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Research Article

Comparison Study of Vibration Control Effects between Suspended Tuned Mass Damper and Particle Damper

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The vibration control performance and its influencing factors of a tuned mass damper and a particle damper are examined by a single degree of freedom structure with such devices. The vibration control effects between these two dampers are also investigated. Increasing the mass ratio of the damper can improve the damping effects; under the condition of tuning frequency, the damping effects are remarkable. However, the more the deviation from the tuned frequency, the less controlling effects can be obtained. The damping effect of a particle damper is generally better than that of a tuned mass damper. For this test model, the particle damper can improve primary structure's equivalent damping ratio 19 times to the original one's, while the tuned mass damper can be 13 times. The reason lies in the fact that the particle damper can dissipate input energy by tuning mass, collision, impact, and friction between particles and the container and the momentum exchange effects between the secondary damper mass and the primary structure.

1. Introduction

The use of damper in structural vibration control area has drawn considerable interests since 1970, based on the theory of energy absorption which was presented by Kelly et al. [1]. The most common used passive approach is tuned mass damper (TMD) [2, 3], which has three main parts: tuning mass, spring, and damper element. Even though TMD has many advantages such as simple manufacturing, high reliability, and low maintenance costs, it can only be used in a small range of resonance frequency and is sensitive to the work environment conditions. If many solid particles are used to replace the single tuned mass, introducing the energy dissipation mechanism of particle collision and friction on the basis of tuning effects, the particle damper (PD) is proposed.

The particle damper, which evolves from the single particle impact damper [4], is a class of auxiliary mass damper, consisting of a container or a structural void partially filled with particles, which can be different materials, like metal, tungsten powder, and so forth. The single particle impact damper dissipates energy by the impacts between the single solid mass and the primary structure. Consequently, it will produce high level of noise during the impact process, and the large contact forces will result in material deterioration and local deformation accompanying plastic collisions. To reduce the noise and material deterioration problems, many small solid particles are used instead of the single mass, hence particle dampers are develop.

In fact, this particle damping technology has been widely used to reduce excessive vibration of mechanical systems, such as turbine blades [5], rocket engine turbopumps [6], and rotary printing equipment [7]. Besides, it is also successfully applied in civil engineering projects. Ogawa et al. [8] introduced a particle damper to suppress wind induced vibration of a single pylon of a cable-stayed bridge. Naeim et al. [9] introduced the performance of a tall building with a particle damper system in Santiago, during the 2010 Chile earthquake. The system performed very well during the earthquake and the building did not suffer any damages above the ground or at any subterranean floors and basements walls.

Although the engineering applications are continually launched, due to the highly nonlinear behaviors of particle collisions and frictions, many theoretical [10–12] and experimental [13–15] studies are still undergoing. However, all the studies are focused on the particle damper itself; the vibration

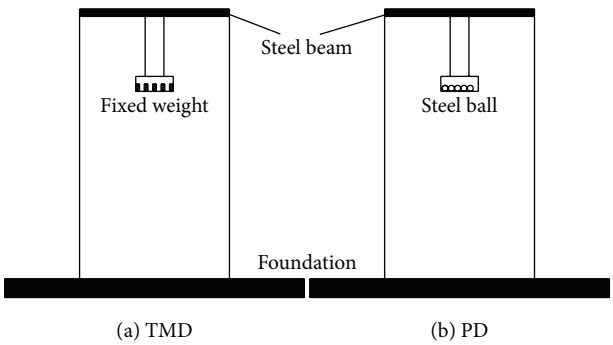


Figure 1: Models.

control effects comparative to other passive control devices, especially the very similar one, TMD, have seldom been carried out. This paper will introduce a preliminary comparative experimental study of both tuned mass damper and particle damper, attaching to a single degree of freedom frame. The influencing factors and energy dissipation mechanisms are also discussed further.

2. Test Model and Test Process

The primary structure of the test model is a single degree of freedom steel frame. The damper is a suspended pendulum, which provides different frequency according to the length of the suspension rope. The mass element of the model is a steel container. For TMD case, weights are fixed to the box, so that they would not move in the test process. For PD case, steel balls are put in the container, which could move freely. The mass of the steel frame in this test is 7.17 kg, the stiffness is 590 N/m, natural frequency is 1.42 Hz, and the damping ratio is 0.55%. These structural characterizations are all obtained from the experiment.

In the test, the vibration control effects of dampers are examined by the decay of the primary structure's response under free vibration, which is formed by giving the primary structure an initial displacement. By changing the frequency ratio and mass ratio between the damper and the primary structure, the working characterizations of suspended TMD and PD can be studied (see, Figure 1).

3. Test Results

3.1. Responses in Frequency Domain. Figure 2 shows the acceleration time histories of the uncontrolled structure and the primary structure attaching TMD and PD, separately, under different frequency ratios. In these cases, the mass ratio (μ) is about 5%, f_0 is the natural frequency of the main structure, and f is the natural frequency of the damper which is determined by the length of the suspension rope. From the figure, the following is shown.

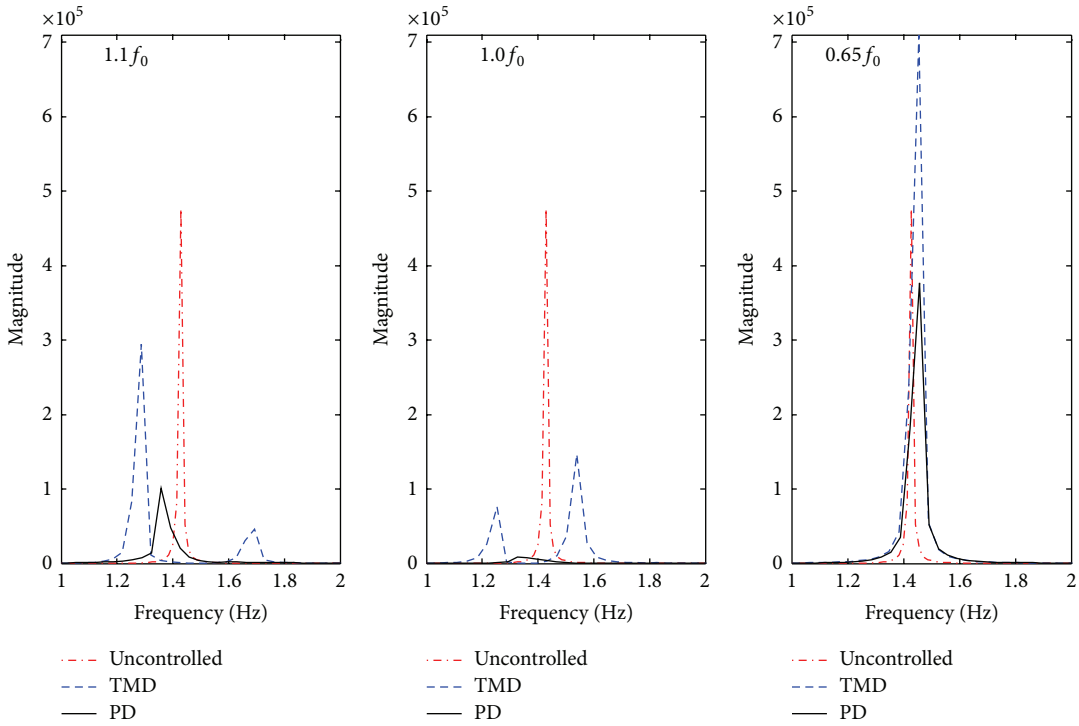
(1) There is only one peak in the Fourier spectra of the acceleration time history of the uncontrolled structure, which is 1.42 Hz, corresponding to its natural frequency of the structure. For TMD case, there are two peaks in the Fourier spectra at tuning frequency ($1.0 f_0$) case and 10% untuning frequency ($1.1 f_0$) case, which locate at both sides of 1.42 Hz, corresponding to the natural frequencies of the primary structure and the TMD device, separately. This is because when the damper is added to the main structure, the mass of that structure would increase, leading to a lower frequency, while there is also only one peak at 1.42 Hz, at 35% untuning frequency ($0.65 f_0$) case. It is shown that TMD has good damping effects in the vicinity of the tuning frequency; however, if untuning too much, such effects cannot be obtained through adjusting the structure's dynamic characteristics.

(2) At the TMD tuning frequency case, the magnitude of the second peak is greater than the first one, which shows that TMD's response is bigger than the primary structure. Thus, the primary structure's response is transferred to the additional mass, achieving a good reduction effect. At the case of $1.1 f_0$, the second peak is lower than the first one, which means that the main structure's response is bigger than the TMD's. Even though it has a certain damping effect, the performance is not as good as the former one. At the case of $0.65 f_0$, the moving mode of TMD is not excited; the amplitude of the primary structure at natural frequency is even larger than that at uncontrolled case, showing that if untuning too much, the TMD device would not play a role in vibration control, on the opposite, it may have the possibility to enlarge the response.

(3) At the cases of PD, the Fourier spectra only have one peak value, corresponding to the frequency which is smaller than 1.42 Hz, but larger than the first peak of the TMD's cases. The reason is that PD adds the weight to the primary structure, compared to the uncontrolled one, leading to a smaller frequency less than 1.42 Hz. On the other hand, the particle mass is not fixed to the container, and the response of the damper is excited by the collision between particles and the inner wall of the container; thus there is only one frequency peak and it is larger than the first one of the TMD's cases.

(4) At the cases of PD, the amplitudes at natural frequency are all smaller than that at the uncontrolled cases and at the TMD cases. This shows that PD has the best vibration control effects, not only at the tuning frequency, but also at the untuning frequencies, which is offset by 10% and 35%. Admittedly, at 35% untuning frequency case, the control effect is not satisfied enough and subject to variable.

(5) The Fourier spectra represent the distribution of vibration power of the primary structure in frequency domain, with the area representing the average vibration power. It is shown that, in the vicinity of the tuning frequency, the areas at TMD and PD cases are both smaller than the uncontrolled one, indicating



(a) $f = 1.1 f_0$ (b) $f = 1.0 f_0$ (c) $f = 0.65 f_0$

Figure 2: Frequency response of the acceleration time history of the primary structure, $\mu = 5\%$.

that the vibration energy of the main structure is obviously reduced by attaching the damper device.

In order to further compare the vibration control effects of TMD and PD, Figure 3 shows the frequency response of the acceleration time history of the primary structure, under different frequency ratios and mass ratios. It is shown that, whether the device is tuned or not tuned, increasing the mass ratio can improve the vibration attenuation effects; the performance of PD is usually better than TMD, especially at the tuning conditions. This is because, at the tuning case, the response of the damper is very large, causing particles in the container moving violently. The particle damper dissipates the input energy not only by tuning mass, but also by the collision, friction, and the momentum exchange between the primary structure and the particles.

3.2. Responses in Time Domain. Corresponding to Figure 2, Figure 4 shows the acceleration time history of the primary structure under different frequencies. From (a) to (c), frequency ratios are 1.1, 1.0, and 0.65, separately, which correspond to the 10% untuning frequency case, tuning frequency, case and 35% untuning frequency case. From the figure, the following is shown.

(1) The acceleration of the uncontrolled structure under free vibration decays slowly. After attaching the TMD, it decays much faster, while after attaching the PD, it decays in the fastest manner. These phenomena show that the damping of the uncontrolled structure itself is very small. The TMD device can provide certain additional damping, while the PD device can provide the most additional damping, thus leading to the best working performance.

(2) At the 35% untuning frequency case, the natural frequency of the primary structure with damper is almost the same as the uncontrolled one, indicating that the movement of dampers is not excited, which cause bad performance. However, at the tuning frequency and 10% untuning frequency cases, the natural period of the primary structure with damper is different with the uncontrolled one; basically it becomes longer. This result also confirms to the discussion in Section 3.1. and will be confirmed again in Section 3.3.

(3) During the very beginning time period, the acceleration of structure with damper is coincident to the uncontrolled one. After that, the former starts to decay much faster than the latter. These phenomena show that both dampers need a short time to trigger. TMD uses this time to complete the start-up process and to get into the large response phase. PD needs this time to create collision patterns; that is, the particles need a certain time traveling to the wall of the container and making impacts, producing damping effects.

In order to further compare the effects of these two damper devices, corresponding to Figure 3, Figure 5 shows

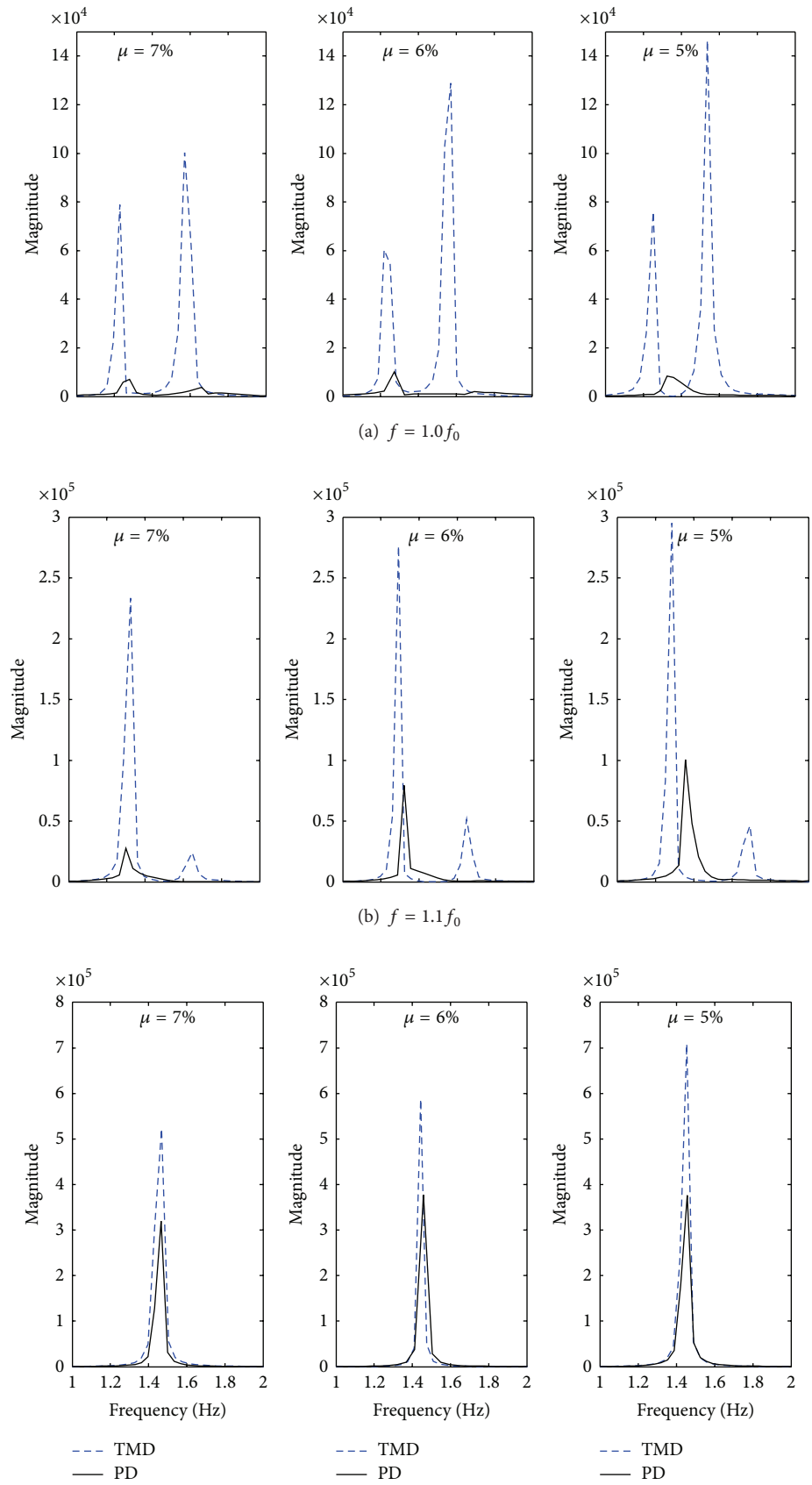


Figure 3: Frequency response of the acceleration time history of the primary structure.

Table 1: Equivalent damping ratios of the primary structure at different cases (%).

Mass ratio	Frequency ratio								
	$0.65f_0$			$1.0f_0$			$1.1f_0$		
	TMD	PD	Improved efficiency	TMD	PD	Improved efficiency	TMD	PD	Improved efficiency
$\mu = 5\%$	0.51	0.59	15.7	0.79	3.90	393.7	0.75	2.16	188.0
$\mu = 6\%$	0.50	0.60	20.0	0.83	3.71	347.0	0.80	3.20	300.0
$\mu = 7\%$	0.51	0.48	−5.9	1.04	10.36	896.2	0.86	3.40	295.3

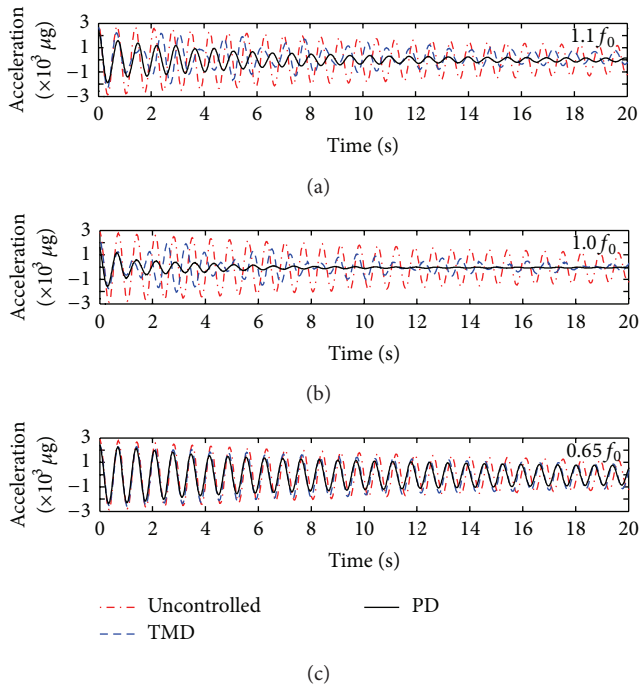


Figure 4: Acceleration time history of the primary structure, μ = 5%.

the comparative time history of the primary structure, under different frequency ratios and mass ratios. It shows that, at the very beginning time period, the response at PD case is almost the same as that at TMD case, while as time goes by, the effect of the former becomes much better than the latter. However, at 35% untuning frequency case, the advantage is not very obvious. Moreover, increasing the mass ratio can improve the damping effects.

3.3. Results of Equivalent Damping Ratio. Table 1 shows the equivalent damping ratio results of the primary structure at all experimental cases. The following can be seen. (1) At 35% untuning frequency case, the equivalent damping ratios of the controlled structure are similar to the uncontrolled one, indicating that no additional damping is achieved by the dampers. (2) At tuning frequency case, both TMD and PD can effectively reduce the response of the primary structure, in which the PD can get even better performance. For example, at the case of 7% mass ratio, the additional damping provided by PD is 10 times that provided by the TMD, and 19 times that provided by the primary structure itself (0.55%). (3) At slight untuning frequency of 10% case, TMD and PD can also reduce the vibration of the primary structure to some extent, in which PD has exponentially better effects than TMD.

4. Theoretical Analysis

The vibration control mechanism of TMD is that the frequency of the damper is tuned to a particular structural frequency so that when that frequency is excited, the damper will resonate out of phase with the structural motion. Energy is dissipated by the damper inertia force acting on the structure. Consequently, the working performance of TMD is influenced by the mass ratio, frequency ratio, and damping ratio. Due to the constraints of the experiment, although the frequency can be tuned to the natural frequency of the primary structure, the damping ratio has not been adjusted to the optimum condition. Consequently, the equivalent damping ratios of the controlled structure are numerically calculated according to the method proposed by [16], with the mass ratio being 7%, the frequency ratio being 1.0, and the damping ratio of the damper device continuous changing, shown in Figure 6. It can be observed that when the damping ratio of TMD is 20%, the device can have the optimum performance, which increases the primary structure's equivalent damping ratio to 7%. In the experiment, the damping ratio of the TMD is tested to be around 1.5%, corresponding to the structure's equivalent damping ratio being 1% in Figure 6. This is also consistent with the test results shown in Table 1.

Particle damper achieves its optimum operation in the experiment. When the mass ratio is 7%, the particles lie at the bottom of the container, occupying about 80% of the bottom area, which provides the severe collisions to dissipate the input energy during the test. If there are too much particles, piling up to three or more layers in the container, then the motion of the lower layers is minimized, and the efficient impact mechanism cannot be established. On the other hand, if there are too few particles, it takes a longer time for particles to move from one wall to the opposite wall after a collision, with fewer impacts occurring, causing an inefficient performance. In fact, the researchers [17] have found that, for a single particle impact damper, the optimum operation can be achieved by two particle-wall impacts per one cycle, and this moving pattern is also the most stable one. This phenomenon has been well verified in the experiment.

Comparing the numerical analysis and experimental results of the two dampers, it is shown that particle damper has better vibration control effects than tuned mass damper,

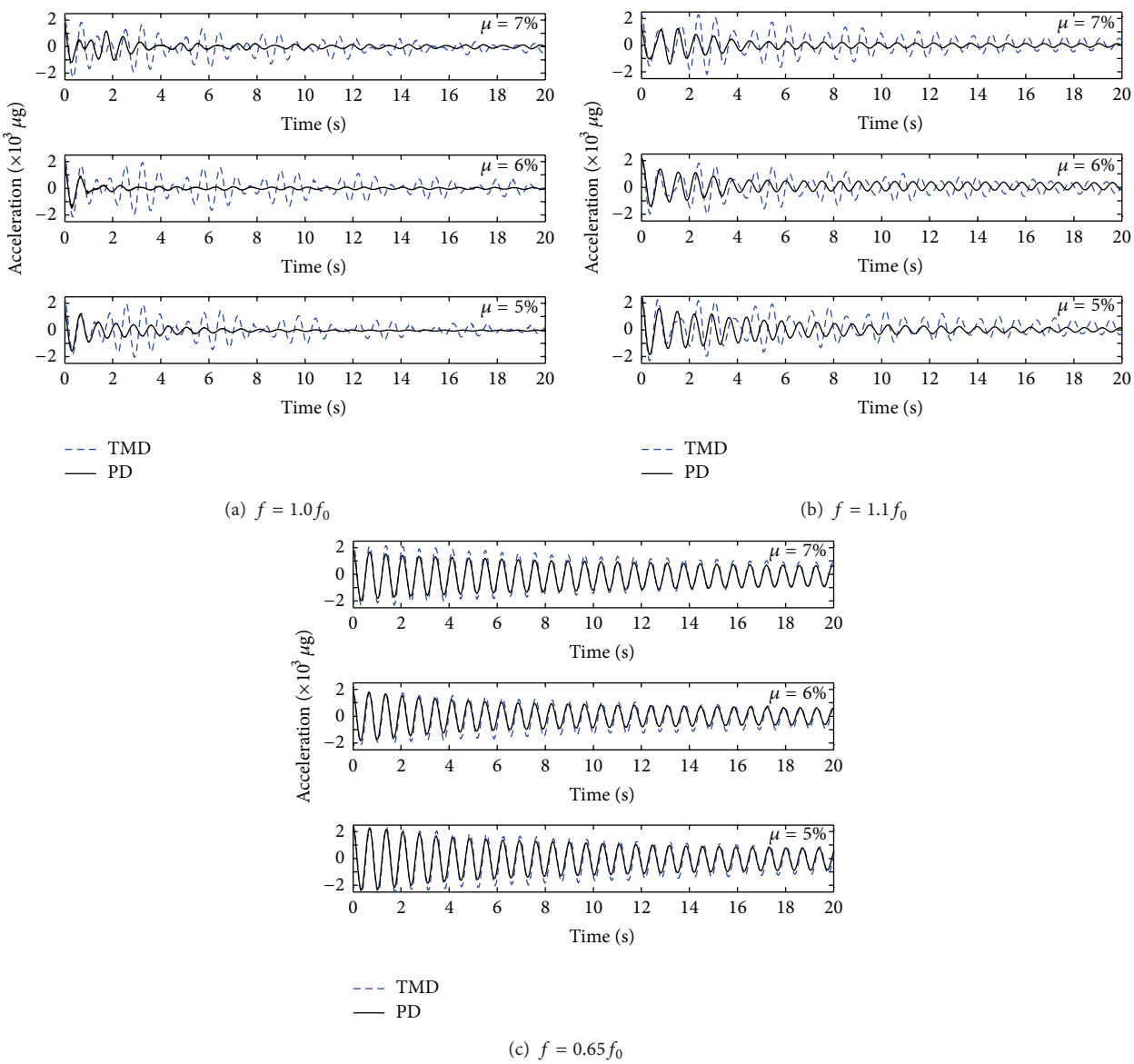


Figure 5: Acceleration time history of the primary structure.

especially in the vicinity of the tuning frequency. At the cases of too much untuning frequency, although both dampers reduce the damping effects obviously, the particle damper seems still better than the tuned mass damper. The reason for this phenomenon may be that the vibration control principle of a suspended particle damper combines tuning mass, collision, impact, and friction between particles and the container and the momentum exchange effects between the additional mass and the primary structure. These various energy dissipation forms improve the vibration control performance.

5. Conclusions

The vibration control performance of a tuned mass damper and a particle damper is tested by a single degree of freedom structure with such devices, and the damping characteristics of these two devices are also compared. Increasing the mass ratio can generally improve the control effects; at the tuning frequency, the damping effects are remarkable. However, the more the deviation from the tuned frequency, the less controlling effects can be obtained. At 35% untuning frequency case, the damping effect is negligible.

The working performance of a particle damper is better than a tuned mass damper. During the experiment, the equivalent damping ratio of the structure is raised up to 10.36% at the optimum operation of the particle damper, which is 19 times that of the primary structure itself. According to the theoretical analysis, the optimum performance of a tuned mass damper can be obtained, raising the structure's equivalent damping ratio up to 7%, which is 13 times that of the structure itself. There is a certain difference between these two passive devices. The reason may lie in the various energy dissipation forms of particle dampers, including tuning mass,

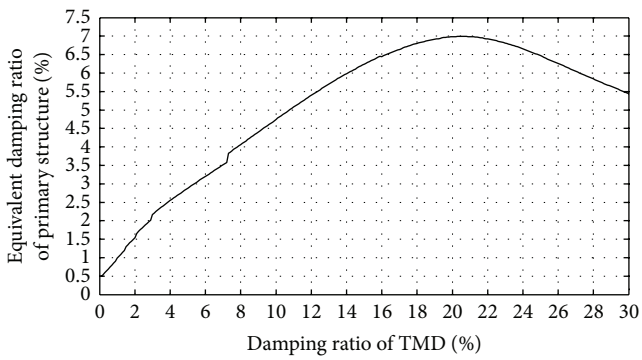


Figure 6: Damping ratios of the primary structure under different damping ratios of TMDs.

collision, impact, and friction between particles and the container and the momentum exchange effects between the additional mass and the primary structure. Based on these preliminary conclusions, the particle damper may have potential wide application prospects in structural control area.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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