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DYNAMIC CHARACTERISTICS AND FATIGUE ANALYSIS OF TURBINE DISK

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Abstract

Fatigue cracking failure of turbine disk was found in hot fires tests of XX rocket engine. A series of calculation and analysis were conducted to determine failure causes. Structural dynamic characteristics in high temperature and high rotating speed environments were emphasized. By comparing the calculated results with the test data of turbo pump in hot fires, it is found that one of modal frequencies of the original structure coincides with the six-times of rotating speed frequency. The dynamic response calculation results show that high stress level exists in the turbine disk when high amplitude resonance occurring. Sometime, it may cause cracking damage of turbine disk. After structural redesign, this kind of coupled mode has been avoided successfully. Also, the improvement design can effectively decrease stress level and increase the fatigue strength of the weak area. The hot fire tests have demonstrated the success of the improved design.

INTRODUCTION

Liquid rocket engine is the main vibration source of a launch vehicle. In operating conditions, the engine itself is also subjected to random vibration loading. Although most liquid rocket engines of launch vehicles are expendable ones, and their accumulated operating time is short, however, failures from structural coupling vibration due to improper design occur occasionally. Turbine disk is a key component of rocket engine operating at high temperature and high rotating speed, which is also subjected to aerodynamic loading and random dynamic vibration loading. Obviously, fatigue cracks in turbine disk will affect the operating safety and reliability of an engine, and sometimes cause catastrophic consequence in hot fires or launch task.

In this paper, modal characteristics of the original turbine structure were investigated firstly in high temperature and high rotating speed environments.

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Furthermore, structural dynamic response was computed. Fatigue strength at the crack initiation area was calculated, and simulation experiments were performed. Finally an overall analysis of the improved structure was given to demonstrate feasibility of the improved design.

STRUCTURAL MODAL ANALYSIS

Structural Modelling

Modal frequency of the turbine disk in assembly conditions for rocket engine is different from that of the free-free conditions. Considering effect of the bearing constraints on modal frequency in actual operating conditions, a passage of shaft is included in the computing model. The connection part is simplified into an integral structure with two stage disks. When the connection strength is strong enough, the above simplification will ensure the accuracy of modal computation.

Loading Conditions and Temperature Field

Modal frequency of the turbine disk in static status is different from that of the rotating conditions. In the rotating conditions, due to the mass centrifugal force, the dynamic frequency is higher than the static frequency. Modal analysis where inertial loading exists belongs to pre-stressed non-linear modal analysis problem.

In combination with data from several hot fire tests, turbine disk dynamic frequency at three typical rotating speeds, namely 9078rpm, 9282rpm and 8670rpm, are selected and calculated.

The turbine disk operates at high temperature and high speed airflow environments, it is difficult to define the boundary of the temperature field, so the following assumption is given: In the high velocity and temperature gas flow, the temperature of the turbine disk is tend to be uniform at a very shot time. In modal calculation, uniform temperature field is selected without temperature gradients.

Determination of the Finite Element Model

Before creating the mesh, the shape of the bladed disk should be simplified firstly. The stiff equivalence method was adopted. The left part of figure 1 shows the cross section of blades for the first stage disk, and the right part shows the simplified rectangular cross section. Therefore, the actual bladed disk is equivalent to a disk without blades. The examining calculation is performed to determine precision of structural equivalence. It is found that the eigenvalues of equivalent disk without blades is nearly equal to that of

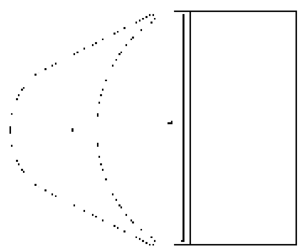

Figure 1. Simple cross section of blade

the actual bladed disk when neglecting those local modes.

Calculation Method

Modes are inherent properties of a structure. Each mode is determined by its natural frequency and mode shape. Lanczos method is an algorithm for computing the eigenvalues and eigenvectors for large symmetric sparse matrices.Because of its good convergence, it is used in the structural modal calculation.

During operation, turbine disk experienced different temperature changes, from normal to high and finally close to uniform. According to temperature field analysis, five kinds of temperature fields were assumed. Temperature correction coefficient, namely the ratio of modal frequency at deferent temperature without rotation and the corresponding modal frequency at normal temperature was introduced. Stability of the mode results can be determined by the temperature correction coefficients

When in operation, turbine disk operates at high rotating speed, the resulting centrifugal force has obvious effect on its natural frequencies. Dynamic frequency coefficient is introduced which can further determine the reliability of modal analysis results. The dynamic frequency coefficient at 9078 rpm and different temperatures for the original disk is shown in figure 2.

Modal Analysis Results

In order to compute modal frequencies at different temperature and turbine rotating speed, three kinds of turbine rotating speed, namely 8670rpm, 9078rpm, and 9282rpm were selected based on measuring data of turbine disk rotating speed and failure phenomena in hot fires.

From comparing of the original with the improved turbine disk modes, it can be found out that the first 30 modes is the entire disk mode and the others are local modes resulting from structural simplification and equivalence, computing error of some modes can occur, those pseudo modes should be eliminated in modal analysis.

By analysing the first 30 modes, we can determine if there is a modal frequency coupling with the gas fluctuation excitation frequency before and after the dynamic modification. Some modes at normal temperature for original and improved structures are shown in table 1. The comparison of computing modes for both kinds of disk corresponding to different temperature and rotating speed is shown in table 2. The mode shape for 920.75Hz frequency is plotted (figure 3), while temperature is 550℃ and rotating speed is 9078 rpm for actual operation conditions.

Table 1. Some modes at normal temperature of turbine disk

Order	Modes of original turbine disk（Hz）	Modes of improved turbine disk（Hz）
15	948.81	997.97
16	967.39	1031.87
28	1844.59	1954.61
29	1844.59	1962.33

Table 2. Comparison of the calculated results for original and improved structures

Rotating speed	0 rpm		8670 rpm		9078 rpm		9282 rpm	
Temperature	original	improved	original	improved	original	improved	original	improved
20℃	967.44	1031.87	1015.43	1079.76	1019.43	1083.87	1021.50	1085.98
300℃	911.28	971.98	961.44	1022.17	966.14	1026.48	967.84	1028.70
500℃	863.64	921.19	916.31	970.04	920.75	978.58	923.04	980.91
600℃	836.22	891.95	890.20	946.16	894.76	950.81	897.11	953.21

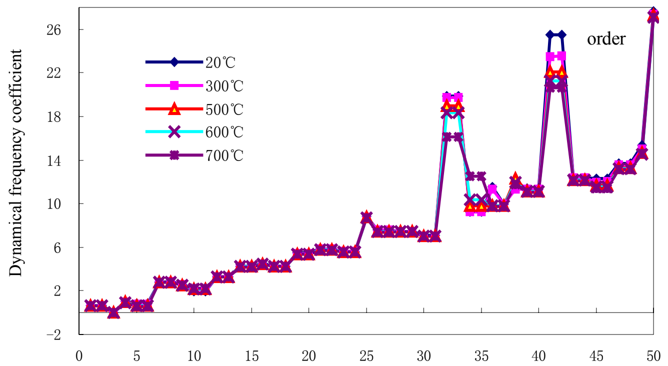

Figure 2. Dynamical frequency coefficient at 9078 rpm and different temperatures

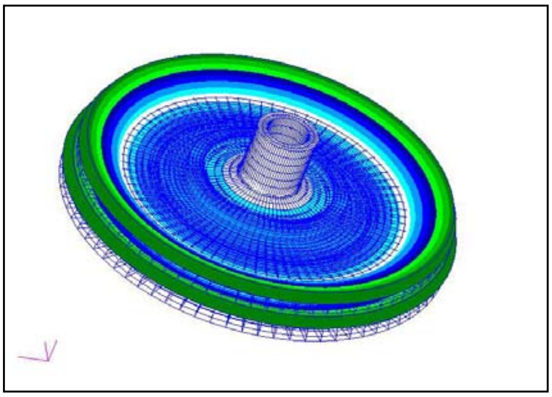

Figure 3. Mode shape at 920.75Hz for original structure

Coupling Mode Analysis

Gas pressure fluctuation is the main excitation source of the turbine disk. Our task is to determine the modal frequency coupling with the six-time or twelve-time excitation frequency. From above-mentioned modal computation and analysis, it is found that the

16th mode of the original and improved disk is numerically near to six-time frequency of rotating speed.

The relationship between frequency and temperature for the original disk at different rotating speed can be studied for this mode. The results are shown in figure 4. It is found that there is one modal frequency coupling with six-time the rotating speed at 550℃ and 9078 rpm , and another appears at 480℃ and 9282 rpm of disk.

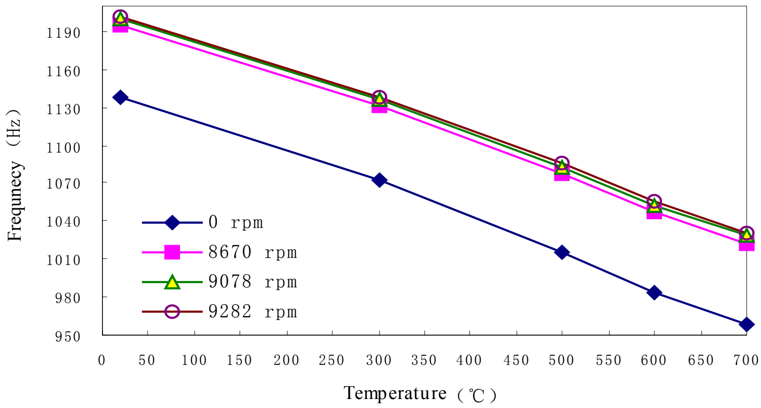

Figure 4. The relationship between frequency and temperature

STRUCTURAL DYNAMIC RESPONSE ANALYSIS

Structural dynamic response was investigated for the original and improved disk with frequency response method respectively. Narrowband random loading was applied at the equivalent areas of actual disk blades simulating the real operation conditions of the turbine disk.

Damping value is a measuring value obtained from experimental modal analysis of the disk in actual turbine pump. Figure 5 shows the corresponding dynamic stress contour of 923.4Hz for the original disk at 500℃ and 9078rpm. It clearly shows the maximum dynamic stress exists in the weak location of turbine disk.

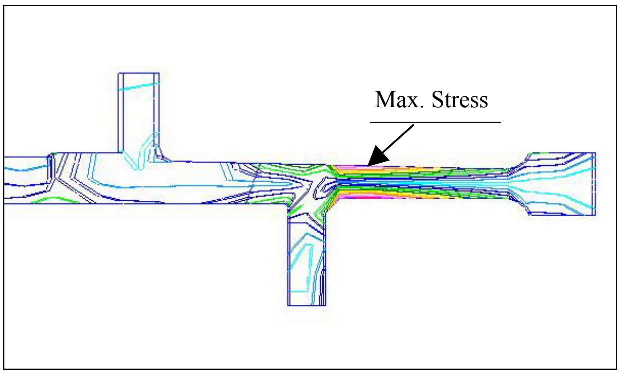

Figure 5. Dynamic stress contour of 923.4Hz mode

FATIGUE STRENGTH ANALYSIS

Simulation Fatigue Test

In order to compare the fatigue behaviour of the original and improved turbine disk, simulation experiments are performed on steel specimens. In fatigue simulation test, the specimen is subjected to harmonic excitation. Meanwhile, the fatigue cracking initiation was determined by vibration response monitoring method.

Large deviation values of the test were checked, and all the test data meet the standard requirements. Test results show that cracks were initiated from the weak area of the specimen, and low cycles fatigue cracks were observed from metallography analysis. Although the comparison results vary with the difference between the specimen's material and the actual disk material, the improved structure has longer fatigue life from test results. It is also proved preliminary that the fatigue life of the improved disk is more than two times life of the original one.

Computation Analysis

Fatigue strength computation was conducted based on failure property of the first turbine disk, dynamic characteristic analysis results and random vibration loading on the turbine disk. It can be seen that disk travelling wave coupling occurs in the turbo pump operation speed. Centrifugal force and thermal stress were computed with finite element method. Thermal stress corresponding to static and transient temperature fields was computed respectively. These initial static stresses were considered as mean stress in fatigue strength computation.

Known the mean stress and stress range, working safety coefficients corresponding to the given fatigue life can be calculated by the following formula:

$$n_\sigma = \frac{\sigma_b' \ \sigma_{-1DN}'}{\sigma_b' \sigma_a + \sigma_{-1DN}' \sigma_m} \tag{1}$$

$$\sigma_{-1DN}' = \sigma_{-1N} \big/ \left(K_{\sigma DN} K_{SN}\right) \tag{2}$$

$$\sigma_b' = \sigma_b \big/ \left(K_{s1}\right) \tag{3}$$

where σ'_{-1DN} is the conditioned fatigue limit of component, K_{SN} is scatter coefficient, $K_{\sigma DN}$ is fatigue notch coefficient, σ_a is stress range, σ_m is mean stress, σ_b is ultimate tensile strength.

The working safety coefficients of disk at different operation conditions is showed in tab.3 in which stress range is about 77.3 MPa . Calculation results show that in the lower temperature field, the structure has a higher working safety coefficient. The coefficient degrades with temperature rise, especially in the fatigue initiation location where great temperature gradient exists. The preliminary fatigue analysis of the improved structure show that fatigue life increased significantly, the redesign of disk is feasible.

Table 3. Working Safety Coefficient

N_f (s) / σ_m (MPa)	10	50	100	200	600
250	2.51	2.24	2.13	2.08	1.98
400	1.77	1.64	1.58	1.55	1.49
550	1.37	1.29	1.25	1.23	1.20

Hot Fire Tests

Analyses of structural dynamic characteristic and rotating speed region coupling with travelling wave resonance were made for the improved disk. Results show that rotating speed coupling region changes obviously, and the lower limit is higher than the maximum rotating speed. The travelling wave coupling resonance does not exist in the improved disk. So the new structure has perfectly improved the structural dynamic characteristics of the disk, preventing from the danger of coupling resonance and the large alternating stress, greatly improving the fatigue life of disk.

After several hot fire tests for rocket engine, no cracks were found in the improved disk. Comparison of the measuring acceleration for the original and improved disk in hot fire test shows the improved structure has little effect on engine vibration environments. The improved disk is sound and does not affect the engine performance.

CONCLUSIONS

Suitable simplification and boundary conditions on turbine disk can increase computing accuracy of the disk modal analysis. For the original turbine disk, there exists two kinds of modal frequencies coinciding with the six-time of rotating speed, one coupling modal frequency appears at 550℃ and 9078 rpm of disk; another appears at 480℃ and 9282 rpm of disk.

Due to the travelling wave coupling resonance and greater static stresses in the weak area of the original disk, it leads to fatigue cracking failure. Comparison of the modal analysis of original and improved disk shows that this kind of coupled mode has been avoided in the improved disk. For the improved disk, its fatigue life increased significantly. The succeeding hot fire tests have demonstrated the success of the improved design.

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