Experimental Investigation of a Gyroscopic Vibration Absorber For Vibration Control of a Vertical Cantilever Beam

[F. Ünker, O. Çuvalcı]

Abstract— **Taking advantage of the angular momentum of the rotating gyroscope, gyrostabilizer systems are expected to become more widely actualized as they provide an effective means of motion control with several significant advantages for various structures. In this paper, a system of the gyroscope coupled via the massless torsional spring and the massless torsional damper mounted on the tip mass of a beam subjected to a vibrating base is considered.**

On the other hand, the effect of the angular velocity of the gyroscope is studied, and it is shown that the angular velocity (spin velocity) of a gyroscope has a significant effect on the behavior of the dynamic motion. Correctness of the numerical results is verified by experimental simulations.

Keywords— **gyrostabilizer, damping, vibration control, gyroscopic vibration absorber**

I. Introduction

The attenuation of vibration caused by dynamic effects is desired in many engineering fields. To reduce or eliminate the undesirable motions, various types of structural control theories have been developed and evolved for different dynamic loads, and quite a few of them have been potentially practically useful. Structural control methods can be classified as passive, active, and semi-active control methods according to their energy consumption (Datta T.K. 2003 and Housner et al 1997). In the last decades, different types of structural control devices have been investigated for the possibility of using active, and semiactive control methods to develop the control forces upon passive approaches for mitigating structural responses (Soong 1990; Spencer and Sain 1997; Housner et al. 1997; Kobori et al. 2003; Soong and Spencer 2002; Spencer 2002). However, the passive control method is activated by the structural motion without requiring external force or energy to reduce structural responses, and utilizes the motion of the structure at the location of the device (Soong and Dargush 1997). Active control method requires considerable amount of external power to operate actuators that supply a control force to the structure. Due to requiring sensors and evaluator / controller equipments, active control devices are more complex, and, furthermore they do not have reliability, low cost, and robustness compared to passive control techniques (Soong and Spencer 2002). On the other hand, the active control is able to adapt to structural changes and to varying loading conditions.
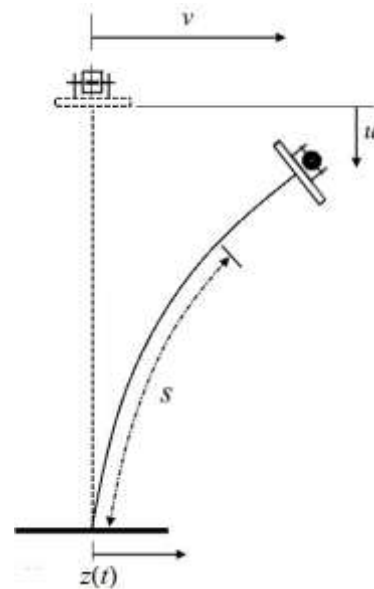

Fig. 1. Cantilever model with end mass, gyroscope, and vertical base excitation.

The gyroscopic moment induced by a rotating object offers attractive means to protect structures against natural hazards in various ways. The engineering community have developed various stabilizer systems as a means for mitigating the effects of dynamic loading on structures. The use of gyrostabilizers has emerged, as they have an ability to control vibration at low frequency, represent a weight and volume saving, and, furthermore, the stored kinetic energy can provide emergency power (Townsend et al. 2007; Thornton et al. 2005; Perez and Steinmann 2009). For roll reduction, Dr. Otto Schlick carried out such a device that has a high speed rotating disk at a constant speed and yields producing a resistive gyroscopic torque as a source of angular momentum (Schlick, 1904a, b). The mechanism does not require any other external source of energy, in which the rotor speed is produced by an electric motor in a rotating gimbal. Therefore, this can be classified as a passive control device in a variation of passive vibration control

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systems. The gyrostabilizer is effective for bending moments rather than shear forces, because the gyrostabilizer utilizes the gyroscopic moment to reduce or eliminate the undesirable motions. Consequently this paper considers the motion of a symmetric gyro with coupling to the free end of a beam subjected to a harmonic base excitation, which is supported with a torsional spring and damper. Correctness of the numerical results is verified by experimental simulations. The comparison with the results shows that the equations of motion are very accurate.

II. Passive Gyroscopic Absorber

Fig. 1 shows the beam as a vertical cantilever of length L with an end mass M_t to which an additional gyro system is attached at the free end and subjected to a harmonic base excitation at the other end of the beam. The horizontal displacement of base is z. The beam is assumed to be initially straight, of length L, with moment of inertia It, at a position L from the free end of the beam. The horizontal and vertical elastic displacements at the free end are v and u, respectively. Due to elastic deformation of the beam, s represents the distance along arc-length of the beam.

A. *Equations of Motion of the Gyro-Beam System*

Equations of the beam and gyro have been expressed in Ref. (Ünker F, and Çuvalcı O.) (see Eqs. (16) and (17)) as follows, respectively;

$$\left\{\begin{aligned} &\rho A G_1 + m + M_t + I_t G_5{}^2 + \left(\rho A G_3 + (m + M_t)G_4{}^2 + I_t G_5{}^4\right)v^2 + \frac{1}{4}I_t G_5{}^6 v^4 \\ &\qquad + \left(G_5 + \frac{1}{2}v^2\,G_5{}^3\right)^2 \left[I_o(\cos\theta)^2 + I_p(\sin\theta)^2\right] \end{aligned}\right\}\ddot{v}$$

$$+ \left[\rho A G_3 + (m + M_t)G_4{}^2 + I_t G_5{}^4 + \frac{1}{2}I_t G_5{}^6 v^2\right]v\dot{v}^2 + \left[EIG_6 - \rho AgG_9 - (m + M_t)gG_4 + 2EIG_7 v^2 + \frac{3}{4}EIG_8 v^4\right]v$$

$$+ \left(G_5 + \frac{1}{2}v^2\,G_5{}^3\right)\begin{bmatrix}(I_p - I_0)\left(\dot{v}G_5 + \frac{1}{2}v^2\dot{v}\,G_5{}^3\right)\dot{\theta}\sin 2\theta \\ + I_o(v\dot{v}^2 G_5{}^3)(\cos\theta)^2 \\ + I_p(v\dot{v}^2 G_5{}^3)(\sin\theta)^2 \\ + I_p\Omega\dot{\theta}\cos\theta\end{bmatrix} = -(\rho A G_2 + m + M_t)\ddot{z} \tag{1}$$

$$I_0\ddot{\theta} - \frac{1}{2}(I_p - I_0)\left(\dot{v}G_5 + \frac{1}{2}v^2\dot{v}\,G_5{}^3\right)^2 \sin 2\theta - I_p\Omega\left(\dot{v}G_5 + \frac{1}{2}v^2\dot{v}\,G_5{}^3\right)\cos\theta + c\dot{\theta} + k\theta = 0 \tag{2}$$

After the rearrangement, Eq.(1) becomes

$$\left\{\begin{aligned} &\alpha_1 + \alpha_2 v^2 + \alpha_3 v^4 \\ &+ \left(G_5 + \frac{1}{2}v^2\,G_5{}^3\right)^2 \left[I_o(\cos\theta)^2 + I_p(\sin\theta)^2\right]\end{aligned}\right\}\ddot{v}$$

$$+ \left\{\alpha_2 + 2\alpha_3 v^2 + G_5{}^3\left(G_5 + \frac{1}{2}v^2\,G_5{}^3\right)\left[I_o(\cos\theta)^2 + I_p(\sin\theta)^2\right]\right\}v\dot{v}^2 + [\alpha_4 + \alpha_5 v^2 + \alpha_6 v^4]v$$

$$+ \left(G_5 + \frac{1}{2}v^2\,G_5{}^3\right)\left[(I_p - I_0)\dot{v}\left(G_5 + \frac{1}{2}v^2\,G_5{}^3\right)\dot{\theta}\sin 2\theta + I_p\Omega\dot{\theta}\cos\theta\right] = -\gamma\ddot{z} \tag{3}$$

in which

B. *Gyroscopic Absorber Design*

The gyro system consists of a disk mass m, which can spin freely about its geometric axis (gimbal) mounted to the tip mass of the beam as shown in Fig. 2. The disk mass, m of gyro at free end is assumed to have angular speed Ω, and the rotation of the gimbal of this disk is also resisted by torsional spring and damping torque defined by $k\theta$ and $c\dot{\theta}$, respectively.
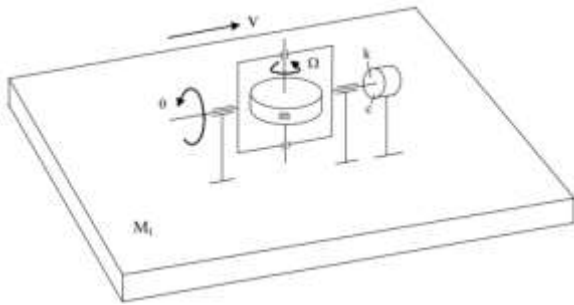
$$\begin{aligned} \alpha_1 &= \rho A G_1 + m + M_t + I_t G_5{}^2 \\ \alpha_2 &= \rho A G_3 + (m + M_t)G_4{}^2 + I_t G_5{}^4 \\ \alpha_3 &= \frac{1}{4}I_t G_5{}^6 \\ \alpha_4 &= EIG_6 - \rho AgG_9 - (m + M_t)gG_4 \\ \alpha_5 &= 2EIG_7 \\ \alpha_6 &= \frac{3}{4}EIG_8 \\ \gamma &= (\rho A G_2 + m + M_t) \end{aligned} \tag{4}$$

Fig. 2. Gyro model with end mass M_t, massless torsional spring and damper.

Table 1: Physical properties of the system

Symbol	Numerical values	Description
E	210×10^9 N/m^2	Young's modulus
h	6 mm	beam thickness
b	30 mm	beam width
L	1.17 m	length of the beam
ρ	7850 kg/ m^3	density of the beam
M_t	2.3 kg	tip mass
I_t/M_t	3.34×10^{-3} m^2	ratio of tip mass moment of inertia
I_t	7.682×10^{-3} kg.m^2	tip mass moment of inertia
I	5.4×10^{-10} m^4	geometrical moment of inertia
g	9.81 m/s^2	gravitational acceleration
A=b.h	1.8×10^{-4} m^2	area of cross section
c	0.002 N.m.s/rad	Torsion damping coefficient of gyro
k	0.04 N.m /rad	stiffness of torsion spring of gyro
c_t	0.6 N.s/m	damping onto the tip of the beam
m	0.1 kg	disk mass of gyroscope
r	0.05 m	radius of disk
$I_p=m.r^2/2$	1.875×10^{-4} kg.m^2	rotary inertia of disk
$I_0=I_p/2$	9.375×10^{-5} kg.m^2	mass moment of inertia of disk
Ω	0-15000 rpm	rotating speed of disk

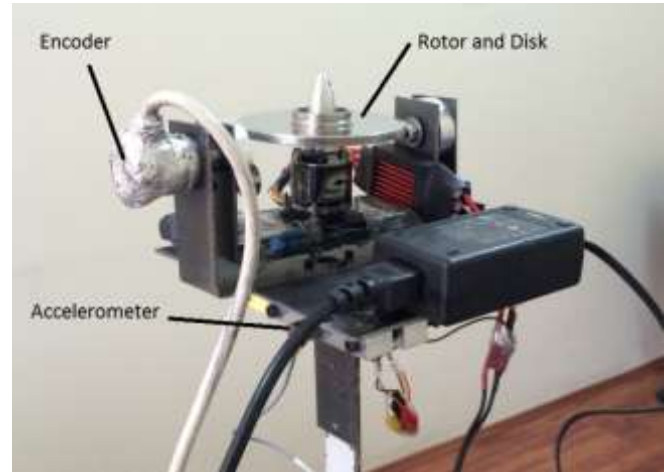Fig. 3. Experimental Gyro-Beam Model

Fig. 4. Experimental Gyroscope Model

III.   Results and Discussion

In the following calculations, a rectangle-cross beam is considered with thickness h =6 mm, width b = 30 mm, length L = 0.95 m, density ρ= 7850 kg/m3, and Young's modulus along the axial direction E=210 x10^9 N/m^2. Eqs. (1) and (2) may be solved by using a Matlab software tool that involves the fourth-order Runge-Kutta method. The parameters of the numerical example are given in Table 1. In order to identify the dynamical behavior, the frequency response is simulated, with the time step size of 0.01 s, and zero initial conditions.

A. Numerical Simulations

Figures 5 represents the frequency response curves of the beam at the free end for the different disk velocities. It demonstrates that the displacement responses of the beam can be considerably reduced when the disk speed is increased. The gyro reduces the maximum top displacement of beam by 100 % by using the disk speed of Ω=15000 rpm. As seen from the figure, the moment of inertia of gyro has the possibility to reduce the vibration by increasing rotating speed Ω of disk.
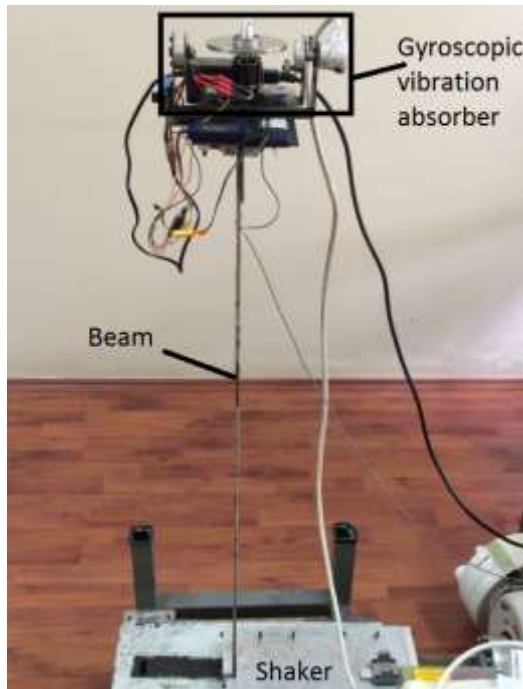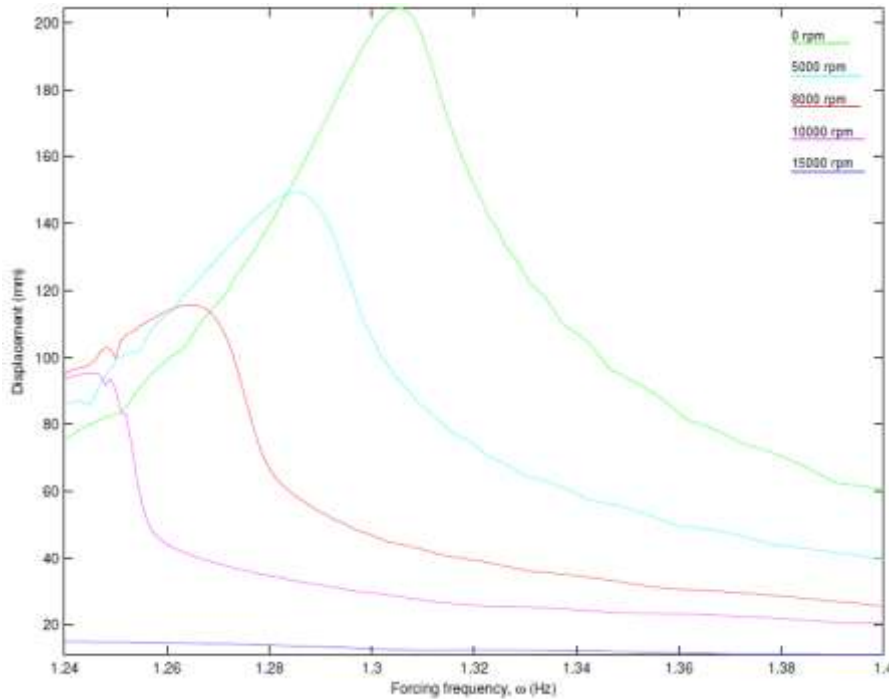
Fig.5. Theoretical frequency response curves of the tip mass for different rotational disk speeds of gyro with the base excitation of z=0.005cos(ωt). (Sweep Rate: 0.001 Hz/Second and Sweep Type : Linear, z_0=5 mm)

B. *Experimental Results*

A Brüel & Kjaer / LDS CometUSB vibration control system was used to examine the frequency responses of the cantilever beam. The controller was setted for Sweep Rate: 0.001 Hz/Second and Linear Sweep Type. Simulated frequency response curves of Figure 5 agree with measured frequency response curves of Figure 6.
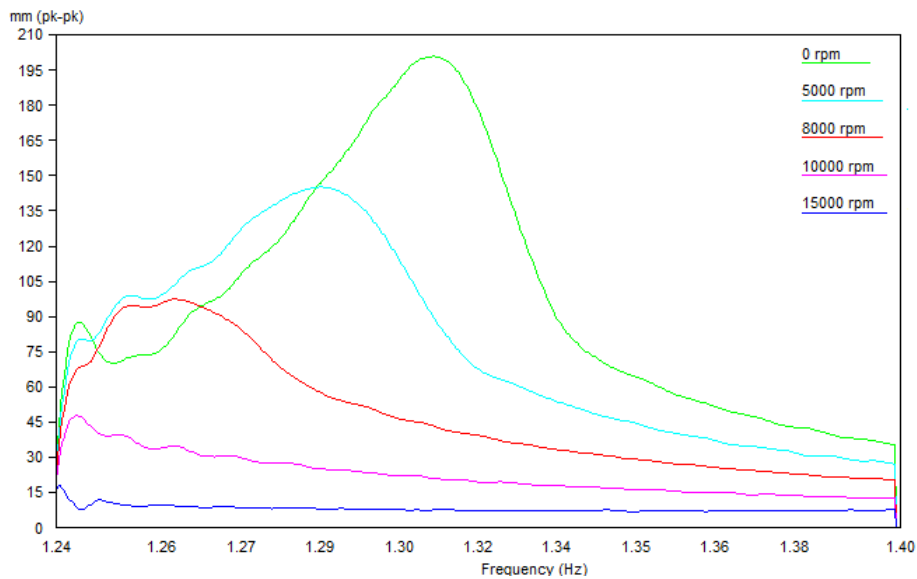

Fig.6. Experimental frequency response curves of the tip mass for different rotational disk speeds of gyro with the base excitation of z=0.005cos(ωt). (Sweep Rate: 0.001 Hz/Second and Sweep Type : Linear, z_0=5 mm)

IV. **Conclusion**

The results presented in these figures all indicate that the gyro system has significantly reduced the vibration of beam at free end by increasing rotating speed Ω of disk of gyro. The effectiveness is presented by computer simulations and the numerical results agree with the experimental solutions. From the consistency of the present study, the following conclusions is found:

- The rotary inertia of the rotor can be reduced by increasing rotor speed, which is proportion to the mass of disk. Thus, the weight and volume of the system can be reduced. The rotor speed for this gyrostabilizer is so effective that this system is more adaptable and has smaller mass than other conventional control devices with the same ability to be employed for vibration control.

Acknowledgment

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