



SHAKE TABLE TESTS OF USING SINGLE-PARTICLE IMPACT DAMPER TO REDUCE SEISMIC RESPONSE

S.M. Zahrai^{1*} and Amir F. Rod²

¹Center of Excellence for Management and Engineering of Civil Infrastructures, School of Civil Engineering, University of Tehran

²School of Civil Engineering, University of Tehran

Received: 20 February 2014; **Accepted:** 12 November 2014

ABSTRACT

Whilst the use of impact damper for single-degree-of-freedom (SDOF) systems has received a significant amount of attention, investigations into their interaction with SDOF systems is rare. In this paper, results of a series of experimental analysis of a horizontal particle impact damper under earthquake and sinusoidally excited primary system are investigated. Specific impact damping is determined for a SDOF. The influence of some system parameters such as clearance, stiffness and specific intensity of excitation are investigated experimentally. Driven by the experimental observation, it has been shown that a value of damping capacity was reached with a particle impact damper. The obtained results proved the efficiency of this process for achieving high structural damping; the results in some instances did not correspond with those found for control of SDOF systems in particular.

One of the main contributions of this investigation involves studying the sensitivity to stiffness. Increased stiffness of the vibration system did not necessarily lead to an increase in damping for all modes. Also, the effect of clearance is unpredictable. In short, the achievements gained in the paper have revealed dynamic characteristics, bringing important engineering application. The paper raises these and other issues which require consideration if impact dampers are to be used to control the dynamic response of SDOF systems.

Keywords: Particle impact damper; vibration response; clearance influence; shaking table test; earthquake.

1. INTRODUCTION

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, and reduces the

*E-mail address of the corresponding author: mzahrai@ut.ac.ir (S.M. Zahrai)

vibration of the primary structure through momentum transfer by collision and dissipation of kinetic energy as heat, noise and high frequency vibrations [1].

It has been shown that impact dampers have proven to be more effective than many other techniques in mitigating the response of a damped structure under dynamic loading [2]. A challenge in the research of impact dampers has been the mathematical complexity in optimizing their design.

In the literature, many applications of impact damper were investigated and accurate approximations have been studied over the years to optimize the design of an impact damper. The idea of reducing the systems vibration by attaching a container, in which a solid particle is constrained to oscillate, was conceived and investigated in 1945 by Lieber and Jensen [3]. They assumed that the motion of an undamped single degree of freedom oscillator with an operating impact damper was still simple harmonic; that the impact of the primary system with the particle was completely plastic and during a period of the sinusoidal forcing function, two impacts occurred at equal time intervals and at opposite sides of the container. Masri (1965) illustrated the general behavior of a single particle impact damper [4]. Results of the analysis were supplemented and verified by experimental studies with a mechanical model and an analog computer. Bapat et al. [5] presented an exact approach to study the number of impacts in a single period oscillation. Hoang and Semercigil [6] proposed a single unit of particle damper to control the excessive transient vibrations of a single-link flexible robot arm for transient excitation, e.g., after a sudden stop or after hitting a stop block. In 1996, Cuvalci and Ertas studied the autoparametric interaction between the first two modes of a pendulum absorber for a nonlinear system of varying orientation. Energy transfer between the modes was shown and the experimental results were in agreement with the theory [7]. The authors demonstrated experimentally that the damper was capable of attenuating up to 95% of the vibrations, with a simple, self-contained construction and without any obstruction to the tasks to be performed by the robot.

The harmonic excitation of a multi-particle impact damper in a single cavity was experimentally studied by Papalou and Masri [8]. They developed an approximate method to predict the damping attained by dampers filled with steel balls of various sizes and showed that, compared with a single-particle damper; a multi-particle impact damper reduced noise, surface deterioration, and improved the effectiveness. Another application of impact damper in motorized hang gliders was conducted by Oledzki et al. [9], where they investigated impact phenomena between metallic bodies with plastic inserts and found optimum values for mass ratio and clearance. It was observed that small amounts of friction force benefited the efficiency, the proposed design reduced the resonance amplitudes by up to 75 percent. Trigul et al. [10] using experimental observation, have shown that a high value of specific damping capacity could be reached with a particle impact damper. The obtained results proved the efficiency of the process for achieving high structural damping. Henmi and Tanaka [11] investigated the impact damper effectively eliminates residual vibration in the step response. An external impact damper was applied to settle transient amplifying-mechanism vibration and to determine differences in damping by setting damper conditions appropriately.

Zahrai and Rod [12] investigated the effect of single unit impact damper on the single degree of freedom system under sinusoidal and impulse excitations. The effects of mass ratio, coefficient of restitution, and gap size were determined using a MATLAB program.

They found impact damper an efficient device for reducing the vibration of structures subjected to impulsive and harmonic excitations and observed that the response of the system varies with small changes in the particle's mass, the coefficient of restitution and particle's clearance. Dehghan-Niri et al. [13] numerically studied the performance of a single horizontal conventional Impact Damper in both wide range frequency and resonance excitations. The optimal parameters were numerically found by discretely varying the clearance and excitation frequency. The vulnerability of the optimized ID versus uncertainties in structural parameters was clearly determined illustrating that less robustness occurs when the performance of the controller is more efficient.

Although there have been few numerical and experimental research projects which indicate the efficiency of ID parameters, none of these studies had clearly addressed the performance of the optimal single impact damper subjected to different values of stiffness.

This paper takes an experimental approach to investigate the dynamic behavior of an impact damping system. For this purpose, a unit-particle ID was designed, fabricated and used to reduce the vibration amplitude of a single degree of freedom under harmonic and earthquake vibrations. The influence of some system parameters, such as clearance, stiffness and intensity of excitation, on the evolution of specific damping capacity were investigated experimentally.

2. EXPERIMENTAL APPROACH

2.1 Design of test specimen

The essential physical properties of any linearly elastic structural system subjected to dynamic loads include its mass, its flexibility or stiffness properties, its energy-loss mechanism or damping, and the external source of loading or excitation [14]. In the simplest model of a SDOF system, each of these properties is assumed to be concentrated in a single physical element.

A mechanical model resembling a SDOF frame structure was designed and fabricated as a single particle impact damper under realistic laboratory conditions. The structural system considered, is shown schematically in Fig. 1. The experimental model was designed to simulate a single degree of freedom linear oscillator, forming the primary system to be controlled.

The single degree of freedom systems are extremely rare in practice and a sort of idealization most often results from simplifications of the distributions of the essential properties of a mechanical or structural system. These properties are mass, stiffness and damping. The water tank in Fig. 1 can be considered as a SDOF structure idealized as a simple idea of all similar single-degree-of-freedom systems. It represents a rigid body of mass constrained to move along X-axis [14]. Simulation considerations were made in order to enable a direct comparison between equivalent frequencies measured on both model and full scale water tank. The mass is attached to a vertical long column. The external-loading mechanism producing the dynamic response of this system, acting on the mass in a horizontal direction is denoted by $z(t)$. In general, it is comprised of the displacement u of the system from an equilibrium position.

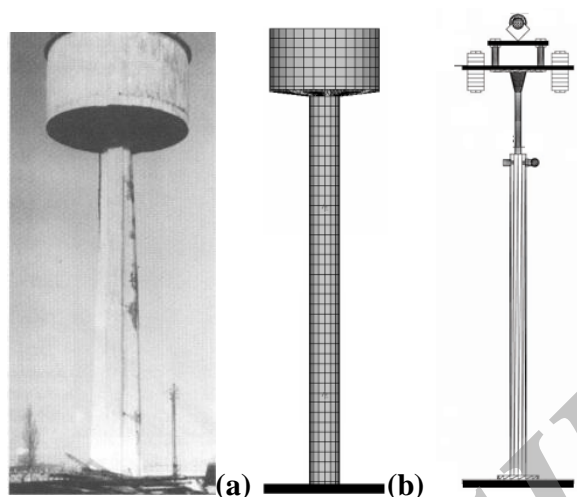


Figure 1. (a) & (b) An air tank, when it is full of water, the structure can be analyzed as a single degree of freedom [15]. (c) Device made using this concept similar to a single degree of freedom system

The primary mass is composed of an adjustable column consisting of three parts made from steel bolted rigidly to a base plate $280 \times 280 \times 6$ mm. The base is fixed to a unidirectional shaking table that is driven by an electro-dynamic shaker to produce base excitation. The main plate on the top is $270 \times 270 \times 5$ mm. The impact mass itself is a steel ball which runs in an 'L'-shaped steel rail/slider angle. This path of the secondary mass is fixed to the top plate that can be leveled by four screws and nuts. Two rectangular shaped brackets are mounted to the stops. The clearance between the stops (and hence, the brackets) is adjustable. It is noteworthy that different materials can be fixed by adhesive to the stops and easily removed and changed. Using a highly deformable stop, the impact noise is greatly reduced. The coefficient of restitution values are changed by different materials.

The primary and secondary masses oscillate in the same line. This assembly acts as a primary mass and its mass can be altered by the external plates attached. Two symmetric holes are placed on main plate for this purpose.

2.2 Test setup and instrumentation

As explained in previous section, a unit impact damper was designed, fabricated and used to conduct the experiments. The design allowed for a variation of parameters to be studied. The test setup (Fig. 2) is composed of no mechanical components. The schematic shape and dimension of components are shown in Fig. 3.

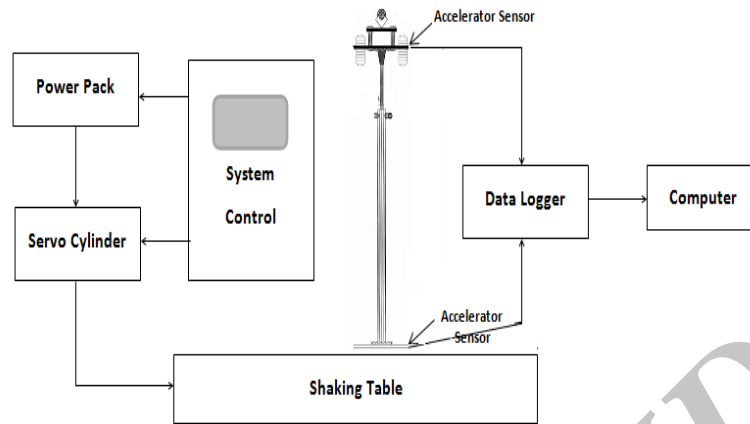
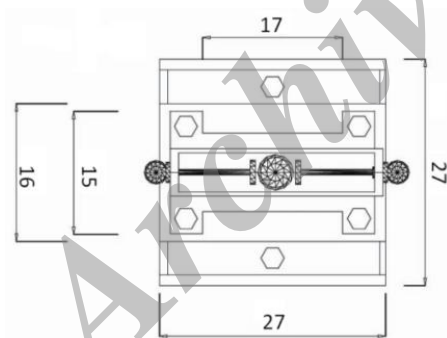


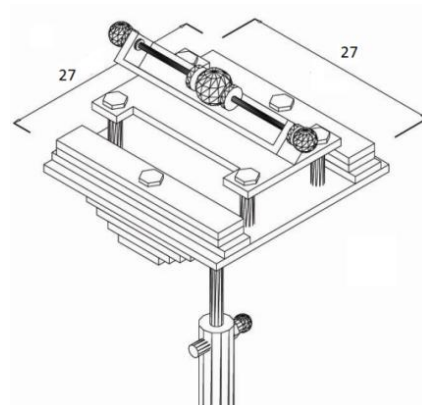
Figure 2. Experimental set-up

Forced-vibration experiments were carried out using both sinusoidal and the Kobe earthquake simulation excitations.

For the swept sine excitation vibration test, the drive signal to the shaker was a sine wave that had excitation amplitude of 0.4g and constant excitation frequency of 7 Hz. The maximum test frequency was the physical limitation of the shaker. The acceleration of the test model was recorded and processed by using “Origin 5”, computer software to obtain response of the system. It is a program appropriate for interactive scientific graphing and data analysis.



(a) Top View



(b) General View

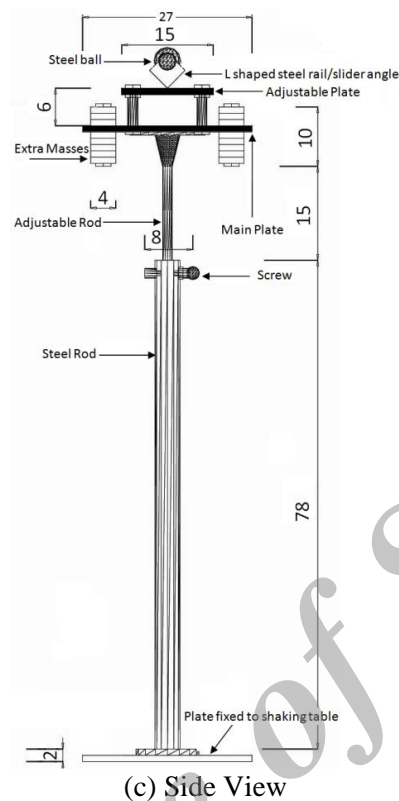


Figure 3. Schematic views of test specimen and setup prepared for experiments (dimensions in cm)

Indicated parameters were changed for each experiment conducted. The parameters studied in this research were the length of impacting walls, and stiffness of the primary system.

The response of the structure was monitored using two accelerometers. Acceleration sensors were used to record the acceleration response of the base. The output signal from the accelerometers was applied to an analyzer via an amplifier. Two piezoelectric low inductance accelerometers were placed on system to measure the output from the software. The acceleration measurement points were located at one of the impact damper ending wall, and bottom of column connected to shaking table to measure the force transmitted to the ground. The signal conditioner was to send the acceleration readings from the accelerometers. The steel ball placed in “L-shape” angle was supposed to impact the steel end walls.

In this experiment, it was possible to measure the acceleration response in order to reduce peak levels utilized for energy analysis. The displacement response and the associated power spectrum could be obtained accordingly.

The coefficient of restitution and contact time both remained approximately constant for the range of velocities occurring in the tests.

A photo of the experimental apparatus is shown in Fig. 4. A sliding still pad was used to adjust the distance through which the impacting mass was allowed to travel. To reduce the friction, this “L shaped” angle could be covered by two pieces of glass.



Figure 4. View of experimental apparatus

Fig. 5 shows a photograph of the experimental setup. Note that the structure of the model is not chain-like, consequently the linearized system stiffness matrix is not banded. The exact values of the system clearance, damping, and stiffness matrices correspond to an infinitesimal (small oscillation) range of the motion in the neighborhood of the position of static equilibrium.

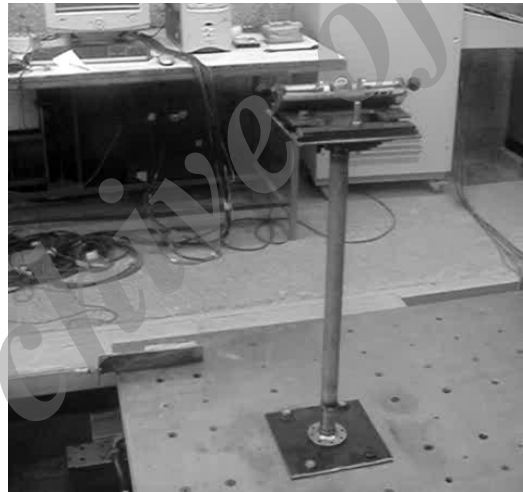


Figure 5. Primary system with impact damper installed on the shake table

3. MATHEMATICAL MODEL

The equation of motion was input into the “*Matlab*” inverse problem algorithm.

We assume that the displacement relative to the ground can be expressed by:

$$u(x, t) = \varphi(x)z(t) \quad (1)$$

The total displacement (Fig. 6) of the system is:

$$u^t(x, t) = u(x, t) + u_g(t) \quad (2)$$

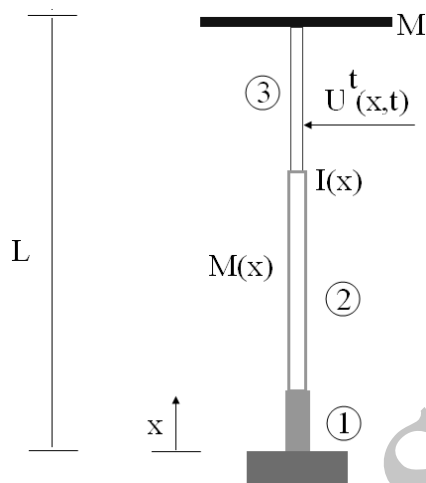


Figure 6. Simplified model of the primary system

The shape function $\varphi(x)$ in Eq. (1) must satisfy the displacement boundary conditions. For this system, these conditions at the base of the system are:

$$\varphi(0) = 0, \quad \dot{\varphi}(0) = 0$$

The deflections of the uniform system with flexural rigidity EI due to a unit lateral force at the top (Fig. 7) are [14]:

$$u(x) = (3Lx^2 - x^3)/6EI \quad (3)$$

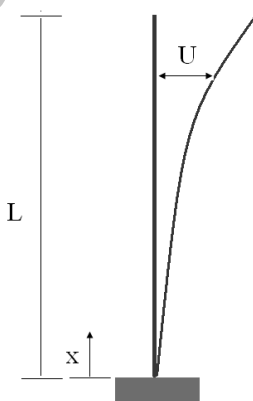


Figure 7. System deflections due to excitation at the base of primary system

If we select the generalized coordinate as the deflection of the some convenient reference point, e.g. the top of the system, then

$$Z = u(x) = L^3/3EI \rightarrow u(x) = \varphi(x)Z \quad (4)$$

$$\varphi(x) = \frac{3}{2} \times \frac{x^2}{L^2} - \frac{1}{2} \times \frac{x^3}{L^3} \quad (5)$$

This $\varphi(x)$ automatically satisfies the displacement boundary conditions at $x=0$ because it was determined from static analysis of the system. The Eq. (5) may also be used as the shape function for nonuniform tower, although it was determined for a static force, and it could be assumed directly; possibilities are [14]:

$$\varphi(x) = 1 - \cos \frac{\pi x}{2L} \quad (6)$$

$$\varphi(x) = \frac{x^2}{L^2} \quad (7)$$

The three shape functions above have $\varphi(L) = 1$, although this is not necessary. The accuracy of the generalized SDOF system formulation depends on the assumed shape function $\varphi(x)$ in which the structure is constrained to vibrate [14].

We proceed to formulate the equation of motion for the system. The equation of dynamic equilibrium of the generalized SDOF system can be formulated conveniently only by work or energy principles. The principle of virtual displacement is used. It states that if the system in equilibrium is subjected to virtual displacement, the external virtual work δW_E is equal to the internal virtual work δW_I .

Having obtained the final expressions for δW_E and δW_I it is obtained:

$$\tilde{r} = \delta z [\tilde{m}\ddot{z} + \tilde{k}z + \tilde{L}\ddot{u}_g(t)] = 0 \quad (8)$$

For this generalized SDOF system, the generalized mass \tilde{m} , generalized stiffness \tilde{k} , and generalized excitation $-\tilde{L}\ddot{u}_g(t)$ are defined by [14]:

$$\tilde{m} = \int_0^L m(x)[\varphi(x)]^2 dx \quad (9)$$

$$\tilde{k} = \int_0^L EI(x)[\varphi''(x)]^2 dx \quad (10)$$

$$\tilde{L} = \int_0^L m(x)\varphi(x)dx \quad (11)$$

Since $\delta z[\tilde{m}\ddot{z} + \tilde{k}z + \tilde{L}\ddot{u}_g] = 0$ is valid for every virtual displacement δz , it is concluded that:

$$\tilde{m}\ddot{z} + \tilde{k}z = -\tilde{L}\ddot{u}_g \quad (12)$$

This is the equation of motion for the system assumed to deflect according to the shape function $\varphi(x)$.

Dividing equation (12) by \tilde{m} gives:

$$\ddot{z} + 2\xi\omega_n\dot{z} + \omega_n^2 z \quad (13)$$

Where $\omega_n^2 = \tilde{k}/\tilde{m}$ and a damping term using an estimated damping ratio has been included. This equation is for an SDOF system, except for the factor:

$$\tilde{r} = \frac{\tilde{L}}{\tilde{M}} \quad (14)$$

The above equations are used to find the optimized system parameters by changing the lateral stiffness as shown in Fig. 8).

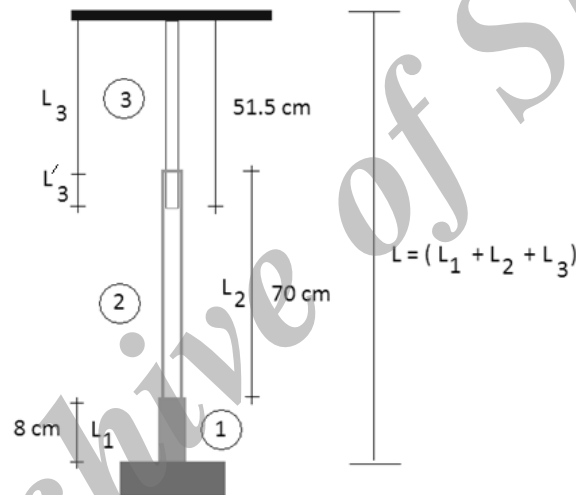


Figure 8. Division of primary system to analyze the lateral stiffness

4. TEST RESULTS & COMPARISON

The test structure was excited through the movement of the shaking table on which the test structure was fixed. A simple Proportional Integral Derivative (PID) feedback controller was used to control the shaking table, ensuring that the correct movement was produced and repeated. The sinusoidal and the Kobe earthquake simulation excitations described above were repeated for the structure with and without an impact damper. The control of a single degree-of-freedom oscillator by using an impact damper is studied experimentally here in three different stiffness values of the system by changing L_3 values. Table 1 shows stiffness of the system in three different values of L_3 .

The experimental results are compared to each other to determine the key parameters of the model under consideration. Experimental results can create the values of system parameters to reproduce the numerical simulation of the equation of motion as closely as possible.

Table 1: System parameters varying by changing length of part L_3

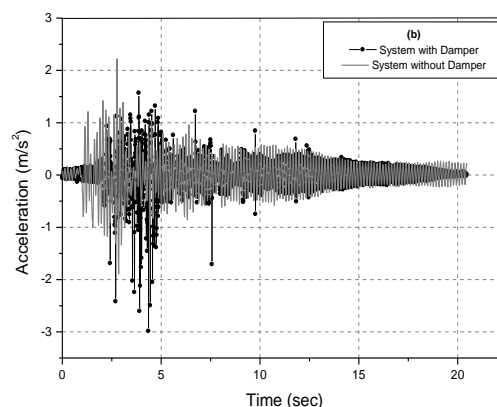
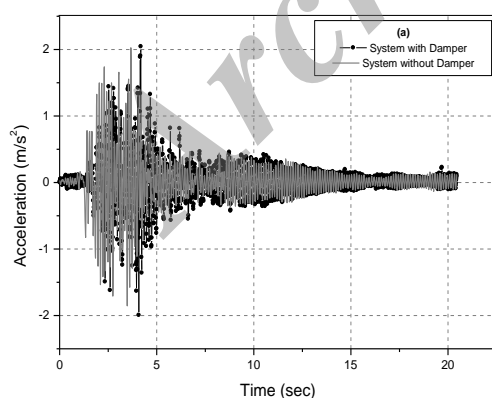
Length of part L_3 (cm)	Lateral stiffness of system \tilde{k} ($\frac{kN}{m}$)	Natural Frequency ω_n ($\frac{rad}{s}$)	Generalized mass \tilde{m} (kg)	Period of the system T (s)
10	69.85	102.0	6.71	0.061
20	49.5	86.2	6.64	0.062
30	36.14	73.0	6.61	0.85

4.1 The Kobe earthquake simulation excitation

The primary structure is excited by setting it to the Kobe earthquake excitation. For this simulation the largest recorded peak accelerations which were about 0.8g was chosen. It was found that, for six aftershocks, horizontal peak acceleration recorded at three site located on a thin layer of soft alluvium (about 10 to 15 meters thick) were about 3 to 5 times larger than those at a reference site on rock, and the Fourier spectral amplitudes for frequencies between 2 and 3 Hz were as much as 20 times higher [16].

The mass ratio is taken as and the clearance (defined as the distance between the stops subtracted by the diameter of the impact mass) is taken as.

Fig. 9 shows the response of primary system with the same clearance and different stiffness values, under the Kobe earthquake simulation. When the stiffness of system is decreased by increasing the length of steel rod, its acceleration will be decreased effectively by utilizing impact damper. While as Fig. 9a shows for stiffer system, the impact damper not only has no proper efficiency but also the acceleration of the system might be even slightly increased too.



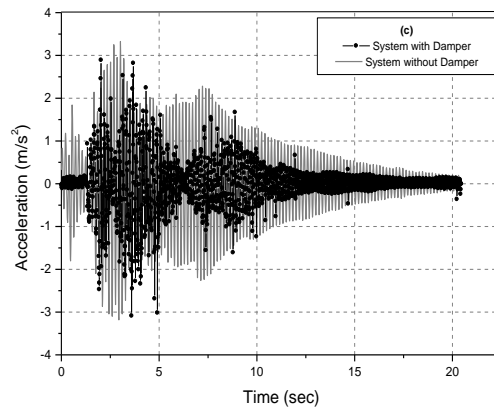


Figure 9. Acceleration response of system under the Kobe earthquake simulation when $d=4\text{cm}$ and a) $L_3=10\text{ cm}$ b) $L_3=20\text{ cm}$ c) $L_3=30\text{ cm}$

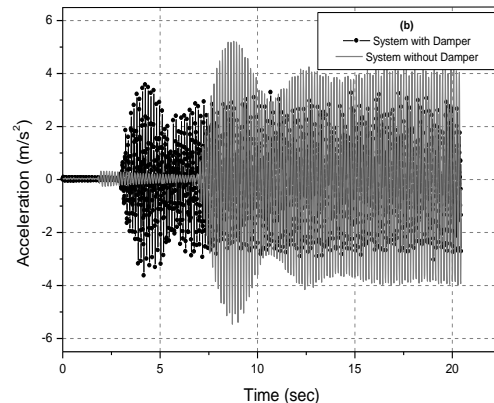
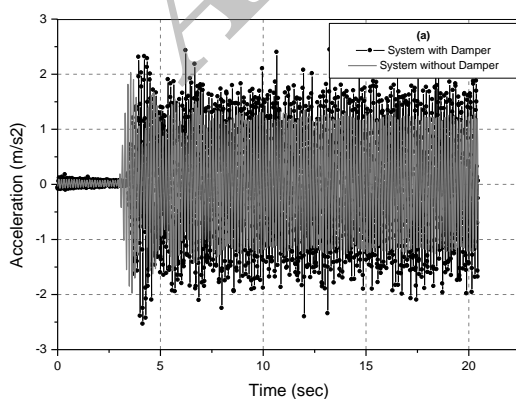
Maximize efficiency of an impact damper between two consecutive impacts was shown in case (c) when the stiffness of the system was in the lowest level. A properly designed single particle impact damper is capable of substantial attenuation of the acceleration response level close to 45%.

4.2 Forced vibration

Sinusoidal base excitation was applied to the structure to investigate the behavior of impact damper further. The mass ratio was chosen to be and optimized clearance was obtained.

The ratio of the mass of the ball to the mass of the primary structure was taken as previous condition and the clearance was taken as.

Fig. 10 shows the time-history of the acceleration responses. High acceleration peaks are observed in Fig. 10, to occur for the primary structure with the impact damper when stiffness is either low or high and impact damper not only shows any proficiency but also the response of the system becomes worse. It is shown that impact damper provides a high level of attenuation in case (b).



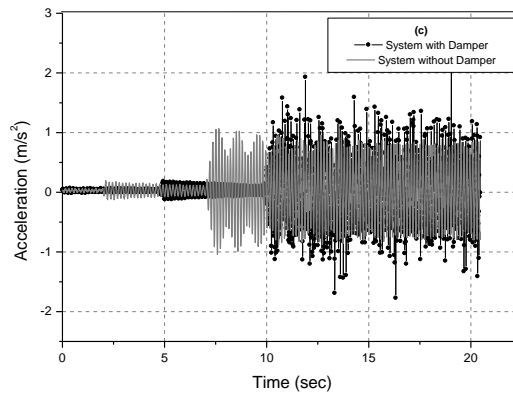
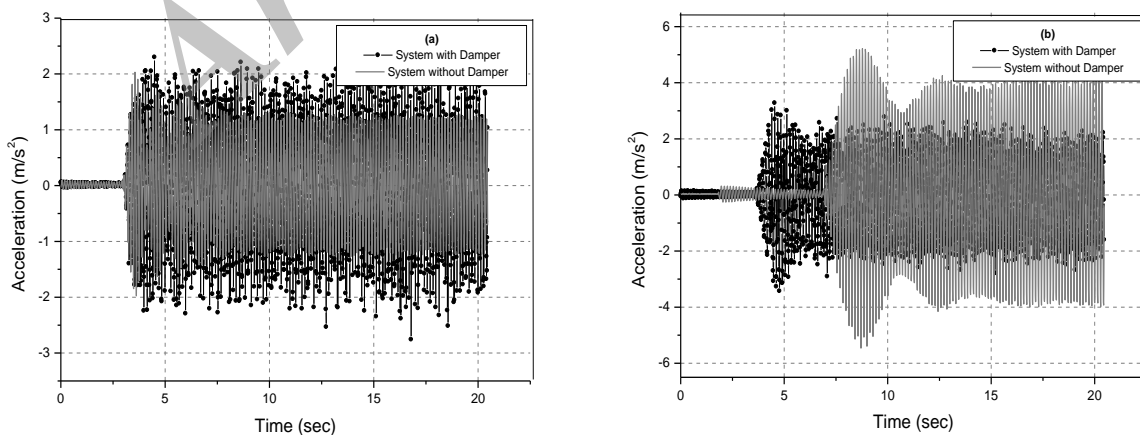


Figure 10. Acceleration response of system under harmonic excitation when $d=1\text{cm}$ and a) $L_3=10\text{ cm}$ b) $L_3=20\text{ cm}$ c) $L_3=30\text{ cm}$

The same results shown in Fig. 10, were obtained when the structure is excited with the different clearance in Fig. 11. In this case the clearance chosen was and other parameters are the same as Fig. 10. The same conclusions can be reached, i.e., the impact damper reduces the response more effectively with the stiffness of (length of $L_3=20\text{cm}$) acceleration response can be reduced in a specific value of stiffness. It was evident during the tests that the efficiency of impact damper was not shown in lower and higher stiffness.

Fig. 12 shows the corresponding time histories of the acceleration responses. The clearance was changed to. By increasing clearance, response of the system will be improved. Comparing to Fig. 11 clearly demonstrates the efficiency of impact damper can be shown with the optimized clearance in the specific stiffness.

Finally, the effect of clearance was investigated with the mass ratio and a clearance of . The results, shown in Fig. 13, demonstrate that the optimized ID controls the structure better (close to 43%) than previous case by clearance parameter of (less than 10%).



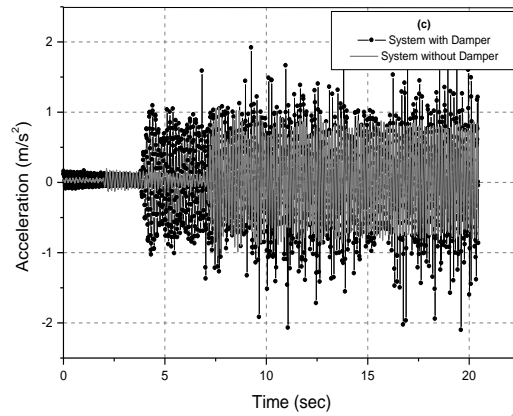


Figure 11. Acceleration response of system under harmonic excitation when $d=3\text{ cm}$ and a): $L_3=10\text{ cm}$ b): $L_3=20\text{ cm}$ c): $L_3=30\text{ cm}$

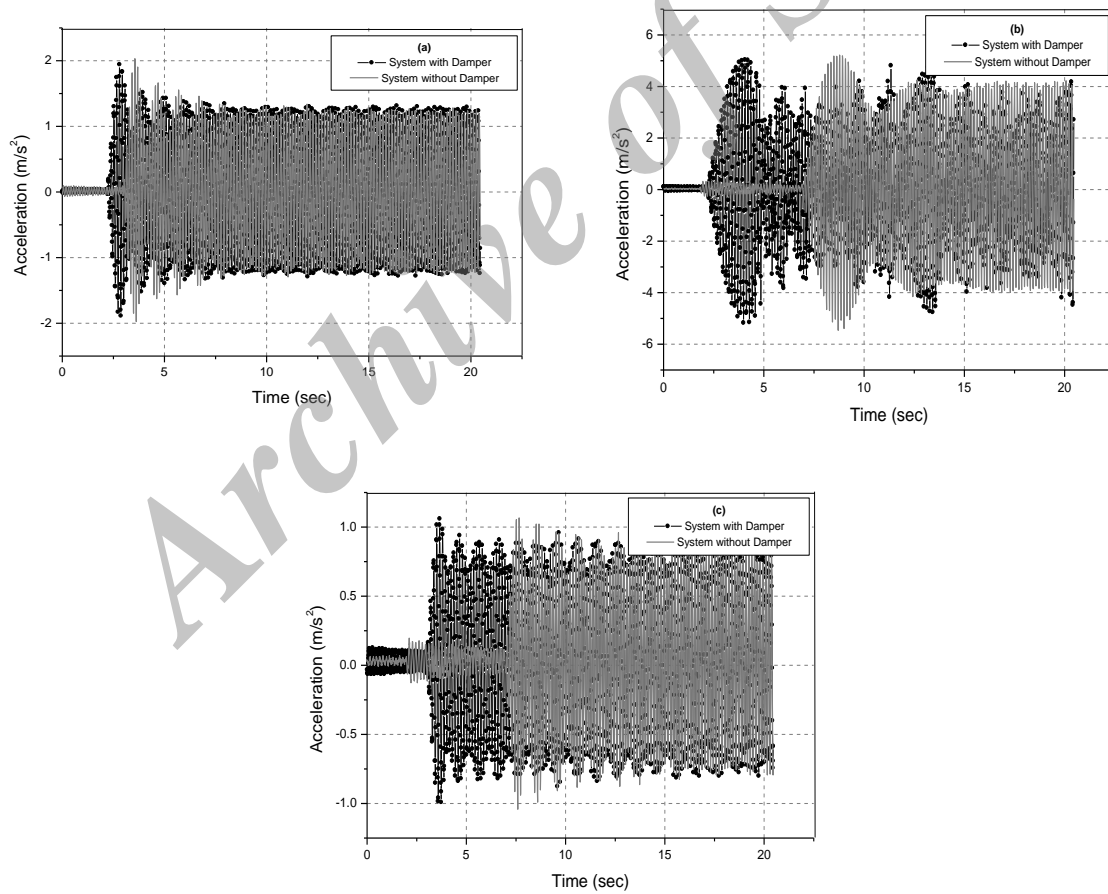


Figure 12. Acceleration response of system under harmonic excitation when $d=10\text{ cm}$ and a): $L_3=10\text{ cm}$ b): $L_3=20\text{ cm}$ c): $L_3=30\text{ cm}$

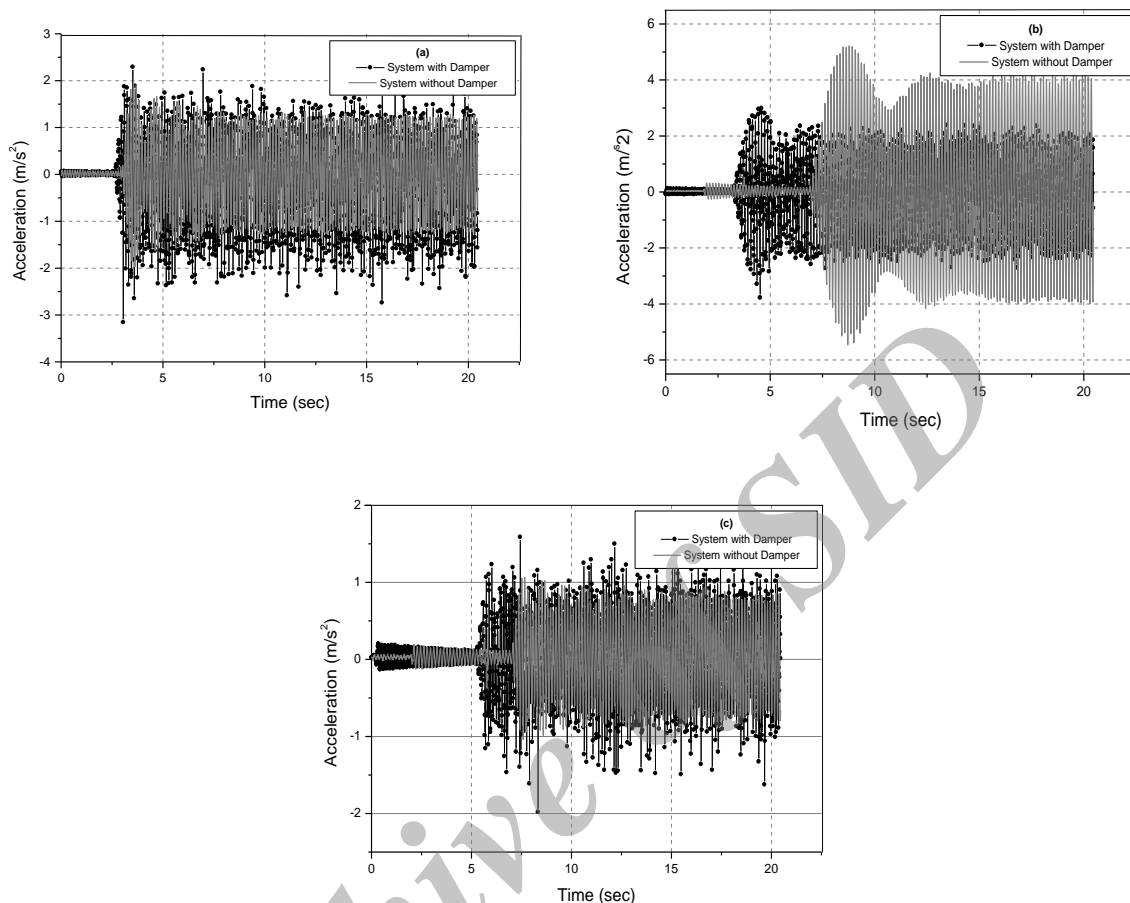


Figure 13. Acceleration response of system under harmonic excitation when $d=15\text{ cm}$ and a): $L_3=10\text{ cm}$ b): $L_3=20\text{ cm}$ c): $L_3=30\text{ cm}$

5. CONCLUSION

In this paper, a passive control is presented to suppress the residual damped vibration for an equivalent single degree of freedom system. Basically, for the SDOF system, a still ball is inlaid in its closed cavity without any restricts, applying the impact damping to the damped vibration of the system. The results show that the inlaid steel ball exhibits a significant impact damping effect on suppressing the acceleration demand of the system, especially at the specific values of clearance and stiffness of the system. By applying sinusoidal excitation at an specific frequency of 7 Hz, and amplitude of harmonic acceleration equals to 0.4g, the effect of clearance and stiffness were investigated.

Previous studies had not examined such stiffness and instead have just focused primarily on the other parameters. This is one of the rare investigations considering the influence of stiffness on conventional impact dampers. There was an optimum clearance for damper whereby the maximum attenuation could be achieved. Comparing four values of clearance

in three different values of stiffness demonstrated that in the harmonic excitation, by reducing the stiffness, the efficiency of impact damper is increased. For efficient operation, the impact damper should have combination of optimized clearance and stiffness values. Accordingly, it was clear from the results that a significant improvement could be achieved in an specific range values of stiffness and clearance. Therefore similar to other system parameters such as coefficient of restitution and mass ratio, increasing or decreasing the optimized values, the efficient configuration could not be achieved through the use of single-particle designed in accordance with the procedure presented in the paper.

Acknowledgement: This research forms part of project sponsored by the college of Engineering at the Univ. of Tehran (project 8108020/1/02). Their support and also help of Soil Mechanic Lab, School of Civil Engineering, University of Tehran for test facilities are gratefully acknowledged by the authors. Any opinions presented in this paper, however, are the authors' alone, and not necessarily the sponsors.

APPENDIX

Nomenclature and Definition of Symbols

ω_n	natural frequency of main system
u	displacement relative to the ground
L	length of the steel column
\tilde{m}	generalized mass
\tilde{k}	generalized stiffness
δz	virtual displacement
\ddot{u}_g	ground acceleration
EL	flexural rigidity
u^{total}	total displacement
Γ	\tilde{L}/\tilde{m}
t	time variable
$z(t)$	generalized displacement
$\varphi(x)$	Shape function

REFERENCES

1. Li K, Darby AP. An experimental investigation into the use of a buffered impact damper, *Journal of Sound and Vibration*, **291**(2006) 844-60.
2. Popplewell N, Liao M. A simple design procedure for optimum impact dampers, *Journal of Sound and Vibrations*, **146**(1991) 519-26.
3. Lieber P, Jensen DP. An acceleration damper: development, design and some

- applications, *Transactions of ASME*, **67**(1945) 523-30.
4. Sami F. Masri. Analytical and Experimental Studies of Impact Damper, PhD thesis, California Institute of Technology, 1965.
 5. Bapat CN, Popplewell N, McLachlan K. Stable periodic motions of an impact pair, *Journal of Sound and Vibration*, No. 1, **87**(1983) 19-40.
 6. Hoang P, Semercigil SE. A passive controller for the flexible robot arm, *Proceedings of the 1992 Engineering Systems Design and Analysis Conference ASME*, (1992) pp. 197-202.
 7. Cuvalci O, Ertas A. Pendulum as vibration absorber for flexible structures: experiments and theory, *ASME Journal of Vibration and Acoustic*, **118**(1996) 558-66.
 8. Papalou A, Masri S. An experimental investigation of particle dampers under harmonic excitation, *Journal of Vibration and Control*, **4**(1998) 361-79.
 9. Oledzki AA, Siwicki I, Wisniewski J. Impact dampers in application for tube, rod and rope structures, *Mechanism and Machine Theory*, **34**(1999) 243-53.
 10. Trigul M, Foltete E, Abbas MS, Fakhfakh T, Bouhaddi N, Haddar M. An experimental study of a multi-particle impact damper, *Journal Mechanical Engineering Science, Part C*, **223**(2009) 2029-38.
 11. Henmi N, Tanaka M. Amplifying mechanism vibration settling by external impact damper, *International Journal of Automation Technology*, No. 3, **3**(2009) 304-7.
 12. Zahrai SM., Rod AF. Effect of Impact Damper on SDOF System Vibrations under Harmonic and Impulsive Excitations, *Journal of Physics*, **181**(2009)012066, doi:10.1088/1742-6596/181/1/012066.
 13. Dehghan-Niri E, Zahrai SM, Rod AF. Numerical Studies of the Conventional Impact Damper with Optimality and Uncertainty Considerations, *Scientia Iranica*, No.2, 19 (2012)166-78.
 14. Clough WR, Penzien J. Dynamics of Structures, Mc-Graw Hill International Editions, Civil Engineering Series, 2003.
 15. Chopra A.K. Dynamics of Structures, Theory and Application to Earthquake Engineering, Third Edition, 2006.
 16. Somerville P. Strong Motion Records from the Kobe, Japan Earthquake of Jan, 17, 1995, and Implications for Seismic Hazards in California, SMIP95 Seminar on Seismological and Engineering Implications of Recent Strong-Motion Data, 1995, pp. 69-82.