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Regular Articles

Theoretical and experimental study on FBG accelerometer based on multi-flexible hinge mechanism

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ABSTRACT

For vibration monitoring, the accelerometers with wider measuring range and greater sensitivity are required. In order to achieve the goal, a novel FBG accelerometer based on three mass blocks and four flexible hinges have been proposed. Then the mechanical model and dynamics equations, the resonant frequency, sensitivity and cross interferences have been investigated. It is observed that the theoretical and experimental data are consistent, the measuring range and the sensitivity of the proposed accelerometer are about 50–800 Hz and 29 pm/g, respectively.

1. Introduction

Fiber optic grating (FBG) accelerometer used in vibration monitoring consists of deformed body and FBG. FBGs are sensitive to temperature and strain at the same time [1–3], so the temperature-insensitive FBG sensors are practical [4,5]. To compensate the effect of temperature, a reference FBG is usually applied. Since FBGs are sensitive to strain, FBGs have been used in varies of sensors, such as shear force sensors [6], displacement sensors [7], and so on.

In many cases, the deformed body in FBG accelerometer is diaphragm, cantilever beam or flexible hinge. Liu reported a diaphragm type accelerometer, whose sensitivity is more than 1000 pm/g within a frequency range of 0.7–20 Hz [8]. Liu presented a double diaphragms based FBG accelerometer with the frequency range of 50–800 Hz and the sensitivity of 23.8–45.9 pm/g [9]. Tan presented a single diaphragm based FBG accelerometer with the sensitivity of 31 pm/g and frequency range of 10–150 Hz [10]. The FBG accelerometer based on cantilever beam has simple structure and stable performance, but the measuring frequency range is narrow and the FBG is easy to chirp due to the bonding of grating on the beam. Khan reported a cantilever beam based FBG accelerometer, its sensitivity is 46 pm/g as the frequency is 5–50 Hz and is 306 pm/g when the frequency is 150–220 Hz [5]. The deformation body they proposed is a beam with varying section, which looks like flexible hinge. The accelerometer based on flexible hinge has smaller size and wider measuring frequency range [11,12]. Dai reported a flexible hinge based FBG accelerometer with the sensitivity of dozen pm/g and resonant frequency of about 3000 Hz [13]. Due to smaller size, the deformation body based on flexible hinge is suitable to fabricate multi-dimension accelerometer. Antunes proposed an FBG accelerometer with the sensitivity of about 90 pm/g and resonant frequency of about 845 Hz in two orthogonal directions [14].

In this paper, we have proposed an FBG accelerometer with three mass blocks and four flexible hinges, and established the mechanical model by using general dynamics equations, and investigated the resonant frequency, sensitivity and cross interferences of proposed accelerometer.

2. Mechanical models

As shown in Fig. 1, the proposed accelerometer is mainly composed of three mass blocks, four flexible hinges and one FBG. With the action of ambient vibration, the middle block with mass m_m will vibrate in the vertical direction, and the other blocks with masses m_j will rotate around the flexible hinge center point A at the same time. The sensitive part of the accelerometer is the FBG which can turn the strain and temperature change into the central wavelength shift.

The dependence relationship of central wavelength shift upon the temperature change (ΔT) and strain (ε) can be expressed as [15]

$$\frac{\Delta\lambda_B}{\lambda_B} = (1-P_e)\varepsilon + (\alpha_\Lambda + \alpha_n)\Delta T \tag{1}$$

where P_e is the effective elasto-optical coefficient, and λ_B is the central wavelength. Besides, α_Λ and α_n present the thermal expansion coefficient and thermal-optic coefficient, respectively. Usually, the

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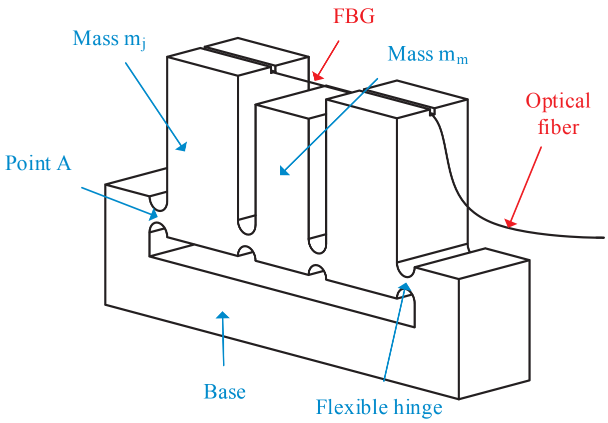

Fig. 1. Mechanism schematic of the accelerometer.

effect of temperature change can be ignored since the signal of accelerometer change very fast.

As the deformation of the flexible hinges is far greater than it of other portions, the masses can be considered as rigid bodies and the flexible hinges are not included in the rigid parts. Meanwhile, the flexible hinges can be replaced by ideal hinges with a pair of bending moments. The ideal mechanical model is shown in Fig. 2.

For the hinge, the moment M is proportional to the rotational stiffness K and the rotation angle α:

$$M = K\alpha \tag{2}$$

There are many researches on the rotational stiffness of the flexible hinges [16]. We have obtained the rotational stiffness of the proposed accelerometer, and it can be expressed as

$$K = \frac{EtR^2}{12\left[\frac{2S^3(6S^2+4S+1)}{(2S+1)(4S+1)^2} + \frac{12S^4(2S+1)}{(4S+1)^{5/2}}\arctan\sqrt{4S+1}\right]} \tag{3}$$

where R is the radius of the flexible hinges, T is the smallest height and S is defined as R/T. E is the elastic modulus and t is the thickness of the accelerometer.

FBG can be simplified as a spring in the mechanical model, and the stiffness is

$$k = \frac{A_f E_f}{l} \tag{4}$$

where A_f and E_f are the cross area and elastic modulus of optical fiber, respectively. l is the length between two bonding point of optical fiber.

2.1. Dynamics equations

To get the expressions of the resonant frequency and the sensitivity, the general equations of dynamics are needed. The D'Alembert's principle and the principle of virtual displacement, having no restraining forces in the equations, are suitable for calculating the resonant frequency and the sensitivity of the accelerometer with multi flexible hinges [17].

D'Alembert's principle clarifies that a particle or a system of particle is held in equilibrium by the motive forces, the restraining forces and the inertia forces. The equilibrium equation can be shown as

$$F_i + N_i + G_i = 0 \tag{5}$$

where F_i, N_i and G_i are the resultant forces of motive powers, restraining forces and inertial forces on particle i, respectively.

Supposing δr_i is the virtual displacement of particle i, the work done by the equilibrium forces on the virtual displacement is zero, which can be expressed as

$$\sum (F_i + N_i + G_i)\cdot\delta r_i = 0 \tag{6}$$

Progressively, supposing constrains are ideal constrains, the work done by restraining forces is zero. That is to say, the work done by motive powers and inertia forces is also zero, so Eq. (6) can be simplified as

$$\sum (F_i + G_i)\cdot\delta r_i = 0 \tag{7}$$

2.2. Resonant frequency

Resonant frequency is an inherent characteristic of a vibration system, and is related to the measuring range directly. To obtain the expression of resonant frequency, the mechanism with inertia forces and motive forces is shown in Fig. 3. Because the whole system is symmetrical, the middle mass m_m is most likely to move up and down with point C. With point A fixed, the mass m_j will rotate around the point A. Generally, the deformation of free vibration is small, so the assumption of small deformation can be used.

In Fig. 3, F_f is elastic force of FBG, G_j and G_m are inertia torque of mass m_j and inertia force of mass m_m, separately. They can be expressed as

$$F_f = 2ky_B \tag{8}$$

$$G_m = -m_m\ddot{z}_m \tag{9}$$

$$G_j = -J\ddot{\alpha} \tag{10}$$

where y_B is the displacement of the point B in y-direction. In addition, the inertia moment of mass m_j around point A and the masses m_m and m_j can be shown as

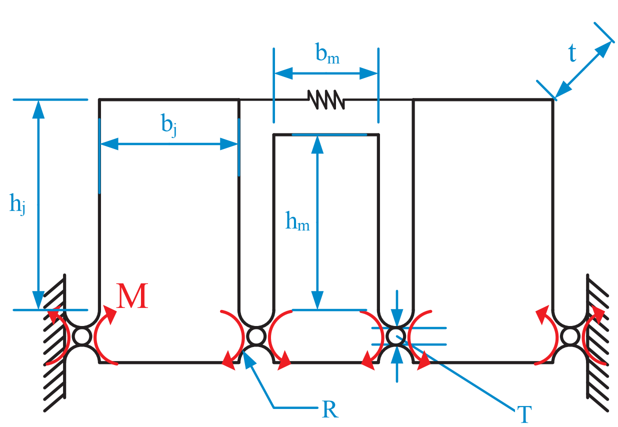

Fig. 2. Ideal mechanical model of the accelerometer.

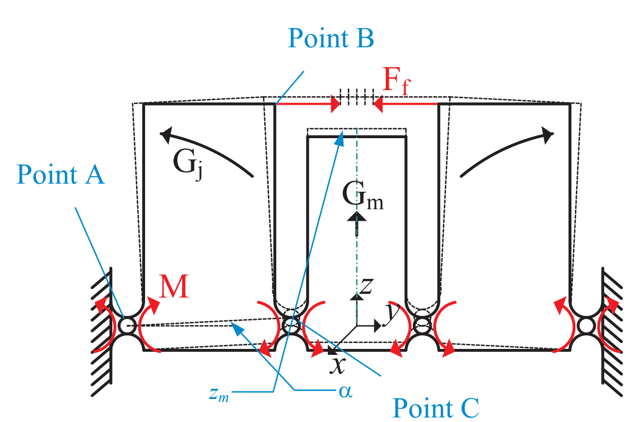

Fig. 3. Mechanical model for undamped free vibration.