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Design of Loading System for Thrust Reverser Ground Simulation Test

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Abstract. Thrust reverser is a necessary device on large passenger plane and large culvert ratio turbofan engine, which has the advantages of shortening the landing distance, reducing the wear of wheel brake, prolonging the service life of braking parts and improving landing safety of aircraft. The thrust reverser ground test is one of the important contents in the design of aircraft nacelle, and generally, it is an effective way to carry out the research test work by semi-physical simulation, in which the analysis of the thrust reverser loading control system are the basis for the design of test loading device. In this paper, take the cascade thrust reverser as the object, firstly, the thrust reverser structure and working principle were specially explained. Then, according the characteristic of the hydraulic actuation system, the loading channel was designed and analyzed theoretically, especially the loading control system. Finally, the model of loading control system was established in Simulink, and the time domain and frequency characteristic were analyzed. For the redundant force in the passive loading system, PID and the principle of the structure invariance were chosen to correct the control system. The simulation results show that the principle of the structure invariance control strategy has a better effect on eliminating the redundant force to meet engineering requirements.

1. Introduction

The increase of aircraft speed and the wing load will make the aircraft take-off and landing slip distance long. If you want to shorten the landing slip distance, some special deceleration devices often are needed, for example the thrust reverser [1, 2, 3]. In military aspect, the thrust reverser shortens the landing distance of the aircraft and greatly improves the aircraft's combat efficiency. In the civil area, the thrust reverser has a significant impact on civil aviation construction. Since the introduction of the thrust reverser, the aircraft landing slip distance has been shortened from 3000m to 450m. Therefore, the thrust reverser plays an important role in the design of civil aircraft.

The thrust reverser driving system drives the thrust reverser structure to deploy and stow by hydraulic. During the landing of the aircraft, the thrust reverser is deployed, the actuator will be subjected to a more complex flow of reverse flow load. If the actuator is affected by air resistance, it cannot be deployed and stowed normally, which will greatly affect the thrust performance and aircraft's safety. Therefore, the load capacity and logic function of the actuator must be tested before the machine is installed. Because of the high cost, big risk and long cycle of physical test, ground simulation loading test that is, semi physical simulation test is usually used to test various load, and to verify the dynamic performance of the reverse thrust system under different aerodynamic forces. In addition, the ground simulation loading test is an important way to evaluate the performance of the designed thrust reverser system, and whether the performance satisfies the requirement of driving is the key technology in the research of the thrust reverser system.

In this paper, the characteristics of the thrust reverser system are introduced firstly, and the method of the ground simulation loading channel is put forward according to the driving characteristics. In addition, the loading control system are analyzed, the control strategy of the loading system is expounded. For the redundant force in the passive loading system, the PID and the principle of the structure invariance are used to correct the system.

2. Thrust reverser device structure and working principle

Thrust reverser is normally mounted in the middle of the engine nacelle under wing, closed during normal flight, opened during landing, reversing jet flow and producing reverse thrust. Its installation is shown in Figure1.

Thrust reverser consists of a set of fixed parts and movable parts uniformly distributed in the circumference, and the structure is schematically shown in Figure2, where the fixing part comprises a ring-shaped cascade, a front frame and a sliding guide; the movable part comprises a ring-shaped translating sleeve, several blocker doors, and blocker door drag links [4, 5, 6].

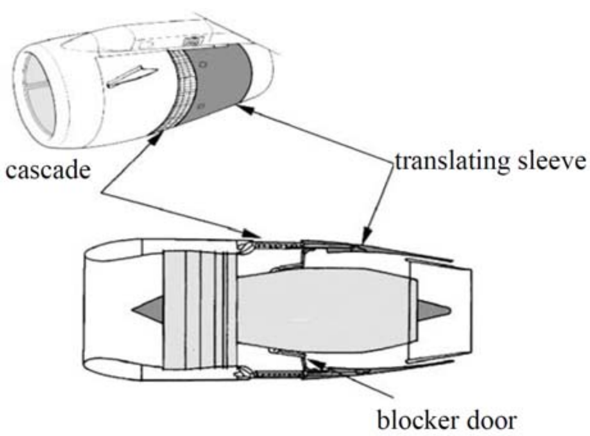


Fig.1 Diagram of thrust reverser installation

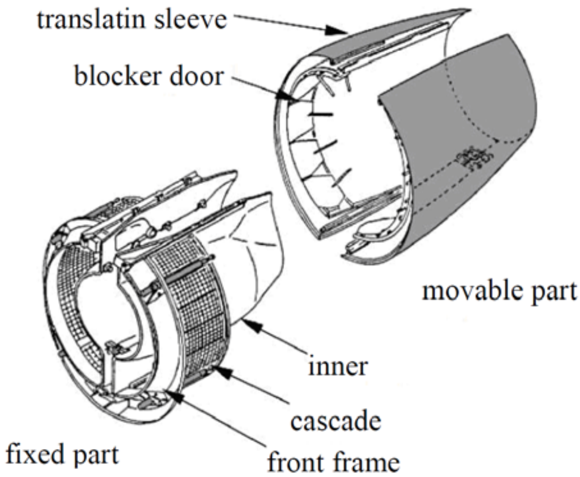


Fig.2 Diagram of thrust reverser structure

Due to the uneven flow field in the airflow at the aircraft engine exit, the load transmitted to the various actuators by the blocker door which blocking the air flow is also unequal. In order to synchronize the movements of the various actuators under different loads, it is also necessary to ensure that the thrust reverser device is in a locked state during flight and to monitor the actuator displacement. Generally, the hydraulic actuation system of the thrust reverser device as is shown in Figure3. It mainly consists of hydraulic pressure source, control valve, actuating cylinder (with locking and non-locking), expansion and retraction hydraulic line, sync flexible shaft, the position feedback mechanism, the sync lock device and so on.

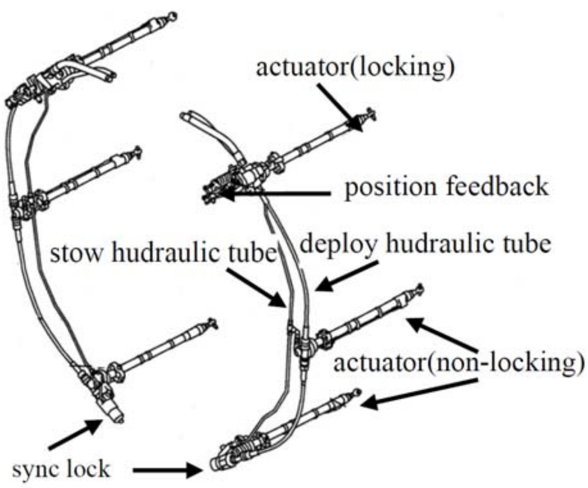


Fig 3. Thrust reverser hydraulic actuation system

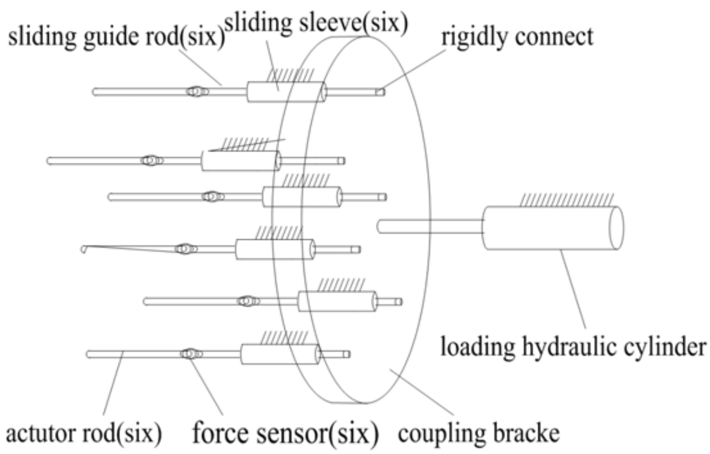
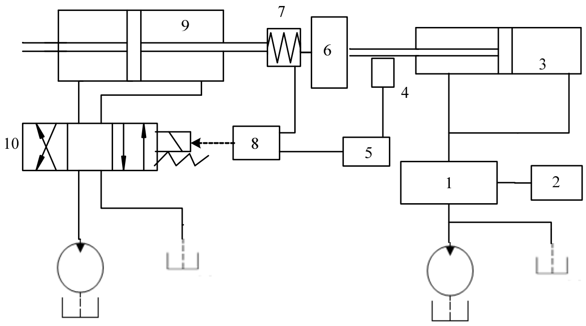


Fig 4. Schematic diagram of loading channel

3. Analysis of thrust reverser loading system

According to the characteristics of the thrust reverser, when designing the loading channel, the sliding guide rod and force transducer can be connected with the piston rod of the thrust cylinder, in this way, the initial installation displacement difference can be compensated by the length of the slide guide rod, so that the output points of the six actuator cylinders which are not in the same plane are on the same plane, and the channel loads can be monitored by force sensor. Also, the sliding guide rod moves in a fixed sliding sleeve to ensure that only the radial loading load of the channel is applied to the thrust reverser actuator, while the other direction of deflection is borne by the sleeve. In addition, each sliding guide rod is rigidly connected to a circular coupling bracket according to the actual distribution of the thrust reverser actuator. On the other side, a loaded hydraulic cylinder is used for centralized loading to ensure the synchronization of loading. The design of the loading channel is shown in Figure4. In the loading process, the circular coupling bracket should be kept perpendicular, so as to ensure that the loading of the hydraulic cylinder can be applied in the radial direction of the thrust reverser actuator [7, 8].



1 thrust reverser hydraulic unit 2 control instruction 3 thrust reverser actuator 4 postion sensor 5 force function generator 6 circular coupling bracket7 force sensor 8 controller 9 loading actuator 10 electrohydraulic servo valve

Fig 5. The schematic diagram of the reverse thrust electro-hydraulic loading channel

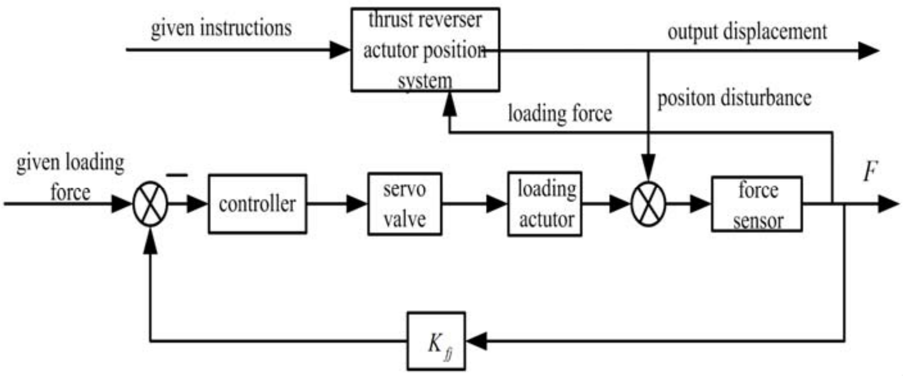


Fig 6. Diagram of loading control system

Thrust reverser electro-hydraulic servo loading system is a force output system, which mainly includes servo amplifier, electrohydraulic servo valve, valve-controlled loading actuator, force sensor and loading controller. As shown in Figure5, the schematic diagram of the reverse thrust electro-hydraulic loading channel is presented. To save space, there should be six cylinders attached to the coupling bracket, but only one representation is drawn. When the thrust reverser cylinder begins to move, force function generator 5 in the loading computer according to the displacement generates theoretical load spectrum signal to the electro-hydraulic loading system. The controller 8 in the system compares the theoretical load spectrum with load force transmitted by force sensor 7. The difference signal is amplified and controls the electro-hydraulic servo valve spool opening degree, thus controlling the flow and pressure flowing into the loading cylinder, resulting in controlling the load force of the output end. The whole system is composed of a closed-loop control system, in the case of the autonomous motion of the thrust cylinder, the servo loading is carried out, the aerodynamic state is simulated, and the motion and control performance of the thrust cylinder is tested. According to the physical parts, you can draw the structure diagram shown in Figure 6.

From Figure6, it is easy to see that the thrust reverser actuator position system and the thrust electro-hydraulic servo loading control system are a coupling system. As a typical passive loading system, during the motion, the thrust reverser actuator position system is the main disturbance source of loading system. The existence of this disturbance makes the loading force in the loading process deviate from the loading requirement of the load spectrum, resulting in superfluous force. The disturbance is directly in the output, so the jamming effect is very obvious, which will seriously reduce the accuracy of the loading process. Therefore, how to ensure that the superfluous force caused by the

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position system to minimize or eliminate the disturbance of the loading system is the first problem to be solved in the loading process.

4. Mathematical Model of Thrust Reverser Loading Control System

Through the above analysis, we can use figure 7 to represent the control block diagram of the thrust reverser loading system.

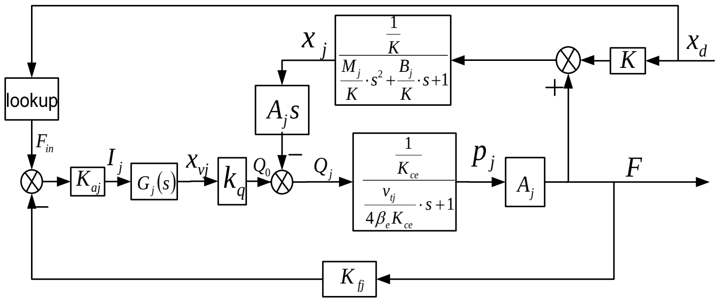


Fig 7. Diagram of loading system control block

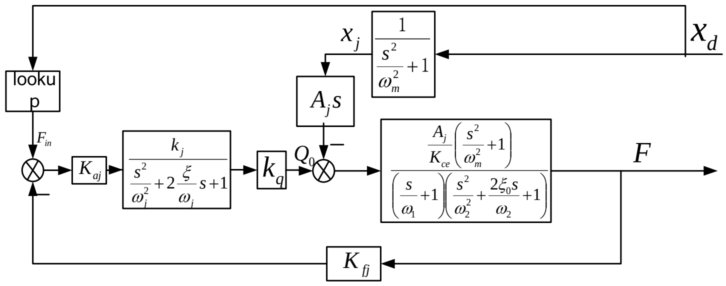


Fig 8. Diagram of simplified control block

It is shown from Figure7 that the transfer function model of the hydraulic drive device is complicated because of the factors such as inertia load, damping, elastic load, oil compressibility and the leakage of actuator. But there is no need to consider so many factors in the theoretical research. For a specific system, under certain conditions, ignoring some of the relatively small impact of factors, but mainly discussing some influential factors, which is more conducive to qualitative analysis. The transfer function of the whole system is formula (1):

$$F=\frac{\frac{A_j}{K_{ce}}\left(\frac{M_j}{K}s^2+\frac{B_j}{K}s+1\right)\cdot Q_0-\frac{A_j^2}{K_{ce}}s\cdot x_d}{\frac{M_j v_{tj}}{\beta_e K K_{ce}}s^3+\left(\frac{B v_{tj}}{\beta_e K K_{ce}}+\frac{M_j}{K}\right)s^2+\left(\frac{v_{tj}}{\beta_e K_{ce}}+\frac{B_j}{K}+\frac{A_j^2}{K K_{ce}}\right)s+1} \tag{1}$$

If the damping coefficient B_j is ignored, and the two-factor is used to overwrite the denominator three-polynomial of the transfer function, the following formula (2) can be obtained:

$$F=\frac{\frac{A_j}{K_{ce}}\left(\frac{M_j}{K}s^2+1\right)\cdot Q_0-\frac{A_j^2}{K_{ce}}s\cdot x_d}{\left(\frac{s}{\omega_1}+1\right)\left(\frac{s^2}{\omega_2^2}+\frac{2\xi_0 s}{\omega_2}+1\right)} \tag{2}$$

Because $\frac{B}{K}<<0$, the mechanical frequency $\omega_m=\sqrt{\frac{K}{M_j}}$, the upper formula is further transformed into formula (3):

$$F=\frac{\frac{A_j}{K_{ce}}\left(\frac{s^2}{\omega_m^2}+1\right)\cdot Q_0-\frac{A_j^2}{K_{ce}}s\cdot x_d}{\left(\frac{s}{\omega_1}+1\right)\left(\frac{s^2}{\omega_2^2}+\frac{2\xi_0 s}{\omega_2}+1\right)} \tag{3}$$

This simplified transfer function can be expressed in Figure8.

5. Simulation of thrust reverser loading control system

Thrust reverser loading system is a force servo system with strong position disturbance. The load is usually called the active loading under the condition of no position jamming, and the system characteristic in this state is called the disturbance-free characteristic. The inherent characteristics of the system are accurately reflected by the disturbance-free characteristic. Therefore, it is necessary to ensure that the loading system has good disturbance-free characteristic, and it is valuable to study the characteristic of the system with disturbing loading. If the active loading performance is poor, then the passive loading state is worse. This section firstly studies the load system disturbance-free characteristic, and designs the controller to make the performance of the disturbance-free system achieve better.

5.1. Frequency characteristic analysis of non-disturbing loading system

Derived from the above, the undisturbed system open-loop transfer function is formula (4):

$$G(s)=\frac{F(s)}{F_{im}(s)}=\frac{K_{aj}\cdot K_{fj}k_j\cdot k_q\frac{A_j}{K_{ce}}\left(\frac{s^2}{\omega_m^2}+1\right)}{\left(\frac{s^2}{\omega_j^2}+2\frac{\xi}{\omega_j}s+1\right)\cdot\left(\frac{s}{\omega_1}+1\right)\left(\frac{s^2}{\omega_2^2}+\frac{2\xi_0 s}{\omega_2}+1\right)} \tag{4}$$

From the formula (4), the open-loop system is a 0-type system, and there is a second order unstable differential link. From the denominator, the system is five-order system, at this time there is a real pole and two pairs of complex poles.

According to the above formula, the open loop bode of the system can be plotted on Matlab, as shown in Figure 9.

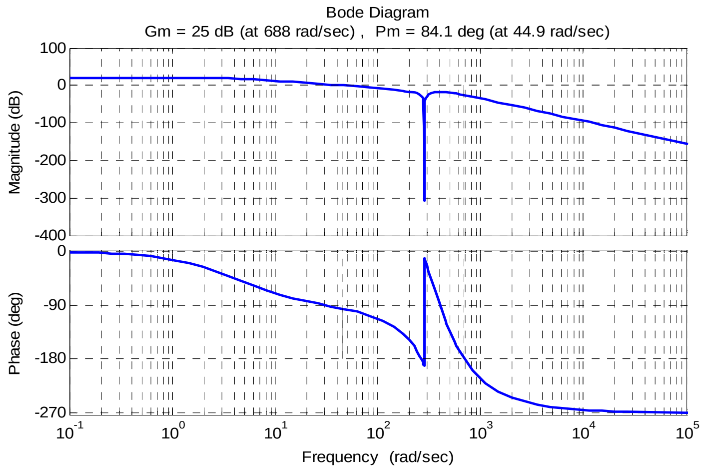


Fig 9. Open loop frequency characteristic of loading system

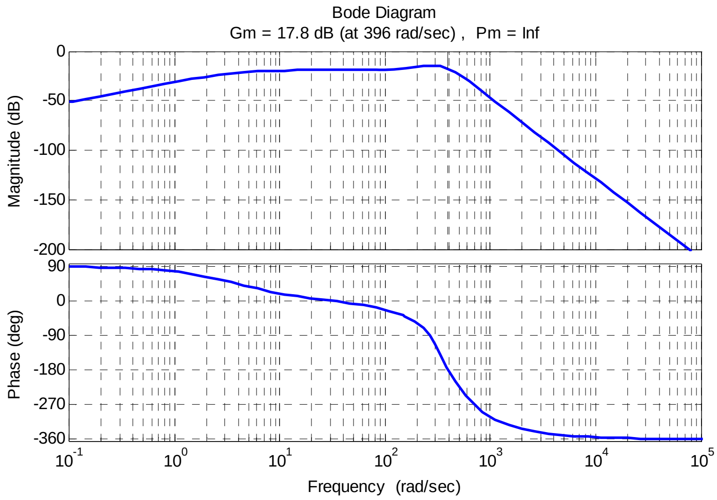


Fig 10. Frequency characteristics of excess force

As can be seen from Figure9, $\omega_j > \omega_2 > \omega_m >> \omega_1$, the characteristics of the system are mainly determined by the lower first-order turning frequencies of the actuator, while the high-frequency segments are mainly determined by ω_j, ω_2 and ω_m. But because of the low loading frequency, it can ignore the influence on the system. In addition, it is noted that the frequency of the load vibration is divided into two parts, in order to ensure the stability of the loading system, the load frequency should be far away from the load vibration frequency, and the load frequency of electrohydraulic servo loading system is also not allowed to exceed ω_m, that is, load vibration frequency ω_m is the upper limit of loading system working frequency. And because the deploy and stow time of the actuator is not greater than 4s, so its highest frequency is in 0.32rad/s, far less than ω_m, so the loading system will not exceed the system closed-loop frequency characteristics, which indicates this feature meets the requirements.

5.2. Analysis of frequency characteristic of redundant force

From the angle of frequency domain, we have deduced the complete block diagram of the loading system. According to the definition of redundant force, we can derive the transfer function of the system output force to the thrust cylinder displacement interference as following formula (5):

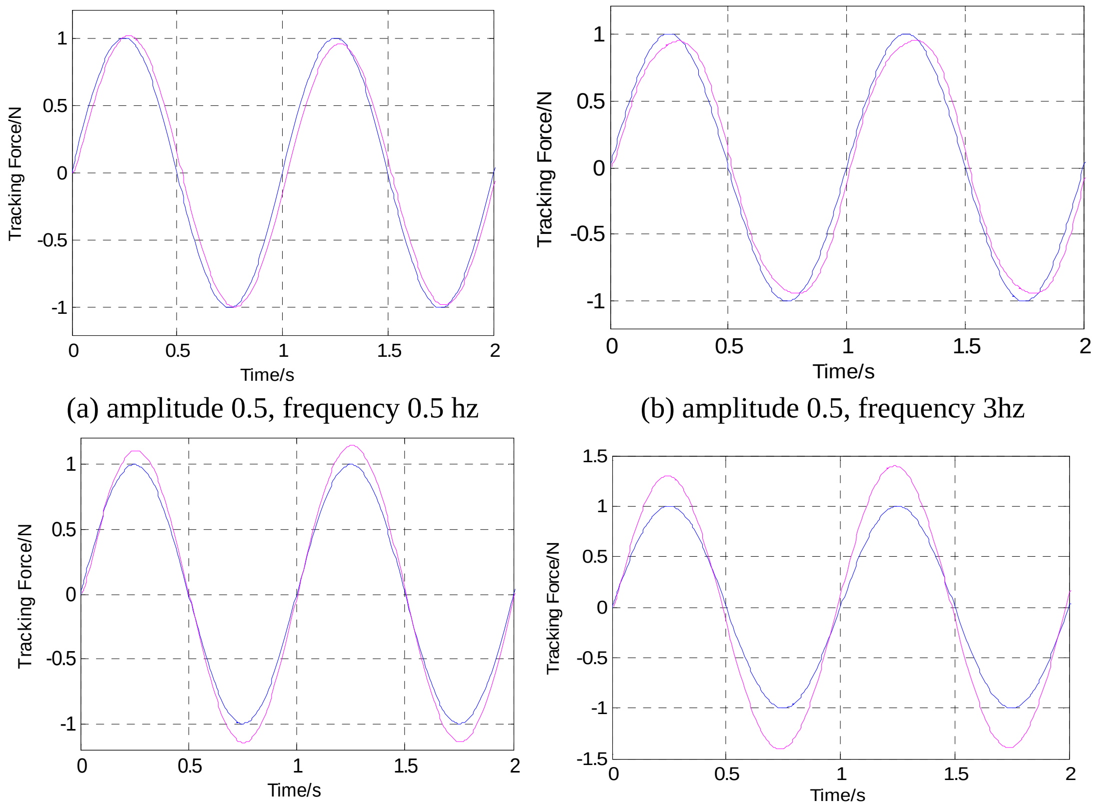
$$G_c = \frac{K_{aj} \cdot K_{fj} k_j \cdot k_q \dfrac{A_j^2}{K_{ce}} \cdot s}{\left(\dfrac{s^2}{\omega_j^2} + 2\dfrac{\xi}{\omega_j} s + 1\right) \cdot \left(\dfrac{s}{\omega_1} + 1\right)\left(\dfrac{s^2}{\omega_2^2} + \dfrac{2\xi_0 s}{\omega_2} + 1\right)} \tag{5}$$

From the above formula (5), the transfer property of redundant force is a type 0-type and five-order system. The molecule is a differential link, which shows that the redundant force caused by the position disturbance of the thrust cylinder is advanced, and is proportional to the two times of the effective area and the velocity of the loaded object, is inversely proportional to the total flow pressure coefficient of the servo valve. Figure10 is the redundant force frequency characteristic curve drawn on Matlab.

It is shown from Figure10 that in the middle and low frequency segment, the differential link has a greater effect on the system, which leads to the increase of amplitude according to the 20dB/10dec, and enlarges the redundant force in the output force. In high frequency segment, due to the strong attenuation effect of the loading system, the influence of the differential link is reduced obviously, and the redundant force decays rapidly.

5.3. Using PID correction to eliminate redundant force

For a stable system, if a certain disturbance is added, the system may become unstable. That is, the dynamic tracking performance of the system becomes worse under the condition of disturbance. Therefore, PI correction is used to improve the response tracking characteristics of the system, and to explore the ability of PI regulation to restrain the excess force generated by position disturbances.



(a) amplitude 0.5, frequency 0.5 hz (b) amplitude 0.5, frequency 3hz

(c) amplitude 2, frequency 1hz (d) amplitude 5, frequency 1hz

Fig 11. Force tracking curves under different amplitude and frequency

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Figure11, It can be seen that the PI correction can suppress the redundant force when the perturbation is low frequency. But when the disturbance is high frequency and large amplitude, the system redundant force far exceeds the command force, the normal command signal is overwhelmed by the redundant force. This shows that PI controller has a certain ability to reduce the redundant force, but this ability is limited, when the disturbance amplitude and frequency increase, PI controller appears powerless. So we must seek other control methods to solve this problem.

5.4. Using the principle of structural invariance to eliminate redundant force

Although the controller with PI correction in the main feedback loop is suitable for most of the control objects, it is difficult to achieve satisfactory control effect if there is a strong position jamming system. Therefore, the principle of structural invariance in the classical control theory is adopted to eliminate the influence of external interference force. The feedforward compensation channel, which is composed of invariance principle, is a kind of feedback controller based on disturbance compensation in theory. If the servo valve nonlinearity is not considered, it can completely compensate the disturbance of the system [9]. The basic principle of structural invariance is shown in Figure12.

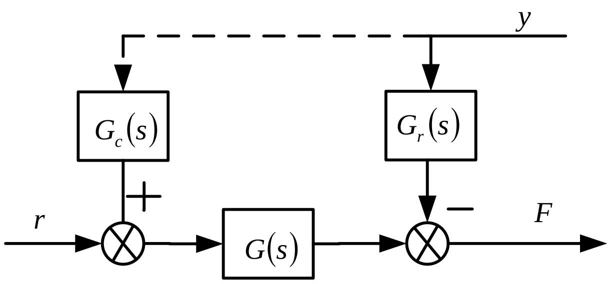
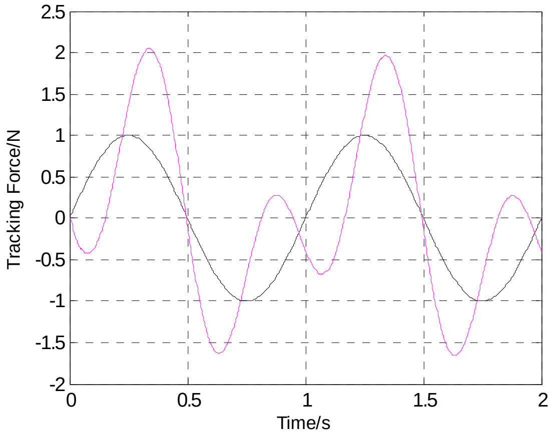


Fig.12 Diagram of principle of structural invariance

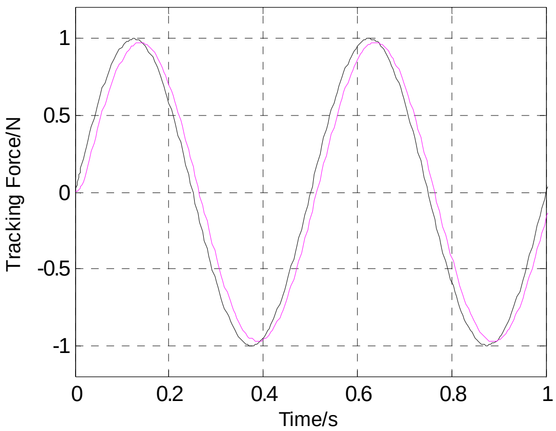
The compensator can be expressed by the following formula (6):

$$G_c(s)=\frac{A_j\cdot s}{K_{aj}k_j\cdot k_q\cdot\left(\frac{s^2}{\omega_j^2}+2\frac{\xi}{\omega_j}s+1\right)\cdot\left(\frac{s^2}{\omega_m^2}+1\right)} \tag{6}$$

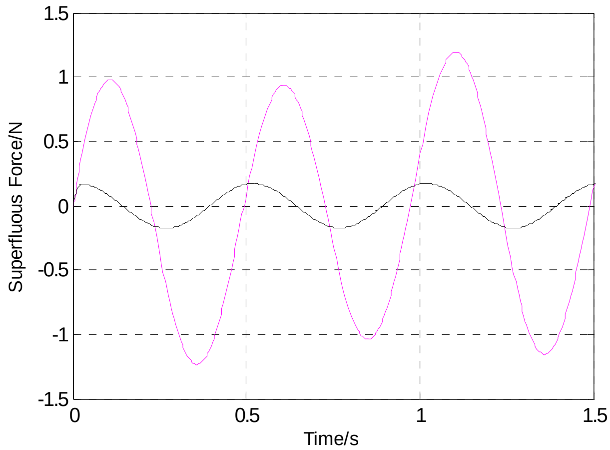
Figure13 is the simulation curve of the redundant force after feed forward compensation in the loading system. Figure13 (a) is the response curve of the system at a disturbance frequency of 2Hz, amplitude 10N. Figure13 (b) is the tracking curve after compensating the system. Figure 13 (c) shows that the designed compensator can greatly reduce the redundant force interference in the loading system and meet the requirements.



(a) Before compensation



(b) After compensation



(c) Contrast curve of excess force before and after compensation

Fig 13. Simulation curve of redundant force before and after compensation

6. Conclusion

In this paper, the structure and working principle of the thrust reverser actuation system were analyzed. According to its characteristics, the thrust reverser test loading scheme was designed, and the load control strategy of the thrust reverser test was studied. So the following conclusions are drawn:

(1) When designing the loading scheme, it is necessary to consider the load synchronization, load initial installation displacement difference, force balance and the redundant force caused by passive loading.

(2) For the redundant force generated by passive loading, the simulation results show that the principle of structural invariance based on feed-forward compensation can better restrain the redundant force and improve the accuracy and accuracy of loading.

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