Experimental study on vibration fatigue of a certain chassis

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Abstract. The chassis is the structure for installing and protecting various circuit units, components and mechanical components inside the electronic device. Vibration fatigue damage is a major form of structural damage, and the vibration fatigue life prediction of the chassis structure is of great significance. The theoretical calculation of structural vibration fatigue life has frequency domain method and time domain method, but it has certain limitations. The theoretical calculation method can not accurately obtain the vibration fatigue life of the structure. According to the actual structure of a chassis, this paper designs and processes a test model of a chassis, applies a random vibration load to the vibration table, and performs vibration fatigue test to predict the vibration fatigue life of a chassis structure. The results show that the vibration fatigue life of the chassis is 40 hours, and fatigue fracture occurs first at the pin.

1. Introduction

The chassis is the structure for installing and protecting various circuit units, components and mechanical components inside the electronic device, which plays a very important role in ensuring the safety, stability and reliability of the equipment [1]. Vibration fatigue damage is a main form of structural failure. Different from the static damage, the damage under the vibration condition is more difficult to predict. Generally, it can be displayed after a period of time, and the potential threat is greater. With the development of airborne radar structure technology, more and more attention has been paid to dynamic strength design [2-3].

There are frequency-domain method and time-domain method in the estimation of structural vibration fatigue life [4-6], The frequency-domain method has the characteristics of simple thinking and small calculation, but the estimation error of fatigue life is large according to the peak distribution method. The time-domain vibration fatigue life estimation method first simulates the response spectral density of the structure in time-domain, obtains the response time history, and then forecasts the fatigue life. The idea of time domain method is the closest to that of general cycle fatigue life estimation method, but it is difficult to apply in engineering calculation because of the large amount of time-domain simulation of random process.

The fatigue life of structure under vibration load is affected by many factors, including average stress, stress concentration and damping of structure [7]. The influence of the average stress and stress concentration on the general cyclic fatigue has been well solved [8], but how to consider the influence of the average stress and stress concentration on the structural vibration fatigue has not been studied, including the influence of damping on the structural vibration fatigue. The feasibility of the new
Neuber formula was also corrected and verified by carrying out simulations and experiments. Chen, Z.-Y. [9] used the stress-strain field strength method to calculate the fatigue life considering the local stress and strain of the wheel notch, and verified the accuracy of the method through theoretical analysis. In the fatigue life prediction, it is often assumed that the random load of the structure is a stationary Gaussian distribution. But in the actual environment, the random load tends to exhibit non-Gaussian characteristics, and in the non-Gaussian process, the super Gaussian random process is the majority [10]. Niu, Q et al [11] proposed a new frequency domain method for used for fatigue analysis of non-stationary processes, and carried out theoretical analysis and experimental verification respectively, and obtained a new fatigue life assessment method taking into account the coupling effect between vibration and fatigue. Pengyu Wang et al. [12] studied the super-Gaussian vibration and obtained the expression of response kurtosis under the single-degree-of-freedom system. It was verified by experiments, and the output response spectrum results were completely within the range allowed by the error. It has a greater significance for the study of super Gaussian vibration.

In order to accurately evaluate the durability characteristics of a chassis under random vibration environment, this paper designs and processes a case test model according to the actual structure of a chassis; uses finite element simulation to analyze the random vibration response of a chassis structure and determine the dangerous point of the structure; applies a random vibration load to the vibration table, and performs vibration fatigue test to predict the vibration fatigue life of a chassis structure.

2. Chassis geometry model
The chassis body is made of aluminum alloy 5A05. The chassis includes three frames of screws, and the rear includes two locating sleeves, as shown in Figure 1. In order to simulate the actual chassis weight, 17 counterweights are added in the chassis. The chassis is installed on the vibration table through tooling as shown in Figure 2(a), and the whole chassis and tooling are installed as shown in Figure 2(b).

![Figure 1. Chassis geometry mode.](a)

![Figure 2. Chassis and tooling model.](b)

3. Finite element simulation analysis
In order to determine the layout of strain measurement points, the finite element fatigue simulation analysis of the case test piece is carried out before the formal test. Through the finite element analysis results, the stress-strain response nephogram of the structure under the given acceleration excitation can be obtained, and then the maximum stress-strain area (i.e. dangerous point) can be found as the reference for the location of strain measuring points.

3.1. Finite element model of chassis
The chassis test piece is simplified and meshed by UG. All the finite element meshes are 10 node tetrahedral elements. In order to apply load conditions and avoid introducing nonlinear contact analysis, the boundary conditions are used to simulate tooling and only the upper case model is established. The components of the chassis model are connected by glue constraints. The finite element model of the chassis is shown in Figure 3.
3.2. Materials and boundary conditions
The chassis material is 5A05 aluminum alloy, and the positioning sleeve material is 1Cr18Ni9Ti stainless steel. See Table 1 for the above specific parameters.

The simulation boundary is given according to the test boundary conditions. The displacement boundary condition is applied to the bolt and locating sleeve to simulate the displacement constraint under the tooling condition. For the thread position surface of the screw that can not be loosened and the lug surface of the peripheral contact tooling, all the degrees of freedom of the grid shall be constrained; for the inner surface of the positioning sleeve, the non axial translational degrees of freedom shall be constrained.

Table 1. Material parameters involved in the analysis.

| Material parameters          | 5A05  | 1Cr18Ni9Ti |
|------------------------------|-------|------------|
| Modulus of elasticity (GPa)  | 70    | 184        |
| Shear modulus (GPa)          | 26.6  | 69.9       |
| Poisson's ratio              | 0.32  | 0.33       |
| Mass density (kg/m3)         | 2660  | 7900       |
| Tensile strength (MPa)       | 290   | 540        |

3.3. Analysis of simulation results
Using Patran & MSC.Nastran software, the random vibration response of the model is calculated by direct method, and the calculation results are shown in Figure 4.

Under the condition of functional random vibration, the maximum stress appears at the position of the long round hole sleeve on the left side of the back (294MPa, where material 1Cr18Ni9Ti, tensile strength 540MPa), the maximum stress of aluminum structure appears at the middle lug of the frame (205MPa, 5A05 (H112) tensile strength 290MPa), and the maximum acceleration appears at the upper center of the back cover plate.
Figure 4. Simulation results of random dynamic response.

Under the condition of durable random vibration, the maximum stress is 186MPa (sleeve) and 130MPa (aluminum structure).

4. Test scheme
In the experiment, the vibration table was used to apply random vibration load, and the strain response of key points was monitored by strain electrical measurement, and the stress response and its evolution process were obtained by principal stress analysis.

4.1. Test loading scheme
The dc6000 electric vibration system is adopted in the test. The loading frequency of the system is 5-2500hz, the maximum sine thrust is 58.8KN, and the maximum acceleration is 100g. It can carry out sine and random loading in horizontal and vertical directions. This test is based on ASTM d4728-2017 test standard. The whole system is shown in Figure 5.

According to the demand and analysis, the most unfavorable loading direction is along the axis of the locating pin. Therefore, use the horizontal slide and cooperate with the tooling to apply random vibration in this direction to the chassis. Its random vibration spectrum is shown in Figure 6, the loading point is below the chassis, and its effective acceleration value is 6.4g. In the process of the test, in order to speed up the test process, increase the amplitude of the random spectrum by 2.25 times, and its effective acceleration values are 9.6g, which eventually leads to failure.
4.2. Response test plan
The strain response of key position was measured by strain electrical method. According to the results of the finite element analysis, it is determined that a 45 ° strain flower is arranged at the three non detachable screw racks of the case and the two locating sleeves at the rear respectively, and the 0 °, 45 ° and 90 ° strain responses at each measuring point are measured, as shown in Figure 7.

Dh5923 dynamic signal test and analysis system is used for strain collection and test, which is completed with strain bridge box. The system has the functions of multi-channel synchronization, high-speed sampling and recording. The sampling frequency of 5KHz is selected in the test to ensure the real and effective data. Strain response data shall be recorded every 0.5/1 hour during the test.

5. Test results and analysis
The test was carried out for 40 hours, in which 0-39.5 hours was a random vibration with an effective acceleration of 6.4g, and 39.5-40 hours was a random vibration with an effective acceleration of 9.6g. In the test, the strain is measured every 5 hours. After the strain response in three directions of each measuring point is measured, the evolution process of the main strain amplitude of each measuring point with time is obtained through the main stress analysis.

No. 1 to No. 6 measuring points are near the loose bolt, No. 7 to No. 10 measuring points are near the locating pin. Some measuring points are damaged in the middle of the test. The amplitude of the principal stress ε 1 at the initial and end time of each measuring point is shown in Table 2.

After 40 hours of random vibration, the pin shaft at the positioning sleeve of the case broke and the test stopped. It can be seen from Figure 8 that the cross section of the pin of the circular locating sleeve has been damaged and it is cut off horizontally after the test. In addition, due to the continuous vibration with high strength, three bolts are bent to a certain extent, as shown in Figure 9. In addition, several bolts used for connecting the front and rear covers of the chassis with the main body of the
chassis are partially fallen off and partially broken during the vibration process, and the rest are bent to a certain extent after being taken off, as shown in Figure 10.

Table 2. Evolution process of main stress $\varepsilon_1$ amplitude of each measuring point

| Measuring point | Initial time | Closing time |
|-----------------|--------------|--------------|
| Point No. 1     | 208.9        | -            |
| Point No. 2     | 167.6        | 204.2        |
| Point No. 3     | 196.3        | 192.1        |
| Point No. 4     | 135.1        | 185.7        |
| Point No. 5     | 120.3        | 169.0        |
| Point No. 6     | 208.2        | 195.9        |
| Point No. 7     | 33.7         | -            |
| Point No. 8     | 188.7        | -            |
| Point No. 9     | 154.4        | 65.7         |
| Point No. 10    | 91.8         | -            |

Figure 8. Failure section of fixed export sales.

Figure 9. Solid bolt.

Figure 10. Bolt failure.

Through analysis, because the random vibration direction is parallel to the axial direction of the pin shaft, and the pin shaft and the fastening bolt are located at the lower part of the chassis, under the condition of the overall weight of the chassis is large, the pin shaft bears a large axial force and finally fatigue fracture occurs in the process of continuous tension and compression. However, the stress level
of the fastening bolt is always at a low level, and there is no obvious fatigue failure before the pin shaft breaks.

6. Conclusion
According to the requirement of durability verification of a case in random vibration environment, random vibration test is carried out. The specified random vibration spectrum is applied to the case through the electric vibration system, and the vibration magnitude is adjusted according to the demand to accelerate the test process. During the test, the strain response of key points is monitored by Strain Electrical Measurement and corresponding acquisition equipment, and the principal stress and its evolution process are obtained by principal stress analysis method. According to the test data and fracture morphology, the failure reason and its rationality were analyzed.

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