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FABRICATION OF A MASS MEASUREMENT DEVICE WITH A DAMPED VIBRATION ABSORBER

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Abstract: A new mass measurement device using a damped dynamic vibration absorber is proposed and fabricated. The frequency response curves of a mass-spring system with a dynamic vibration absorber pass through a point that is independent of damping in the absorber. The proposed device utilizes this fixed point to estimate mass independently of damping. The principle of measurement was described on the basis of a mathematical model. An experimental apparatus was designed and fabricated. The fundamental properties necessary for precise measurement were certified experimentally.

Keywords: mass measurement, vibration absorber, fixed point, phase-locked loop, space technology.

1. INTRODUCTION

The completion of the International Space Station (IIS) increases the opportunities for experiments in space. The necessity for mass measurement under weightless conditions increases accordingly. Various methods of measuring mass under weightless conditions have been proposed and studied [1, 2]. The authors have proposed to use a dynamic vibration absorber simultaneously as measurement device and vibration control device for mass measurement under weightless condition [3-5]. Mass measurement systems using vibration absorber are classified into centrifugal and vibration types according to the motion of the measurement object [5]. In the centrifugal type, the object to be measured is fixed to a rotating table (rotor) at a distance from the rotational axis [3]. Because the object unbalances the rotor, centrifugal forces cause the supporting structure to vibrate during rotation. An undamped dynamic vibration absorber attached to the structure is tuned to stop the vibration. When the structure does not vibrate, the absorber vibrates in such a way that the product of the absorber mass and the amplitude of its vibration equals the amount of unbalance, that is, the product of the mass to be measured and its distance from the rotational axis. Therefore, the mass of the object can be determined from the motion of the absorber mass. In the vibration type, an object is attached to the absorber mass of the undamped dynamic vibration absorber [4]. One of the advantages of the vibration type over the centrifugal type is that it can estimate mass independently of the place at which a measurement object is fixed.

We fabricated a vibration-type measurement system with an undamped vibration absorber and carried out measurements [5]. However, inevitable damping in the absorber was thought to cause some measurement error. In addition, the automatic tuning of the absorber was not achieved.

A new method of measuring mass has been proposed to overcome these problems [6]. The frequency response curves of a mass-spring system with a dynamic vibration absorber pass through one point independent of damping in the absorber. The proposed method utilizes this fixed point to estimate mass independently of damping. In this paper, the measurement principle is described on the basis of a mathematical model. The outline of measurement system is also presented. It is characterized by the use of the phase-locked loop (PLL) to achieve the tuning condition automatically. An experimental apparatus is designed and fabricated to demonstrate the efficacy of the proposed method.

2. PRINCIPLE

A basic model of the proposed mass measurement system with a damped dynamic vibration absorber is shown in Fig.1 [6]. The primary system consists of a mass m_p and a spring k_p. A dynamic vibration absorber consisting of an absorber mass m_a, a spring k_a and a damper c_p is attached to the primary mass. When the force acting on the primary mass is represented by $f(t)$, the equations of motion are derived as

$$m_p\ddot{x}_p = -k_p x_p - k_a(x_p - x_a) - c_a(\dot{x}_p - \dot{x}_a) + f(t), \quad (1)$$

$$m_a\ddot{x}_a = -k_a(x_a - x_p) - c_a(\dot{x}_a - \dot{x}_p). \quad (2)$$

From Eqs.(1) and (2), we get

$$\frac{X_p(s)}{F(s)} = \frac{t_a(s)}{t_p(s)t_a(s) + (c_a s + k_a)m_a s^2} (\equiv G_p(s)), \quad (3)$$

$$\frac{X_a(s)}{F(s)} = \frac{c_a s + k_a}{t_p(s)t_a(s) + (c_a s + k_a)m_a s^2} (\equiv G_a(s)), \quad (4)$$

where

$$t_p(s) = m_p s^2 + k_p, \quad (5)$$

$$t_a(s) = m_a s^2 + c_a s + k_a. \quad (6)$$

To obtain the stationary vibration displacements when the force $f(t)$ is harmonic, we represent the variables as

$$f(t) = Fe^{j\omega t}, \quad x_p(t) = \overline{X}_p e^{j\omega t}, \quad x_a(t) = \overline{X}_a e^{j\omega t}. \quad (7)$$

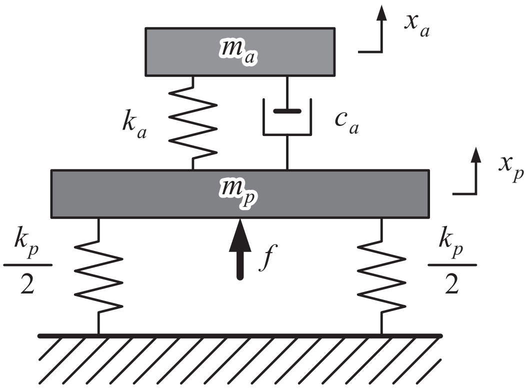

Fig.1 Basic model of mass measurement system with a damped dynamic vibration absorber

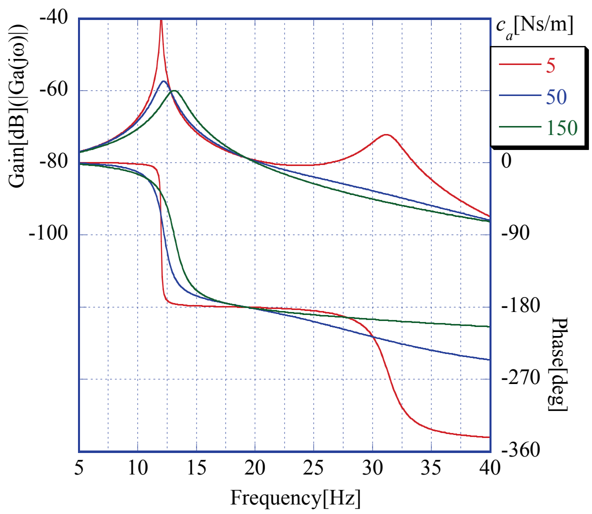

Fig.2 Calculated frequency responses of the absorber mass displacement to the applied force

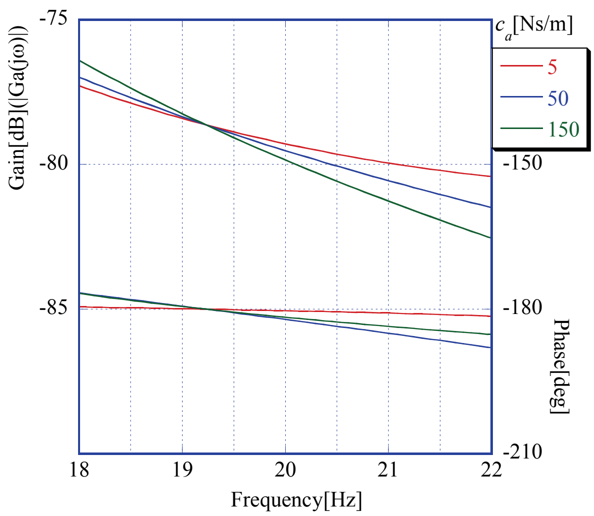

Fig.3 Extended figure near the fixed point

The complex constants (phasors) $\bar{X}_p$ and $\bar{X}_a$ are given by

$$\bar{X}_p = G_p(j\omega)F\,, \tag{8}$$

$$\bar{X}_a = G_a(j\omega)F\,. \tag{9}$$

The behaviour of the primary mass has been studied extensively to design the absorber. It is well known as Fixed Points Theorem that all the curves in the frequency response diagram pass through two points independent of damping [7]. This theorem has been utilized to optimize the absorber. In contrast, we focus on the behaviour of the absorber mass. It will be proved that all the curves pass through a point independent of damping in the frequency diagrams for the displacement of the absorber mass.

From Eqs.(4) and (9), we get

$$X_a = \frac{j\omega c_a + k_a}{t_p(j\omega)t_a(j\omega) - (j\omega c_a + k_a)m_a\omega^2}F\,. \tag{10}$$

We define the resonant frequency of the primary system as

$$\omega_p \equiv \sqrt{\frac{k_p}{m_p}}\,. \tag{11}$$

The frequency of the harmonic force is assumed to equal this frequency, that is

$$\omega = \omega_p\,. \tag{12}$$

From Eq.(4), we get

$$t_p(j\omega_p) = 0\,. \tag{13}$$

From Eqs.(10) and (13), we get

$$\begin{aligned}\bar{X}_a &= -\frac{F}{m_a\omega_p^2}\\ &= -\frac{m_p}{m_a}\cdot\frac{F}{k_p}\,.\end{aligned} \tag{14}$$

Equation (14) indicates that

- The vibration amplitude of the absorber mass is independent of the damping c_a.

It shows that all the curves pass through a point independent of damping in the absorber. In addition,

- The phase difference between the force and the displacement of the absorber mass is just 180 degrees.

This property is critical for constructing the proposed mass measurement system shown in the next section.

Figure 2 shows a plot of the vibration amplitude of the absorber mass as a function of the angular frequency for various values of the damping c_a. Figure 3 shows an expanded plot near the fixed point. The values of the parameters are selected to correspond to those of the fabricated apparatus (see Table 1). This figure shows that the fixed point exists for all the curves regardless of damping in the absorber.

When Eq.(12) is satisfied, it is derived from Eq.(14) that

$$m_a = \frac{m_p}{|\bar{X}_a|}\cdot\frac{F}{k_p}\,. \tag{15}$$

Equation (15) indicates that the mass of an object attached to the *absorber mass* can be determined by measuring the vibration amplitude of the absorber mass.

Mass can also be determined by attaching an object to the *primary mass* instead of the absorber mass because the following equation is derived from Eq.(14).

$$m_p = m_a \left| \overline{X}_a \right| \frac{k_p}{F} . \quad (16)$$

In this case, mass can also be determined by measuring the tuned frequency because the following equation is derived from Eqs.(11) and (12).

$$m_p = \frac{k_p}{\omega^2} . \quad (17)$$

3. MEASUREMENT SYSTEM

To estimate mass according to Eq.(14), the condition given by Eq.(11) have to be satisfied. To achieve the condition, the phase-locked loop (PLL) is applied in this research.

A typical PLL is a circuit that synchronizes an output signal generated by an oscillator with a reference or input signal in frequency as well as in phase [8]. It consists of three functional blocks:

- Voltage-controlled oscillator (VCO)
- Phase detector (PD)
- Loop filter (LF)

The PLL has broad industrial applications in telecommunications, electrical measurement and frequency synthesis. It is also applied to the precise control of motor speed [9], and self-sensing control of magnetic suspension [10]. However, mass measurement using a PLL has not been reported.

As mentioned above, the phase difference between the force and the displacement of the absorber mass is just 180 degrees when Eq.(11) is satisfied. The frequency of the force is tuned to realize this phase condition according to the output of a phase detector (PD), which is one of the key components of PLL.

Figure 4 shows the configuration of the proposed mass measurement system. The harmonic force is produced by an actuator with force output, which is mounted directly on the base. Such excitation was referred to as direct excitation method. In this configuration, it is assumed that no phase difference exists between the force and the output of VCO, another key component of PLL.

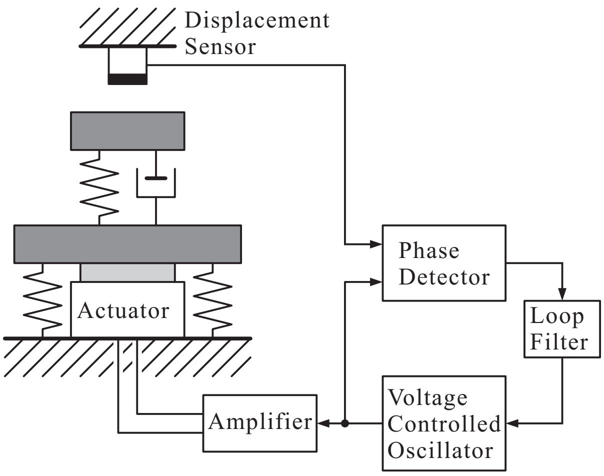

Fig.4 Measurement system

4. EXPERIMENT

Apparatus: An apparatus is fabricated for verifying the principle of measurement. Figure 5 shows a photo of the fabricated apparatus. Figure 6 shows its schematic drawing. Both the primary mass m_p and the absorber mass m_a vibrate horizontally. The T-shaped primary mass m_p is suspended and guided to move horizontally by a pair of leaf springs that are attached to the base. The absorber mass m_a is also suspended by a pair of leaf springs that are attached to the primary mass. The absorber mass is driven by a voice coil motor (VCM). This VCM is used to adjust the stiffness k_a and damping c_a by feeding back the displacement and velocity of the absorber mass. Another VCM fixed to the base produces force f acting on the primary mass. The parameters of the apparatus are shown in Table 1.

Experimental results: Figure 7 shows frequency response curves of the absorber mass displacement as a function of the angular frequency for various values of c_a.

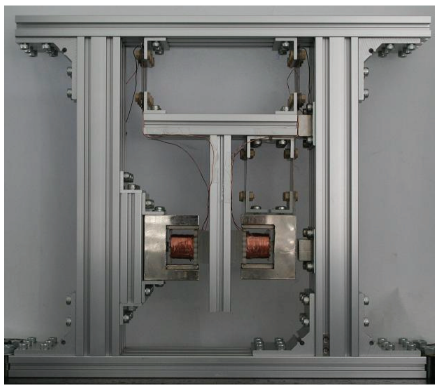

Fig.5 Photo of the fabricated apparatus

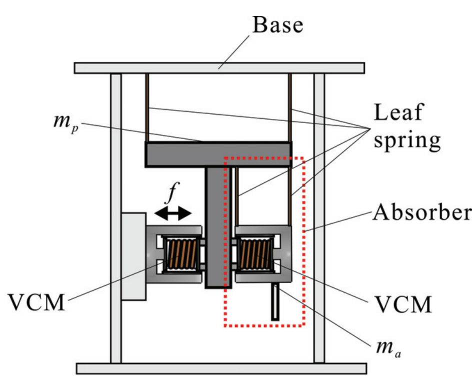

Fig.6 Schematic drawing of the apparatus

Table 1 System parameters

Primary system		Absorber	
m_p	0.599 kg	m_a	0.585 kg
k_p	8.74×10^3 N/m	k_a	8.85×10^3 N/m
		c_a	0.764 Ns/m
		ξ	0.003

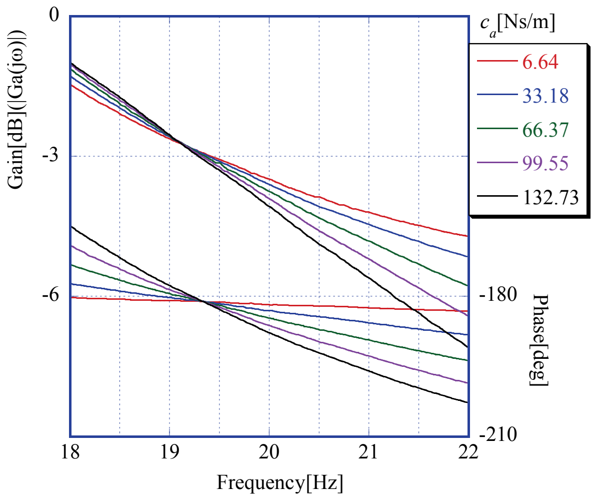

Fig.7 Measured frequency responses of the absorber mass displacement to the applied force

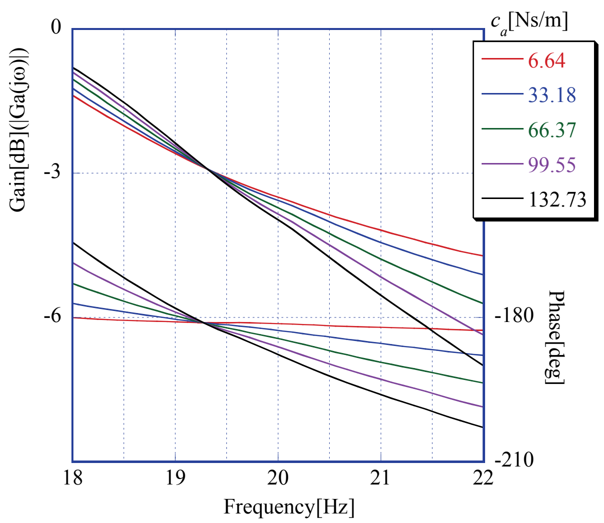

Fig.8 Measured frequency responses when damping in the primary system was reduced

Fixed points were observed. However, the fixed point of the phase curves is at 19.4Hz whereas that of the gain curves is at 19.1Hz. This difference in the fixed points is caused by the primary system damping c_p.

The primary system damping c_p was reduced as much as possible by the VCM for generating harmonic force; a feedback for the velocity of the primary mass was added. Figure 8 shows the frequency response curves of the absorber mass as a function of the angular frequency for various values of c_a. The fixed points of the gain and phase cureves appear at almost the same frequency. Therefore, the conditions for measurement have been realized, and measurement can be conducted according to the principle.

5. CONCLUSIONS

A new mass measuring device was proposed and fabricated. It is characterized by the use of a damped dynamic vibration absorber. The frequency response curves of a mass-spring system with a dynamic vibration absorber pass through a point that is independent of damping in the absorber. The proposed method utilizes this fixed point to estimate mass independently of damping. The principle of measurement was described on the basis of a mathematical model. A measurement apparatus was designed and fabricated. Frequency responses were measured to study the effect of damping in the absorber. A difference in frequency between the fixed points of the gain and the phase was observed. It was caused by damping in the primary system. Finally, the necessary condition for precise measurement was achieved by reducing damping in the primary system.

5. REFERENCES

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