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Finite element analysis of a large flexible knuckle spring tube in aviation

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Abstract: As a standard part, spring is the core component of a satellite flexible mechanism. Its mechanical properties and fatigue life are important factors in determining the stability and safety of satellite flight. According to the actual demand of the coil spring used in this satellite and the fatigue failure of the coil spring at high frequency, the life of the coil spring is reduced. The stress, stiffness and fatigue life of the large coil spring in the flexible joint are analyzed by finite element method. The analysis shows that the maximum equivalent stress of the coil spring at the allowable torsion angle of 300° is less than the yield limit strength of the 316L stainless steel material. The error between the stiffness analysis result and the theoretical calculation value is 1.9%, and the minimum fatigue at a frequency of 1 Hz. The life span is 140,000 times, and the results meet the performance requirements of the actual flight of the satellite, providing a reliable basis for actual production and use. At the same time, the analysis results also provide a data basis for the establishment of the coil spring test bench.
Keywords: finite element; stress; stiffness; fatigue life.

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1. INTRODUCTION

The spring is a key component in a satellite system of aerospace. It not only stores and releases energy, but also mitigates shocks to some extent [1]. Cylindrical spiral torsion spring has the advantages of simple structure, convenient production and good mechanical properties [2], so it is widely used in aerospace. The performance of the spring directly affects the safety factor of satellite flight stability and operational safety. Ensuring the performance requirements of the spring is critical in the aerospace industry. Therefore, in the design and processing process, the spring has higher performance requirements.

At present, domestic and foreign scholars have done a lot of research on the performance of springs: the calculation formulas used in the design of the strength and stiffness of torsion springs generally assume that the torsion angle and torque of the spring are linear and ignore the twist of the spring wire. The effects of shearing and tensile deformation make the results of the mechanical properties of the spring slightly biased. Li Baofu [3] et al. discussed the variation law of cylindrical helical spring assemblies and proposed a more accurate calculation method, but the factors considered are difficult to control in actual production. Li Hongyan [4] used the spring geometry model established by APDL language to analyze the strength of the coil spring considering the spring base, and explained that the influence of the mechanism on the torsion spring should be analyzed and considered in the analysis. Yu Dongman [5] pointed out the defect of traditional mechanical design, based on the stress and strength interference model, introduced the reliability design of the fatigue strength of cylindrical helical spring, but it is necessary to obtain the real data in the actual scene before analyzing the design results. And judgment. At present, many scholars mainly carry out extensive and in-depth research on how to improve the fatigue life and anti-relaxation properties of springs under high stress, improve the heat treatment process, and improve the surface quality and dimensional accuracy of materials [6][7].

This paper conducts finite element analysis on the coil spring of a large flexible joint of aviation satellite, including the stress, stiffness and fatigue life of the coil spring. According to the complex use environment and strict performance requirements of the coil spring, the experimental results analyzed can meet the specified performance indicators, and also provide the basis for the establishment of the vacuum coil spring test bench.

2. PRETREATMENT OF THE COIL SPRING

2.1 Geometric model of the coil spring

The coil spring material studied in this paper is 316L stainless steel, which has an elastic modulus of 200 GPa, a shear modulus of 82 GPa, a density of 7850 kg/, a Poisson's ratio of 0.3, and a yield strength of 175 MPa. The specific size requirements are: the inner diameter of the coil spring tube is 425mm, the outer diameter is 3mm, the inner diameter is 2.6mm, the effective number of turns is 11 turns, and the twisted arms of each end are 100mm long. The condition of the coil spring tube is fixed at the upper end, and the lower end is twisted by 300° with the chassis. According to the specified size requirements, the geometric model of the torsion spring is established by using the three-dimensional modeling software CATIA as shown in Fig. 1 and the model of the spring mechanism is shown in Fig. 2.

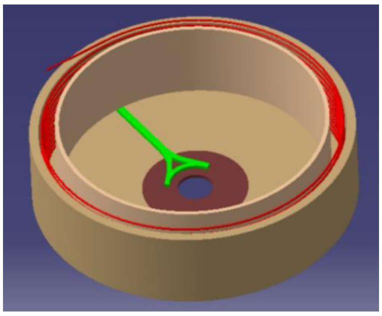

Figure 1 Working mechanism model of the coil spring

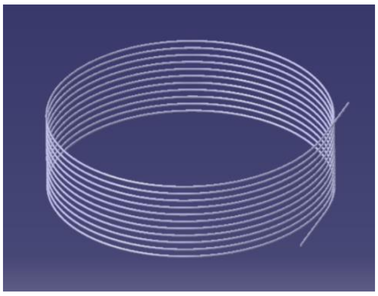

Figure 2 Clarinet geometry model

2.2 Finite element model

The authenticity and reliability of the finite element model analysis results are closely related to the degree of simulation of the actual working state. When creating a finite element model, it is necessary to realistically simulate the actual working state of the design object, and also take into account the calculation accuracy, convergence and computational efficiency of the model [8]. Therefore, the use of the working conditions for the model is conducive to reducing the huge amount of calculation, and it is also helpful for the convergence of the result process.

2.3 Meshing

The coil spring tube studied in this paper is a cylindrical spiral spring with a hole with a large diameter and a thin wall. In order to avoid excessive angles and deformed meshes, a more regular finite element mesh is divided, and a certain number of meshes is required to ensure the accuracy of the calculation results [9], so SOILD186 high-order 3D 20 nodes are used. The solid element, which uses the quadratic interpolation function, has better precision for irregular shapes and can control the arc curve boundary more accurately. The number of finite element units finally obtained was 603.28 million and the number of nodes was 486.251 million. The meshing is shown in Figure 3 and Figure 4.

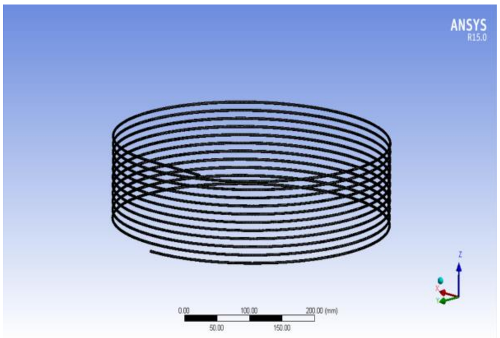

Figure 3 Overall mesh division of the coil spring

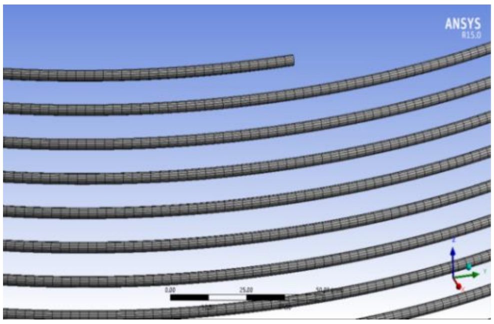

Figure 4 Partial mesh refinement diagram of the coil spring

2.4 Boundary conditions and load application

The boundary condition in the analysis is the specific expression of the actual assembly and use condition of the coil spring in the finite element model [10]. The upper end portion of the coil spring tube is restrained and fixed, and its freedom of movement and rotational freedom in three directions of X, Y, and Z are restrained, and then a concentrated load is applied to the lower end portion of the coil spring tube, and a twist angle of 300°is applied. 5 substeps are evenly loaded.

3. STRESS ANALYSIS OF COIL SPRING

3.1 Stress analysis of coil spring

The displacement diagram of the coil spring tube is shown in Fig. 5. It can be seen that the maximum displacement appears on the sixth circle and the value is 421.56 mm. The coil spring tube is in the range of 300° of the predetermined working torque, and the distribution trend of the maximum equivalent stress is: as the torsion angle becomes larger, the equivalent stress value gradually increases, and when the twist angle reaches the maximum angle of 300°， it appears. The equivalent stress maximum. Figure 6 shows that the maximum equivalent stress of the outer ring is 168.77 MPa, and Figure 7 shows that the maximum equivalent stress value of the inner ring is 139.9 Max. The maximum equivalent stress value is less than the yield limit of the material of 175 MPa, so the coil spring tube is safe in the working state of twisting 300°.

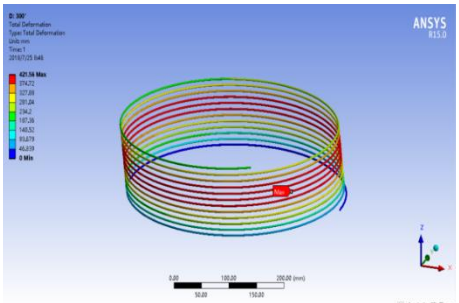

Figure 5 Total deformation displacement diagram of the coil spring

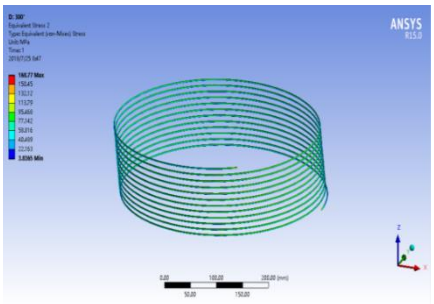

Figure 6 Stress distribution of the outer ring of the coil spring

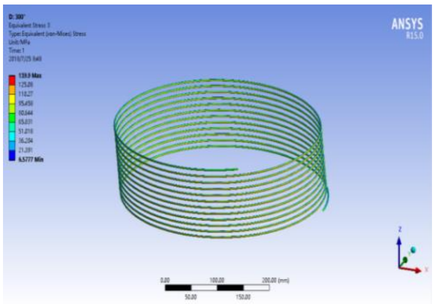

Figure 7 Stress distribution of the inner ring of the coil spring

3.2 Stiffness analysis of coil spring

Cylindrical coil springs generally only bear the effect of torque, and the coil helix angle of the spring can be approximated as 0, so only the bending moment acts on any section of the spring material $M = T$. Therefore, if the influence of the torsion arm on the stiffness is not considered, the theoretical stiffness of the torsion spring is calculated as:

$$K = \frac{Ed^4}{3667Dn}$$

If the influence of the torsion arm on its stiffness is considered, the theoretical stiffness of the torsion spring is calculated as:

$$K = \frac{Ed^4}{3667Dn + 389(a_1 + a_2)}$$

Where: E is the modulus of elasticity, d is the diameter, D is the median diameter, n is the effective number of turns, a_1 and a_2 the length of the torsion arm, respectively. Because the length of the torsion arm only plays a fixed role in the mechanism, and generally judge whether to ignore the influence of the torsion arm on the stiffness:

$$(a_1 + a_2) \geq 0.09\pi Dn$$

For this example, the length of the large torsion spring torsion arm is less than the value of the right side of the equation, so the influence of the torsion arm on the stiffness is not considered in this paper. The stiffness results are calculated after taking the specific parameters: $K=0.945N \cdot mm$

According to the finite element analysis, the relationship between the torque of the coil spring and the torsion angle is as shown in Fig. 8:

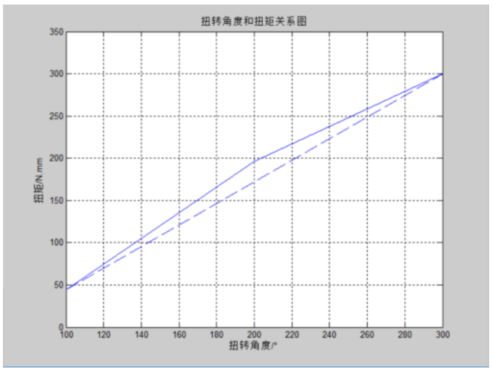

Figure 8 Torque and Torsion Angle Diagram

From the above relationship curve, the slope of the curve can be found to be 0.96, which is the stiffness value of the coil spring. The error with the theoretical calculation formula is 1.9%, and the error range is within the practical allowable range, so the simulation results have credibility.

3.3 Fatigue life curve

In this paper, the number of load cycles that the coil spring management wants is 100,000 times, which is a high cycle fatigue. The ultimate tensile strength of 316L stainless steel is 480MPa, and the fatigue life is 100,000 times. According to the allowable stress value requirements of the material, the maximum stress of the work as shown in Table 1 shall be less than 55%-65% of the tensile limit. between. When the smaller value is 55%, it is 264 MPa, and the maximum equivalent stress value in this example is 168.77 MPa, which is only 63.9% of the tensile limit. Meet the requirements of its regulations.

The fatigue performance of the high-cycle fatigue crack formation stage is often characterized by the SN curve, S is the stress level, N is the fatigue life, and the logarithmic relationship of S with N is obtained under the symmetric stress cycle (cycle characteristic value $\frac{\sigma_{\min}}{\sigma_{\max}} = -1$)

3.4 Fatigue life analysis

Table 1 Allowable stress values of materials

<table>
<tr><td colspan="3">Wire type</td><td>Stainless steel wire for spring</td></tr>
<tr><td rowspan="4">Torsion spring</td><td colspan="2">Test bending stress</td><td>$0.75\sigma_b$</td></tr>
<tr><td colspan="2">Allowable bending stress under static load</td><td>$0.68\sigma_b$</td></tr>
<tr><td rowspan="2">Allowable bending stress of dynamic load</td><td>Finite fatigue life</td><td>（0.55-0.65） σ_b</td></tr>
<tr><td>Infinite fatigue life</td><td>（0.45-0.55） σ_b</td></tr>
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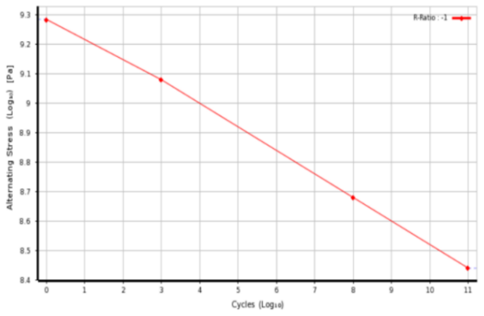

Figure 9 S-N curve logarithmic plot

The distribution of the stress analysis of the coil spring tube is obtained, and the maximum stress of the coil spring tube is 168.77 MPa, which is located in the sixth working circle. If the fatigue life of the position is within the design range, the overall fatigue life of the coil spring meets the engineering application requirements.

The simulation results are shown in Fig. 10. The minimum number of cycles at a frequency of 1 Hz is 140,000 times, and the result is greater than the required 100,000 times, which can meet the requirements for use.

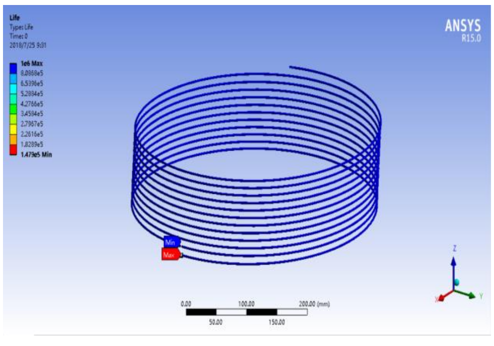

Figure 10 Foil spring fatigue life diagram

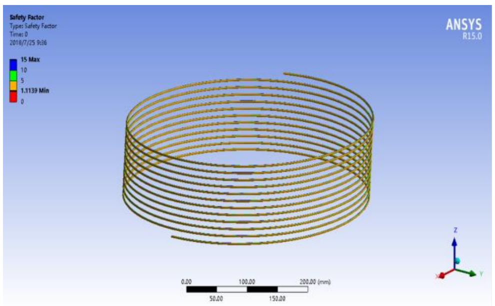

Figure 11: Minimum fatigue safety factor diagram of the coil spring

Fatigue safety factor can reflect the relationship between fatigue strength and maximum stress, and is also an important factor to measure component safety and fatigue life. The simulation results are shown in Fig. 11. The minimum fatigue safety factor is 1.1139, which is greater than the standard value of 1, so the working state of the torsion spring is safe.

The stress intensity factor is an important parameter to evaluate the safety of the component. Damage, crack, corrosion and other factors will cause certain damage to the safety of the component, which directly affects the service life of the component. The simulation results show that the minimum strength safety factor of the torsion spring is 2.24, which is greater than the standard value 1, as shown in Fig. 12. Therefore, the safety of the coil spring meets the requirements.

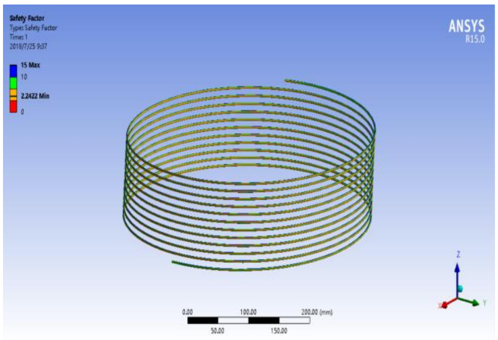

Figure 12: Minimum strength safety factor diagram of the coil spring

4. SUMMARY

The maximum stress of the coil spring occurs at the sixth coil value of 168.77 MPa, which is less than the yield limit of the 316L stainless steel material, and satisfies the stress range of the coil spring torsion 300°.

Comparing the results obtained by finite element analysis and theoretical calculation, it can be found that the error value of the coil spring stiffness calculation value is only 1.9%, the calculation result satisfies the actual working condition requirement, simplifies a large number of manual calculations, and also guarantees the result. Reliability.

The dangerous part of the coil spring is located outside the inner side of the sixth ring. The fatigue life of the coil spring is at least 140,000 times at a frequency of 1 Hz, which is greater than the fatigue life requirement of the class II load of 100,000 times.

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