Development of a Semi-Active Electromagnetic Vibration Absorber and Its Experimental Study

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In this work, a semiactive electromagnetic vibration absorber has been developed based on a proposed electromagnetic stiffness adjustable spring model, which presents a new solution for adjusting stiffness in the field of vibration absorber devices. Simulation study on the electromagnetic spring has been performed to determine the structural parameter of the semiactive vibration absorber. An experimental rig is also built up to investigate its practical vibration control effectiveness. Firstly, the finite element model of the test bench is used to analyze its vibration characteristics. Then, the vibration reduction effect is predicted through the simulation analysis, from which the optimal control positions are found. Finally, the experimental studies are also conducted, and the results show that this semiactive electromagnetic vibration absorber has a frequency adjustment range from 21 Hz to 25 Hz, in which considerable vibration reduction from 5 dB to 10 dB can be achieved. [DOI: 10.1115/1.4023952]

Keywords: electromagnetic semiactive vibration absorber, electromagnetic spring, variable stiffness, simulation, test bench

1 Introduction

Dynamic vibration absorbers (DVAs) can be classified into three categories: passive, active, and semiactive [1]. An excellent review on the research work of these three types of DVA has been made by Sun et al. [2]. Traditionally, passive DVA's are used for the vibration attenuation, while they are just effective for certain single frequency. As a result, active DVAs have been proposed by several researchers [3,4]. However, the development of this type DVA is limited by its high energy consumption and complex structure [5]. Then, semiactive DVA is attracting more and more research interest since they combine the advantages of both passive and active DVA's [6]. There are three types of semiactive DVA's, namely variable stiffness, variable damping and variable inertia DVA's.

Megahed and Abd El-Razik [1] presented the design of a variable inertia vibration absorber, as well as its dynamic modeling and simulation, for the vibration control of two degrees primary systems. Franchek et al. [7] reported a stiffness tunable vibration absorber, in which variation of absorber stiffness was achieved by varying the effective number of coils in a helical spring. Keye and Keimer [8] designed a vibration absorber with variable eigenfrequency for turboprop, frequency tuning is realized through a piezo-electric stack actuator, which applies a pressure force to a pair of leaf springs; thus reducing their effective bending stiffness. Dong et al. [9] proposed a new variable stiffness absorber based on magneto-rheological elastomer, and its vibration control performance was also studied. A virtual tuned mass damper used to suppress the self-excited vibration of maglev vehicle-girder was reported by Zhou et al. [10]. Through a theoretical study, Gao and Hao [11] developed an active vibration control scheme for controlling torsional vibration of a rotor shaft caused by electromagnetic disturbances or unsteady flow in large steam turbine generator sets. Diaz and Reynolds [12] explored the use of on-off nonlinear velocity feedback control (NLVFC) to attenuate the

human-induced floor vibrations. Nagaya et al. [13] designed a vibration absorber consisting of a flexible cantilever beam with a mass attached to its free end. Through the variation of beam length, the absorbing frequency can be changed accordingly. However, it should be noted that all of the above mentioned devices face a common problem, namely their reaction speed is too low. Then, various studies based on magnetism or electromagnetic phenomena have been performed for the vibration control. Liu and Liu [5] proposed a newly designed electromagnetic vibration absorber (EMVA), whose stiffness is on-line tunable. The EMVA is composed by a clamped-clamped aluminum beam and a permanent magnet, which is embedded at the center of the beam and placed between two poles of a C-shaped electromagnet. By varying the current of the electromagnet, stiffness of the EMVA can be adjusted instantaneously. Gospodaric et al. [14] designed an active electromagnetic damping of lateral vibration of a clamped steel cantilever. The electromagnetic system includes a switching current amplifier and a pair of electromagnets, which are mounted laterally at a right angle to the plane of the steel cantilever.

In this work, an electromagnetic semiactive vibration absorbing system is proposed, and enough detail has been described on its designing process. Experimental study is then carried out to demonstrate its practical vibration control effectiveness, and its future applications for engineering vibration attenuation are also discussed.

2 Structural Design of an Electromagnetic Semi-Active Vibration Absorbers

During the design of a variable stiffness semiactive vibration absorbing system, which can always work under the adjustable status, the concept of a controllable electromagnetic spring is introduced. The natural frequency of a dynamic vibration absorber can be modified by changing the electromagnetic spring stiffness, then it is able to track the external excitation frequency variation, and the dynamic vibration absorber can work effectively in a relatively wide frequency range.

Under the action of a magnetic field, a single small tooth has the operation characteristic line similar to the sine curve along the direction parallel with the tooth surface. As illustrated in Fig. 1,

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Fig. 1 Schematic of single tooth operation characteristics

when the relative displacement between the tooth surfaces is less than $S/2$, the relationship of electromagnetic force with the displacement is linear; when the relative displacement is larger than $S/2$, the magnetic force will decrease as the displacement increases. In this figure, δ is the air gap between small teeth, t means the distance between the two teeth, and S denotes the liner range, during which the displacement of small teeth is proportional to the generated tensile force. S generally takes the value of $t/2$. The x -axis is the displacement between the small teeth above and down, and the y -axis is the generated force parallel with the tooth surface between the small teeth.

Shown in Fig. 2 is the designed model of an electromagnetic dynamic vibration absorber based on the electromagnetic spring principle. The static magnetic field solver (2D magneto-static field solver) is employed for the magnetic circuit calculation in order to determine the required electromagnetic stiffness. The basic size parameter of the whole model is given by the design demand, such as the weight is 30 Kg, and the frequency range of vibration absorption is from 21 Hz to 25 Hz. So the range of the total stiffness need be from 473,260 N/m to 739,470 N/m at least. In this paper, the total stiffness is the sum of the electromagnetic spring stiffness and two mechanical spring stiffnesses with its value as 350,000 N/m, so the range of the electromagnetic spring stiffness need be from 123,260 N/m to 389,470 N/m.

The largest electromagnetic tensional force of the actuator can be calculated from the following simplified formula, namely

$$F = \frac{8CP(NI)^2\mu_0D}{\delta} \quad (1)$$

where C is the influence coefficient of the magnetic saturation, P is the number of tooth pairs, NI denotes the ampere-turns of the incentive coil, D denotes the diameter of the active cell, δ is the air gap distance, and μ_0 is a constant defined as $\mu_0 = 1.256 \times 10^{-8}$ H/cm.

In order to obtain much larger electromagnetic force under the same exciting current, a reasonable choice of the tooth structure size is necessary. According to the existing standards, the largest travel distance of the electromagnetic actuator is determined by the tooth space t , and the maximum amplitude L should not be larger than $t/4$, otherwise the electromagnetic actuators will not work normally. The reason underlying this requirement is that the relationship between the electromagnetic force and displacement will be nonlinear if the amplitude is larger than $t/4$. Theoretically, a smaller air gap will generate much bigger static electromagnetic tension force. Considering the processing technology and the reli-

Fig. 2 Calculation model of the designed electromagnetic dynamic vibration absorber

ability, δ is usually chosen from $t/30$ to $t/100$. According to these basic criteria of designing the vibration absorber, the parameters are finally determined as: the pairs of tooth are 3, the teeth space is 12 mm, and the air gap is 0.3 mm.

Based on the determined parameters, the model is built for the electromagnetic force simulation analysis, and the calculated results from Maxwell SV are presented in Figs. 3 and 4, which correspond to the magnetic field distribution and the magnetic induction intensity B distribution respectively, when the current is 2A and the displacement is 3 mm. It can be seen that this model has no magnetic flux leakage, and the most experienced magnetic induction is all in the tooth, so the model has been designed reasonably.

Making use of the developed model, the maximum electromagnetic forces under different exciting currents can be predicted when the maximum displacement of the active cell x is $t/4 = 3$ mm, namely the relationship is still in the linear range. The comparison of the theoretical calculation results is tabulated in Table 1. From Table 1 we can see that the range of the designed electromagnetic stiffness can meet the demand.

The structure diagram and real photo of the electromagnetic dynamic vibration absorber designed based on the simulation results are shown in Figs. 5 and 6, respectively. From Fig. 5, it

Fig. 3 Distribution of magnetic field

Fig. 4 Distribution of magnetic induction intensity B

Table 1 Comparison of the predicted electromagnetic force and stiffness with its experimental data while the displacement is 3 mm

Current (A)	0.8	1.2	2	3	4	5
Prediction force (N)	56.196	222.420	795.993	997.005	1174.074	1285.995
Experiment force (N)	83	283	651	959	1202	1316
Prediction stiffness (N/m)	18732	74140	265331	332335	391358	428665
Experiment stiffness (N/m)	27666.67	94333.33	217000	319666.7	400666.7	438666.7

1-magnet exciting coil; 2-upper end cover; 3-spring plate; 4-shaft; 5-liner bearing; 6-iron core; 7-lower cover; 8-mechanical spring

Fig. 5 Structure diagram of the electromagnetic semiactive vibration absorber

can be seen that the electromagnetic dynamic vibration absorber is composed by shells 2 and 7, magnet exciting coil 1, mechanical spring 8, liner bearing 5 and etc., while 4 and 6 are active cells, the remaining parts are stators.

The stator and active cell constitute a moving conjunction through bearing and mechanical spring, and the magnetic circuits of the stator and active cell are made of good magnetic materials (namely, 10[#] steel or electrician pure iron), which is excited by the field coil. A small gap is kept between the work surface of the stator and active cell. These two covers and bearing block are made of nonmagnetic material (aluminum alloy). The corresponding stiffness of the electromagnetic spring can be calculated using the simulation results of electromagnetic force presented in Table 1. Its comparison with that stiffness from the real measurement is illustrated in Fig. 7. It can be observed that very good agreement between these two results is achieved, and then the reliability of such an approach for electromagnetic force prediction has been validated, which will be further employed for the design of more complex electromagnetic vibration absorbers.

Fig. 6 Photo of the electromagnetic semiactive vibration absorber

Fig. 7 Comparison of stiffness from the simulation and experimental measurement

Fig. 8 System diagram of the vibration absorbing experiments

Fig. 9 Real photo of the test bench

Fig. 10 Finite element model developed using ANSYS

Table 2 The first 20 natural frequencies of the test bench

Mode	1	2	3	4	5	6	7	8	9	10
Frequency (Hz)	3.49	5.45	6.98	7.68	8.84	9.41	9.99	10.23	12.45	13.86
Mode	11	12	13	14	15	16	17	18	19	20
Frequency (Hz)	14.83	20.79	23.13	24.64	25.13	27.68	27.81	67.80	68.37	71.04

(a)

(b)

(c)

(d)

Fig. 11 Test bench mode shapes within the range of 21 Hz–25 Hz

Fig. 12 Arrangement schemes of evaluation points, excitation point and vibration absorbing positions

3 Experimental Study of Such Semi-Active Vibration Absorber

Experimental study will be performed by using the designed electromagnetic semiactive vibration absorber, whose vibration absorbing frequency is set within the range from 21 Hz to 25 Hz.

3.1 System Diagram for the Experiments of Semiactive Vibration Absorber. As shown in Fig. 8, the signal generator sends a single-frequency sine excitation signal to the excitation source, which further excites the test bench to vibration. Meanwhile, the excitation signal is also sent to the control system which calculates out the current value in correspondence with the excitation signal frequency, and sends the current signal to the vibration absorbers through the power amplifier, in which the natural frequency of the vibration absorber is the same to the excitation signal frequency; thus the vibration of the test bench can be absorbed. For a diesel engine, which generally operates at steady-state condition, its vibration can be approximated as sine excitation with harmonic frequency. Such a vibration absorbing system can be applied to the diesel engine plant, and the engine speed signal can be chosen as the reference signal, which will be fed forwardly to the control system. Natural frequency of the vibration absorber can be then controlled and adjusted according to different speeds, it means that the vibration caused by the diesel engine can be effectively suppressed by these vibration absorbers.

Fig. 13 Vibration levels of points 1 ~ 20 at different frequencies and different vibration absorbing points

3.2 Design of the Semiactive Vibration Absorber Test Bench. The test bench used in this study is composed of the channel steel, steel plate and mass block, whose real photo is presented in Fig. 9. Main geometrical parameters are 2.5 m long, 1.38 m wide, 0.294 m tall, and 0.01 m thick. The finite element model is also developed by using ANSYS, as shown in Fig. 10. Two beams are added to the test bench for suspending the exciter, which will generate the primary vibration disturbance of the test bench. To simulate the external excitation in practical application, here, the type JSK-100 vibration generator weighed 68 kg is attached to the middle beam through a linear spring with its stiffness as 3000 N/m or so.

In order to achieve a good vibration reduction of the target test bench, simulation predictions are first carried out for the modal characteristics analysis of the test bench, and the first 20 natural frequencies from the FEA calculations are listed in Table 2. It can be found that natural frequencies of the 12th, 13th, 14th, and 15th modes fall in the range of 21 Hz ~ 25 Hz, with their modal frequencies as 20.79 Hz, 23.13 Hz, 24.64 Hz and 25.13 Hz, respectively. To better understand the modal behavior of these four modes, their corresponding modal shapes are also presented in Fig. 11. It can be observed that the local modes of beams, on which the exciter is suspended, have been introduced in the 13th and 15th modes, and these factors will not be taken into account in the semiactive vibration absorbing simulation of the test bench.

3.3 The Arrangement Scheme of the Vibration Absorbers and the Simulation Results. In this work, the vibration transmission through the isolator to its foundation is of great concern, and ten positions on the lower stage corresponding to these isolators are selected as the evaluation points of the test bench vibration control effectiveness. At the same time, another ten points on the upper stage are also chosen as the evaluation positions as a reference. As illustrated in Fig. 12, the points on the upper and lower stages are respectively numbered as 1~10 and 11~20.

As for the position selection of the vibration absorber points, 8 different schemes have been considered in detail, namely (1) no vibration absorber; (2) one absorber applied at the excitation point; (3) one vibration absorber at point 3; (4) two vibration absorbers at points 3 and 8; (5) two vibration absorbers located at points 4 and 7; (6) two vibration absorbers located at points 2 and 9; (7) two vibration absorbers at points 5 and 6; (8) two vibration absorbers applied at points 1 and 10.

Simulation studies have been, respectively, conducted for these eight schemes mentioned above, and the results are given in Fig. 13. From this figure, as we can see that the best control effectiveness can always be obtained under all frequencies for the scheme that two vibration absorbers are applied at points 3 and 8.

In the real test bench, points 3 and 8 are occupied by the beam for suspending the vibration exciter, the location of the vibration absorbers are actually chosen near these two points. Here, four schemes for application of vibration absorbers around the points 3 and 8 are considered in the FEA simulation of vibration attenuation. Shown in Fig. 14 is the comparison of vibration reduction at point 3 under four different application schemes of vibration absorbers around points 3 and 8. It can be observed that the largest difference is not more than 3.3 dB, and the arrangement scheme in which the vibration absorber is located on the left side seems better than the other schemes, one possible reason for this phenomenon is that the left-side response of the opening is more considerable than that on right. Actually, this is also relevant to the test bench structure, here, the mass distribution of this test bench is extremely nonuniform and both sides are attached with big mass blocks, while the mass on the left is much larger than that of the right. Then, the right side of points 3 and 8 are the softest components in the test bench. Equipment of vibration absorbers at this part can effectively isolate the vibration transmission, and lead to the best effectiveness of vibration reduction.

Fig. 14 Vibration reduction of point 3 when vibration absorbers located at right and left sides of points 3 and 8

1. mass block; 2. acceleration sensor; 3. rubber isolator; 4.

base; 5. electromagnetic semi-active vibration absorber; 6. Excitation force.

Fig. 15 Schematic diagram of the experimental system

Fig. 16 Physical photo of the real experimental system

3.4 Experimental Effectiveness of the Vibration Absorbers. Based on the aforementioned simulation results of an electromagnetic semiactive vibration absorbing system, the installed positions of two vibration absorbers are determined as to the right of points 3 and 8 in the real test bench, and the schematic diagram and the real photos of the experimental system are, respectively, presented in Figs. 15 and 16, in which the excitation points located at the middle of the upper stage, and the two vibration absorbers are equipped at the opening on the left of the supporting beam structure.

In Fig. 17, the vibration amplitudes before and after control are measured and compared with the simulation results for all the 20 evaluation points under different disturbance frequencies. From this figure, it can be seen that although the experimental and simulation results have some difference, considerable vibration reduction can be obtained for all the evaluation points in the whole frequency band, except for a slight increase (1.855 dB) at 21 Hz for the point 20. The vibration reduction at all evaluation points

Fig. 17 Comparison of experimental and simulation results at different frequencies

Table 3 Vibration reduction observed at each evaluation point (dB)

Evaluation points	Frequencies				
	$f_1 = 21 \text{ Hz}$	$f_2 = 22 \text{ Hz}$	$f_3 = 23 \text{ Hz}$	$f_4 = 24 \text{ Hz}$	$f_5 = 25 \text{ Hz}$
1	8.87395	0.660005	9.118639	12.15676	15.5169
2	13.27611	20.78387	17.48253	15.26856	13.2782
3	17.70521	24.8404	27.02495	29.26499	27.2730
4	13.4065	10.53991	11.90088	13.46656	17.2031
5	7.981932	13.06425	13.82409	13.2524	14.1286
6	13.23049	12.29091	15.29575	14.55917	14.3810
7	19.32283	8.800496	11.30377	13.01331	17.6282
8	20.52518	21.84979	23.75107	23.93539	26.8551
9	17.33418	15.96709	20.3048	18.39318	14.8072
10	6.0206	4.165519	6.375175	10.9675	16.9622
11	11.85833	3.312517	12.0412	11.71161	11.3059
12	8.602494	10.3905	15.96709	20.172	17.5012
13	15.5025	26.44439	22.1102	17.89572	13.5753
14	11.10816	10.60462	15.65121	15.82649	12.5648
15	10.67205	20	15.56303	13.68493	11.9539
16	25.94791	18.90758	15.56302	15.61853	14.4436
17	14.09445	9.876539	15.40764	14.84626	12.2956
18	18.7218	16.85218	23.26472	19.46833	13.7684
19	14.17031	17.30603	18.21554	18.0618	17.2339
20	-1.85508	6.490222	8.222029	11.39751	11.8445

corresponding to these excitation frequencies are tabulated in Table 3. The average decrease of vibration level is 14.95 dB, and the largest attenuation can be 29.265 dB for point 3 at 24 Hz.

4 Conclusions

In this paper, an electromagnetic semiactive vibration absorbing system is designed, and simulation and experimental studies are then conducted to investigate its vibration control effectiveness. In order to design a reasonable structure of the electromagnetic semiactive vibration absorber, magnetic circuit calculation is carried out to determine the main parameters of the variable stiffness electromagnetic spring, namely the number of tooth pairs is 3, the teeth space is 12 mm, and the air gap is 0.3 mm. During the experimental study, the single frequency sine signal of 21 Hz, 22 Hz, 23 Hz, 24 Hz and 25 Hz are, respectively, chosen as the excitation inputs. The measured results show that the electromagnetic semiactive vibration absorbing system can accurately follow the variation of external disturbance in the whole exciting frequency range. For the majority of the evaluation points, good vibration reduction of 10 dB-29 dB can be achieved, while more than 5 dB vibration reduction is obtained at several excitation frequencies for some individual evaluation points, namely the total vibration level of the whole test bench and the vibration transmission through the isolators to the base can be effectively controlled.

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