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Design and Research on the Lifting Push Rod for a Radar Antenna Array

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Abstract. A lifting electrical push rod mechanism which is suitable for the upper load pitch lifting has been designed based on a particular mobile radar in this paper. Through the theoretical modeling analysis of the lifting force of the electric actuator and the stability of the screw during the whole Lifting process, this article aims to investigate the influence of the compound load on the force and stability of the push rod during the lifting process of the screw, and to verify the accuracy of the calculation results by combination with the actual measurement results. The research results show that: in the lifting process of the upper load from the lifting angle 0 to 90 °, the self-weight moment of the upper load is the main influencing factor of the force on the electric push rod, and in the lifting process, the maximum force on the electric push rod appears in the middle state of the lifting process, the lifting angle with the most dangerous screw stability coefficient and the position of the maximum force on the screw do not overlap. In the lifting process, the measured results and the theoretical calculation values are in good agreement, which proves the accuracy of the calculation model and method.

Keywords. Lifting mechanism, electrical push rod, lifting force, stability of the screw

1. Introduction

Mobility is one of the key factors affecting the survivability and rapid deployment capability of vehicle-mounted radar on the battlefield. The rapid and reliable transition of the antenna array from the mobile transport state to the operational state is an important factor affecting the mobility of the on-board radar [1-4]. The size of the radar antenna directly affects its action distance, and in order to reduce the influence of ground clutter and reduce the influence of shielding angle, the radar antenna needs to be lifted to a specified height and particular angle (0~90°) when used [5]. The radar antenna needs to overcome the influence of compound loads such as its own heavy moment, moment of inertia and the impact of wind moment during the lifting process. For these reasons, the lifting mechanism must have a high degree of reliability and stability.

In the engineering design, the lifting mechanism is generally arranged between the antenna and the turntable or between the turntable and the base bracket to complete the angular transformation of the antenna relative to the horizontal direction [6-7]. The lifting electric actuator in this paper is designed based on a vehicle-mounted radar. By establishing the physical model and simulation analysis, the force situation and the

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stability of the electric actuator in the whole lifting process have been analyzed in this paper, and the effects of the compound load on the force and stability of the rod in the lifting process have been discussed.

2. Overall Structure Design

The electric push rod mechanism designed in this paper is used to realize the pitch angle change of the upper loads of the turntable and the antenna array, and its relative location is shown in figure 1. The electric actuator mechanism is placed between the turntable and the bottom bracket. With the variation of the extension length of the electric actuator screw, the upper load is pitched clockwise around the rotary shaft with a pitch angle of 0~45°~90°.

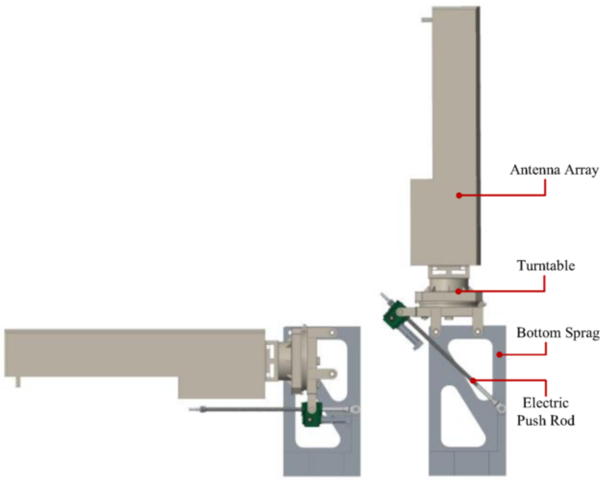


Figure 1. Layout diagram of electric actuator mechanism.

The electric actuator mainly includes a gear motor, a hand crank interface, a lead screw, a sliding filament, a set of equal-diameter spur gears, a set of bevel gears, some cover plates and sheaths. As shown in figure 2, the lead screw adopts the form of a self-locking trapezoidal screw, which can effectively ensure the safety of the system. The gear motor drives the linear motion between the sliding filament and the filament through an equal spur gear, and the axial force transmitted by the filament is supported by a set of tapered roller bearings at the sliding filament. The motor is not with the gate. In the case of motor failure, manual drive of the reducer bevel gear set can be made by hand cranking the interface external rocker, which drives the relative rotation between the equal-diameter spur gears. As a result, an emergency lifting or landing of the screw mechanism will be done.

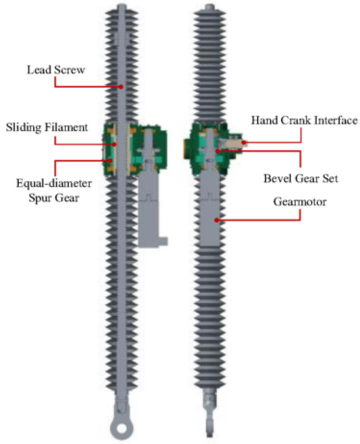


Figure 2. Schematic diagram of electric actuator structure.

3. Analysis of the Force on the Screw

As shown in figure 3, the upper load is mainly subjected to the comprehensive effect of its self-gravity, the wind resistance and the support force of the screw rod during the lifting process. During the lifting process, the upper loads are subject to balanced moments according to the law of moment balance, which means that the supporting torque of the screw to the upper loads relative to the rotating shaft is as equal as the sum of the torque of self-weight and wind resistance of the upper loads. In the figure, A is the coordinate point of the rotating shaft position, B is the coordinate point of the hinge position of the electric thruster and the bottom support, C is the coordinate point of the articulation position of the electric thruster and the turntable, C_1 is the coordinate point of the initial position when the lifting angle is 0°, and C_2 is the coordinate point during the lifting process. G_z is the self-weight of the upper loads, L_2 is the force arm length of the upper loads from the rotating shaft, F_b is the pushing force of the electric thruster, F_w is the wind resistance of the upper loads, and L_1 is the force arm length of the wind resistance.

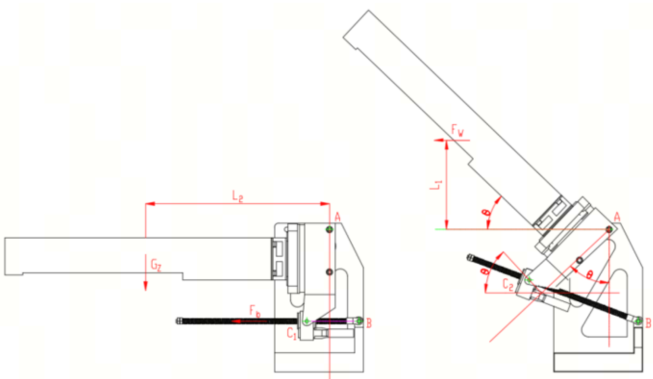


Figure 3. Schematic diagram of the force in the moving process.

Defining the axis of rotation position coordinates point A (0, 0), it is easy to get the linear equation composed of B (X_b, Y_b) and C_2 (X_c, Y_c) as following.

$$(X_c - X_b)(y - Y_b) = (Y_c - Y_b)(x - X_b)$$

According to the formula for the distance from a point to a straight line, it is easy to obtain the electric push rod thrust arm that the distance Lb from A to the straight line BC_2。

$$L_b = \frac{|X_b Y_c - X_c Y_b|}{\sqrt{(X_c - X_b)^2 + (Y_b - Y_c)^2}}$$

The torques of the upper loads are balanced according to the moment balance law, i.e.

$$F_b \bullet L_b = G_Z \bullet L_2 + F_w \bullet L_1$$

At the time of startup, the electric actuator needs to drive the upper loads to accelerate in order to obtain the required rotational speed, and at this time the electric actuator needs to overcome the moment of inertia of the upper loads F_β .

$$F_\beta = \frac{I_{zg}\beta}{L_b}$$

Where, I_{zg} is the rotational inertia of the upper loads with respect to the rotating shaft, and β is the angular acceleration. In the case involved in this paper, the rotational inertia and angular acceleration are small enough relative to the self-weight load, thus the effect caused by the moment of inertia can be ignored.

The force on the screw is mainly impacted by the gravity of the upper loads during the lifting process as shown in figure 4, and the force increases and then decreases with the variation of the lifting angle from 0 to 90°, and reaches the maximum value at about 11.46°.

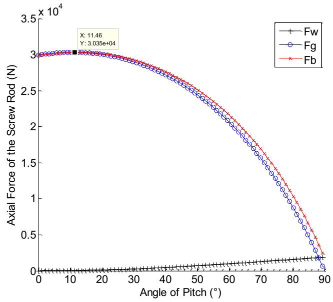


Figure 4. Variation curve of screw force in relation to the lifting angle.

4. Analysis of Stability

In case of pressurized screws with large L/D ratio, when the axial pressure is greater than a certain critical value, the screw will suddenly bend laterally and lose its stability. Therefore, it is necessary that the axial force on the screws should be less than the critical load with a certain safety factor under normal conditions. In conventional cases, it is common to adopt the stability analysis in the initial and the final states instead of the stability analysis of the whole process. But obviously, the electric actuator mechanism involved in this paper reaches the maximum force state in the intermediate lifting state (lifting angle of about 11.46°). Therefore, it cannot effectively illustrate the force stability of the screw by adopting the starting and final state analysis.

The critical load of the longitudinal bending damage of the lifting mechanism screw can be calculated according to Euler's formula.

$$F_{\mathrm{cr}} = \frac{\pi^2 EI}{(\mu l)^2}$$

where: E is the modulus of elasticity, taken as 206000 MPa. I is the moment of inertia of the dangerous interface of the screw, mm^4. d_1 is the small diameter of the screw. μ is the length factor of the screw, taken as 1; l is the working length, take the distance of the hinge point BC^2, mm.

$$\eta = \frac{F_{cr}}{F_b}$$

The change curve of the safety coefficient of screw stability in the lifting process with the change of lifting angle is shown in figure 5. As can be seen from figure 5, the coefficient of screw stability with the lifting angle from 0 to 90 ° decreases and then increases, and reaches a minimum value of about 4 around 60 °, which meets the requirements of the coefficient of screw stability 3.5~5.0 for the force transmission screw drive.

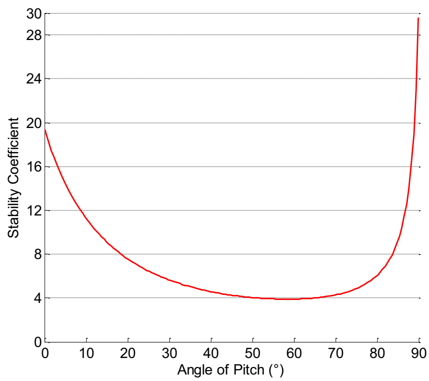
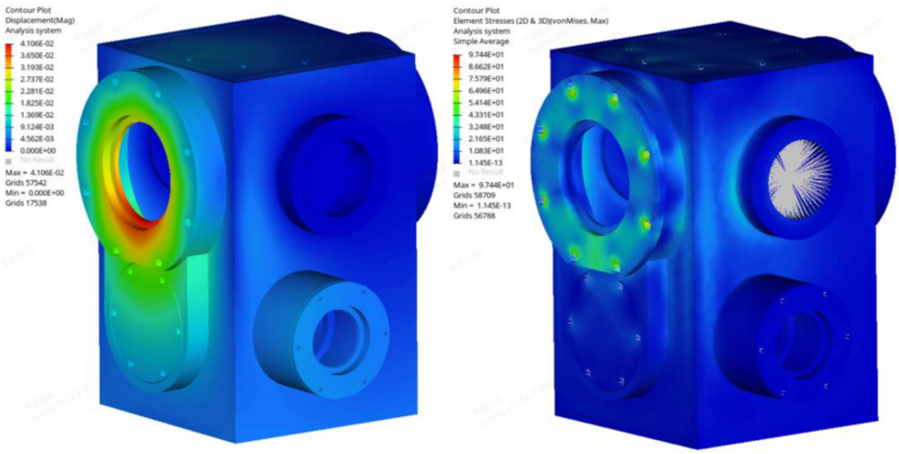


Figure 5. Change curve of the coefficient for screw stability during the lifting process.

5. Modelling Analysis

In the use of electric actuators, the shell is the mainly load-bearing structural member, except for the screw rod and gears which need to be calculated and certified. The electric actuator shell is welded with Q355D and then machined, and the simulation result is shown in figure 6. The simulation results show that the shell has a maximum deformation of 0.042mm and a stress of 97.44MPa, which meets the requirements of use.



(a) Strain (b) Stress

Figure 6. Stress-strain diagram of the shell under the maximum stress condition.

6. Tests

This study was tested in actual use of an electric lift actuator system on a radar array in order to test the accuracy of the computational method and the model. The comparison results of the measured and calculated values with the change of lifting angle are shown in figure 7. As seen in figure 7, the calculated results are in good agreement with the measured values. From the figure, the measured value is slightly larger than the theoretical calculation when the lifting angle is from 0 to 25°, which is mainly due to the processing and assembly errors as well as the control algorithm; from 55° to 90°, the measured value is slightly smaller than the theoretical calculation and has a certain fluctuation, which is due to the reduction of the lifting angle speed in the later lifting period, the friction torque at the rotating shaft and the friction torque of the joint bearing caused by assembling and other torques. This is due to the influence of the noise caused by the frictional moment at the rotating shaft and the frictional moment of the joint bearing caused by the assembly, which leads to the relatively large difference between the measured value and the calculated result in the late lifting period. From 55° to 90°, the measured value is slightly smaller than the theoretical calculation and has certain fluctuation, which is due to the reduction of lifting angular speed in the late lifting period. The clutter caused by the frictional torque at the rotating shaft and the frictional torque of the joint bearing caused by the assembly causes the measured values to be relatively different from the calculated results in the late lifting period.

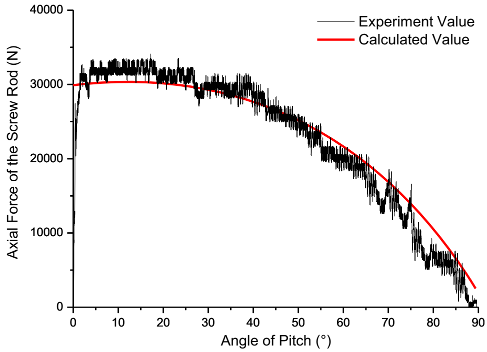


Figure 7. Comparison of the measured and calculated values.

7. Conclusion

An electric push rod mechanism for upper loads pitch lifting has been designed based on a particular mobile radar in this paper. Considering the comprehensive effects of self-weight, wind load and support reaction force, force and screw stability for the push rod of the whole lifting process are analyzed in this paper, and the accuracy of the calculation results is verified by the actual measurement results. The conclusions are as follows:

- In the lifting process, the maximum force on the screw happens in the intermediate state of the lifting process, at about 11.46°.
- The lifting angle where the screw stability coefficient is minimum and the position of the maximum force on the screw do not overlap in the lifting process, and the screw stability coefficient reaches a minimum of about 4 when the lifting angle is about 60°.
- Based on the above 2 conclusions, the whole lifting process should be subjected to force analysis and pressure rod stability calculation and verification for a lifting mechanism of the same type, instead of simply adopting the analysis of the initial state and the final state.

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