VIBRATION CHARACTERISTIC OF METALLIC THIN-WALLED STRUCTURES UNDER HIGH TEMPERATURE AND HIGH INTENSITY NOISE

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Thin-walled structures of aeroengine are under the ever growing threats from the combined excitation of elevated thermal load and high level acoustic load which can reach 1500°F in temperature and 170dB in sound pressure level. The consequent large amplitude nonlinear responses reduce the fatigue life of structures greatly. An acoustic experiment on a turbofan aeroengine is firstly implemented to investigate noise characteristic in the combustion chamber. The noise signal under several adjustment status of the engine is obtained which served as the acoustic load in the numerical calculation. The coupled Finite Element Method and Boundary Element Method (FEM/BEM) is used to calculate dynamic responses of metallic thin-wall rectangular plate with four edges clamped. The influence of progressive wave acoustic field and the effects of temperature rising on the vibration response of thin-walled structures are discussed. Time-domain stress results under different thermal-acoustic loads are extracted. The characteristics of thermal-buckling and snap-through of structures are analyzed. And at a special buckling coefficient, stress response of the structure shows intermittent snap-through motion. The calculation method and analysis results obtained in this paper can contribute greatly to the thorough understanding of thermal-acoustic response characteristics of thin-walled structures and the anti-acoustic fatigue design in aeroengine.

Keywords: combustion chamber, thin-walled structure, acoustic load, vibration, snap-through

1. Introduction

With continuous strengthening of national defense requirements and the rapid development of the aviation industry in all countries in the world, the performance of modern aeronautical vehicles and engines have also been constantly enhancing. In order to meet high standards of performance requirements and guidelines, thin-walled structures are widely used in aircraft parts, such as nacelle structure, combustion chamber and nozzle of engine [1]. Especially, thin-walled structures of aeroengine are under the ever growing threats from the combined excitation of elevated thermal load and high level acoustic load which can reach 1500°F in temperature and 170dB in sound pressure level. Under these loads, the thin-walled structure will show common features such as thermal buckling, rapid
alternating stresses inside the structure [2] and snap-through motion [3][4][5]. Such large deflection nonlinear dynamic response will seriously affect mechanical properties of structures, weaken the fatigue performance of structures and further reduce the fatigue life of thin-walled structures [6]. Therefore, in order to improve structural strength design requirements and enhance mechanical properties of aeronautical thin-walled structures under severe loads, it is significant to carry out related simulation analysis of metallic thin-walled structures to thermal-acoustic load.

Several experimental researches on the structure under harsh thermal-acoustic load have been performed. An acoustic excitation test on the aluminum alloy plate was carried out by C. Mei and K.R. Wentz. And the strain response of aluminum plate at different sound pressure levels was measured [7]. J.R. Ballentine investigated the influence of composite loads including temperature, noise and low-frequency vibration on the fatigue life of the structure [8]. Structural acoustic fatigue design criteria was presented at high temperature by C.W. Schneider [9]. An acoustic fatigue testing technique at high temperature was expounded by M.H. Hieken [10]. Thermal-acoustic experiment was studied to examine the thermal protection system of the space shuttle in the German Space Research Center. As the next-generation launch technology research project had been proposed in NASA, the design of thin-walled structure had also been gradually shifted from thermal protection to thermal structure [11].

Due to high cost and getting reliable data is difficult in experimental test for severe thermal-acoustic loads, numerical calculation has received particular attention. Besides, predicting the response and life of complex structures under thermal-acoustic conditions has become a huge challenge. The single-mode motion equation of a rectangular isotropic plate under thermal-acoustic loads was obtained based on the Galerkin method. And the mean and effective values of stress and strain responses were calculated [12]. With the improvement of calculate ability, Finite Element Method (FEM) which can analyze nonlinear response analysis of complex structures has gradually become a mainstream method. FEM was first extended to the simulation of isotropic beams and slabs under thermal-acoustic loads, and the response analysis was completed [13].

In this paper, an acoustic experiment on a turbofan aeroengine is firstly implemented to investigate noise characteristic in the combustion chamber. The noise signal under several adjustment status of the engine is obtained which served as the acoustic load in the numerical calculation. The coupled FEM/BEM method is used to calculate dynamic responses of metallic thin-wall rectangular plate with four edges clamped [14]. The influence of progressive wave acoustic field and the effects of temperature rising on the vibration response of thin-walled structures are discussed. Time-domain stress results under different thermal-acoustic loads are extracted. The characteristics of thermal-buckling and snap-through of structures are analyzed. And intermittent snap-through motion of the plate is caught at a special buckling coefficient. The calculation method and analysis results obtained in this paper can contribute greatly to the thorough understanding of thermal-acoustic response characteristics of thin-walled structures and the anti-acoustic fatigue design in aeroengine.

2. Acoustic experiment system

An acoustic experiment on a turbofan aeroengine is firstly implemented to investigate noise characteristic in the combustion chamber, where the static pressure is up to 4 MPa, the temperature is about 2200 °F. Therefore, an acoustic experiment system is adopted for the combustion noise testing. It is essentially an acoustic waveguide system. A quarter inch condenser microphone is internally installed in the waveguide system. One end of the system is connected to a semi-infinite tube which applied to avoid the acoustic wave reflection in the system, and another end is connected to the combustion with the pipe which installed flush with inner surface of the connotation. The acoustic experiment system connection diagram is presented in Figure 1. The static calibration of the waveguide system is implemented by the pistonphone with 250 Hz / 124 dB. The digital closed-loop acoustic calibration system is used to calibrate the dynamic characteristics of the waveguide system. The
acoustic calibration system connection diagram is presented in Figure 2. The frequency response data of various length pipe at different frequencies are obtained through the calibration system.

![Acoustic calibration system diagram](image)

**Figure 1: Acoustic measurement system.**  **Figure 2: Acoustic calibration system.**

### 3. Characteristic of combustion noise

During the testing, noise in the combustion chamber is obtained with the rotating speed of aeroengine synchronously. When the rotating speed of aeroengine is operated on the design condition, the time-domain wave of noise is transformed into frequency spectrum by Fast Fourier Transform. It is shown in Figure 3. Under twelve typical working conditions of the aeroengine from the lowest continuous speed with load (LS) to the maximum no load governed speed (MAX), the noise spectrum distribution in the combustion chamber is basically the same. And the peak value of the noise spectrum mainly appears in the low frequency band below 2000Hz. With the increase of rotation speed, the noise spectrum presents characteristic frequency peak in the frequency range from 655 Hz to 870 Hz. The characteristic frequencies shift to the right of the spectrum as the rotation speed increases. When the rotating speed reaches to 96% and higher, another characteristic frequencies appear in the range from 5305 Hz to 5475 Hz, and which shift to the left of the spectrum as the rotation speed increases.

![Evolution law of noise frequency spectrum](image)

**Figure 3: Evolution law of noise frequency spectrum.**
4. Vibration response of thin-walled structure

4.1 Theory basis

The dynamics responses of thin-walled structures in a complex thermal-acoustic environment are considered to be large deformation nonlinear vibration. When the effect of large deformation is taken into consideration, the equations of motion of rectangular plate can be expressed as

$$DV^4w + D\alpha (1 + \mu) \nabla^2 \theta = \frac{\partial^4 F}{\partial x^4} \frac{\partial^2 w}{\partial x^2} + \frac{\partial^4 F}{\partial y^2} \frac{\partial^2 w}{\partial y^2} - 2 \frac{\partial^2 F}{\partial x \partial y} \frac{\partial^2 w}{\partial x \partial y} + R$$

(1)

where $D = Eh^3/12(1 - \mu^2)$, $D$ is the bending rigidity, $h$ is the thickness of the plate. $\theta$ is the temperature gradient across the thickness $h$. $w$ is the deflection of the plate in $z$ coordinate. $\mu$ is the Poisson’s ratio. $E$ is the Young's modulus. $\alpha$ is the coefficient of thermal expansion. $F$ is the Airy stress function, $F = F_h + F_p$, composed of harmonic solution $F_h$ and particular solution $F_p$.

For the clamped plate, the deformation of the edge is zero, the curvature along coordinate axis is zero. And the stress boundary condition is

$$\frac{\partial^2 F}{\partial x^2} - \mu \frac{\partial^2 F}{\partial y^2} = 0$$

$$\frac{\partial^2 F}{\partial y^2} - \mu \frac{\partial^2 F}{\partial x^2} = 0$$

(2)

Based on the boundary conditions, the deflection for the plate can be expressed by trigonometric functions

$$w(x, y, t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} A_{mn}(t) \phi_{mn}(x, y)$$

(3)

where $A_{mn}$ is the $(m, n)$ order mode amplitude, $\phi_{mn}$ is the $(m, n)$ order mode shape.

When the structure is heated by high speed flow, it will perform expansive deformation with thermal stress and thermal moment, resulting in thermal buckling. The temperature corresponding to thermal buckling state is called the critical buckling temperature. In this paper, the bifurcation of the thermal-buckling is mainly investigated, which is the quasi-static process. It’s assumed that the temperature distribution of the plate is uniform and the plate is subjected to no other applied pressure, and the equation (1) is simplified to be

$$DV^4w + D\alpha (1 + \mu) \nabla^2 \theta = \frac{\partial^4 F}{\partial x^4} \frac{\partial^2 w}{\partial x^2} + \frac{\partial^4 F}{\partial y^2} \frac{\partial^2 w}{\partial y^2} - 2 \frac{\partial^2 F}{\partial x \partial y} \frac{\partial^2 w}{\partial x \partial y}$$

(4)

The first order critical temperature of clamped plate $(m = 1, n = 1)$ can be determined as

$$T_c(1, 1) = \frac{\pi^2 h^2 (\beta^4 + 2\beta^2 / 3 + 1)}{3\alpha b^2 (1 + \mu)(1 + \beta^2)}$$

(5)

The nonlinear large deflection equation under thermal-acoustic loadings is treated with Galerkin’s method, which employs the trigonometric functions as weight functions to integral the governing equation, to obtain the Duffing equation

$$\ddot{q} + 2\alpha_0 \xi \dot{q} + \omega_0^2 (1 - S)q + kq^3 = f + f_0$$

(6)

where $\omega_0$ is the natural frequency. $k$ denotes the nonlinear stiffening coefficient arising from the large deformation. $S$ is the ratio of the panel temperature to its buckling temperature. $f$ represents the acoustic excitation. $f_0$ models the effects of the thermal moments, $q$ denotes single-mode displacement of the plate.
Due to thermal-buckling amplitude is static amplitude, for the simply supported plate and clamped plate, the differential term about the time variable in the modal equations is zero, the mode equation can be written as:

$$kQ^3 + \omega_0^2 (1-S)Q = f + f_0$$  \hfill (7)

Assuming $f$ and $f_0$ are constant, as the thermal load is respectively weak, $S < 1$, the solution of the equation can be determined

$$Q_1 = \left[\frac{f + f_0}{2k} + \left(\frac{f + f_0}{2k}\right)^2 + \left(\frac{\omega_0^2 (1-S)}{3k}\right)^{1/3}\right] + \left[\frac{f + f_0}{2k} - \left(\frac{f + f_0}{2k}\right)^2 - \left(\frac{\omega_0^2 (1-S)}{3k}\right)^{1/3}\right]$$ \hfill (8)

If $f$ and $f_0$ have the same value 0, the solution can be obtained, $Q_1 = 0$. And when $f$ and $f_0$ are not zero simultaneously, the real solution can be determined. When the plate is in the post-buckling region, $S > 1$, the solution is

$$Q_2 = \left[\frac{f + f_0}{2k} + \left(\frac{f + f_0}{2k}\right)^2 - \left(\frac{\omega_0^2 (S-1)}{3k}\right)^{1/3}\right] + \left[\frac{f + f_0}{2k} - \left(\frac{f + f_0}{2k}\right)^2 - \left(\frac{\omega_0^2 (S-1)}{3k}\right)^{1/3}\right]$$ \hfill (9)

4.2 Modal analysis of thin-walled structure

In order to further analyze the vibration behavior of thin-walled structures under thermal-acoustic loads, a geometry model as shown in Figure 4 is chosen for simulation. The clamped constraint is imposed on shadow part of a thin-walled structure. Because the peak value of the combustion noise spectrum mainly appear below 2000Hz, acoustic load is gaussian white noise with limited bandwidth (0-2000Hz). Thermal load is a uniform temperature field in a steady state. Then through calculating dynamic responses of the structure in pre-buckling, critical buckling and post-buckling regions, the response results of the structure are mainly extracted for analysis.

In order to clearly illustrate the response results, a buckling coefficient $S = T/T_{cr}$ indicates the temperature change trend. Here $T_{cr}$ is the critical buckling temperature. For example, (1.3, 172) represents that the buckling coefficient is 1.3 and the sound pressure level is 172 dB. The critical buckling temperature of the structure is 341.6 K, and the first order thermal-mode frequencies in different temperatures are depicted in Figure 5. In pre-buckling, the structure is in a softened region, and the fundamental frequency decreases with the temperature increases. And the fundamental frequency approaches the lowest value in critical buckling. The structure is in hardened region and the fundamental frequency gradually increase in post-buckling.

![Figure 4: Thin-walled structure.](Image 330x90 to 545x243)

![Figure 5: The modal frequency at different temperature](Image 67x90 to 309x243)
4.3 Stress analysis

Time-domain stress results of the structure under different thermal-acoustic loads are extracted as depicted in Figure 6 - 9. It can be seen that with the increase of buckling coefficient in pre-buckling region, the nonlinear response increases. In critical buckling region, the non-linearity of stress response is obvious. And in post-buckling region $S=1.3$, the stress response shows a persistent snap-through motion. When $S=1.6$, thermal-acoustic load is comparable. And the stress response shows intermittent snap-through motion. When $S=1.8$, thermal load plays a leading role, and the stress response appears as a linear random vibration around a new balanced position in post-buckling region.

Figure 6: The time history of stress (0, 172)

Figure 7: The time history of stress (1.3, 172)

Figure 8: The time history of stress (1.6, 172)

Figure 9: The time history of stress (1.8, 172)

5. Conclusion

In this paper, an acoustic experiment on a combustion is implemented. The analysis results show that the noise spectrum distribution in the combustion chamber is basically the same under different working conditions of the aeroengine. The peak value of the noise spectrum mainly appears in the low frequency band below 2000Hz. With the increase of rotation speed, the noise spectrum presents characteristic frequency peak in the frequency range from 655 Hz to 870 Hz. The characteristic frequencies shift to the right of the spectrum as the rotation speed increases. When the rotating speed reaches to 96% and higher, the characteristic frequencies appear in the range from 5305 Hz to 5475 Hz, and which shift to the left of the spectrum as the rotation speed increases.

A coupled FEM/BEM method is used to calculate dynamic responses of the thin-wall rectangular plate with four edges clamped. The acoustic load in the simulation is chosen according to the
experiment result of the combustion noise. In pre-buckling, the structure is in a softened region, and the fundamental frequency decreases with the temperature increases. And the fundamental frequency approaches the lowest value in critical buckling. The structure is in hardened region and the fundamental frequency gradually increase in post-buckling.

Time-domain stress results of the structure under different thermal-acoustic loads are calculated. With the increase of buckling coefficient in pre-buckling region, the nonlinear response of the plate vibration increases. In critical buckling region, the non-linearity of stress response is obvious. And in post-buckling region, the stress response shows a persistent snap-though motion. The stress response shows intermittent snap-though motion in S=1.6. And the stress response appears as a linear random vibration around a new balanced position in post-buckling region. All these analysis can contribute greatly to the thorough understanding of thermal-acoustic response characteristics of thin-walled structures and the anti-acoustic fatigue design in aeroengine.

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