction load and effectively improve the stabilize accuracy of EOSPs. The results show that the compensation strategies of coordinating with one another can constrain effectively the impact of control performance caused by load change, and improve the platforms’ stabilization accuracy. According to experimental verification, the control method is effective and applicable in variable load conditions, and can provide reference resources and basis for engineering application of the control algorithm.

1. Introduction

Friction is an important factor affecting the control performance of the photoelectric servo mechanism [1][2]. In the photoelectric stabilization platform, it is difficult to use mechanical means to eliminate or reduce the influence of friction interference from the source, but more from the control means to suppress the friction, through the adoption of appropriate control algorithm, to some extent to suppress the influence of friction on the servo control performance.

Disturbance observation means that by establishing the nominal model of the control object, according to the output error between the actual object and the nominal model as well as the control input information, the inverse operation of the model is used to estimate various disturbing moments including friction. The disturbance observer is proposed based on this principle. Since friction changes nonlinearly in the mechanical system, Wu Xiaomin tried to compensate the joint motion with the Stribeck friction model considering the temperature factor to improve the control accuracy, while Kun-Yung Chen used the control gain automatic regulation mechanism combined with the disturbance observer to suppress the uncertain external interference from the disturbance observation point of view [3]. However, the disturbance observer is not perfect, and there are some problems such as inaccurate compensation, difficulty in filter design, and great influence by noise and model accuracy. Based on the principle of disturbance observer, Sangjoo Kwon and Choong Woo Lee et al respectively proposed two multi-loop disturbance compensation structures to achieve effective compensation control [4][5]. Addisu Tesfaye studied the optimal design of Q filter from the matching of system sensitivity function and target sensitivity function, and obtained the reasonable compromise of instruction tracking, disturbance suppression and noise suppression [6]. Aiming at the problem of actuator saturation in
DOB control system, Gyujin Na proposed a design guide to avoid performance degradation caused by noise and verified its effectiveness in brushless DC motor system [7].

For high frequency noise suppression performance is poor, model matching demand is high, consider disturbance observer as a two degree of freedom controller and independently design the characteristics of the forward and feedback channel controller. When model mismatch and output noise are present, because of the lack of the former controller and disturbance observation information of global adjustment which cause system instability, disturbance suppressor based on model reference is proposed, and combined with disturbance observer for a new type of disturbance observer, to compensate the main disturbance. The difference between the real object and the nominal model is constrained by the disturbance function of the disturbance suppressor model to compensate the residual disturbance and suppress the influence of noise. Based on the proposed observer design method, this paper uses the motor experimental device to experimentally verify the command tracking performance of the proposed compensation control method under different frictional load conditions. The experimental results and theoretical analysis are mutually verified, providing a reference and basis for the engineering application of the control algorithm.

2. Disturbance suppression principle

The first paragraph after a heading is not indented (Bodytext style). Figure 1 is the implementation schematic diagram of the proposed interference suppressor. The controller to be designed is shown in the figure, which is the nominal velocity closed-loop transfer function.

![Perturbation Rejector Schematic Block Diagram](image1)

Fig.1 Perturbation Rejector Schematic Block Diagram

And the output equation of the system can be obtained as follows:

$$G_{by}(s) = \frac{G_p(s)G_f(s)(1+G_f(s)G_m(s)+G_p(s)G_f(s))}{(1+G_f(s)G_m(s))(1+G_f(s)G_p(s)+G_f(s)G_f(s))}$$  \hspace{1cm} (1)

Formula (1) shows that the characteristic equation of the closed-loop control system is:

$$(1+G_f(s)G_m(s))(1+G_f(s)G_p(s)+G_f(s)G_f(s))=0$$

Obviously, even when the model mismatch is serious, the closed-loop system will not be unstable because of the characteristic polynomial.

This indicates that the tracking characteristics of the closed-loop system will remain unchanged after the disturbance suppressor is added, and will not be affected by the changes in the characteristics of the actual object.

The influence of external disturbance and noise on the output can be obtained according to the transfer function $G_{by}(s)$ and $G_{by}(s)$.

Relative to the single velocity closed loop, its ability to suppress external disturbances
And, its ability to suppress measurement noise

\[
\begin{vmatrix}
G_p(s) \\
1 + G_c(s)G_p(s)
\end{vmatrix} \Rightarrow
\begin{vmatrix}
G_p(s) \\
1 + G_c(s)G_p(s) + G_f(s)G_p(s)
\end{vmatrix}
\]

Both have increased by times of

\[
\begin{vmatrix}
1 \\
1 + G_c(s)G_p(s)
\end{vmatrix} \Rightarrow
\begin{vmatrix}
1 \\
1 + G_c(s)G_p(s) + G_f(s)G_p(s)
\end{vmatrix}
\]

This shows that the disturbance suppressor has the ability to suppress not only the disturbance moment but also the noise interference. Obviously, if \( G_f(s) = kG_c(s) \), the interference suppression ability will be increased by \( k + 1 \) times.

3. **Design of a new disturbance compensator**

By analyzing the disturbance suppression principle of disturbance observer and disturbance suppressor, it can be seen that the two control methods have their own advantages. At the same time, when the nominal model is appropriate, both methods can achieve good control effect. Considering that the disturbance observer is mainly affected by the inaccuracy of nominal model and noise interference factors in application. For the disturbance suppressor, its advantage is that it is less affected by the inaccuracy and noise of the nominal model, and will not cause the system instability due to the model mismatch.

Through the above analysis, it can be seen that disturbance suppressor can be used to make up for the shortcomings of disturbance observer to some extent, and the performance of disturbance suppressor can also be improved by adding accurate disturbance compensation. Fig.2 shows the schematic diagram of disturbance compensation with disturbance observer and disturbance suppressor. The disturbance observer is used to compensate the main components of the external disturbance. The disturbance suppressor is used to suppress the residual disturbance and noise of the system.

The output of the control system

\[
G_{do}(s) = \frac{G(s)G_p(s)G_m(s) + G_f(s)G_p(s)G_m(s)}{G_m(s)[1 + G(s)G_p(s) + G_f(s)G_p(s) + Q(s)[G_p(s) - G_m(s)]}
\]

According to (6), the closed-loop transfer function of the system is

\[
G_m(s)[1 + G(s)G_p(s) + G_f(s)G_p(s) + Q(s)[G_p(s) - G_m(s)] = 0
\]

Obviously, the stability of the whole closed-loop system depends not only on the stability condition of the disturbance observer, but also on the design of \( G_f(s) \) in the interference suppressor. \( G_f(s) \) can be selected to ensure that the root of the characteristic equation is in the left half plane of \( s \) even when the model difference is large. In the sense of the model, the difference between the real object and the nominal model can be constrained by the similar function of the model reference control of the disturbance suppression device, so that the disturbance observer can avoid the stable performance degradation or even instability caused by the large model difference.

So, you can think of it in the lower frequencies as \( G_p(s) = G_m(s) \), and \( G_f(s) = k(1 + G_c(s)G_p(s)) \).

Obviously, it can be seen from the above formula that the tracking performance of the whole control system does not change much under the interaction of disturbance observer and disturbance suppressor. However, the low frequency disturbance can be well suppressed because there is \( Q(s) = 1 \) in the low frequency band and the overall disturbance gain decreases \( -20\log(1 + kG_p(s))dB \). For high-frequency signals, it will also attenuate \( -20\log(1 + kG_p(s))dB \), so it will also have a certain suppression effect on measurement noise.
4. Experimental verification of disturbance observation compensation method

In order to test the performance of the proposed disturbance observation method in command tracking and disturbance suppression, this section will take the direct drive electromechanical system experiment platform as the object, give a design example, and verify the performance of the control algorithm through command tracking experiments. The friction load is acted on the experimental system by sealing ring and magnetic powder brake. The measured static friction characteristics are shown in Figure 3. Then $F_s = F_c = 0.7394 \text{ N} \cdot \text{m}$, $\sigma = 0.0235 \text{ N} \cdot \text{m} \cdot \text{s} / \text{rad}$.

Fig. 3(b) shows the frequency response curve of the experimental system obtained by frequency sweeping under the excitation of large-value signals. The nominal model of the experimental system can be identified by using the identification toolbox in Matlab:

$$G_{pn}(s) = \frac{9.231e006 + 1.428e008}{s^3 + 557.4s^2 + 7.311e004s + 4.363e005}$$

According to the nominal model $G_{pn}(s)$, the following lagging lead controller $G_c(s)$ is designed:

$$G_c(s) = \frac{0.013s + 2.4}{s}$$

Choose $Q_{31}(s)$ as a low pass filter of disturbance observer, which take $\tau = 0.003s$, the filter bandwidth is about 88Hz. According to the nominal model and forward channel controller, then

$$\Phi_{te}(s) = \frac{1.218e005(s + 182)(s + 15.46)}{(s + 176.6)(s + 15.93)(s^2 + 364.9s + 1.218e005)}$$

Design of disturbance suppressor $G_f(s) = k \left(1 + G_c(s)G_{pn}(s)\right)$, we can get

$$G_f(s) = 0.06 \left(\frac{s^6 + 557.4s^5 + 1.99e005s^4 + 2.48e007s + 3.427e008}{s^3 + 557.4s^2 + 7.311e004s + 4.363e005}\right)$$

In order to test the control performance of three disturbance compensation algorithms DOB, PR and PR+DOB under variable load conditions, this section realizes the load change by randomly changing the control voltage of magnetic powder brake based on the friction characteristics shown in Fig.3. Speed instruction is $V_{cmd}(t) = 0.1096(\text{rad} / \text{s}) \sin(t)$.

Fig. 4 shows the control effect of the interference suppressor under variable load. It can be seen from the figure that the speed response noise is small and it can track the change of input instruction well. Since the purpose of the interference suppressor is to constrain the difference between the real object and the nominal model, and to play an auxiliary compensation role, the tracking error peak is reached 0.8 mrad.

Fig. 5 shows the control effect of the disturbance observer under variable load. It can be seen from the figure that although the speed output can track the change of the input instruction well, the noise of the speed signal is large. According to the driving force curve in
Fig. 5(c), although the load fluctuation compensated by the disturbance observer is smaller than that of the other two algorithms, its peak position error reaches 0.4 mrad and there is a large noise. This is consistent with the conclusion that the disturbance observer has no good suppression effect on high frequency noise.

Fig. 6 shows the compensating control effect by combining the disturbance suppressor and disturbance observer. It can be seen from the figure that the speed signal of this compensation algorithm inherits the feature of low noise under the control of interference suppressor. Moreover, its position error inherits the advantage of accurate compensation of the disturbance observer, so that the position error can still obtain the error curve of approximate sinusoidal signal with the peak value of position error less than 0.2 mrad under the circumstance that the frictional load varies greatly (see Fig. 6(c)).

Through the instruction tracking and variable load experiments, it is shown that although the disturbance observer's control performance will decline rapidly or even become unstable due to the inaccuracy of the nominal model, the auxiliary compensation of disturbance suppressor can effectively make up for the deficiency of the disturbance observer and obtain a better disturbance suppression effect.
5. Summary
From the perspective of control, accurate and effective estimation of disturbance and compensation control are effective means to improve disturbance suppression performance of stable servo mechanism. The influence of friction, model parameter change and load torque change on the control performance of the photoelectric stabilized platform is considered as the control error caused by disturbance. When the disturbance observer is combined with the disturbance suppressor, even when the system characteristics are affected, better control performance can be achieved by the combination of the two control methods.

References
[1] WANG Yi, HE Zhen. Friction compensation for servo system[J]. Electric Machines and Control, 2013, 17(8): 107-112.
[2] WANG Zhengxi, ZHANG Bao, LI Xiantao. Friction compensation strategy of high performance for aerial photoelectrical stabilized platform[J]. Acta Aeronautica et astronautica sinica, 2017, 38(12): 321350.
[3] Chen, Kun-Yung. Model Following Adaptive Sliding Mode Tracking Control Based on a Disturbance Observer for the Mechanical Systems[J]. Journal of Dynamic Systems, Measurement, and Control, 2017, 140(5): 051012.
[4] Choong Woo Lee, Chung Choo Chung. Design of a New Multi-Loop Disturbance Observer for Optical Disk Drive Systems[J]. IEEE Transactions on Magnetics, 2009, 45(5): 2224–27.
[5] SangJoo Kwon, Wan Kyun Chung. Robust Performance of the Multiloop Perturbation Compensator[J]. IEEE/ASME Transactions on Mechatronics, 2002, 7(2): 190–200. https://doi.org/10.1109/TMECH.2002.1011257.
[6] Tesfaye, A., Ho Seong Lee, M. Tomizuka. A Sensitivity Optimization Approach to Design of a Disturbance Observer in Digital Motion Control Systems. IEEE/ASME Transactions on Mechatronics, 2000, 5(1): 32–38. https://doi.org/10.1109/3516.828587.
[7] Na, Gyujin, Nam Hoon Jo, Yongsoon Eun. Performance Degradation Due to Measurement Noise in Control Systems with Disturbance Observers and Saturating Actuators[J]. Journal of the Franklin Institute, 2019, 356(7): 3922–47. https://doi.org/10.1016/j.jfranklin.2019.03.001.