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An experimental investigation into the use of a buffered impact damper

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Abstract

An impact damper is a freely moving mass, constrained by stops, located on a dynamic structural system to be controlled. As the system is excited, the impact mass moves relative to the structure resulting in impacts between the mass and the stops, transferring momentum from the structure to the impact mass, and dissipating energy as heat, noise and high frequency vibrations. At the point of impact, large accelerations are imparted to the structure, which may be undesirable, particularly for occupied structures. In order to reduce these high accelerations, it is proposed to incorporate a buffer region between the mass and the stop. Free and forced vibration tests of a system equipped with a buffered impact damper are used to study the resulting damping effect and impact characteristics. The performance of the buffered impact damper is compared with that of a conventional rigid impact damper. It is found that the buffered impact damper not only significantly reduces the accelerations, contact force and the associated noise generated by a collision but also enhances the level of vibration control. A possible reason for the enhanced control is postulated by examining impact behaviour. The investigation shows that the effective reduction of the vibration response depends not only on the magnitude of the contact force but also upon the contact time.

1. Introduction

Controlling the dynamic response of structures has an important role to play in engineering. In particular, passive control methods are favoured where possible due to their mechanical simplicity and lack of power requirement. An impact damper is a passive control device which takes the form of a freely moving mass, constrained by stops attached to the structure under control, i.e. the primary structure. The damping results from the exchange of momentum during impacts between the mass and the stops as the structure vibrates. Energy is dissipated as heat and noise together with the development of high frequency vibrations in the structure.

It has been shown that impact dampers can be more effective than comparable tuned-mass-dampers in mitigating the response of a lightly damped structure under dynamic loading [1]. Many other features, such as simple maintenance free construction, also make impact dampers attractive for practical implementation.

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They have been successfully used to reduce excessive vibration of turbine blades, radar antenna, electrical relays, light poles, tall slender structures and machine tools. However, there are a number of issues which prevent more widespread use of impact dampers, particularly in civil engineering. One such obstacle is the high level of acceleration caused by collision of the rigid mass with the rigid stops attached to the structure together with the corresponding high noise level, particularly when collisions occur between metal objects. Additionally, the large contact force may cause local damage to the structure of the mass or the stops. If the structure under control is an occupied building, then the high accelerations and noise will cause discomfort to the building's occupants. Another obstacle to more widespread use is that the performance of an impact damper is particularly sensitive to system parameters and type of loading. The performance can drop off significantly if the parameters are not optimised for a particular excitation type.

Attempts have been made to decrease the contact force and the corresponding high noise level, by introducing the idea of multi-unit impact dampers where a single mass is replaced by a number of smaller masses resulting in smaller contact forces for impact of each mass, but with a similar overall effect as an equivalent single mass [2]. A particular form of multi-unit impact damper is a bean-bag damper [3]. This consists of a bag filled with small, spherical lead shots. The flexibility of the bag can be modified to change the resilience by adjusting the tightness of the bag. The performance of a bean-bag impact damper was studied for controlling a single-degree-of-freedom (sdof) structure under sinusoidal excitation. The investigation showed that the bean bag impact damper is not only a good attenuator of the displacements at resonance but the contact force and the noise generated by collision are also reduced. This investigation also revealed that the performance of the bean-bag impact damper was significantly affected by the tightness of the bag, since it is one of the main factors governing the contact characteristics. As the author pointed out, the contact force is the key to the control effect and in turn the key to the design of such an impact damper. Unfortunately, the contact forces cannot be predicted easily because they evolve over time and change nonlinearly with the level of the external excitation force. This may prove to be an obstacle to the widespread application of this kind of impact damper. Moreover, further investigations are necessary to ascertain the response of structures under other forms of excitation, such as random excitation.

Another similar system, called a particle, or granular impact damper, has been developed by Araki et al. [4] with the same aim of reducing or eliminating the acceleration and noise problems of a rigid impact damper mass. This damper consists of a bed filled with granular material which is fixed to the primary structure. The performance of a particle impact damper on a sdof primary structure under random excitation has been investigated [5]. The influence of particle size, container dimensions, mass ratio and intensity of excitation were investigated experimentally. One problem with this kind of impact damper is that its performance is significantly influenced by the intensity of the excitation. When the excitation is of low intensity, the control effect of this kind of impact damper is poor since the particles are not mobilised. The level of damping of particle impact dampers has also been shown to depend upon the geometry of the device [6]. It should be noted that, for such a damper, damping occurs primarily from friction between the particles rather than impacts and, therefore, a particle damper is not a true impact damper although it shares many of the same characteristics. Another problem posed by particle impact dampers is precise modelling of the dynamic system. This, like the bean-bag impact damper, is difficult to resolve satisfactorily, although an attempt has been made to capture the physics of the main energy dissipation mechanisms [7].

Recently, work has been carried out on resilient impact dampers, i.e. a damper where the collision between the impact mass and the stop takes deformability into account [8]. Using a highly deformable stop, the impact noise is greatly reduced. However impact time must be taken into account in the modelling of the behaviour so coefficient of restitution models are unsuitable. An analytical model has been developed for such a system under free vibration using a spring-damper to model the impact surface. The investigation examined the effect of clearance for such a freely vibrating system and concluded that the clearance should be less than twice the initial displacement of the structure (when the free vibration is initiated by an initial displacement of the structure) if the damper is to work effectively.

This paper takes a similar approach to not only reducing impact noise but also reducing the high accelerations and contact forces associated with a conventional rigid impact damper by introducing a flexible buffer zone between the freely moving damper mass and the stop fixed onto the primary system. The buffer cushions the impact, increasing contact time whilst reducing contact force. The advantage of this system over bean-bag or particle impact dampers is that it has the potential for simple modelling of the system dynamics. This paper uses experimental results to investigate the performance of a buffered impact damper compared with that of a rigid impact damper. Not only are the effects of clearance investigated, but also excitation type, frequency and amplitude, size of damper mass and buffer stiffness. The damping characteristics of a buffered damper are also investigated.

2. Experiments

2.1. Experimental structure and set-up

The experimental structure, illustrated in Fig. 1, was built to simulate a SDOF linear oscillator, forming the primary system to be controlled. The test model is composed of two sets of flexible columns made from steel strips with a 40×1 mm cross section and a beam made from aluminium alloy with a 40×30 mm cross section. The columns are bolted rigidly to the beam and the base such that the ends are rotationally fixed. The base is fixed to a unidirectional shaking table that is driven by an electro-dynamic shaker to produce base excitation. The natural frequency of the primary structure is $f_n = 4.03$ Hz.

The impact mass itself is a steel ball which runs in a groove along the top of the beam. Two triangular shaped brackets are mounted on top of the beam to act as motion limiting stops for the freely moving impact mass. Buffers, made from various plastic, rubber and sponge materials, can be fixed by adhesive to the stops and can be easily removed and changed. The damping and stiffness parameters, c_b and k_b, of the buffers under impulse loading (which differ greatly from the same parameters under quasi-static loading) are calculated, based on a technique developed by the authors [9], using the measured coefficient of restitution, e, and contact time, T_c (which both remain approximately constant for the range of velocities occurring in the tests) as summarised in Table 1.

The clearance between the stops (and, hence, the buffers) is adjustable. The response of the structure is measured using an accelerometer fixed to the end of the beam. The base excitation is also measured using an

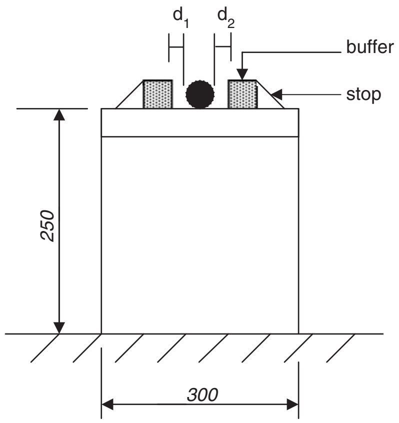

Fig. 1. Test structure.

Table 1
Parameters of buffer/stop materials

Impact mass	Buffer/stop material	e	T_c (s)	k_b (kN/m)	c_b (Nm/s)
Steel ball	Buffer 1 (sponge)	0.61	0.025	1.65	4.0
Steel ball	Buffer 2 (soft rubber)	0.44	0.019	2.99	8.8
Steel ball	Buffer 3 (hard rubber)	0.53	0.004	65.7	32.5
Steel ball	Buffer 4 (hard plastic)	0.49	0.003	118.0	48.7
Steel ball	Steel (no buffer)	0.46	0.0003	11900.0	529.6

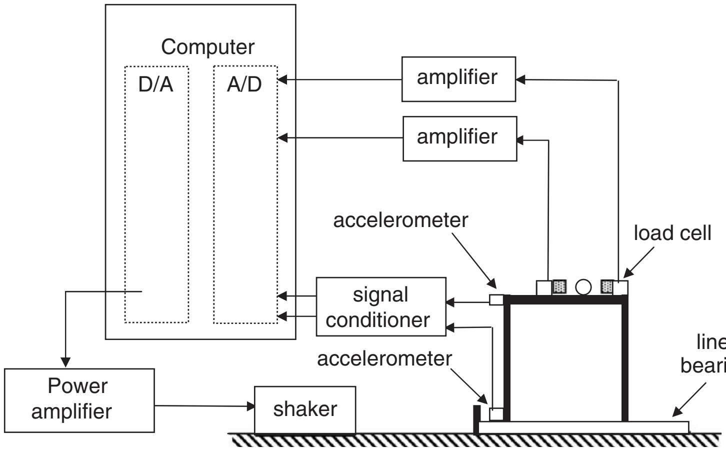

Fig. 2. Experimental set-up.

accelerometer. Impacts between the stops and the impact mass are measured using force transducers fixed between the stops and the buffers. The complete experimental set-up is shown diagrammatically in Fig. 2.

2.2. Experimental procedure

Free-vibration experiments were carried out by initially exciting the primary structure in two different ways. The first is by applying an initial displacement, x_0, to the structure and then releasing it. The second is by releasing a pendulum from a predetermined distance to strike the primary structure and, hence, give it an initial velocity, v_0. During these free-vibration experiments the shaking table was fixed to a stationary base. Both of these two simple excitation methods have been found to be reliable in producing a consistent and repeatable transient disturbance.

Forced-vibration experiments were also carried out using both sinusoidal dwell and random excitation. The test structure was excited through the movement of the shaking table on which the test structure was fixed, to provide base excitation. A simple PID feedback controller was used to control the shaking table, ensuring that the correct movement was produced and repeatable. The free and forced excitations described above were repeated for the structure without an impact damper, with a conventional rigid impact damper and with a buffered impact damper.

3. Results and comparison

3.1. Free vibrations

For free-vibration experiments, the primary structure is excited by setting it to an initial displacement of $x_0 = 10$ mm and then releasing. The mass ratio (i.e. the ratio of the mass of the ball to the mass of the primary structure) is taken as $\mu = 0.082$ and the clearance (defined as the diameter of the impact mass subtracted from the distance between the stops or buffers, or, referring to Fig. 2, $d = d_1 + d_2$) is taken as $d = 15$ mm (less than twice the initial displacement as suggested by Chen and Wang, [8]) for both the conventional rigid impact damper and the buffered impact damper. Fig. 3 shows the power spectral density (PSD) of the acceleration response of the primary structure when without an impact damper, with the conventional rigid impact damper and with the buffered impact damper. It can be seen that both the rigid and buffered impact dampers provide a high level of attenuation, with the buffered impact damper resulting in slightly better control compared to the rigid impact damper. Fig. 4 shows the time history of the acceleration responses. Very high acceleration peaks are seen to occur for the rigid impact damper at the moment of each collision with the stops whereas, for the buffered impact damper the accelerations remain small and at a lower level than the acceleration response without an impact damper. It was also evident during the tests that the impacts of the rigid impact damper

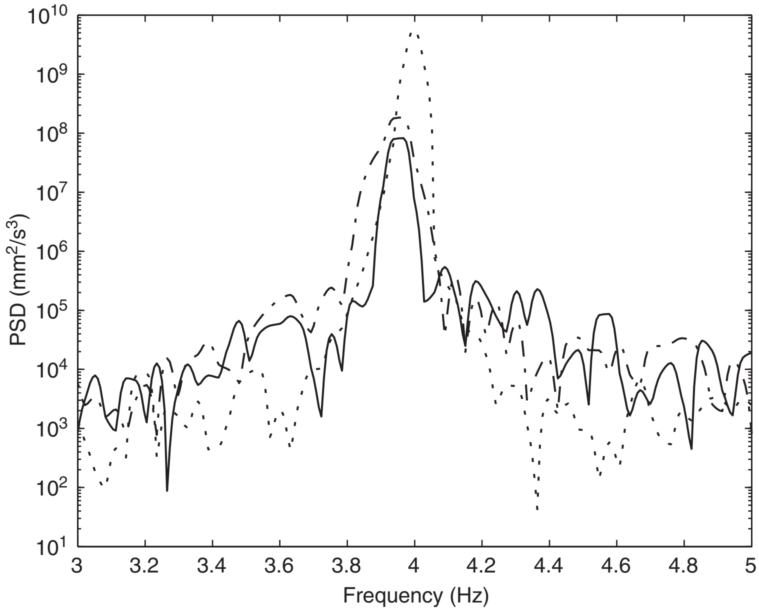

Fig. 3. Power structural density (PSD) of acceleration response, free vibration (initial displacement). …… Without damper; – · – · – with damper; —— with buffered damper.

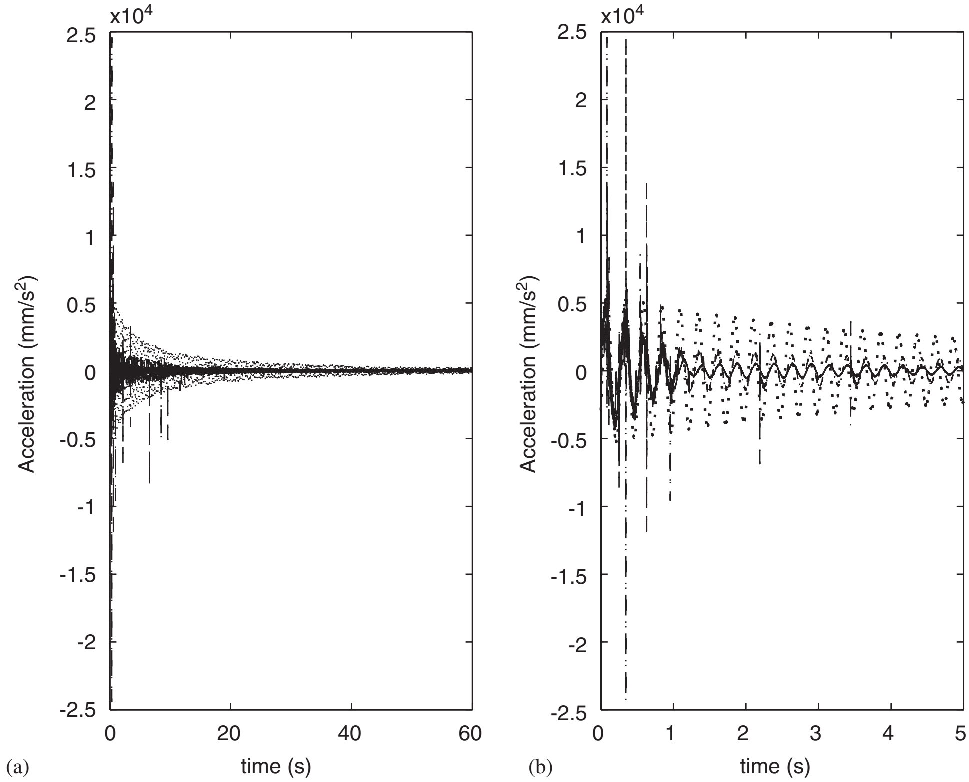

Fig. 4. Time history of accelerations (a) first 60 s, (b) first 5 s, free vibration (initial displacement). …… Without damper, – · – · – with damper, —— with buffered damper.

resulted in a much higher level of noise than for the buffered impact damper. It can also be seen from Fig. 4 that the buffered impact damper reduced the acceleration response much more quickly than the rigid impact damper.

Similar results are obtained when the structure is excited by an initial velocity (by release of a pendulum from a predetermined distance and striking the primary structure), as presented in Figs. 5 and 6. Fig. 5 shows the comparison between the PSDs of the acceleration response when without an impact damper, with a rigid impact damper and with a buffered impact damper. Fig. 6 shows the corresponding time histories of the acceleration responses. The same conclusions can be reached, i.e. the buffered impact damper reduces the response more effectively than the rigid impact damper, with no peaks in acceleration or related noise at the moment of impact.

The effect of mass ratio was investigated by means of initial velocity excitation. In this case the clearance chosen was $d = 20\,\mathrm{mm}$ and the three mass ratios investigated were $\mu = 0.19$, 0.82 and 0.05. The results, shown in Fig. 7, demonstrate that the buffered impact damper controls the structure better than the rigid impact damper for all three mass ratios investigated. Moreover, the difference between the response with the rigid impact damper and the buffered impact damper becomes more significant as the mass ratio becomes smaller. This makes the buffered impact damper even more attractive for use in practice since it is usually desirable for the impact mass to be as small as possible.

Finally, the effect of the amplitude of the excitation was investigated with a mass ratio of $\mu = 0.05$ and a clearance of $d = 20\,\mathrm{mm}$. Excitation amplitude was increased by releasing the impact pendulum from increasing height, resulting in increased impact velocity and, hence, increased amplitude impulse force, although the actual excitation impulse force was not measured. The results, shown in Fig. 8, are in the order of increasing excitation amplitude (and, therefore increased amplitude response). It can be seen that for the buffered impact damper, the PSD is damped to approximately the same level irrespective of the amplitude of excitation, and in all cases results in a significantly smaller response than the conventional unbuffered impact damper.

3.2. Forced vibrations

Sinusoidal dwell base excitation was applied to the structure to investigate the behaviour of buffered impact dampers further. The mass ratio was chosen to be $\mu = 0.082$ and clearance $d = 20\,\mathrm{mm}$. By applying sinusoidal

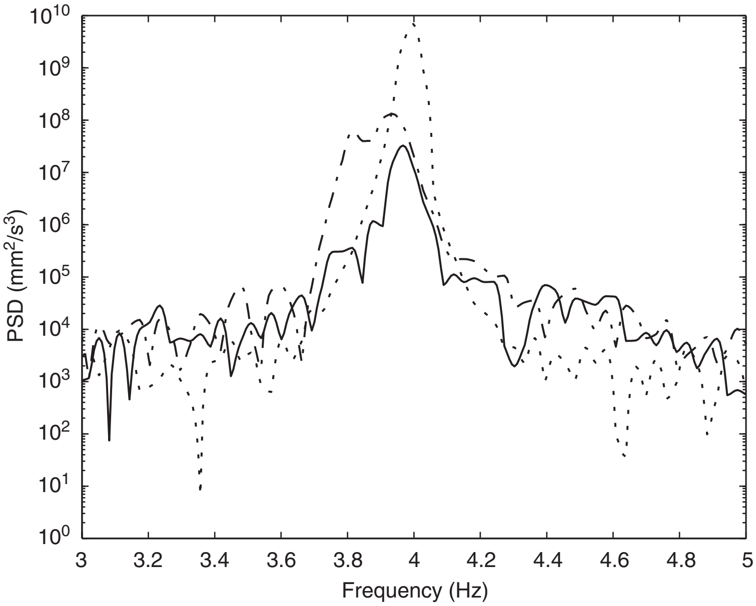

Fig. 5. Power spectral density (PSD) of acceleration response, free vibration (initial velocity). ······ Without damper, – · – · – with damper, —— with buffered damper.

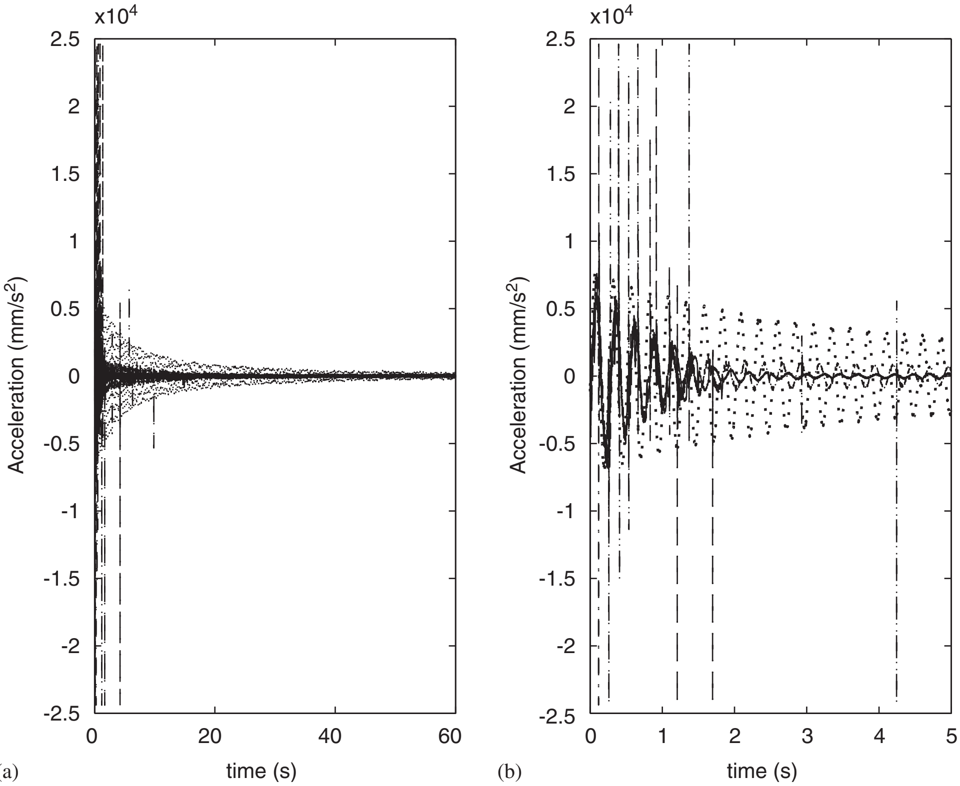

Fig. 6. Time history of accelerations (a) first 60 s, (b) first 5 s, free vibration (initial velocity). ······ Without damper, – · – · – with damper, —— with buffered damper.

excitation at frequencies above and below the natural frequency of the structure, the effect of excitation frequency was investigated. The results are shown in Fig. 9 where frequency ratio, r, is defined as the ratio of excitation frequency, f, to that of the natural frequency of the primary structure, f_n, i.e. $r = f/f_n$, and P/P_0 represents the ratio of P, the peak value of the PSD of acceleration with an impact damper, to P_0, the peak value of the PSD of acceleration of the structure without an impact damper. Hence, a PSD ratio of less than 1.0 represents a control effect, whilst a ratio greater than 1.0 represents a detrimental effect. It can be seen that when the frequency ratio is less than 0.9 both the buffered impact damper and rigid impact damper result in an increased response of the primary system with the buffered impact damper increasing the response of the primary system more than the rigid impact damper. When the frequency ratio is between 0.9 and 0.95 the control effect of the buffered impact damper and that of the rigid impact damper is almost identical. However, when the frequency ratio is between 0.95 and 1.25 the control effect of buffered impact damper is significantly better than that of rigid impact damper. It should be noted that for the lightly damped structure under investigation, the response of the structure, whether controlled or not, is relatively small outside the range $0.95 > f/f_n > 1.05$ and therefore the effect of the damper increasing the response for $f/f_n < 0.9$ is not of major concern. The overall response at these frequencies is still less than, or equal to, the controlled response when excited at the resonant frequency. It should also be noted that the reduction in response for the case of base excitation is not as significant as for the free vibration case. This is due to the damper having to dissipate the constant energy input of the system under sinusoidal base excitation as opposed to the transient energy input of the free vibration response.

In the general situation, a structure may be excited over a wide frequency range, rather than at a specific frequency. Therefore, further tests have been performed using random base excitation with a band-limited frequency content between 0 and 15 Hz. Figs. 10(a), (b) and (c) show comparisons between the transfer function (between the base excitation and the acceleration response of the primary structure) when without an

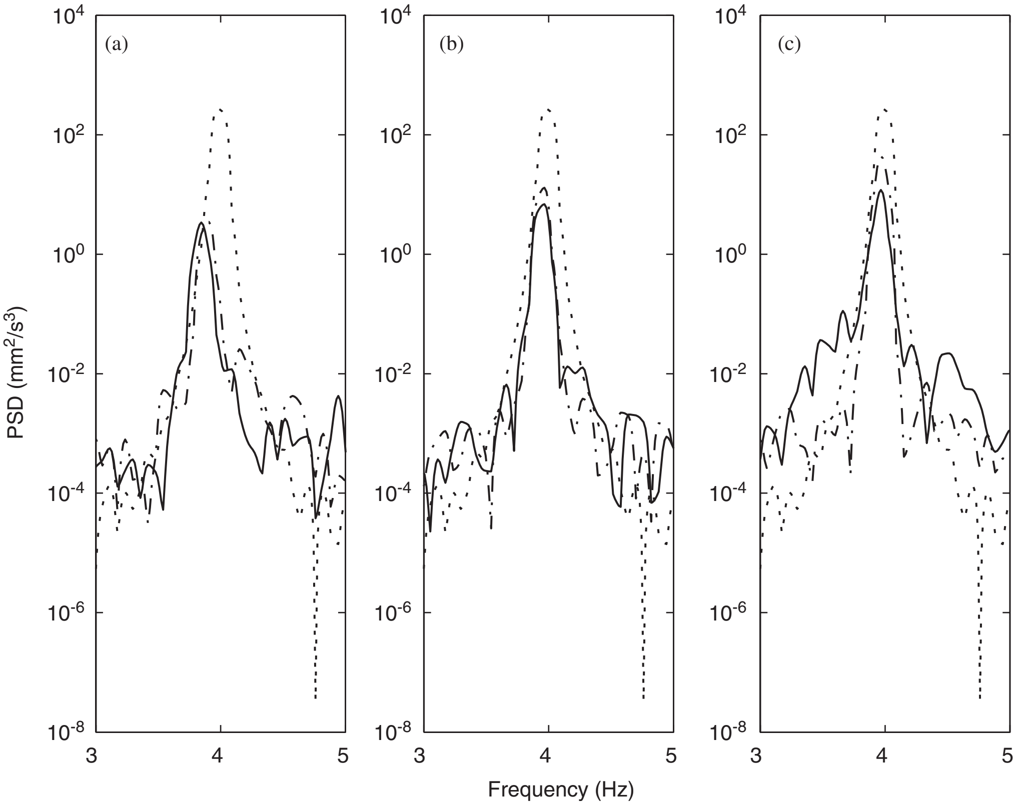

Fig. 7. Effect of mass ratio of (a) 0.19, (b) 0.082, (c) 0.05, free vibration. ⋯⋯ Without damper, – · – · – with damper, —— with buffered damper.

impact damper, with a conventional rigid impact damper and with a buffered impact damper for three different mass ratios ($\mu = 0.05$, 0.082 and 0.192). The clearance taken in this case is $d = 20$ mm. It can be seen that for both buffered and rigid impact damper, a higher mass ratio results in better control. However, the buffered impact damper produces better control of the resonant peak than the rigid impact damper in all cases. This is similar to the results seen in Fig. 7 for free vibration. However, the buffered impact damper results in worse control at frequencies below the natural frequency, which is as expected from the results presented in Fig. 9. Again, a similar trend is seen in the results shown in Fig. 7 for free vibration. As stated before, the increase in response is still very small compared with the peak response, and is therefore not critical in most applications.

Fig. 11(a) shows a segment of a typical time history of the acceleration response for a given random excitation input. The graph shows a comparison between the response of the primary structure without a damper, with a rigid impact damper of mass ratio $\mu = 0.05$ and with a buffered impact damper of mass ratio $\mu = 0.192$. It can be seen that even though the mass of the buffered impact damper is almost four times that of the rigid impact damper, the peak accelerations at impact are much smaller. Additionally, as would be expected, the general control effect of the large buffered impact damper mass is significantly better than for the unbuffered smaller damper mass, particularly for large amplitude vibrations. Fig. 11(b) shows the corresponding measured contact force between the mass and the stops for the rigid impact damper and the buffered impact damper. It can be seen that the contact force of the buffered impact damper is much smaller than that of the rigid impact damper, even though the mass ratio of the former is much larger. However, it can also be clearly seen that the contact time of each collision is much longer for the buffered impact damper than for the rigid impact damper. This is discussed further in Section 3.3.

The effect of clearance under random excitation is shown in Fig. 12 where A_0 is the integral of the PSD of the acceleration response of the primary structure without an impact damper while A is the integral of the PSD

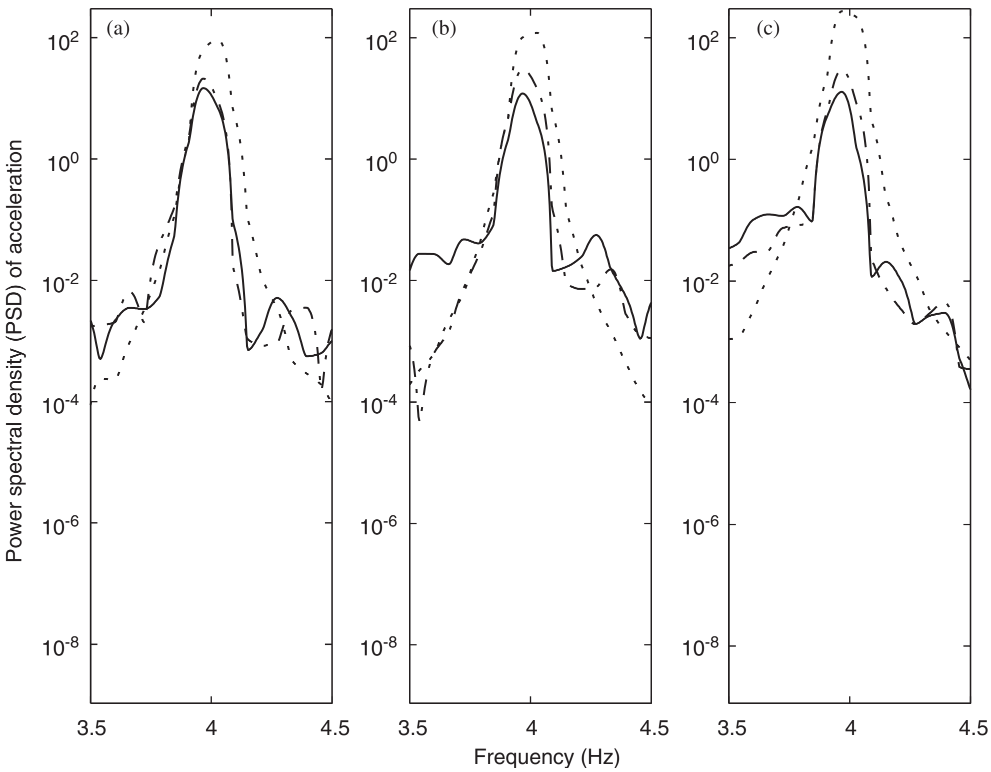

Fig. 8. Effect of increasing excitation, free vibration (a) small amplitude, (b) medium amplitude, (c) large amplitude. ...... Without damper, – · – · – with damper, —— with buffered damper.

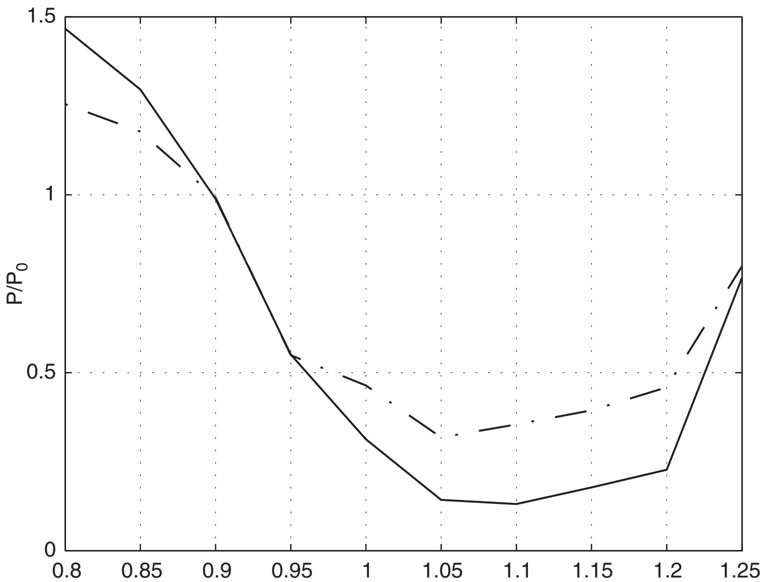

Fig. 9. Effect of excitation frequency, sinusoidal excitation. —— Ratio of peak PSD of response without damper to that with buffered impact damper. – · – · – Ratio of peak PSD of response without damper to that with conventional impact damper.

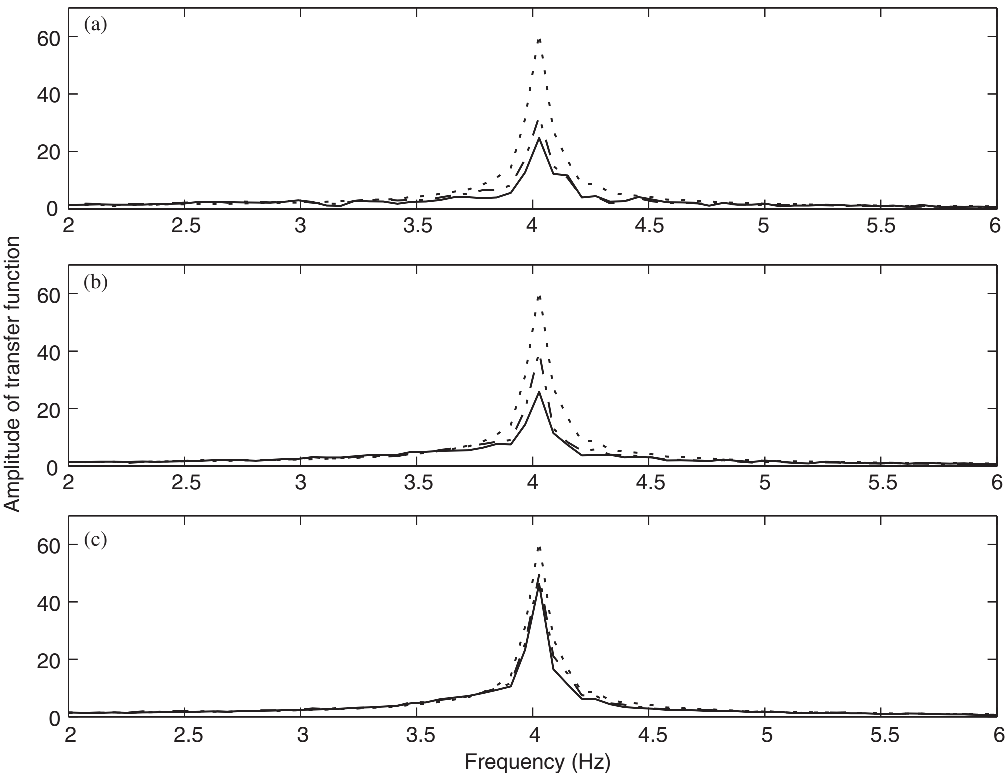

Fig. 10. Effect of mass ratio, under random vibration, with mass ratio of (a) 0.19, (b) 0.082 and (c) 0.05. ⋯⋯ Without damper, – · – · – with damper, —— with buffered damper.

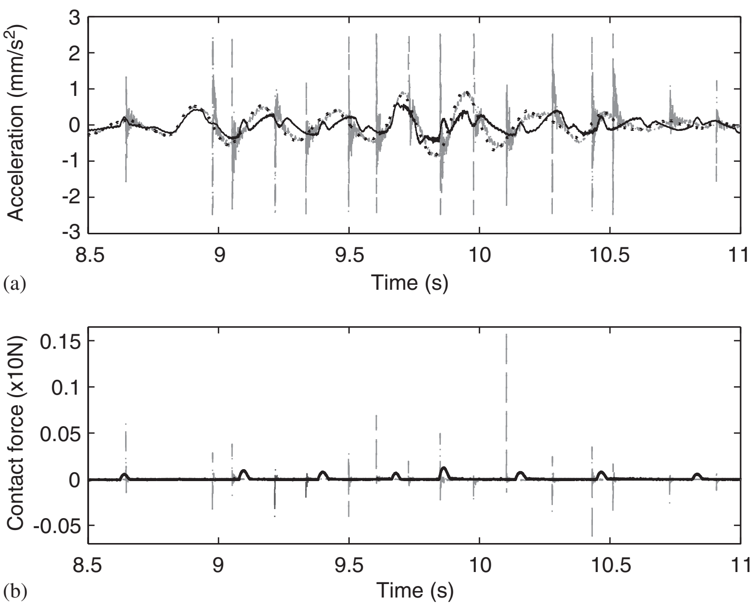

Fig. 11. (a) Acceleration response and (b) contact force, under random vibration random vibration. —— Buffered damper and mass ratio 0.192, – · – · – with damper and mass ratio 0.05, ⋯⋯ without damper.

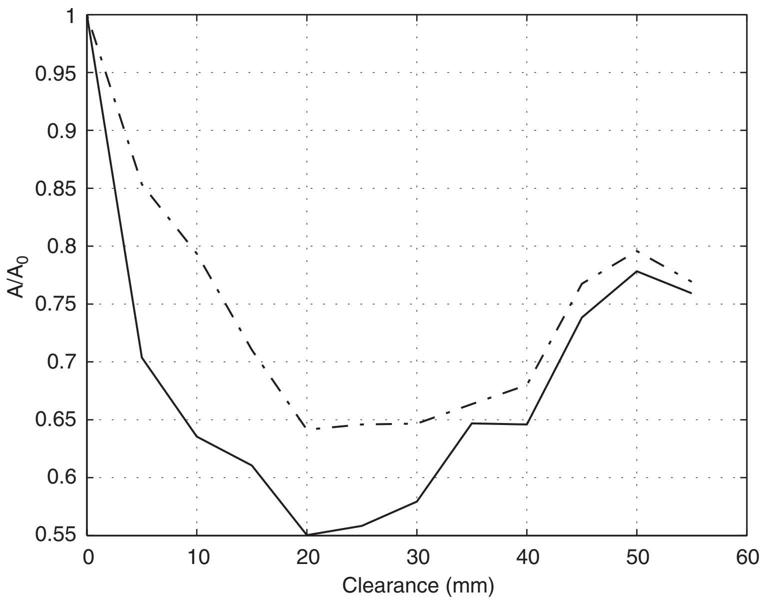

Fig. 12. Effect of clearance under random excitation. —— Ratio of area under PSDs between undamped response and response with buffered damper. – · – · – Ratio of area under PSDs between undamped response and response with conventional impact damper.

of the acceleration response of the structure with either a rigid or buffered impact damper (the integral of the PSD was used since, as can be seen from Fig. 10, the response of the structure is significant over a wide frequency range rather than concentrated at the natural frequency, so the integral provides a better indication of the degree of control over the whole frequency range than the peak value would). In this case the mass ratio used is $\mu = 0.082$. It can be seen that over the whole clearance range of clearances $0 < d < 55$ mm, the control effect of buffered impact damper is always better than that of conventional rigid impact damper with the difference becoming more significant when the clearance is small.

Finally, the effect of the amplitude of random excitation on control effect is investigated. Figs. 13(a), (b) and (c) present a comparison of the PSDs of the responses in order of increasing excitation amplitude. The mass ratio and clearance taken in this case are $\mu = 0.082$ and $d = 20$ mm, respectively. It is clear that as the amplitude of excitation increases, so the response increases over the whole frequency range. It can be seen that for different levels of excitation the control effect of the buffered impact damper is always better than that of rigid impact damper at the resonant frequency. However, the response becomes slightly worse at frequencies below the resonant frequency, as has been found previously (Figs. 7, 9 and 10).

3.3. Buffer and contact characteristics

The macroscopic experimental results already presented demonstrate that the buffered impact damper is more effective in vibration attenuation and less sensitive to variations in excitation type and the parameters of the damper itself than a conventional rigid impact damper. Additionally, the peak accelerations (and hence the collision force) of the primary structure equipped with a buffered impact damper are much smaller than with a rigid impact damper and, indeed, smaller than the uncontrolled structure with no impact damper.

There is no doubt that the improved performance of the buffered impact damper must arise from the buffer itself. The characteristics of the contact between the impact mass and the stop are altered by the introduction of a buffer. To understand this behaviour further, buffers of different materials (as defined in Table 1) have been investigated and compared with the behaviour with no buffer, i.e. the conventional rigid impact damper. For these tests, sinusoidal dwell excitation was provided to the base of the structure at a frequency of $f = 4.03$ Hz (i.e. at the natural frequency of the structure). The mass ratio and clearance used are $\mu = 0.082$ and $d = 20$ mm, respectively. Fig. 14 shows the PSDs of the acceleration response of the primary structure

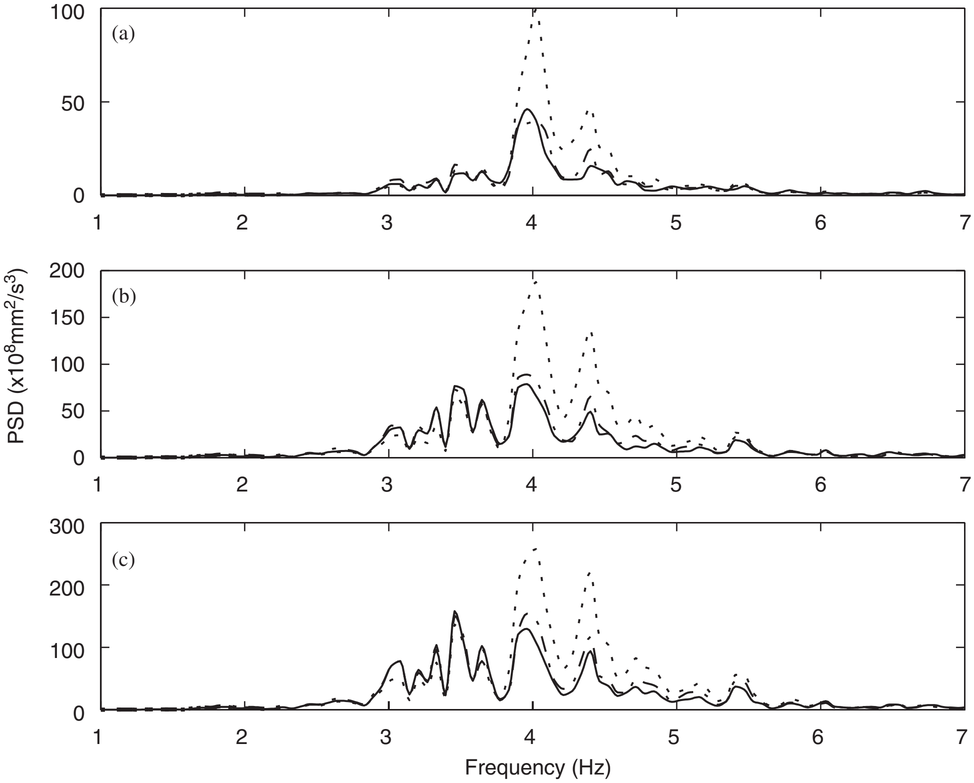

Fig. 13. Effect of increasing amplitude, random excitation (a) small amplitude, (b) medium amplitude, (c) large amplitude. ⋯⋯ Without damper, – · – · – with damper, —— with buffered damper.

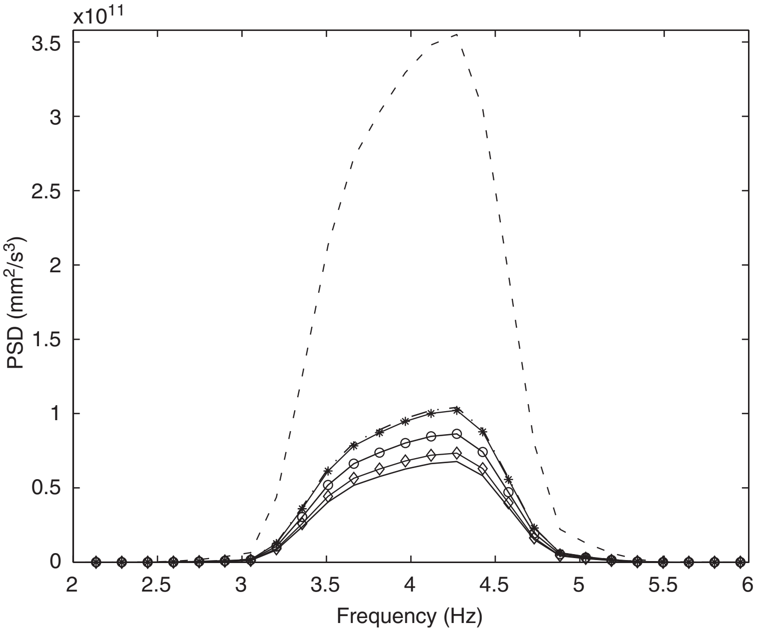

Fig. 14. Performance of different buffers under sinusoidal excitation at natural frequency. ⋯⋯ Without damper, ——damper with buffer 1, ◇ damper with buffer 2, ○ damper with buffer 3, * damper with buffer 4, – · – · – damper without buffer.

when without a damper, with the conventional rigid impact damper and with a series of different buffered impact dampers, as defined in Table 1. Buffers 1–4 are made of progressively stiffer materials. The rigid impact damper can be thought of as a limiting upper bound case for the buffered impact dampers (although, in reality, there is clearly some elasticity/plasticity associated with the stop). It can be seen that with buffer 1, the least stiff buffer, the control effect is the best whilst with buffer 4 the control effect is only slightly better than the rigid impact damper. However, all give a significant level of control over the case with no impact damper. Figs. 15(a), (b), (c), (d) and (e) give the time history of the acceleration response of the primary structure when with buffers 1, 2, 3, 4 and no buffer (rigid impact damper), respectively, compared with the response with no damper. Acceleration spikes can be seen in the sinusoidal acceleration profile which correspond to collision between the impact mass and a stop. These spikes get progressively larger as the stiffness of the buffer material is increased. However, it can be seen that of the four buffers, only in the case of buffer 4 are the accelerations of the primary structure bigger than that without a damper. The accelerations for the case of the rigid impact damper are significantly greater at the moment of impact than for the uncontrolled case.

Figs. 16(a)–(e) show the contact force measured with buffers 1–4 and when without a buffer (the rigid impact damper), i.e. with an increasing stiffness of impact surface. Figs. 16(a1)–(e1) focus in on a single impact for the five cases. It should be noted that, for clarity, both the time and force scales are different for each plot. It is clear that as the stiffness of the buffer decreases, the maximum collision force decreases. The force in the case of buffer 1 is two orders of magnitude smaller than for the rigid impact damper. However, the contact

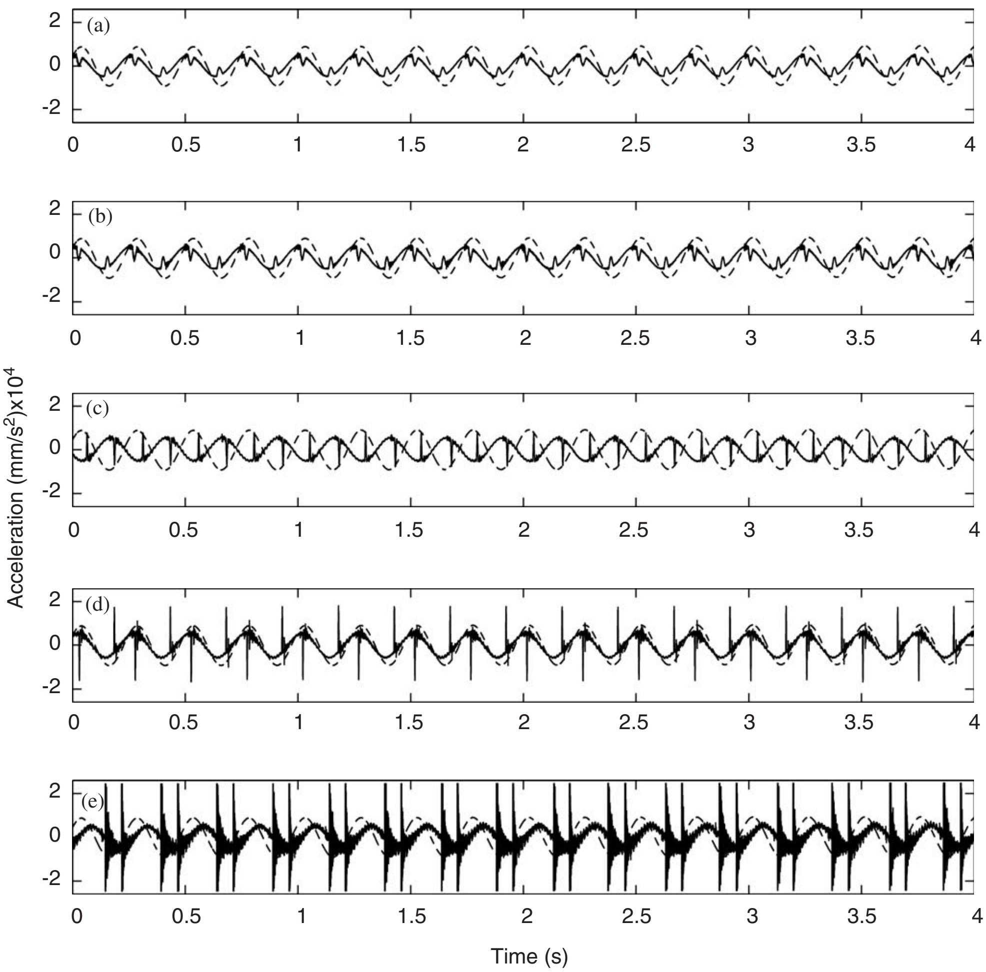

Fig. 15. Acceleration response under sinusoidal excitation with (a) buffer 1, (b) buffer 2, (c) buffer 3, (d) buffer 4, (e) no buffer.---- Without damper, —— with damper.

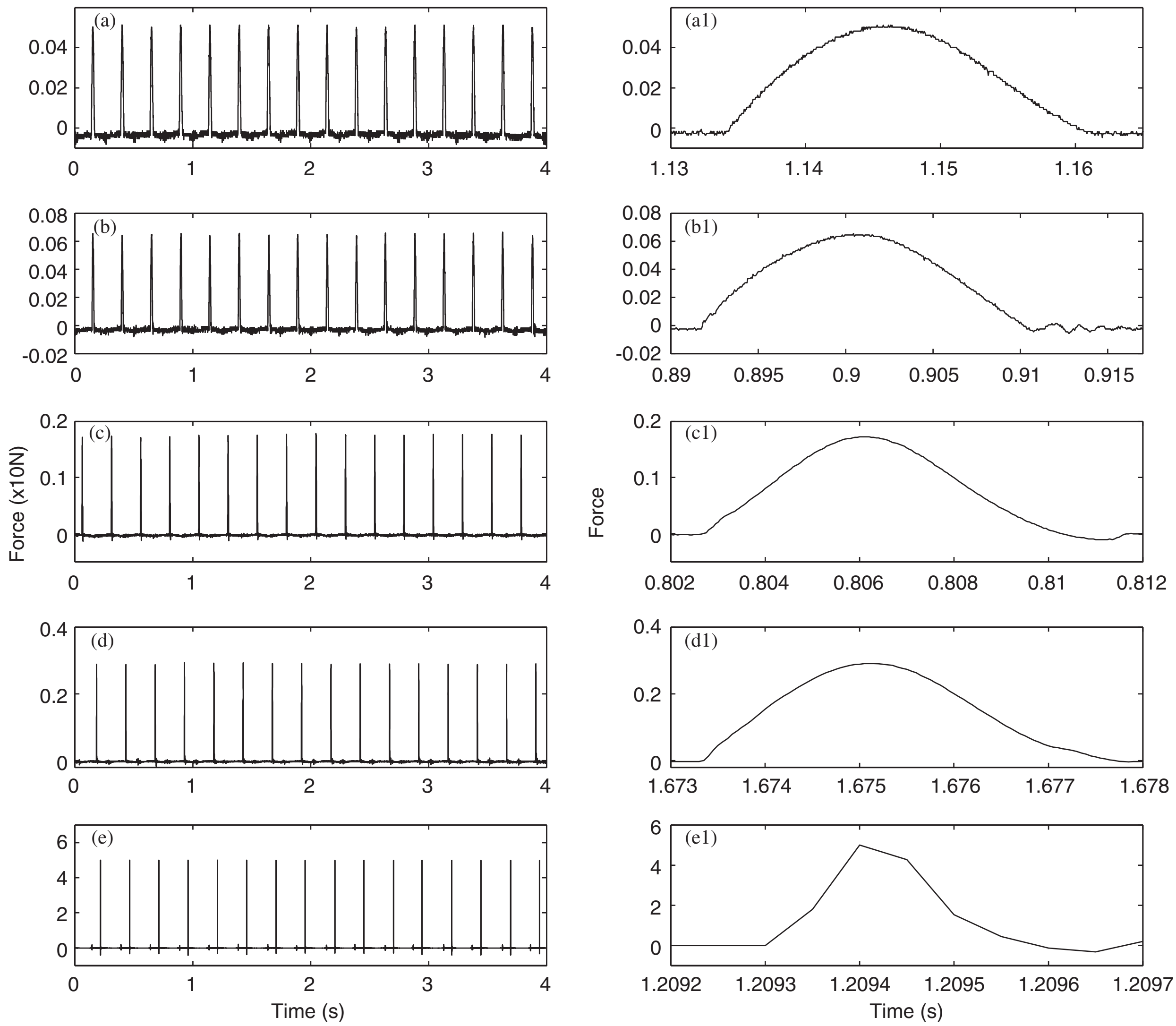

Fig. 16. Contact force under sinusoidal excitation with (a) buffer 1, (b) buffer 2, (c) buffer 3, (d) buffer 4, (e) no buffer. (a1) to (e1) are the corresponding contact force for a single impact.

time, defined as the duration of time while the impact mass stays in contact with the stop, increases as the buffer stiffness decreases, as might be expected. The contact time for the rigid impact damper is approximately 0.0003 s, while for buffers 1–4 the contact times are approximately 0.0255, 0.019, 0.004 and 0.003 s, respectively. Hence, the contact time for buffer 1 is two orders of magnitude greater than for the rigid impact damper.

The vibration control effect of an impact damper comes from collisions of the impact mass with the stops, resulting in exchange of momentum. For a single collision, the impulse momentum relationship is given by

$$I = M(V^{+} - V^{-}) = \int_{0}^{T_c} f_c(t)\,\mathrm{d}t. \tag{1}$$

Here M is the mass of the primary structure and V^{+} and V^{-} represent the velocity of the primary structure immediately before and after the collision, respectively. T_c and $f_c(t)$ represent contact time and contact force, respectively. The effect of a collision upon the structure depends upon the impulse momentum, I. This, in turn, depends not only upon the contact force, but, importantly, also on the contact time. According to Hertzian impact theory [10], for an elastic collision the contact time T_c is proportional to $(X_1 + X_2)^{2/5}$ where X_1 and X_2 are elastic coefficients for the two bodies (i.e. the impact damper mass and the stop). The elastic coefficient of a buffered stop, say X_2, is much higher than that of an unbuffered stop (usually metal). Therefore, unsurprisingly, there is an increase in contact time by the addition of a buffer and hence a corresponding reduction in contact force resulting in lower accelerations and reduced damage during impacts.

To make further comparison, Fig. 17(a) shows the ratio of the impulse momentum of one collision (obtained by numerical integration), where I_0 is without a buffer and I is with a buffer, and Fig. 17(b) the ratio of the control effect, where A_0 is the integral of the PSD of the acceleration response of the primary structure with the rigid impact damper and A is with a buffered damper. It can be seen that the impulse momentum ratios get progressively higher as the buffer stiffness decreases, resulting in better control effect (i.e. lower A/A_0 ratios).

The improved performance of a buffered impact damper (i.e. the high I/I_0 ratio) possibly stems from the elastic deformation of the buffer. This can be seen by examining the coefficient of restitution, e, defined as

$$e = \frac{v^- - V^-}{V^+ - v^+}, \tag{2}$$

where v^+ and v^- are the velocities of the impact mass immediately before and after impact, respectively. A coefficient of restitution of 1.0 represents perfectly elastic collision with no damping occurring by the impact process itself. At the other extreme, a coefficient of restitution of zero represents a perfectly plastic collision where all energy is dissipated in the form of plastic deformation (and only a single impact would occur). In between these two extremes, a coefficient of restitution less than 1.0 is caused by a combination of elastic, plastic and/or viscoelastic deformation. Chatterjee et al. [11] found that, in the case of an impact damper for control of a forced oscillator, the value of coefficient of restitution should be as high as possible to achieve a maximum attenuation.

To calculate the coefficient of restitution between the impact mass and the buffer, a series of experimental measurements were made by dropping the impact mass from a series of predetermined heights onto the buffer and measuring the corresponding rebound height, using a high-speed video camera. From this information the coefficient of restitution for each buffer could be calculated. It was found that, for the drop heights and, hence, impact velocities examined, the coefficient of restitution remains approximately constant. These impact

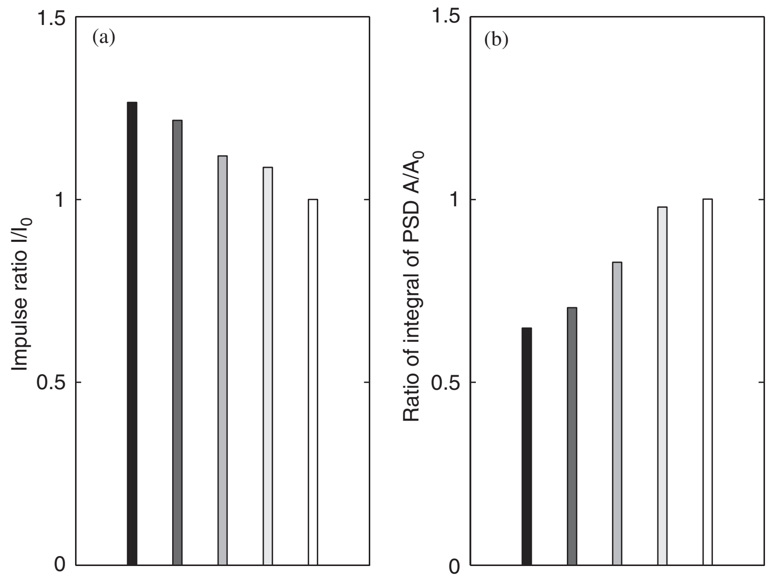

Fig. 17. (a) Ratio of impulse momentum between buffered and unbuffered damper and (b) ratio of the integral of the PSD of the response between a buffered and unbuffered damper. ■ Buffer 1, ■ buffer 2, ■ buffer 3, ■ buffer 4, □ no buffer.

velocities were of the order encountered by the impact mass for the tests described in this paper. The results show that, for the system under investigation, the coefficient of restitution for impact between a metal ball and buffer 1 is 0.61 while the coefficient of restitution for impact between a metal ball and the unbuffered metal stop is 0.46. Therefore, significant plastic deformation occurs during impact of the unbuffered damper (as one might expect for metal on metal impact) whilst impact with the buffered damper is more visco-elastic in nature. Whilst it might be thought desirable that energy is dissipated during the impact itself, it is more important that the impact mass has a high velocity following impact. This means that more kinetic energy is transferred from the structure to the impact mass (and, thus, the dynamic response of the structure is reduced) and also results in a high impulse momentum will be imparted at the next impact. The impulse momentum of the impact of the damper mass can be defined according to the following equation:

$$I = m(v^+ - v^-), \tag{3}$$

where m is the mass of the damper mass. The impulse momentum defined in Eq. (3) is equivalent to impulse momentum defined in Eq. (1). Assuming v^+ and V^+ act in the opposite sense before collision and v^- and V^- act in the same sense following collision (i.e. the impact mass starts moving in the same direction as the structure) then it follows that if v^- remains high following collision then, equating Eqs. (1) and (3), V^- must become correspondingly smaller.

If significant energy is absorbed during the impact (i.e. if the coefficient of restitution is small) then the post impact velocity is small and the subsequent impulse momentum will be smaller and less kinetic energy is transferred to the impact mass. The more elastic contact of the buffered damper provides a higher coefficient of restitution and therefore higher post impact velocity of the impact mass, greater impulse momentum and lowers the velocity of the structure more efficiently at each collision.

It should also be noted that, due to the rolling behaviour of the impact mass, there will be frictional energy loss during impact. At the moment of impact the rolling motion results in friction between both the mass and the stop or buffer, and the mass and the groove in the beam, until the mass begins rolling in the opposite direction. The friction will result in a slight (usually negligible [12]) change in the coefficient of restitution and energy dissipation due to sliding. This frictional component will depend upon the velocity of the ball, the material of the stop or buffer and the stiffness of the buffer. Whilst this energy loss might potentially be significant, it is neglected in this study. This appears to be justified by comparison between experimental results and theoretical models developed by the authors [9] which neglect frictional effects. The theoretical and experimental results appear to match very well despite neglecting these frictional effects. It can therefore be assumed that the frictional energy loss is indeed negligible compared with the visco-elastic or plastic deformation losses within the buffer/stop material itself.

4. Conclusions

Free and forced-vibration experiments demonstrate that a buffered impact damper results in better vibration control than a conventional rigid impact damper. Additionally, with such an impact damper, the accelerations and contact forces are significantly reduced (the accelerations in the case of a buffered impact damper can be smaller than that without a damper even at the point of impact), together with a corresponding reduction in generated noise. Moreover, the buffered impact damper is less sensitive to variation of excitation type and clearance and mass parameters of the damper itself, and results in quicker attenuation in the free vibration response. All these features make the buffered impact damper not only attractive but also practical for civil engineering applications.

Preliminary investigations into the contact characteristics show that the impulse momentum and duration of a collision, rather than the contact force itself, are the important factors in the design and behaviour of such a damper. A higher coefficient of restitution, given by an elastic buffered damper results in higher impulse momentum and increased transfer of kinetic energy from the structure to the damper mass. However, since collision duration is also important, a simple coefficient of restitution model is not suitable for modelling such impact behaviour. Rather, a spring-damper model of the buffer is more appropriate, provided that the stiffness and damping characteristics can be defined. This is the subject of further research.

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