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Application of the Electromagnetic Dynamic Vibration Absorbers to Suppress Undesired Vibrations and Harvest Energy

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Application of the Electromagnetic Dynamic Vibration Absorbers to Suppress Undesired Vibrations and Harvest Energy

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Abstract
In this study, an electromagnetic energy harvester is considered to suppress undesired vibrations. For this reason, application of a dynamic vibration absorber system, which is equipped with an electromechanical device, is studied. In doing so, a theoretical model is developed to study the electromechanical behavior of the system. Then the results, which are obtained by solving the theoretical governing equation of motion and the Simscape toolbox in MATLAB software, are experimentally validated while the vibration absorber is tuned for the first frequency of the main system. Results show the dynamic vibration absorber can effectively mitigate undesired vibrations of the main vibratory system. Furthermore, it is shown that the average angular speed of the electromechanical generator is equal to 375 rpm, which can be increase in case of the system with higher natural frequencies. Finally, it is shown that average of the harvested power by the presented system, which can be categorized as a small energy harvesting system, is equal to 0.17W.
Keywords: Experimental Study; Dynamic Vibration Absorber; Energy Harvesting; Electromagnetic

Introduction

Dynamic vibration absorbers are combination of masses, springs and dampers, which are attached to the main vibratory system, to suppress undesired vibrations. This concept is one of the first strategies to attenuate the dynamically excited mechanical systems [1]. The main aim of widespread use of these systems is due to well established design approaches seeking to find the best properties of them [2]. In the last years, the energy harvesting from the vibratory systems is an interesting subject, which is studied by many researchers. Three different methods are generally used to convert vibratory energy into useful electrical energy are electrostatic [3; 4], piezoelectric [5-8] and electromagnetic-based energy harvesting methods. Through all these three type of energy harvesting, electromagnetic method is popular for capacity of producing high power electrical energy and can be used in large scale structure.

Tang and Zuo replace energy-dissipating element in tuned mass dampers with an electromagnetic energy harvester to harvest energy and suppress vibrations of high-rise buildings, simultaneously. They used a three story building prototype with tuned mass damper including a rack and pinion mechanism and rotational brushed DC motor as an electromagnetic transducer to harvest electrical energy. Moreover, they utilized different control system and showed that self-powered active and regenerative semi-active controls can reduce vibration of the main system better than the passive one [9]. Wang *et al.* designed and tested a speed bump energy harvester (SBH) to harvest electrical energy from vehicles pass through the speed bump. They used energy conversion mechanism of mechanical motion rectifier (MMR) to overcome the impulse shape excitation. To use both downward and upward impulses, unidirectional bearing is utilized to rotate the generator in the one direction. The results showed that harvester system including MMR mechanism can harvest energy three to four times more than the harvester system without MMR. An experimental test using a passenger car demonstrated near to 1.3 KW peak electrical powers, which can be produced when only one wheel-axle of vehicle passes through the speed bump harvester at low speed. Furthermore, interaction between the vehicle and speed bump energy harvester is studied by assuming vehicle as a 4-DOF two-axle model [10].

Zhang *et al.* introduced a portable energy harvester including super capacitors, which exchanges the railroad track vibrations into electrical energy. The produced energy is kept in the super capacitors and used in safety devices or in the rail-side equipment. They also used one-way bearings to convert bidirectional transitional motion of the railroad to unidirectional rotation. Numerical results showed an efficiency of 55.5% in experimental test with excitation frequency between 1Hz and 4 Hz like the rail track vibrations [11]. Gonzalez *et al.* introduced an electromagnetic vibration absorber involving a damper with electromagnetic transducer, which works while the electrical resistance is connected to the terminals of the device. The results showed the maximum displacement of the main structure can be decreased up to 20% in comparison with a passive device, while the harvested power is more than the power losses in the coil [12]. Salvi and Giaralis presented a dynamic vibration absorber, which tuned for a low-frequency structure to reduce the vibration and harvest electrical energy. This system contains classical linear TMD, which connected to an electromagnetic energy harvester. The results showed that damping of the main structure has a great influence on the harvested energy [13]. Shen *et al.* suggested a macro scale pendulum-type electromagnetic harvester to harvest electrical energy from structures under earthquake excitations. They validated their model with a single-story steel frame model under scaled El Centro earthquake [14]. In another study, Shen *et al.*

investigated a dual functions tuned mass damper, which can control vibration and harvest energy in a tall building. The dual system includes a pendulum-type TMD, an energy-harvesting circuit, and an electromagnetic damper. The models disclose power averages from hundreds to thousands of watts harvested when the average wind speed varieties between 8 and 25 m/s [15]. Takeya *et al*. suggested a Tuned Mass Generator consist of the tuned dual-mass system with a linear electromagnetic transducer as damper to harvest energy from bridge vibration. Results showed that TMG need to have a strong design against uncertain bridge vibration [16]. Pirisi *et al*. announced a system for harvest electric energy for traffic by tubular permanent-magnet brushless linear generator. Hybrid evolutionary algorithm is utilized to optimize general effectiveness of the system and the influence on the environment and transportation systems [17].

In this study, application of an electromechanical system for suppressing vibrations and harvesting energy is studied. In doing so, in this research, to improve application of the dynamic vibration absorbers, an electromagnetic device is designed and attached to the main vibratory system.

Mathematical Modeling

Consider DC motor with an eccentric rotating mass, which is connected to the middle of a simply supported beam. Using the classical dynamic vibration absorbers, consist of a linear spring and a mass, which coupled to the energy harvesting system, forced vibration of the beam is attenuated. Schematic of the discussed system is shown in Figure 1. In this figure, M_{motor}, $m_{\text{unbalance}}$ and m are respectively the mass of the DC motor, the eccentric rotating mass and the mass of the simply supported beam. Also L and r are respectively length of the main beam and radius of the rotating mass. Furthermore, x_1 and x_2 represent displacement of main and absorber masses. Also M_2 and K_2 are equivalent mass and stiffness of the dynamic vibration absorber.

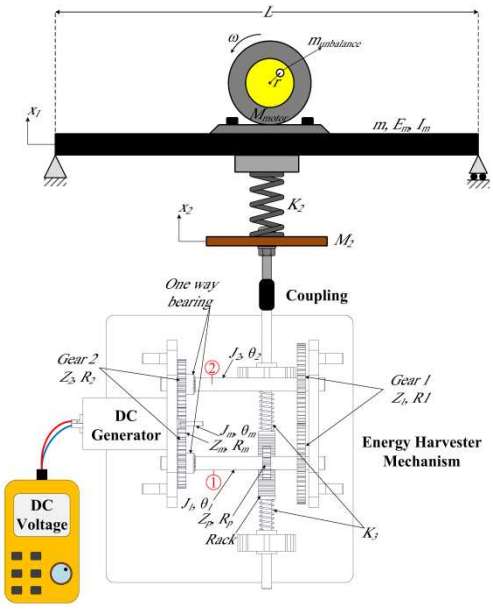


Figure 1. Schematic of the main vibratory system with electromagnetic dynamic vibration absorber

In the above figure, Z_1, Z_2, Z_p and Z_m and R_1, R_2, R_p and R_m are number of teeth and radius of the parts 1, 2, pinion and motor gears, respectively. Furthermore, J_1, J_2 and J_m and θ_1, θ_2 and θ_m are mass moment of inertia and angle of rotation of the parts 1, 2 and motor shafts, correspondingly. K_3 is equivalent spring stiffness of two same parallel springs, which helps rack moves back. The relation between angle of rotation of the shafts and downward and upward motion of the rack can be written as follows:

$$\begin{cases} \theta_P = \theta_1 = \dfrac{x_2}{R_P} \ , \ \theta_2 = -\theta_1 \ , \ \theta_M R_M = -\theta_2 R_2 \ ; \text{Downward} \\ \theta_P = \theta_1 = -\dfrac{x_2}{R_P} \ , \ \theta_M R_M = -\theta_1 R_2 \ ; \text{Upward} \end{cases} \tag{1}$$

In this system two one-way bearings are mounted between first and second shafts and gears 2, which caused unidirectional rotation in the generator shaft. Continues and more smooth unidirectional rotation are advantages of using one-way bearings in this system, which cause to harvest more electric power. Equation of motion for both downward and upward motion of the system are shown as follows:

$$M_1\ddot{x}_1 + (C_1 + C_2)\dot{x}_1 - C_2\dot{x}_2 + (K_1 + K_2)x_1 - K_2 x_2 = F \tag{2}$$

$$M_2\ddot{x}_2 - C_2\dot{x}_1 + (C_2 + C_3)\dot{x}_2 - K_2 x_1 + (K_2 + K_3)x_2 = -\frac{k_b R_2}{R_m R_p} I \tag{3}$$

$$L_a\dot{I} + R_a I = \frac{k_b R_2}{R_p R_m}\dot{x}_2 \tag{4}$$

where F, M_1 and M_2 are excitation force, equivalent mass of the main system and dynamic vibration absorber, separately. K_1, C_1, K_2 and C_2 are spring stiffness and inherent damping of the simply supported beam and dynamic vibration absorber, respectively. C_3 is the equivalent damping of the energy harvester system because of the bearings and gears used in the system. Moreover k_b and L_a are back emf constant and inductance of the DC motor and I is the output current of the harvester system. Note that this system for each upward and downward motion have a separate dynamic equation and just for this special system with the specific arrangement, these equations can be written as the discussed equations. Unlike previous studies, in this system because of the almost high vibration frequency (over than 10 HZ), the inductance of the DC motor cannot be neglected. Coefficients, which are presented in the above equations, are given in the follows relations:

$$K_1 = \frac{48 E_m I_m}{L^3} \tag{5}$$

$$M_1 = M_{motor} + 0.5m + M_{accelerometer}$$

$$K_2 = \frac{G_2 d_2^4}{8 N_2 D_2^3}$$

$$M_2 = 0.4075 + M_{accelerometer} + M_{coupling} + M_{shaft} + \frac{1}{3} M_{spring}$$

$$K_3 = 2 \times \frac{G_3 d_3^4}{8 N_3 D_3^3}$$

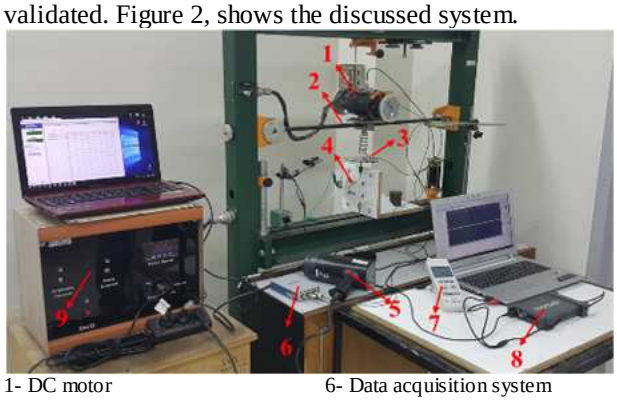
$$F = 2 m_{unbalance} r \omega^2 \sin(\omega t)$$

where E_m and I_m are the Young's modulus and area moment of inertia of the main beam. $M_{\text{accelerometer}}$,

$M_{coupling}$, M_{shaft} and M_{spring} are the masses of the accelerometer, dynamic vibration absorber and harvester coupling, the rack shaft and dynamic vibration absorber spring, respectively. G_2, G_3, N_2, N_3, d_2, d_3 and D_2, D_3 are the modulus of rigidity, number of turns of the spring's wire, diameter of the wire and average diameter of the spring of the dynamic vibration absorber, and energy harvester system, respectively.

Experimental validation

In this section the electromechanical equation of motion, which is derived in previous section, is experimentally validated. Figure 2, shows the discussed system.



1- DC motor
2- Main vibratory system
3- Dynamic vibration absorber
4- Energy harvesting system
5- Stroboscope
6- Data acquisition system
7- Portable vibrometer
8- Oscilloscope
9- Speed controller

Figure 2. Experimental test setup

Properties of the main beam, dynamic vibration absorber and energy harvester system are listed in Table 1. The main system is excited using a DC motor with an eccentric mass. The Root Mean Square (RMS) acceleration of the main mass and dynamic vibration absorber mass are measured by an accelerometer (Global Test AP2037-100). The signal acquisition is performed using National Instruments hardware with sampling rate 12.8 kHz, which is higher than necessary to ensure any higher harmonic content is considered (frequency range of interest ≤25Hz). Note that the added masses due to the sensors are considered in M_1 and M_2 coefficients. Furthermore, Lutron VB-8203 portable vibration meter is used to double check the RMS acceleration of the main system. The output electrical voltage is measured using Hantek 6022BL oscilloscope. Stroboscope is used to read the angular velocity of the generator during the test.

Results and Discussion

RMS of the acceleration of the main system versus frequency is demonstrated in Figure 3. Also, RMS of the main beam acceleration without the dynamic vibration absorber is shown in this figure. Regarding to this figure, natural frequency of the main vibratory system is equal to ω=108.6 rad/s. Furthermore, RMS of the acceleration of the vibration absorber and RMS of the output electrical voltage versus frequency are respectively shown in Figure 4 and Figure 5. Agreement between the theoretical and experimental results guarantees accuracy of the presented electromechanical equation of motion.

Table 1. Properties of the main system and electromagnetic dynamic vibration absorber

Parameters	Values
L_m (mm)	805.50
E_m (GPa)	190
I_m (mm^4)	4543.8
M_{motor} (kg)	5.6195
G (GPa)	79
M_1 (kg)	6.6876
M_2 (kg)	0.5531
$M_{unbalance}$ (kg)	0.0143
r (mm)	36
k_b (V/rad/s)	0.05178
R_1 (mm)	24.5
R_2 (mm)	15
R_m (mm)	10.5
R_p (mm)	10
Z_1	47
Z_2	28
Z_m	19
Z_p	18
K_1 (kN/m)	79.29
K_2 (N/m)	3479
K_3 (N/m)	2023

In these figures two methods, which are utilized to calculate theoretical results are shown. In the first method, the MATLAB code, which is written based on the discussed electro-mechanical equations, is used. In the other method, system is numerically modeled in Simscape part of the SIMULIK toolbox of the MATLAB Software.

Note that inherent damping of the vibration absorber and mass moment inertia of gears and shafts are neglected because the small amounts of these parameters. Effective application of the dynamic vibration absorber to suppress undesired vibrations can easily be observed in this figure.

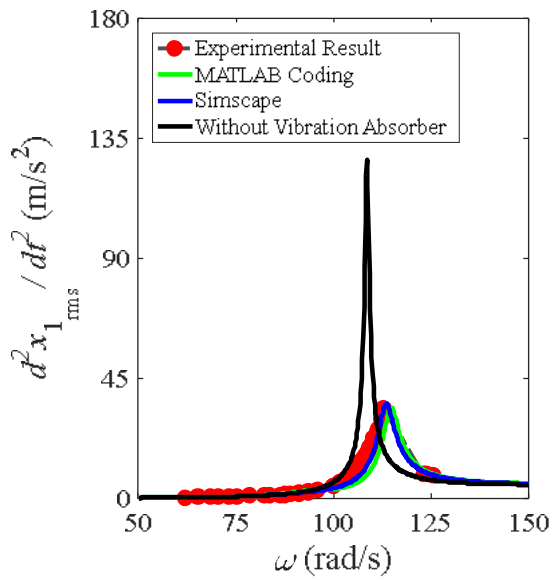


Figure 3. RMS acceleration of the main beam

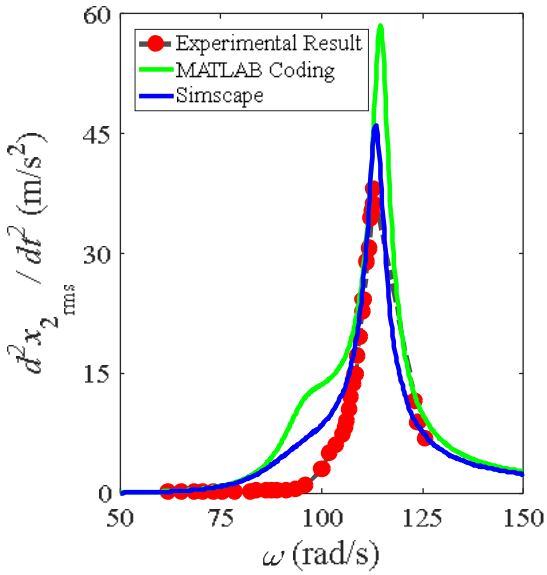


Figure 4. RMS acceleration of the dynamic vibration absorber

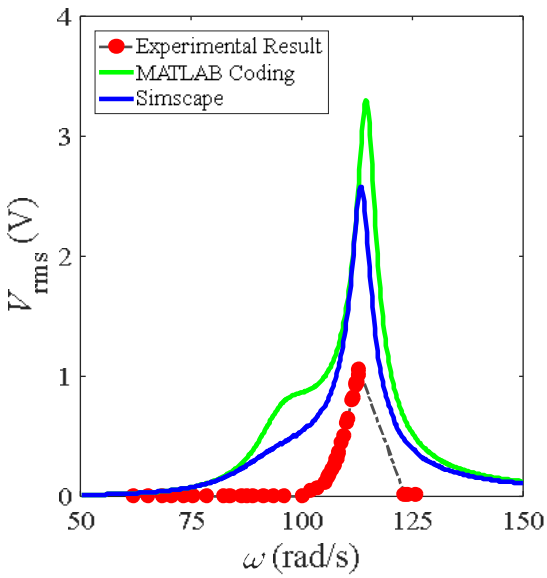


Figure 5. RMS output voltage from energy harvesting system

As it can be seen, using dynamic vibration absorber effectively decreased the RMS acceleration of the main beam and two method of solving the problem have near answer to each other. Reduction percentage of the RMS acceleration of the main beam is 90.5%. It is obvious that the second frequency peak of a two-DOF system, particularly for the experiment test and Simscape results, is very small because of the high damper coefficient of the energy harvester system. Two frequency of the two-DOF system are 94 and 115.2 rad/s.

Displacement time response of the main beam is proved in the Figure 6. The effect of using dynamic vibration absorber, in the resonance frequency of the main beam, to decrease the displacement of the beam is revealed in this figure.

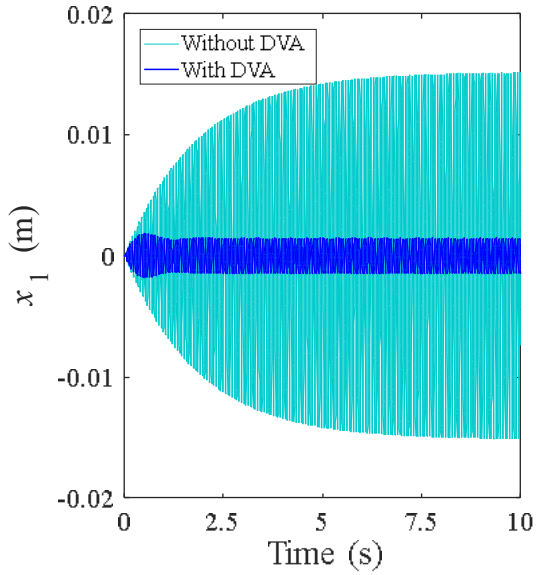


Figure 6. Displacement of the main beam with and without dynamic vibration absorber at ω=108.6 rad/s

Displacement of the dynamic vibration absorber and electrical power output in the resonance frequency are shown in Figure 7 and Figure 8, correspondingly. The maximum pick power of the harvester at the steady time is about 0.34 W. As it can be seen in the Figure 8 average electrical power is about 0.17 W and maximum angular speed of the generator is 375 RPM during this moment, according to the Figure 9. According to this figure the average angular velocity is about 220 RPM. This behavior in speed of the generator can be improved and smoothed if the mass moment of inertia of the generator is increased and then the average value of the angular velocity can be closed to the maximum angular velocity of the generator shaft.

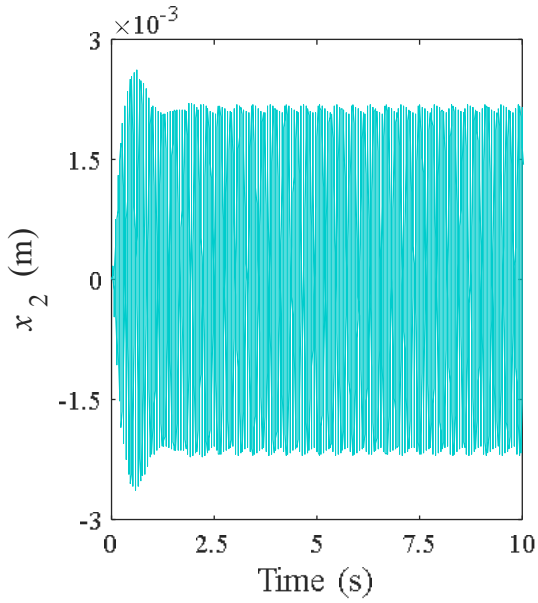


Figure 7. Displacement of the dynamic vibration absorber at ω=108.6 rad/s

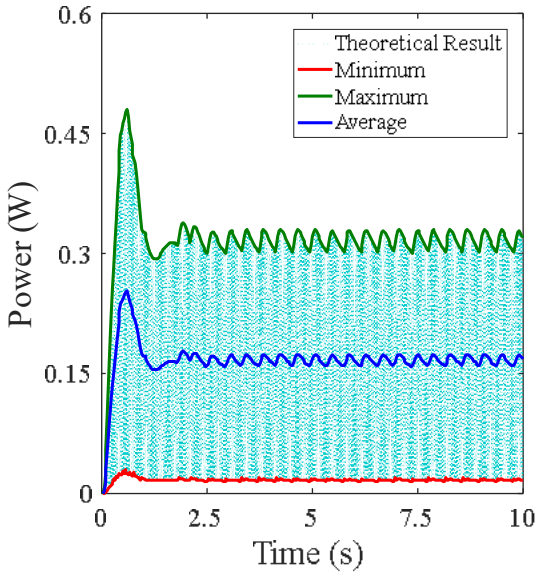


Figure 8. Electrical output power harvested at ω=108.6 rad/s

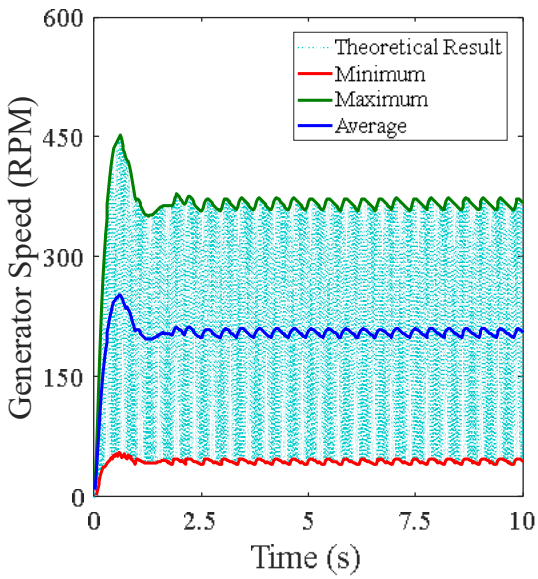


Figure 9. Angular speed of the generator at ω=108.6 rad/s

Conclusions

In the presented study, a vibration suppressor, which can convert undesired vibratory energy into useful electrical energy, is studied. Using Newtonian method electro-mechanical governing equations is derived. Two method of solving including MATLAB coding and Simscape environment is utilized to calculate theoretical results of the system. Also, governing equations are validated by an experimental test setup consist of a simply supported beam which excited by an eccentric rotating DC motor and a dynamic vibration absorber which tuned on the first frequency of the beam attached to the first system. Results have a good accuracy which shown the precision of the theoretical solving. Furthermore, results show that dynamic vibration absorber has a great effect to mitigate the undesired vibration of the main beam. Using theoretical computation, the time response of the resonance frequency of the main system and dynamic vibration absorber are calculated. Results revealed a 0.34 W maximum output electrical power which proved this system can reduce the vibration of the main system and harvest electrical energy, simultaneously.

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