50KN Compression Spring Fatigue Testing Machine Design

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Abstract

Spring fatigue test is the key process of spring performance testing, especially for automotive suspension springs, train damping springs, engine valves and other critical parts of the spring, must do the reliability assessment of spring fatigue performance. Because the requirements of different springs are different, the frequency and amplitude used in the test also have different requirements. In this case, a compression spring fatigue tester was developed and designed to test the maximum number of cycles of a spring under a given failure condition by applying a cyclic variable load to the spring. Through the design of the compression spring fatigue testing machine, the test prototype is finally developed.

Keywords: spring, fatigue test, tester, structural design

1. Introduction

The spring is an elastic element that can transform mechanical work or kinetic energy into deformation energy, or deformation energy into mechanical work or kinetic energy when it deforms and returns to its original state.

In actual engineering, most springs work under variable loads, and their working stress is often lower than the yield strength of the material. Spring in the role of this variable load, after a longer period of operation and the phenomenon of operational failure is called the spring fatigue damage[1,2]. Fatigue damage is the main form of spring failure, according to statistics, about 80% or more of the spring failure is caused by fatigue damage [3]. With the development of modern machinery in the direction of large-scale, many springs in high temperature, high pressure, heavy load and corrosion and other harsh operating conditions, fatigue damage accidents are numerous [4].

Therefore, experimental verification of the fatigue strength design of springs, correct evaluation of their true fatigue characteristics, and verification of the true effect of the fatigue design are of great importance to improve the reliability and service life of mechanical products [5]. The compression spring fatigue tester designed in this paper can effectively determine the mechanical properties, process properties and fatigue strength of coil springs. It can help spring manufacturers to improve the quality of their products and ensure the safety and reliability of their products.

2. Main Technical Indexes

The main technical indicators of the spring fatigue tester are.

- Maximum load: ±50KN.
- Maximum loading displacement: ±15mm.
- Test frequency: 0.1~10Hz.
- Load displacement deviation amount: ≦3%.
- Test length space: 60-300mm.
- Counting range: 1~100000 times.

3. System Composition and Working Principle

The spring fatigue tester designed in this paper is realized by plane mechanical type, and the tester is composed of motor, transmission mechanism, frame, fixture and other parts. The structure of the spring fatigue tester is shown in Fig.1.

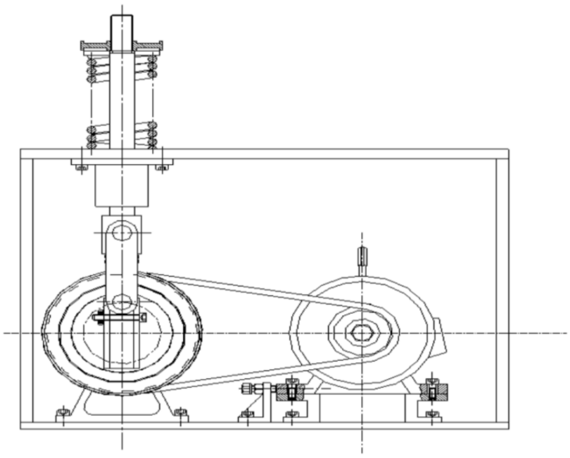

Fig.1 Test machine structure

At the beginning of the test, the motor drives the eccentric wheel to rotate, and at the same time drives the driving rod to do round-trip motion to apply a certain frequency of dynamic load to the spring under test. The motion system uses a crank slider mechanism to allow the driving rod to move in the guide rail, and the motor frequency control is used to control the loading frequency of the test, and sensors are used to count the number of tests. At the same time, in order to ensure the safety of the test, the tester can achieve automatic alarm and stop when the specimen fatigue fracture.

4. Drive Train Design

The scheme design of the mechanical drive system is an important part of the overall design of the fatigue testing machine. Its fundamental task is to convert and transmit the power generated by the motor to the actuating part according to the needs of the system. The design of the transmission scheme is to complete the power distribution ratio, calculate the motion and power parameters of the transmission device, and the selection of the motor.

4.1. Motor selection

The total efficiency of the system is assumed to be $\eta = 0.8$ in the design because of the power loss in the transmission process. Calculate the work required for the tester to make one stroke.

$$\omega = \frac{1}{2}sf = \frac{1}{2} \times 15 \times 50 = 375j \quad (1)$$

If the transmission ratio is $i = 2.9$ and the motor speed is $n_1 = 1400\mathrm{r/min}$, the spindle speed is $n_1 = 480\mathrm{r/min}$. The time required for one rotation of the spindle is 0.125s. The spindle power can be calculated.

$$\mathrm{P} = \frac{\omega}{\mathrm{t}} \approx 3000\mathrm{W} \quad (2)$$

motor power:

$$P_E = \frac{P}{\eta} \approx 3750W \quad (3)$$

Therefore, we can temporarily choose the servo motor model Y2-112M2-4, whose technical parameters is shown in Table 1

Table 1. Motor parameters

Model	Y2-112M2-4
Power Rating(kw)	4
Rotational Speed(r/min)	1400
Stall torque/rated torque	2.2
Maximum torque/rated torque	2.3

4.2. Drive train design

Because the machine overload easily damage the motor, in order to protect the motor, so the mechanical drive form of belt drive is used.

The belt drive is an extremely widely used flexible drive. Advantages of belt drive: smooth operation without noise; can moderate shock and absorb vibration; can prevent damage to other parts when the machine is overloaded; suitable for transmission with large center distance; simple structure and low cost.

According to the smooth load of V-belt, 16-hour two-shift working system, take $\mathrm{K}_A = 1.3$. The calculated power can be obtained.

$$P_{ca} = K_A P = 5.2KW \quad (4)$$

According to the calculated power $P_{ca} = 5.2KW$ and the small pulley speed $n_1 = 1400\mathrm{r/min}$, by checking the belt type chart, we can know that we should choose the A type V belt. The belt selection diagram is shown in Fig.2.

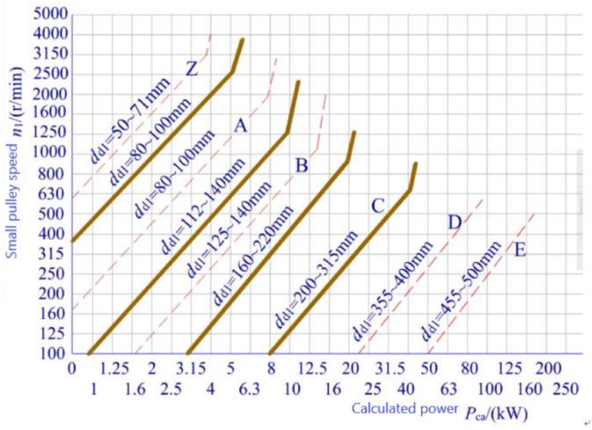

Fig.2. Belt selection diagram

A type V-belt small pulley diameter range is $\mathrm{d_{d1}} = 80\sim100\mathrm{mm}$. We take the small pulley diameter as $\mathrm{d_{d1}} = 100\mathrm{mm}$, and since the transmission ratio is $i = 2.9$, the large pulley diameter is derived as $\mathrm{d_{d2}} = 290\mathrm{mm}$, and finally the large pulley diameter is selected as $\mathrm{d_{d2}} = 280\mathrm{mm}$ according to the actual situation.

To check the correctness of the drive belt selection, a ratio error check and a belt speed check are performed.

(i) Transmission ratio error check.

$$i_e = \frac{\mathrm{d_{d2}}}{\mathrm{d_{d1}}\times(1-\varepsilon)} = 2.86 \tag{5}$$

ε is the elastic sliding rate. The percentage error of the transmission ratio is.

$$i = \frac{i-i_e}{i}\times 100\% = 1.3\% \tag{6}$$

The error percentage is less than the allowable error percentage and meets the requirements.

(ii) Band speed check.

$$\mathrm{v} = \frac{\pi\times\mathrm{d_{d1}}\times n_1}{60\times1000} = 7.33\ m/s \tag{7}$$

The belt speed satisfies $\mathrm{v} > 5\ m/s$, so the belt speed is appropriate.

5. Mechanical Component Design

In the design of the mechanical components of the spring fatigue tester, the design of the spindle and eccentric wheel mechanism is the most important.

5.1. Main shaft

The spindle is one of the important parts that make up the machine. The main function of the spindle is to support rotating parts and transmit torque and motion. The working condition of the spindle directly affects the performance and quality of the whole machine.

When designing spindles generally the main focus is on designing the structure, diameter and strength of the spindle.

(i) Structure of the spindle.

The structure of the spindle depends mainly on the type, number, location and method of mounting and positioning of the fixtures, transmission parts, bearings and seals mounted on the spindle, as well as on the processability of spindle machining and assembly. In order to meet the stiffness requirements and to obtain sufficient thrust surface as well as to facilitate assembly, the spindle is often designed as a stepped shaft, i.e., the shaft diameter decreases from the front shaft diameter to the back. The spindle of this design, also designed as a stepped shape, was also designed as a solid shaft while meeting the stiffness requirements. The structure of the spindle is shown in Fig.3.

Fig.3. Main shaft structure

(ii) Spindle diameter.

In this design, the spindle is made of 45# steel. Its tensile strength limit is $\sigma_b = 600MPa$, the allowable bending stress is $[\sigma_{-1b}] = 55MPa$, the allowable shear stress is $\tau_T = 30\sim40MPa$, and the coefficient is $\mathrm{C} = 106\sim108$. From this, the diameter of the spindle can be calculated.

$$\mathrm{d} = \mathrm{C}\sqrt[3]{\frac{P}{n_2}} = 19.7\sim21.7\ mm \tag{8}$$

Considering that there is an eccentric wheel installed on the shaft and a keyway at the end of the shaft, in order to ensure sufficient stiffness. Therefore, the estimated diameter range is increased by 3% to 5% and taken as 20.3~22.8mm. Final take $\mathrm{d} = 22\mathrm{mm}$.

(iii) Spindle strength check.

In this design, the spindle only transmits torque. Therefore, the strength check of the spindle can be calculated according to the torsional strength.

$$\tau = \frac{T}{W_T} = \frac{9.55\times10^6 P}{0.2d^3 n_2} \le \tau_T \tag{9}$$

In the above equation: τ is the torsional shear stress of the spindle; T is the torque; W_T is the torsional cross-sectional coefficient, for a circular section shaft $W_T \approx 0.2d^3$.

The final calculation yields $\tau = 37.37\text{MPa}$, which satisfies the strength requirement.

5.2. Eccentric wheel mechanism

When the stroke of the crank slider mechanism is small and the crank is short, the crank slider mechanism is usually designed as an eccentric wheel structure, using the eccentric pitch as the crank length. The spring fatigue tester in this design requires amplitude adjustment, i.e., the travel of the slider can be adjusted as needed, so the eccentric wheel is designed is shown in Fig.4.

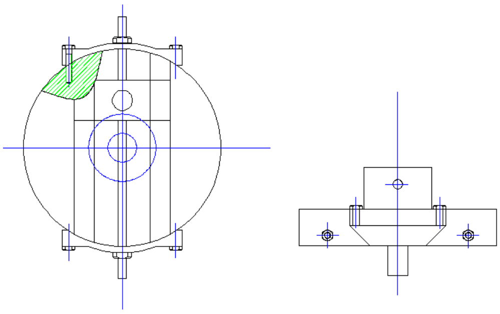

Fig.4. Eccentric wheel mechanism

In order to ensure the stability of the eccentric wheel mechanism, strength calibration of the eccentric wheel is required. The eccentric wheel is made of 45# steel and the slider is modulated. The circular shaft at the upper end of the slider is connected to the slider by welding, and the eccentric wheel is hinged between the circular shaft and the connecting rod. The maximum load transmitted to the drive rod by the eccentric wheel is $\text{F}_{max} = 250KN$. The articulated part of the slider and the connecting rod is a short circular shaft with a length of $\ell = 200\text{mm}$ and a diameter of $\text{d} = 20\text{mm}$. Calculate the maximum bending moment stress on the slider section.

$$\sigma_{max} = \frac{M_{max}}{W} \quad (10)$$

$$M_{max} = \text{F}_{max} \times \frac{\ell}{2} \quad (11)$$

$$W = \frac{\pi d^3}{32} \quad (12)$$

The maximum bending moment stress on the slider section is calculated to be $\sigma_{max} = 64MPa$. After the modulation treatment, the flexural fatigue limit of 45# steel is $\sigma_{-1} = 270MPa$. The comparison of the results shows that the strength of the eccentric slider meets the design requirements.

6. Conclusion

This design of spring fatigue tester realizes the fatigue test of compression spring. The machine has a simple structure, accurate and rapid running action, stable and reliable performance, easy to operate, etc. At the same time, the machine has good reliability, stability and versatility, and is suitable for use in small and medium-sized spring manufacturing plants and professional laboratories.

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Authors Introduction

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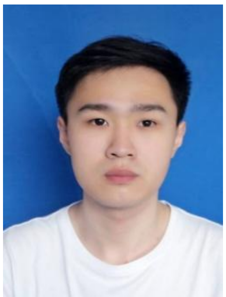

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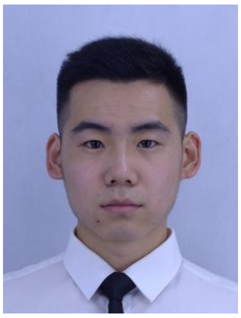

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