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Offset Prediction of an Asymmetric Force Damper on Harmonic Base Excitation Isolation System

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Abstract

An asymmetric force damper, commonly known as a dual-rate shock absorber, is basically designed to improve ride-comfort quality of the vehicles. The damping forces for extension and compression direction are not equal. In general, the damping force for compression is much less than that for extension. This is to absorb immediate shock caused by road input during the compression stroke and to prevent the car body or isolated mass from oscillating during the extension stroke. However, the asymmetric force damper can be found in the application of the harmonic base excitation isolation systems. This is because the damper can be easily found commercially. Such the applications, the isolated mass will be shifted and oscillating about a new equilibrium position. This occurrence may cause failure to other connected component due to misalignment. This study focuses on seeking simply analytical method in order to predict the level of offset resulting from the difference of damping forces. The analytically predicted offset is compared to the that obtained using numerical simulation. The results are all in good agreement. It found that the greater in the difference of damping force as well as the higher excitation frequency cause the greater level of offset. This information can be used prior the decision in using the automotive shock absorber in the harmonically base excited vibration isolation.

Keywords: Asymmetric force damper, Offset prediction, Base excitation isolation

1. INTRODUCTION

The passive hydraulic type automotive shock absorber can be found as a part of automotive vehicle suspension system. The shock absorber connects the sprung mass to the unsprung mass. The sprung mass includes masses above the suspension whereas the unsprung mass refers to masses below the suspension as shown in Fig. 1. The suspension system is expected to provide good ride-comfort performance. By means of comfort quality, the main function of the suspension system is to isolate the sprung mass from vibration or forces caused by the unsprung mass. Ideally, the shock absorber should transmit no force to the sprung mass during the compression stroke or no resistant force by the shock absorber. This is to prevent the sprung mass from the exciting force. In contrast, for the ride quality, the shock absorber should against or slow down the motion of the sprung mass during the extension stroke. This is to prevent the sprung mass from oscillating.

These requirements result in the hydraulic type shock absorbers for the automotive vehicles working in two stages, compression and extension. These two stages provide different damping force which achieved by using different sizes of valve as the structure of shock absorber shown in Fig. 2 as described in Dixon (2007). These

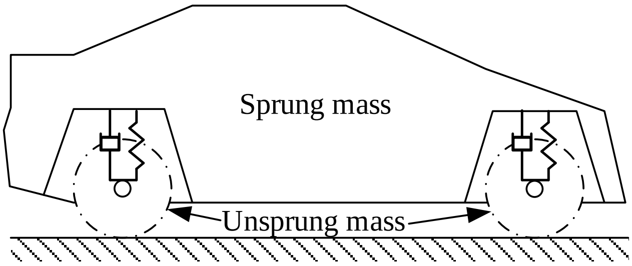


Fig. 1. Half car model with sprung mass and unsprung mass.

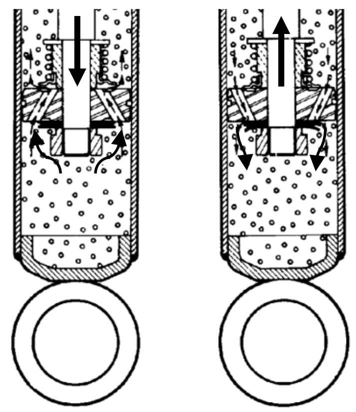


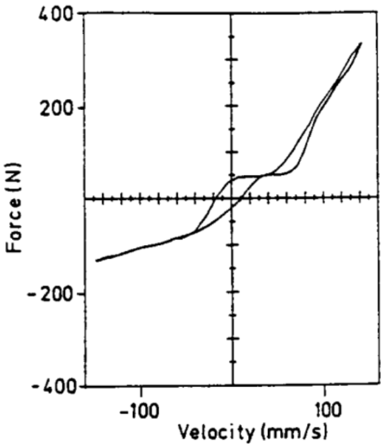
Fig. 2. Valves during compression (left) and extension (right).

valves are located in the piston. During the compression stroke, these valves should allow greater volume of fluid in the cylinder to flow than that during in the extension stroke. This characteristic is known as the asymmetric force damper.

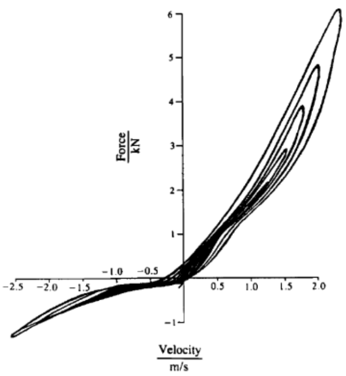
There are numbers of publications presenting the test results of the asymmetric force damper. For example, Wallaschek (1990) employed a standard twin-tube shock absorber of Ford Sierra for the test. Single frequency between 1 – 10 Hz and range of velocities from 0 – 500 mm/s were applied. The force-velocity characteristic is as shown in Fig. 3 (a).

Surace et al. (1992) also tested shock absorbers of FIAT using single frequency harmonic excitation. The force-velocity of these shock absorber appeared similar to that published in Wallaschek (1990) as shown in Fig. 3 (b). The characteristic can be considered as bilinear damping with high damping for extension and low damping for compression.

Furthermore, in many publications such as the studies of Wallaschek (1990), Rajalingham and Rakheja (2003), Calvo et al. (2009), Mahumud (2003) and Wang et al. (2007), the characteristic of automotive shock absorber has been simplified. One of considerations is simplify as an asymmetric linear piece-wise force-velocity characteristic as shown in Fig. 4. The graph shows that the damping force during the extension stroke is higher than that during the compression stroke. This characteristic of the automotive shock absorber is found to improve the ride comfort of the automotive vehicle.



(a)



(b)

Fig. 3. Asymmetric damping force-velocity characteristic

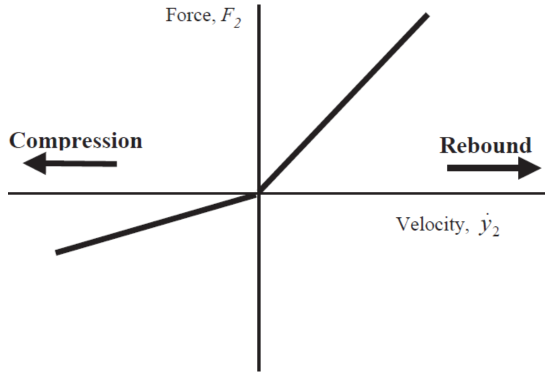


Fig. 4. Asymmetric damping force-velocity characteristic

In addition to the passive asymmetric characteristic of the automotive shock absorber, there are also publications which extend to the possibility to control the response of the damper to be asymmetric or the so-called in the paper as the dual-rate damping. For example, Verros et al. (2000), deliberately applied semiactive system to switch the damping coefficient between two values for a quarter car model under the harmonic motion. So that for the automotive shock absorber, the characteristic needs to be asymmetric.

As the automotive shock absorber can be easily found in the market. In some situations, the asymmetric force automotive shock absorber may be used in the suspension system to support the power plant engine. Such the system operates at the constant speed. As such, the force acting on the shock absorber is harmonic or periodic-like. Once the shock absorber generates the offset, it may cause damages due to occurrence of misalignment. Thus using the automotive shock absorber or the so called asymmetric force damper as the damping component for other isolating system is tricky. So that, the level of the offset should be pre-defined if it is acceptable for the system.

There are numbers of publications found that using the asymmetric force damper under the harmonic base excitation causes the isolated mass or the system mass to be shifted to the new position, for example in Rajalingham and Rakheja (2003). As the author concerns, determining the level of offset of the isolated mass for the isolation system that uses the automotive shock absorber is interesting. The report on the prediction in the offset level have not been found. Thus, in this paper, the method to predict the level of offset is proposed in the comparison to the results obtained from numerical simulation.

2. Mathematical Modelling

2.1 Linear Damping System Configuration

In this paper, the asymmetric force damper is assumed to be applied in the base excited vibration isolation system other than the automotive vehicle suspension system. The model used in this paper is as shown in Fig. 5. The model consists of an isolated mass (m), a linear stiffness (k), an asymmetric force damper (c) and an excitation base (x_0).

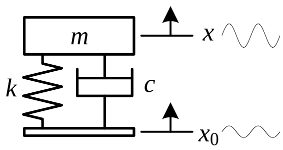


Fig. 5. Single Degree of Freedom base excited vibration isolator

Once harmonically base excitation, $x_0 = X_0 \sin(\omega t)$, is applied, motion of the isolated mass by means of displacement can be represent by $x = X \sin(\omega t + \phi)$.

The input displacement normalisation is applied for more convenient explanation. So the application yields $w = \frac{x}{X_0}$ and $w_0 = \frac{x_0}{X_0}$ with X_0 an amplitude of base excitation. The commonly known definitions of mass normalised parameters for vibration system are also applied, $\frac{c}{m} = 2\zeta\omega_n$ and $\frac{k}{m} = \omega_n^2$. The input displacement normalised equation of motion for the mass can be written as

$$w'' + 2\zeta\omega_n w' + \omega_n^2 w = 2\zeta\omega_n w_0' + \omega_n^2 w_0 \quad (1)$$

For harmonic excitation, the normalised motions are $w = W\sin(\omega t + \phi)$ and $w_0 = \sin(\omega t)$ with W a normalized amplitude of the mass. For some more convenience, the ratio between the excitation frequency and the natural frequency is defined, i.e. $\Omega = \omega/\omega_n$. The non-dimensional time is also defined as $\tau = \omega_n t$. Substitution of normalised motions, division of ω_n^2, frequency ratio and the non-dimensional time leads Eq. (1) to become

$$(1 - \Omega^2)W\sin(\Omega\tau + \phi) + 2\zeta\Omega W\cos(\Omega\tau + \phi) = 2\zeta\Omega\cos(\Omega\tau) + \sin(\Omega\tau) \quad (2)$$

In addition, equation (1) can be considered by means of relative motion, which is the difference of amplitude of motion between the isolating mass and the base excitation, i.e., $u = w - w_0$. This leads Eq. (1) to become

$$u'' + 2\zeta\omega_n u' + \omega_n^2 u = -w_0'' \quad (3)$$

By considering the harmonic response, the relative motion can be expressed as $u = U\sin(\omega t + \phi_u)$. The assumption of harmonic response yields the closed form solution for Eq. (3) as

$$U = \frac{\Omega^2}{\sqrt{(1-\Omega^2)^2 + (2\zeta\Omega)^2}} \quad (4)$$

For the linear system, the exact solution for Eq. (3) can be found in general. The closed form solution of Eq. (3) is shown in Eq. (4). For this occasion, the value of damping ratio is assumed to be underdamped, i.e., $0<\zeta<1$.

Eq. (4) can be used to predict the amplitude response of the linearly underdamped vibrating system for three regions of excitation frequency.

For the very low that the frequency ration is very much less than unity, $\Omega \ll 1$, amplitude of the relative motion in Eq. (4) can be estimated as zero, $U \approx 0$. This means that the isolating mass is moving in unison with the base excitation.

When the excitation frequency is close to the natural frequency of the system, $\Omega \approx 1$, amplitude of the relative motion is about the inverse of two time the value of damping ratio, $U = \frac{1}{2\zeta}$.

Once the excitation frequency is much higher than the natural frequency, $1 \ll \Omega$, amplitude of the relative motion becomes frequency dependence. The greater the excitation frequency results in the greater amplitude of the relative motion. This refers to the isolation region. Vibrating amplitude of the isolating mass is much less than that of the base excitation.

However, Eq.(4) does not inform effect of damper with asymmetric force. So that, for the asymmetric force damper, the solution needs to be extract in two parts for compression and extension strokes of the damper.

2.2 Asymmetric force damper configuration

The value of damping ratio for the asymmetric force damper can be considered two different linear damping ratios, i.e., compression and extension. These two values are given by means of the ratio as

$$\zeta_c = n\zeta_e \quad (5)$$

where ζ_c and ζ_e are damping ratios for compression and extension, respectively. n is the different percentage of the damping ratios, $0 \leqslant n \leqslant 1$. The definition in Eq. (5) becomes zero when there is no damping force during compression stroke. For n = 1, the damper becomes classical linear damping case.

3. Numerical Simulation

In order to see the effects of the asymmetric force damper for the harmonic excitation, numerical simulation was implemented using governing equation shown in Eq. (1). The Scilab/Xcos with RK45 solver was applied

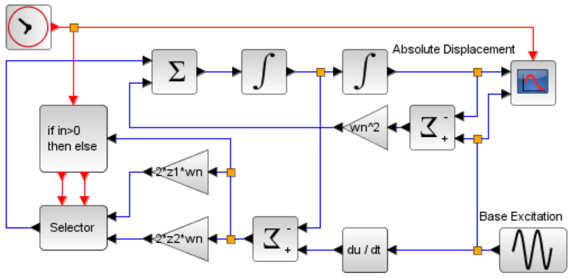


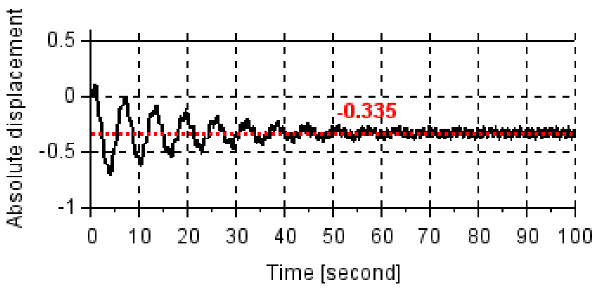
Fig. 6. Simulation model applied in Scilab/Xcos

to obtain the numerical solutions. The simulation model is as shown in Fig. 6.

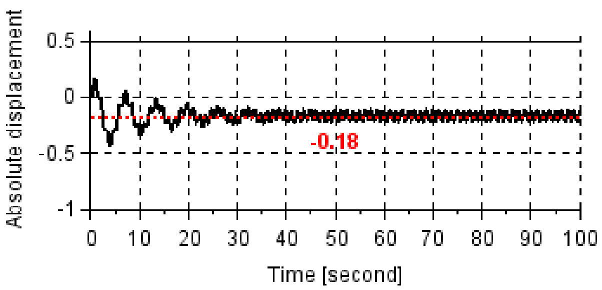
The values of damping between the compression and extension strokes were chosen dependently upon the direction of relative velocity across the damper. The sign of relative velocity indicates the motion of the damper, i.e., positive sign refers to extension and negative sign refers to compression.

The value of damping ratio for the extension stroke was set as constant with the value of ζ_e = 0.1. This is to see the effects of varying damping ratio for the compression stroke. The simulation results obtained from using ζ_e = 0.1 with n = 0 and 0.5 are shown an example in Fig. 7.

It is seen from Fig. 7 that, the greater the difference in damping ratio results in the greater level of the offset. This implies that the isolated mass is oscillating in the lower level than its equilibrium position. Occurrence of offset can be detrimental to the connecting equipment, such as, bending to the connect shaft etc.



(a)



(b)

Fig. 7. Numerical responses for ζ_e = 0.1 with n = 0 (a) and 0.5 (b)

4. Prediction of the Offset

The level of offset, caused by the different values of damping ratio during compression and extension strokes, can be predicted by considering the offset to be a new equilibrium position of the mass. Then assume that, at this position, the summation of forces acting on the isolated mass equals to zero, i.e.,

$$\Sigma F = F_s + F_d = 0 \tag{6}$$

where F_s is stiffness force and F_d is the difference of damping force. F_d is considered in two difference circumstances, i.e., damping forces during extension stroke, F_{de}, and compression stroke, F_{dc}. Thus, F_d can be defined as

$$F_d = F_{de} - F_{dc} \tag{7}$$

The levels of F_{de} and F_{dc} are respectively determined by

$$F_{de} = 2\zeta_e \Omega U_e \tag{8}$$

and

$$F_{dc} = 2\zeta_c \Omega U_c = 2n\zeta_e \Omega U_c \tag{9}$$

The definitions in Eqs. (8) and (9) lead Eq. (7) to become

$$F_d = 2\zeta_e \Omega (U_e - U_c) \tag{10}$$

where U_e and U_c are the normalized relative motion of the relative motion during extension and compression strokes, respectively.

By considering the force normalised term, stiffness force, F_s, Eq. (6) can be reduced to absolute displacement, W. Thus the resulting expression can be used to determine the absolute displacement of the isolated mass in conjunction with the application of Eq. (10), i.e.,

$$W = 2\zeta_e \Omega (nU_c - U_e) \tag{11}$$

However, this force is acting on the mass only a half of cycle. Then Eq. (11) is reformed to

$$W = \frac{2\zeta_e \Omega (nU_c - U_e)}{\pi} \tag{12}$$

Equation (12) can then be used to predict the level of offset by means of the normalised absolute position of the sprung mass.

5. Comparison and Discussions

5.1 Comparison

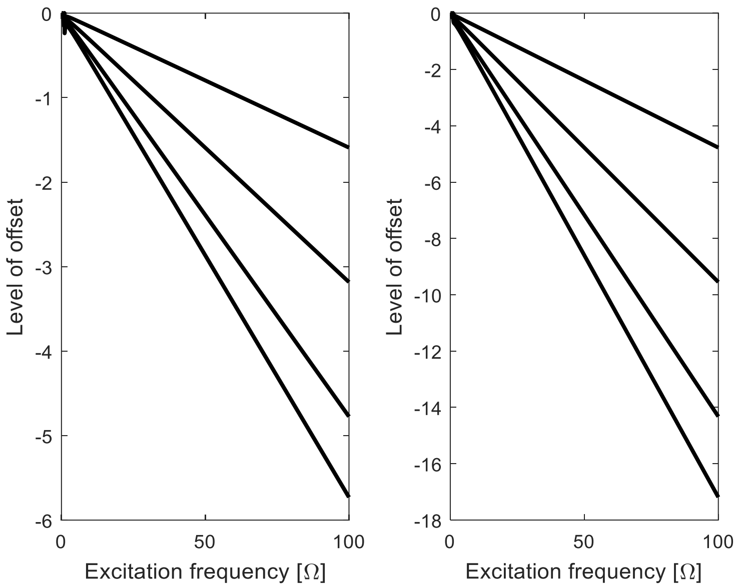
Two types of results are compared, i.e., numerical results obtained from numerical simulation (ODE) and analytical results obtained from calculation using Eq. (12). Two values of damping ratio for the extension

Table 1 Comparison of numerical simulation and prediction for ζ_e = 0.1

Ω	***n* = 0**		***n* = 0.1**		***n* = 0.25**		***n* = 0.5**		***n* = 0.75**	
	ODE	Predict	ODE	Predict	ODE	Predict	ODE	Predict	ODE	Predict
0.1	0	0	0	0	0	0	0	0	0	0
0.3	-0.0018	-0.0019	-0.0016	-0.0017	-0.0013	-0.0014	-0.001	-0.001	0	0
0.5	-0.011	-0.011	-0.01	-0.009	-0.008	-0.008	-0.005	-0.005	-0.003	-0.003
1.0	-0.631	∞	-0.516	0	-0.378	0	-0.211	0	-0.091	0
1.5	-0.169	-0.167	-0.153	-0.150	-0.127	-0.124	-0.084	-0.082	-0.042	-0.040
5.0	-0.335	-0.331	-0.296	-0.298	-0.246	-0.248	-0.180	-0.166	-0.083	-0.083
10	-0.642	-0.643	-0.577	-0.578	-0.452	-0.482	-0.319	-0.321	-0.160	-0.161

Table 2 Comparison of numerical simulation and prediction for ζ_e = 0.3

Ω	***n* = 0**		***n* = 0.1**		***n* = 0.25**		***n* = 0.5**		***n* = 0.75**	
	ODE	Predict	ODE	Predict	ODE	Predict	ODE	Predict	ODE	Predict
0.1	0	0	0	0	0	0	0	0	0	0
0.3	-0.0053	-0.0056	-0.0048	-0.0050	-0.0039	-0.0041	-0.0026	-0.0027	-0.0013	-0.0014
0.5	-0.033	-0.029	-0.029	-0.026	-0.024	-0.021	-0.016	-0.014	-0.008	-0.007
1.0	-0.637	∞	-0.522	0	-0.383	0	-0.214	0	-0.091	0
1.5	-0.478	-0.418	-0.425	-0.367	-0.349	-0.292	-0.226	-0.176	-0.109	-0.078
5.0	-0.969	-0.987	-0.874	-0.887	-0.727	-0.738	-0.481	-0.491	-0.244	-0.244
10	-1.926	-1.926	-1.732	-1.733	-1.442	-1.443	-0.959	-0.961	-0.481	-0.480



(a) (b)

Fig. 9. Effects of excitation frequency to the offset
(a) ζ_e = 0.1 and (b) ζ_e = 0.3

stroke were applied, i.e., ζ_e = 0.1 and 0.3. The value of damping ratio for the compression stroke, ζ_c, is varied for n = 0, 0.1, 0.25, 0.5 and 0.75.

The offset level for the numerical results are calculated from the mean value of the time response. That for the prediction are calculated using Eq. (12) by means of relative displacements, U_e and U_c, calculated from (4) for the value of damping aforementioned. The comparisons for Ω = 0.1, 0.3, 0.5, 1.0, 1.5, 5.0, and 10.0 are listed in Table 1 and Table 2 for ζ_e = 0.1 and 0.3, respectively.

The levels of offset calculated from numerical results and analytical prediction are similar with the difference of around 5% in total. However, the levels of offset for Ω = 0.1 are seen to be zero. This occurrence can be described as it is very low excitation frequency, compared to the natural frequency, the response can be said as quasi static. In other words, the isolated mass is moving in unison with the base excitation

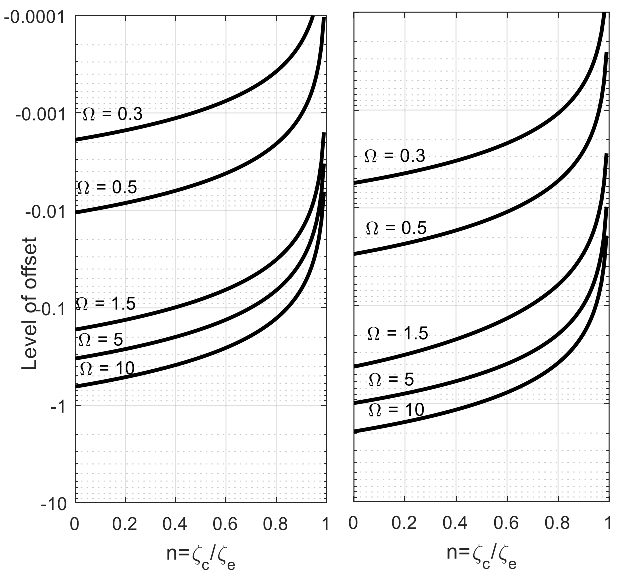
For that at Ω = 1, the prediction could not perform due to the resonance phenomenon. The relative displacement for both U_e and U_c are very similar. As a result, the level of offset could not be predicted. Thus, the data for Ω = 0.1 and 1 will be omitted and will not be discussed in detail using analytical prediction.

5.2 Discussions

In this section, the reader may be reminded that this study focuses on proposing the possible approach in determining level of offset causing from the asymmetric characteristic of the automotive shock absorber. The analytical results are compared to the numerical results listed in Table 1 and Table2.

The predicted data listed in Table 1 and Table 2 are reproduced to increase number of data and plotted in Fig. 8. It shows that the difference in the value of damping ratio affects the level of offset. By considering the excitation frequency Ω = 5, for example, it is seen that greater the difference causes the greater level of offset. When the value of is getting closer to that of ζ_e, the offset tends towards zero and becomes zero when $\zeta_c = \zeta_e$ regardless the excitation frequency.

This information also reveals the effect of excitation frequency when the asymmetric force damper is applied. The higher excitation frequency also results in the greater level of offset as also shown in Fig. 9. The level of offset appears to change linearly proportional to the excitation frequency. The maximum level of offset at Ω = 5 for ζ_e = 0.3, is seen to be about 3 times greater than that for ζ_e = 0.1. Therefore, from the predicted data, the level of offset caused from using the asymmetric force damper can be estimate.



(a) (b)

Fig. 8. Effects of difference between damping ratios to the offset (a) ζ_e = 0.1 and (b) ζ_e = 0.3

6. CONCLUSIONS

The effects of asymmetric force damper, normally used in the automotive vehicle suspension, is examined when used in the base excited vibration isolation. The number of publications and numerical results reveal the level of offset occurred for the harmonic excitation. This causes the isolated mass to be oscillating around the new equilibrium position. The level of offset is found to change when the ratio of value of damping ratio for compression and extension stroke changes. The level of offset, however, can be predicted by equating the forces acting on the isolated mass. The method provides similar level of offset to that obtained from numerical simulation which reveals the effects of the asymmetric force damper. The level of offset is found to be affected by the difference of damping ratio during compression and extension as well as the excitation frequency. The obtained information can be very useful to understand and prevent the mechanical system connected to the base excited vibration isolation having the asymmetric force damper.

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8. BIOGRAPHIES

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