MATERIAL DAMPING TEST OF HIGH TEMPERATURE ALLOY

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Abstract

This study presented a high temperature material damping test method and developed an experiment apparatus. A resonance dwell technique using a double-reed specimen was proposed to test the material damping of high temperature alloy. This test system consists of five subsystems including heating oven and heating control system, vacuum control system, temperature monitoring system, water cooling system and vibration signal measurement system. Using this testing system, the material damping of Ti-6-4 alloy was tested, and the relations between material damping and temperature or frequency were obtained. Test results shows that Ti-6-4 damping increases with temperature, but it is very nonlinear dependence; Ti-6-4 damping ratio keeps nearly constant from room temperature to 600 F°, and then rapidly increase at higher temps above 600 F°; very slight dependence of damping for Ti-6-4 on frequency was observed.

1 Introduction

Material damping is a common topic in material engineering and structure engineering, especially in the field of structural dynamics. The mechanism of material damping is rather complicated and nearly it can not be described by mathematical formulations. All the material damping data used in engineering are obtained from experiments [1,2]. Usually, material damping varies with its environmental temperature and operation frequency, sometimes the effect of strain needs to be considered. Many efforts have been made by researchers in this field for decades. It found that according to the working condition and the problem to be solved, literatures about material damping experiments mainly focus on the research of testing technique and effects of various factors on material damping.

Modern measurement techniques of damping can be categorized to three ranks: low frequency technique (<200Hz), intermediate frequency technique (<20kHz), high frequency technique (>20kHz).

The most popular low frequency damping test apparatus are the inverted torsional pendulum [3] and the dynamic mechanical thermal analyzer (DMTA)[4]. The test frequency of the inverted torsion pendulum apparatus is less than 20 Hz. The advantages of DMTA is convenient to study the influence of temperature, frequency and strain amplitude on the damping factor of a material; and the accuracy of damping test is high with the damping resolution of 10^{-5}. Its disadvantage is that the test frequency of apparatus is less than 300 Hz, moreover, when the apparatus is used to measure the damping of metal material, the controllable strain amplitude will be a very small value because of high stiffness.

The intermediate frequency damping test technique(<20kHz) are defined as the techniques that use a resonance apparatus to measure damping, and typical apparatus include the bending resonance experiment set-up and axial resonance experiment set-up. The suitable frequency scope of bending resonance apparatus[5] is from 50Hz to 5000Hz, and the suitable frequency scope of axial resonance apparatus[6] is from 5 kHz to 30 kHz. The bending resonance experiment set-up is the most popular apparatus for measuring vibration-damping properties. Existing bending resonance method fall into two categories: the cantilevered bending resonance apparatus and the free-free bending resonance apparatus. For the cantilevered bending resonance apparatus, The vibration of fixtures is inevitable, resulting in a loss of mechanical energy that manifests itself as a positive systematic error in the measured value of damping. Free-free bending resonance apparatus are known to have low parasitic losses.

Ultrasonic composite oscillator technique[7] is a common technique for measuring mechanical damping of solids in frequency range of approximately 20 kHz~200 kHz. The technique depends on mechanically vibrating the solid in the longitudinal or torsion mode by a piezoelectric crystal or magneto-strictive transducer. The main disadvantage of these apparatus is difficult to add a temperature chamber around the specimen. Moreover, the strain amplitude excited by this apparatus is small, and it is unsuitable to measure the damping of the strain dependent materials.

This study presented a high temperature material damping test method to measure the damping of turbo machinery materials in high temperature environment. This method can quantitative analysis the effect of vibration frequency (50~2500Hz), temperature and strain level on the material damping.

2 Test Method

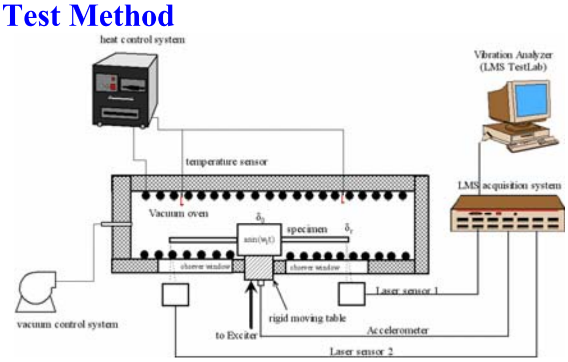

Fig 1 Material damping measuring scheme basing on a resonance dwell technique

A resonance dwell technique for a double-reed cantilever beam (see fig 1) was presented to test material damping. The double-reed cantilever specimen can eliminate the effect of rocking motion and joint damping. An electromagnetic exciter is used to sinusoidal drive the center rigid block of specimen at first or second resonant frequency of the specimen.

Assume the center block is driven sinusoidally with acceleration amplitude a_0 at a resonant frequency of the specimen.
Specimen damping coefficient:

$$\eta_s = \frac{1}{Q} \tag{1}$$

$$Q = \frac{\delta_{res}(x,t)}{\delta_0(x)} \tag{2}$$

Where, Q is amplification factor and δ_0 is the deflection produced by the statically distributed exciting forces proportional to the inertia forces of the mode, which will be obtained by the theoretical calculation. δ_{res} is the deflection produced by the same pattern of harmonic force at the natural frequency, which can be measured.

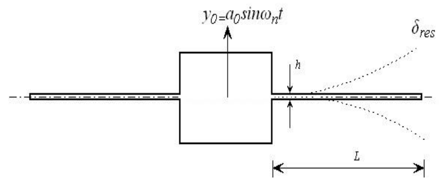

Fig 2 Double reed cantilever sample excited by base motion

Now, consider the single reed shown in Figure 2, we will derive explicit relations for δ_0. Let $v(x,t)$ be the relative transverse deflection of the reed beam, and the differential equation is

$$EI\frac{\partial^4 v}{\partial x^4} + \rho bh\frac{\partial^2 v}{\partial t^2} = 0 \tag{3}$$

where, EI is the flexural modulus, ρ is the mass density, b is reed width, and h is the thickness of the reed.

The natural modes of the single reed can be expressed as

$$\phi_m(x) = \sin\lambda_m x + D_m\cos\lambda_m x + E_m\sinh\lambda_m x + F_m\cosh\lambda_m x \tag{4}$$

where λ_m, D_m, E_m, F_m have been listed in Tab1,and the corresponding natural frequency is given by

Tab1. Value of λ_m, D_m, E_m, F_m

m	λ_m	D_m	E_m	F_m
1	$\frac{1.875}{L}$	-1.3622 -	1	1.3622
2	$\frac{4.694}{L}$	-0.9819 -	1	0.9819

Now, we consider the static deflection of the beam when the base is steadily accelerated with the constant acceleration a_0 . The inertia force per unit length is ρbha_0 , and the governing equation for the deflection $v(x)$ is

$$EI\frac{\partial^4 v}{\partial x^4} = \rho bha_0 \qquad (6)$$

This deflection may be considered as a superposition of all the natural modes

$$v(x) = \sum C_m \phi_m(x) \qquad (7)$$

In order to measure the amplification factor at the resonance of mode ϕ_1 , we need to know the magnitude of c_1 in Equation 7. This is obtained by inserting 7 into 6, multiplying both sides by $\phi_1(x)$, and integrating over the length of the beam.

$$\begin{aligned} &EI\int_0^L \sum C_m \frac{d^4\phi_m(x)}{dx^4}\phi_1(x)dx \\ &= EI\int_0^L \sum C_m(\lambda_m^4\phi_m(x))\phi_1(x)dx = \int_0^L \rho bha_0\phi_1(x)dx \qquad (8) \\ &= \int_0^L \rho bha_0\phi_1(x)dx \end{aligned}$$

Using the orthogonality of the modes, we know $\int_0^L \phi_m(x)\phi_n(x)dx = 0$, when $m \neq n$. So the integrating equation 8can be simplified as

$$C_1\int_0^L EI\lambda_1^4(\phi_1(x))^2 dx = \int_0^L \rho bha_0\phi_1(x)dx \qquad (9)$$

From equation (9), we can get

$$C_1 = \frac{\int_0^L \rho bha_0\phi_1(x)dx}{\int_0^L EI\lambda_1^4(\phi_1(x))^2 dx} \qquad (10)$$

The first-mode tip deflection under this steady state loading is

$$\delta_{10} = C_1\phi_1(L) \qquad (11)$$

Similar, the second-mode tip deflection under steady loading is

$$\delta_{20} = C_2\phi_2(L) \qquad (12)$$

$$C_2 = \frac{\int_0^L \rho bha_0\phi_2(x)dx}{\int_0^L EI\lambda_2^4(\phi_2(x))^2 dx} \qquad (13)$$

Then, according cantilever beam vibration theory, the modal mass is defined as

$$M_m = \int_0^L \rho bh(\phi_m(x))^2 dx \qquad (14)$$

and modal stiffness is

$$K_m = \int_0^L EI(\phi_m(x))^2 \lambda_m^4 dx = \omega_m^2 M_m \qquad (15$$

Substitute (15) and (14) to (10) and (13), we can get

$$\delta_{10} = C_1\phi_1(L) = 1.566\frac{a_0}{\omega_1^2} \qquad (16)$$

$$\delta_{20} = C_2\phi_2(L) = 0.868\frac{a_0}{\omega_2^2} \qquad (17)$$

Now , the material damping can be expressed as:

$$\eta_1 = \frac{1.566a_0/\omega_1^2}{|\delta_{res}(L,t)|} \qquad (18)$$

when exciting frequency is equal to first natural frequency ω_1 .

$$\eta_2 = \frac{0.868a_0/\omega_2^2}{|\delta_{res}(L,t)|} \qquad (19)$$

when exciting frequency is equal to second natural frequency ω_2

Where $|\delta_{res}(L,t$ is the tip displacement amplitude of specimen which is measured by a laser displacement sensor, and a_0 is the acceleration amplitude of center block which is measured by a high temperature accelerometer installed on the bottom surface of center block of specimen. The application of base exciting and laser sensors can avoid the limit of high temperature. Damping measurement is arranged to perform in a vacuum oven, so the effect of air damping also can be eliminated.

3 High temperature material damping measurement system

A high temperature material damping test setup was designed and manufactured to perform damping measuring experiments for high temperature alloy (see Fig. 2). This system consists of five subsystems including heating oven and heating control system, vacuum control system, temperature monitoring system, water cooling system and damping measurement system.

In this test scheme, vacuum oven can be divided into two vacuum chambers. The upper vacuum chamber is used to install a test specimen, and an electric coil furnace around the test specimen is used to heat the reed beam. Two thermocouples can monitor the temperature of oven chamber and transfers the temperature signal to heating control system. The under vacuum chamber is a room temperature vacuum chamber which is used to install the exciter. The test specimen is fixed on a rigid ceramic table by four blots, and the rigid ceramic table is connected to the exciter rod. In the neck of vacuum oven, a water cooling system is designed to decrease the temperature of rigid ceramic table. On the other hand, diathermancy of ceramic material also is low. So the temperature of bottom surface of ceramic table will be lower than the reeds of specimen. An accelerometer will be installed on the bottom surface of ceramic table to measure base motion amplitude.

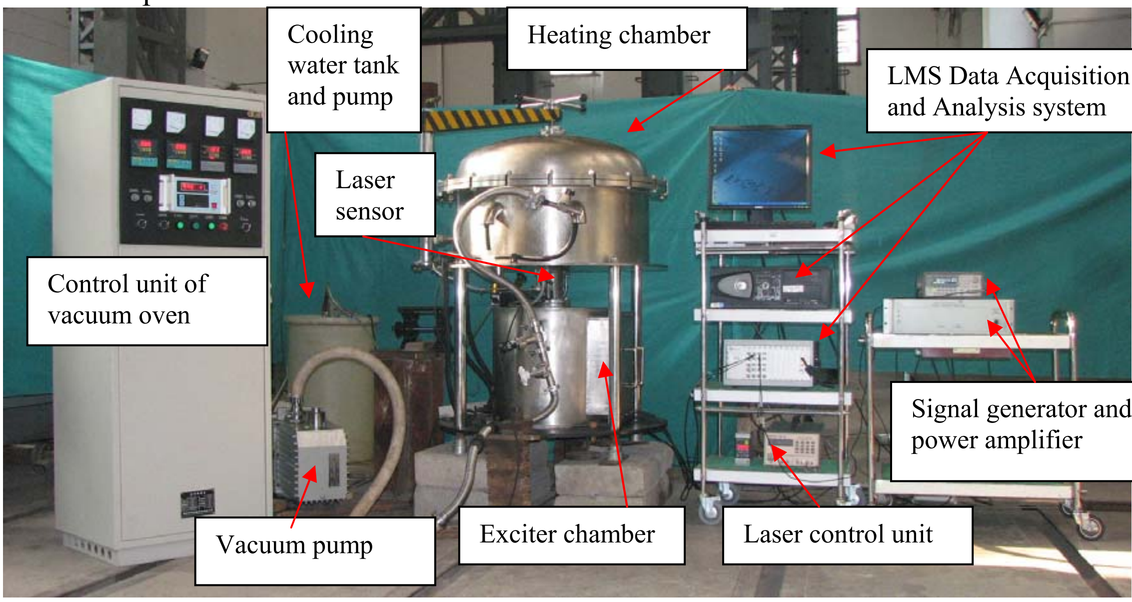

Fig 3 High temperature material damping measurement system

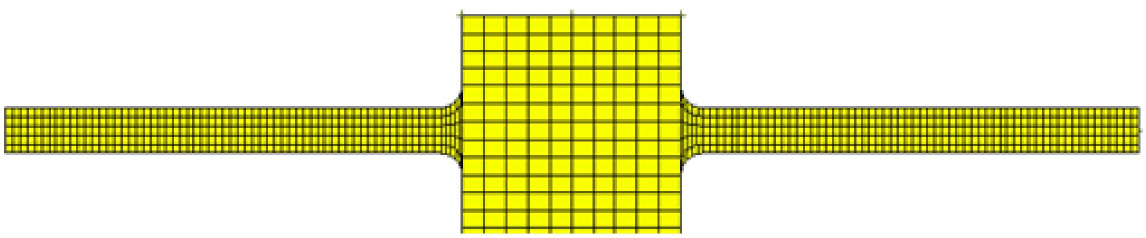

Fig.4 Finite element model of a double reed specimen

4. Numerical simulation for damping experiment of a double-reed specimen

In this section, we establish a finite element model to verify the above damping formulas with numerical simulations, and the effect of unbalanced double reeds on damping measure also has been discussed. Fig2 shows the finite element model of Ti-6-4 specimen, whose first bending natural frequency is 202.663Hz, and second bending natural frequency is 1266Hz. Two different load cases are studied by this finite element model. Modal superposition method is used to calculate the time history response of specimen. Fig 4. Finite element model of a double reed specimen

For load case 1, we assume the modal damping ratio is 0.0003, and the center block of specimen is driven sinusoidally with displacement amplitude of 0.00025mm at its fist bending resonant frequency. For load case 2, the modal damping ratio is set as 0.005, and the center block of specimen is driven sinusoidally with displacement amplitude of 0.00025mm at its second bending resonant frequency. In order to consider the effect of unbalanced double reeds on damping measure, the density of the element on the tip is changed to make two reeds have different resonance frequencies. Table 1 shows the simulation results for load case1. Table3 shows the simulation results for load case 2. These simulation results demonstrate that: .

Tab2. Simulation result of load case 1

	δ_0 /mm	**natural freq. of reed /Hz**		δ_{tip} **/mm**		**Phase difference**	$\varsigma = \frac{1}{2}\eta_s = \frac{1.566\delta_0}{\bar{\delta}_{tip}}$
		Reed 1	Reed 2	Reed 1	Reed 2		
Balance	2.5e-4	202.663	202.663	0.6412	0.6412	0	3.0e-4
Unblance1	2.5e-4	202.663	202.704	0.6023	0.5984	5%	3.26e-4
Unblance2	2.5e-4	202.663	202.763	0.6381	0.5123	10%	3.40e-4

Tab3. Simulation result of load case 2

	δ_0 /mm	**natural freq. of reed /Hz**		δ_{tip} **/mm**		**Phase difference**	$\varsigma = \frac{1}{2}\eta_s = \frac{0.868\delta_0}{\bar{\delta}_{tip}}$
		Reed 1	Reed 2	Reed 1	Reed 2		
Balance	2.5e-4	1266.065	1266.065	0.02163	0.02163	0	0.005015
Unblance1	2.5e-4	1266.065	1267.165	0.02154	0.0216	2.53%	0.005030
Unblance2	2.5e-4	1266.065	1268.265	0.02130	0.02141	5.06%	0.005082

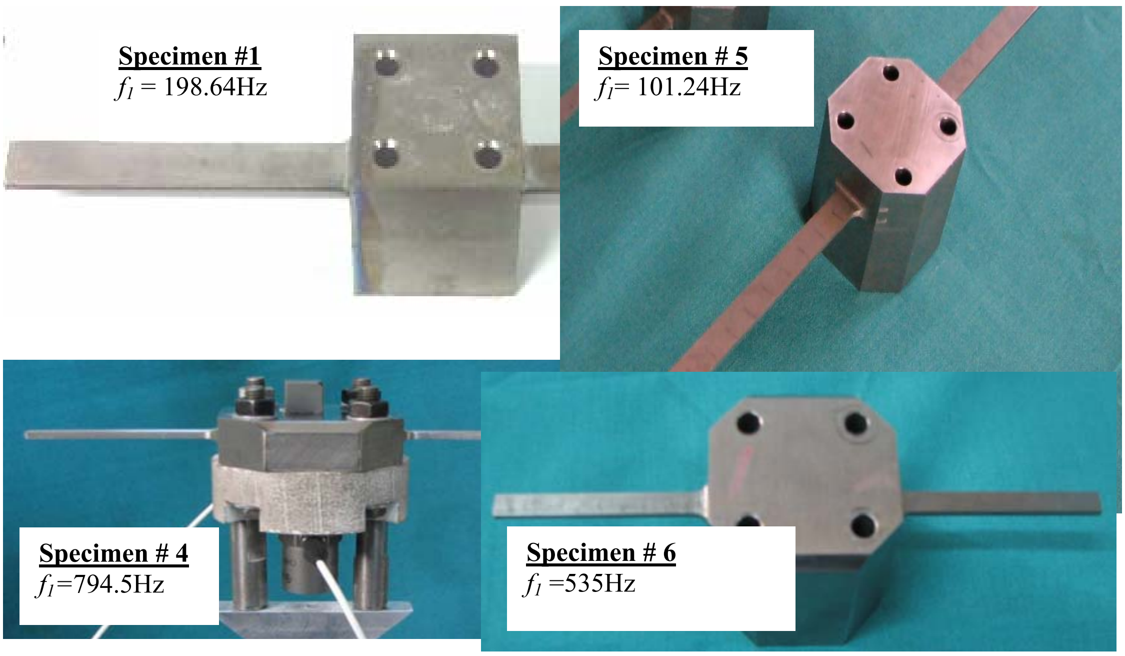

Fig 5. Ti-6-4 alloy specimens

1. Above-mentioned damping test formulas are feasible and with high accuracy.
2. When the resonance frequency and damping ratio is low, time history of tip displacement is very sensitive to the balancing status of double reeds. 0.05% resonance frequency difference between double reeds can make 13% damping "measure" error. In order to ensure the measure more precisely, the double reeds should be manually tuned to hold a precise identical vibrating behavior.
3. When damping ratio and resonance frequency is relative high, a 0.2% resonance frequency difference between double reeds just make a 1.3% damping "measure" error. That is to say, the balancing status of double reeds has weak effect on damping "measure" in this case.

5 Test Results

Using this testing system, the material damping of four Ti-6-4 alloy specimens was tested. Four specimens are shown in fig5.

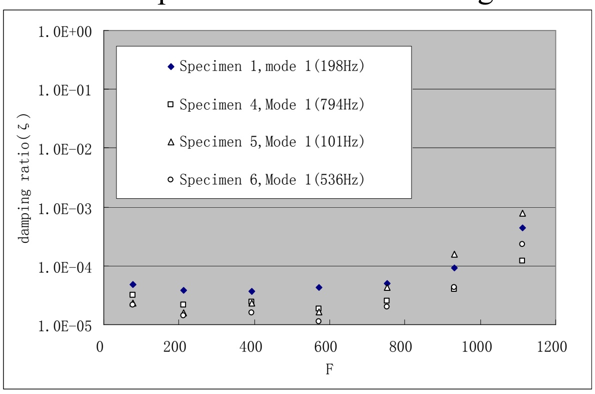

Fig. 6 Damping test result of Ti-6-4

Fig 6 shows the relations between measured material damping and temperature or frequency. Several preliminary conclusions were obtained. 1. Ti-6-4 damping increases with temperature, but it is very nonlinear dependence. 2. Ti-6-4 damping ratio keeps nearly constant from room temperature to 600 F°, and rapidly increases at higher temps above 600 F°. 3. Very slight dependence of damping on frequency was observed.

Acknowledgments

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