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SEISMIC COUNTERMEASURES FOR RAIL-COUNTERWEIGHT SYSTEM IN ELEVATORS

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SUMMARY

Elevators in hospitals and other essential facilities serve very important functions. Several design and sensing improvements have been made to improve the performance of the elevators, but their malfunctions have still been reported in recent earthquake events. The counterweights, being the heaviest component in the system, are most likely to cause damage to their guiding system consisting of the guide rails, bracket supports, and roller guide or guide shoes. In this paper, three protective measures are examined for reducing the seismic response of the rail-counterweight system. The first method is to enhance the damping of the system. This approach has its limitations because of the tight clearances and spaces around the counterweight. By providing small dampers in the roller guide assemblies, the maximum stress in the rail can only be reduced by about 7.5%. The second approach consists of utilizing a part of the dead weights so that it can act as a tuned vibration absorber for the rest of the system. Although this approach can only be used to reduce the in-plane motion of a counterweight, this direction also happens to be the most critically stressed of the two directions of the motion in which counterweights respond. There are some issues concerning the tuning of these devices as the stiffness of the rail-counterweight system can change dramatically due to the nonlinearities of its supporting system, but a tuning frequency ratio between 0.8 and 1.0 is found to be effective. The performance of a passive tuned vibration absorber can be further improved by using it in the active mode by installing an actuator. To regulate such a system, herein a simple linear quadratic regulator is used to define the control algorithm. The optimal control force is calculated using the constant pre-contact system stiffness matrix' although the system stiffness changes constantly with time. The results show that the maximum stress in the rail can be reduced to about half of its original value using an active vibration absorber. The required control force is also within the reasonable limits even for high intensity ground motions.

INTRODUCTION

Almost all modern buildings higher that four stories are likely to use elevators. Several different types of elevators can be found, but the traction elevators with roller guide assemblies are most commonly used in

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multi-story buildings. These elevators have been observed to get damaged in spite of several changes made in the codes which are used to design these systems. Some damages are more serious than others, but they all lead to malfunction of the system. Although no report of serious human casualties have been made, the malfunction of these systems can, indeed, pose serious problems in hospital facilities which are needed immediately after an earthquake event to provide much needed medical care to the victims of the event.

An elevator system consists of several mechanical and electrical components that can get damaged during an earthquake. However, the rail-counterweight systems used in traction elevators are among the most vulnerable components to earthquake-induced motions, primarily because the counterweights are also the heaviest of all components. The counterweights are designed to compensate for the weight of the car plus 40% of the rated payload to be carried by the car.

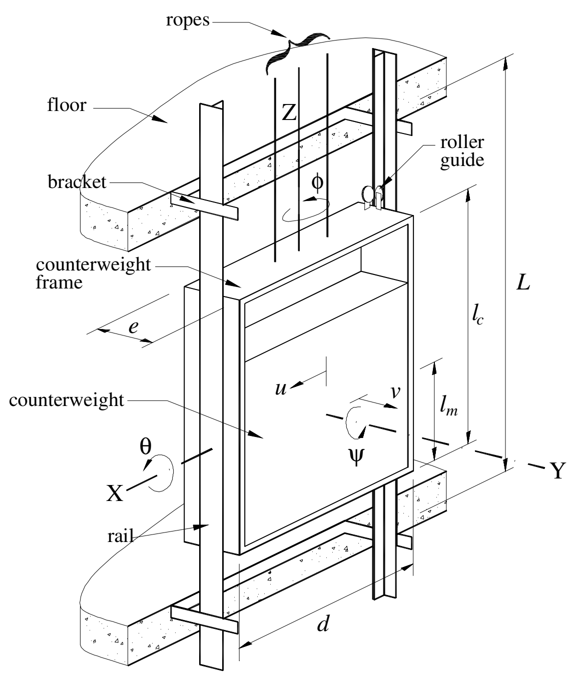


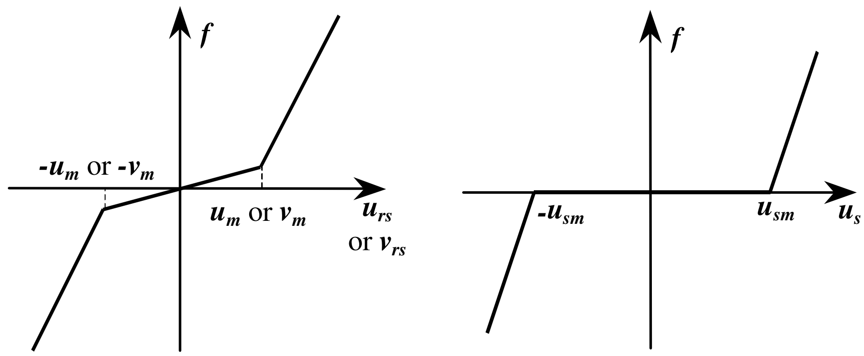
Figure 1. Schematic layout of rail-counterweight system of an elevator.

Figure 1 shows the schematics of the counterweight along with rails and supporting brackets. The brackets are usually supported at each building floor level; they can also be supported at other elevations along the building height if an elevator shaft with side walls is used. The counterweight consists of deadweights stacked over each other in a frame usually made of welded steel channel cross sections. The counterweight is pulled up and down by cables that are operated by a traction motor, usually placed on the top of the elevator shaft. The counterweight frame is guided along the rails by roller guide assemblies provided at each corner of the counterweight frame. The roller guide assemblies consists of usually three roller wheels with rubber tires that are kept in constant contact by pre-loaded helical springs. Some roller guide assemblies with six roller wheels are also available and used. Similar roller guide assemblies, but usually with bigger roller wheels are used to guide the car on separate guide rails. The rubber tires ensure a noise-free comfortable ride for the elevator occupants. To limit excessive deformations of the rubber tires, and also to avoid the tendency of derailments of the roller guide assemblies, the A-17 ASME Code [1] requires the use of restraining plates in each roller guide assembly. The restraining plate has a U-shaped cut to accommodate the guide rail head. The code permits a clearance of 3/16 inch between the rail head and the three edges of the U-shaped cut in the restraining plate.

The physical characteristics of the counterweight-rail system mentioned in the previous paragraph are very important to consider in the dynamic analysis of counterweight system exposed to earthquake induced ground motions. The seismic motions are transmitted to the counterweight from the supporting building to the bracket supports and then to the counterweight through the rails. Thus the seismic motions are transmitted through several points on the systems to the counterweight, and this spatial variability of the multi-support excitation is an important characteristic that must be considered in the analysis. In the dynamic analysis of the counterweights, it is also necessary to consider the flexibility of the support brackets, rails, helical springs, and rubber tires on the roller wheels. Furthermore it is also extremely important to consider the effect of the contact that can possibly occur between the rails and the restraining plates provided under the rollers at each roller guide assembly because of closing of the 3/16 inch gap between the two. This can occur if the ground motion intensity is high. This contact between the rail and the restraining plate changes the stiffness of the system at that contact point dramatically from a low value, which is primarily governed by the low stiffness of the rubber tires, to a very high value primarily governed by the flexibility of the rail. This sudden change can occur at any or all four corners where the roller guide assemblies are placed. This sudden change introduces nonlinearity in the dynamic system analysis, and must be considered as it has a significant effect on the system response and the stresses and deformations in the rails and roller guide assemblies.

Besides this source of nonlinearity in the system analysis, there is another source which is also associated with closing and opening of the gaps between the counterweight frame and the rail at the bracket supports. One could avoid this contact by increasing the gap size, but to alleviate the possibility of getting large bending stresses in the rails during a strong earthquake, the ASME code recommends that this gap not be more than ½ inch. When a bracket support is between the upper and lower roller guides, the gap between the frame and the rail at the support can close during the in-plane of the counterweight-frame, allowing the transfer of inertial forces directly to the support and thus reducing the forces on the roller guide assemblies which cause bending in the rails. The closing of this gap changes the system model by introducing a large stiffness element at the point where the contact occurs. This discontinuity in the system characteristics must also be considered as it also affects the system response significantly.

Figures 2a and 2b show the force-deformation characteristics of the spring at a roller guide assembly and at the bracket support. The initial flat part in Figure 2a represents the no-contact case, with the low value of the slope associated with the flexibility of the rubber tires and helical springs primarily. The higher slope part is associated with the flexibility of the rail support on the bracket supports and is realized only when the 3/16 inch gap is closed. Similarly, in Figure 2b, the initial part with zero force is due to no contact of the frame with the support but once a contact occurs, the force is increased significantly due to large stiffness of the bracket support. It has been observed that the fundamental frequency of the system is increased three times when the gaps at all four roller guide assemblies are closed, and this higher frequency is further increased by a factor of two when the frame also comes in contact with the rail at the bracket support. This change in the fundamental frequency is relevant in the context of tuning mass dampers for response reduction. It is, however, mentioned that not all these gaps may close at the same time, and thus these extreme frequency values may only be realized rather infrequently.



(a) (b)

Figure 2. Bilinear force-deformation diagram for (a) equivalent spring with contact between the rail and restraining plate, and (b) contact between the counterweight frame and the rail.

The writers have examined [2, 3] the response characteristics of the counterweight elevator systems considering their unique features and the system nonlinearities caused by the closing and opening of the gaps in the system. The nonlinearities in the system could also arise when the stresses in the rails or support brackets exceed their yield values. These post-yielding, however, are not of much operational interest as when the yielding occurs the functionality of the system is already compromised. Thus, considering only the nonlinear effects caused by intermittent opening and closing of the gaps mentioned in the previous discussion, the response of a typical counterweight rails system was thoroughly examined by the writers in earlier studies. Besides several response characteristics that were observed, one of the main conclusions was that during a strong ground excitation it is quite possible to overstress the rails and other components, and thus compromise the functionality of an elevator system.

In Figure 3 are plotted a set of curves showing the maximum stresses in a typical counterweight of an elevator was housed in a 10-story building. The maximum stresses were obtained in the rails at several locations for several different positions of the counterweight along the height. The seismic input was defined by two well-known recorded earthquake motions: 1940 El Centro and 1994 Northridge earthquakes. These earthquake time histories were normalized to different peak ground acceleration values. For these inputs of different intensities applied to the building, the maximum global stresses induced in the rails are plotted in Figure 3. Also shown in the figure is the maximum allowable stress in the A-17 ASME Code [1]. It is noted that the maximum stress could go beyond the allowable value for higher levels of ground motion intensities. For example at the acceleration level observed in one of the Northridge record, shown in Figure 3, the induced stresses are higher than the allowable. Thus, if this elevator system were on the site where this record was obtained, it would likely to be overstressed in this seismic event. It is, thus, of interest to develop protective systems that can reduce these high stresses, and enhance the availability of the elevator systems after a major earthquake event such as the 1994 Northridge event. This paper presents some of the approaches that can be used for mitigating seismic effects on the counterweights.

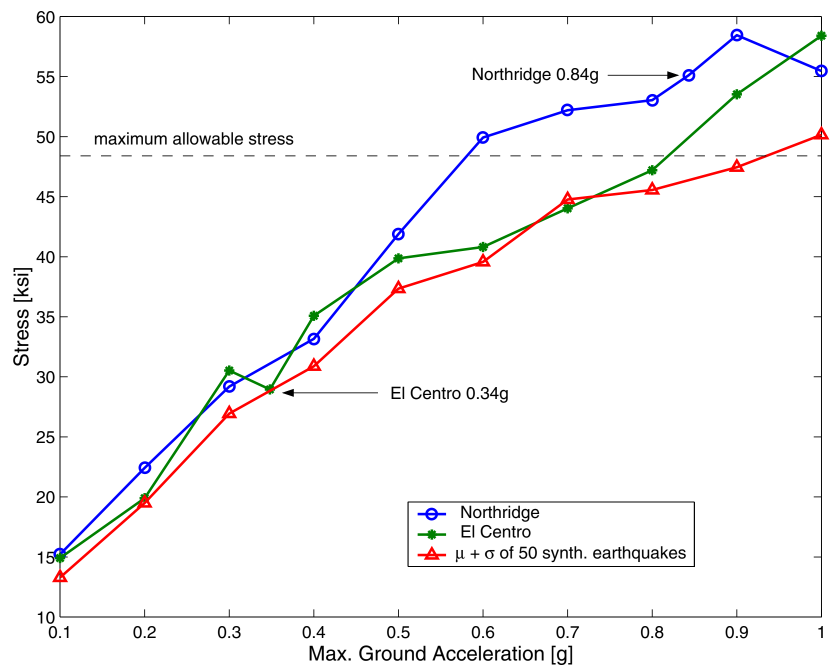


Figure 3. Maximum stress in the rail for increasing intensity of Northridge and El Centro accelerograms.

PROTECTIVE SYSTEMS FOR ELEVATOR COUNTERWEIGHTS

To reduce the dynamic effects of earthquakes in structural systems several passive and active response control schemes have been considered in the past. The passive schemes consist of energy dissipation devices, base isolation devices, and tuned vibration absorber systems. Among these passive schemes, the base isolation and energy dissipation systems have found good acceptance in the profession. The tuned vibration absorbers have received a mixed report for application in earthquake structural engineering. The active schemes in which external forces are applied to counteract the seismic vibrations so far have been found to be infeasible, primarily because they need large forces and a large power source to work with large civil structures. The active schemes such as active tuned vibration absorbers have also been considered, but primarily for the mitigation of wind induced vibrations for the comfort of the occupants. The semi-active methods that do not need large power sources, and which primarily regulate the stiffness and damping characteristics of the system through installation of devices that can be managed on a real-time basis such as magnetorheological dampers, however, seem to have promising applications in earthquake structural engineering. For application to counterweights, we have investigated the applications of (1) damping enhancement schemes through the use of viscous dampers, (2) passive tuned vibration absorbers, and (3) active tune vibration absorbers. Some results demonstrating the application of these three methods are described in the following sections.

Damping Enhancement

The equations of motion for the in-plane and out-of-plane motion of a counterweight supported by rails can be described by the following equations of motion consisting of usual mass, stiffness, and damping matrices and forcing function terms:

$$\boldsymbol{M}_i\ddot{\boldsymbol{q}}_i + \boldsymbol{C}_i\dot{\boldsymbol{q}}_i + \boldsymbol{K}_i\boldsymbol{q}_i = -\boldsymbol{M}_i\ddot{\boldsymbol{y}}_i + \boldsymbol{f}_i \quad (1)$$

$$\boldsymbol{M}_o\ddot{\boldsymbol{q}}_o + \boldsymbol{C}_o\dot{\boldsymbol{q}}_o + \boldsymbol{K}_o\boldsymbol{q}_o = -\boldsymbol{M}_o\ddot{\boldsymbol{y}}_o + \boldsymbol{f}_o \quad (2)$$

where the subscript i and o denotes the quantities associated with the in-plane and out-of-plane motions with the usual notation $\boldsymbol{M}$ for the mass, $\boldsymbol{C}$ for the damping , and $\boldsymbol{K}$ for the stiffness matrices. The response vector $\boldsymbol{q}$ in the in-plane motion consists of two elements, where as the response vector for the out-of-plane motion contains three generalized displacement elements. Thus, the motion of the counterweight, considered as a rigid body can be described by five-degrees of freedom. The vectors $\ddot{\boldsymbol{y}}_i$ and $\ddot{\boldsymbol{y}}_o$ are the input motion vectors, and $\boldsymbol{f}_i$ and $\boldsymbol{f}_o$ are the nonlinear force vectors associated with the nonlinear behavior of the supporting systems. The elements of these vector as well as the elements of the stiffness matrices change as the system gaps open and close. The details of these matrices and vectors are given in explicit forms by the writers in References 2 and 3. It is noted that the in-plane response of the counterweight is more critical than the out-of-plane response, and thus the critical response quantities shown in the following section usually pertain to the in-plane response of the system.

To demonstrate the effect of increasing the damping in the systems in simple terms, the modal damping ratio of each mode was increased from 2% to 5%. This change in the modal damping ratios caused the damping matrices in Equations (1) and (2) to change. It was observed that this small increase in the damping ratio could reduce the maximum stresses by about 20% for Northridge input motion. Thus damping ratio increase can be effective if the dampers can, indeed, be installed on the system in the very tight spaces around the counterweight. One place where viscous dampers can be installed is in parallel with the helical springs which are used to maintain contact of the rollers with the rails. To investigate this, two small size dampers with linear damping coefficients values of 10 lb-s/in and 50 lb-s/in were installed in parallel to each helical spring to see what changes in the maximum response they could cause. It was noted that the damper with smaller coefficient reduced the maximum stress in the rail by about 5%, but the five times large damper only reduced it by 7.5%. The primary reason for this was that this damper was not fully effective; when the restraining plates came in contact with the rails damper did not experience any relative motion to dissipate the energy. The installation of discrete dampers at locations where they can experience relative motion to dissipate energy is possible, but it will require special arrangements and, thus, it was not investigated. Another possible method of increasing the energy dissipation is through installation of viscoelastic patches on the rails, and this approach is currently under investigation.

Passive and Active Tuned Vibration Absorbers

The tuned vibration absorbers have been used with mechanical systems subjected to sustained harmonic inputs. Their use with civil structures exposed to irregular motions such as earthquake inputs has received mixed support from the researcher about their effectiveness. Some researchers indicate that they can be effective in reducing seismic effects, yet there are others who say that that they are not effective, and may even be detrimental in seismic application. See Singh et al [4] for further discussion. In this study we have tried to explore the possibility of using these systems with the counterweights. Since the counterweight consists of a stack of filler dead weight, it seems tempting to use the top portion of these filler weights as the mass of a tuned vibration absorber. For this the top filler weight of the counterweight can be separated from the remaining filler blocks, attached to a suitably selected spring (and also a damper), and allowed to move on the bottom blocks in the plane of the counterweight. This arrangement is shown schematically in Figure 4. This arrangement can also be made active by installing an actuator between the frame and the tuned vibration absorber mass. It is noted that this approach is effective only in reducing the in-plane vibrations which, however, also happen to be the more critical than the out-of-plane vibrations.

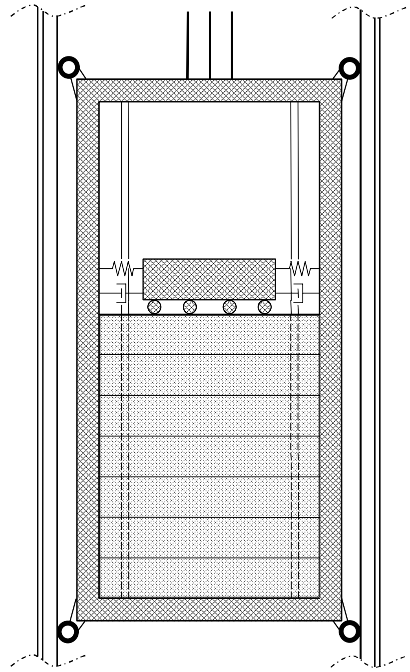


Figure 4. Counterweight showing a part of the weights acting as tuned-mass damper

The equation of motion of a tuned vibration absorber can be written as

$$\mu m_c \ddot{u}_T + c_T\left[\dot{u}_T - \dot{u} - (1-\mu)\dot{u}_e\right] + k_T\left[u_T - u - (1-\mu)u_e\right] = -\mu m_c \ddot{x}_c + f_T \tag{3}$$

where μ is the ratio of the mass damper to the total mass m_c of the counterweight, k_T and c_T are stiffness and damping attached between the mass and the frame, and u and u_T are, respectively, the displacement of the counterweight and mass damper relative to the building structure. The displacement u_e is related to the rotational motion ψ of the counterweight, as shown in Figure 1, through

$$u_e = l_m \psi \tag{4}$$

where l_m is the distance from the bottom of the counterweight to its center of mass, also shown in Figure 1. The control force f_T is added if active mass driver is used.

The tuned mass damper is assumed to have translation degree of freedom only. That is, the whole counterweight, including the mass damper, rotates as a single body. Combining Eq. (3) with the in-plane equation of the counterweight Eq. (1) we obtain

$$\boldsymbol{M}_i \ddot{\boldsymbol{q}}_i + \boldsymbol{C}_i \dot{\boldsymbol{q}}_i + \boldsymbol{K}_i \boldsymbol{q}_i = -\boldsymbol{M}_i \ddot{\boldsymbol{y}}_i + \boldsymbol{D} f_T + \boldsymbol{f}_i \tag{5}$$

where the mass, damping and stiffness matrices are now 3 × 3 matrices and the displacement vector has three displacement degrees of freedom. The location vector $\boldsymbol{D}$ is used to add the active control force into the equations of motion of the active tuned vibration absorber. To obtain the active control force, the equations of motion is written in the state forms as:

$$\dot{z} = Az + Bf_T + f^* \tag{6}$$

where

$$A = \begin{bmatrix} 0 & I \\ -M_i^{-1}K_i & -M_i^{-1}C_i \end{bmatrix} \quad ; \quad B = \begin{Bmatrix} 0 \\ D \end{Bmatrix} \quad ; \quad f^* = \begin{Bmatrix} 0 \\ -\ddot{y}_i + M_i^{-1} f_i \end{Bmatrix} \tag{7}$$

$$z = \begin{Bmatrix} q_i \\ \dot{q}_i \end{Bmatrix} \tag{8}$$

To obtain the optimal control force the standard LQR approach is used in which the following cost function is minimized:

$$J = \int_0^{t_f} \left(z^T Q z + r f_T^2 \right) dt \tag{9}$$

where Q and r are the weight factors used in the cost function. The optimal control force is then defined by the standard LQR procedure as;

$$f_T = -Gz \tag{10}$$

where G is the gain matrix defined as

$$G = \frac{1}{r} B^T P \tag{11}$$

in which P is the solution of algebraic Riccati equation

$$PA + A^T P - \frac{1}{r} PBB^T P + Q = 0 \tag{12}$$

Since the stiffness matrix of the system constantly changes as the motion proceeds, a question arises as to what matrix one should use to calculate the control gains. In this study we used the pre-contact stiffness matrix to define matrix A and to calculate the gains. The effectiveness of this assumption is confirmed the by numerical examples presented later.

In the design of a tuned vibration absorber, its tuning is of some concern. As mentioned earlier, the natural frequency of the rail-counterweight system can change during earthquake-induced vibration. From the initial non-contact region to the region where contact occurs at both upper and lower supports, the natural frequency increases about three times its original value. Furthermore, if there is another contact between the frame and the rail, the frequency again increases by a factor of about two. To obtain an effective frequency ratio for the tuned mass damper, the effect of tuning frequency is studied for a broad range of the frequency ratio values. The frequency ratio is defined as the ratio of the vibration absorber frequency to the fundamental frequency of the counterweight system when the gaps close at the top and bottom roller guide assemblies. A damping ratio of 5% is used in the mass damper system. The excitations at the base of the buildings are defined by the as-observed Northridge and El Centro earthquakes motions.

Figures 5 and 6 show the ratios of the maximum stress obtained with and without the tuned vibration absorber for the two earthquake motions — Northridge and El Centro earthquakes, respectively. The maximum displacement of the mass damper is shown in Figure 7. The effect of using different mass ratios (defined as the ratio of the mass of the vibration absorber to the total mass) is also shown in these figures. These figures show that the maximum stress in the rail is reduced by the tuned vibration absorber when the frequency ratio is between 0.7 and 1.0. The absorbers with mass ratios of 20% are quite effective. The larger mass ration does not necessarily means a larger response reduction. The stress reductions of about 20% or more are observed. The displacement of the mass damper is less than 2 inches for Northridge and less than 1 inch for El Centro. Increasing the mass ratio beyond 40% was also studied but the trend in response reduction was somewhat erratic which made it difficult to identify the tuning frequency range. The results for higher mass ratios are thus not presented.

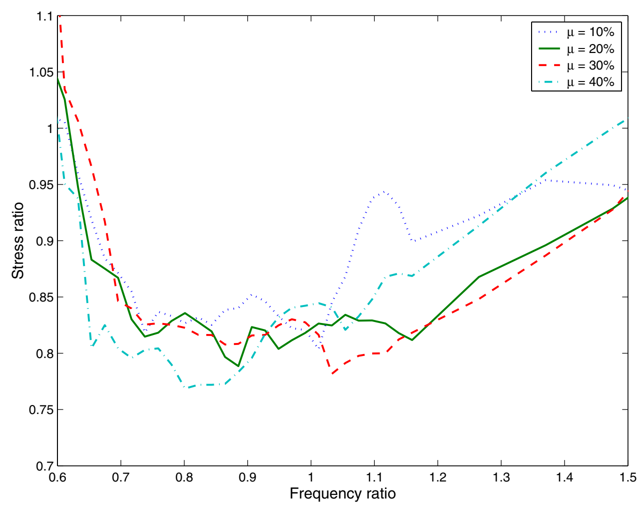


Figure 5. Ratio of maximum stress in the rail for different frequency ratios of the vibration absorber for Northridge earthquake

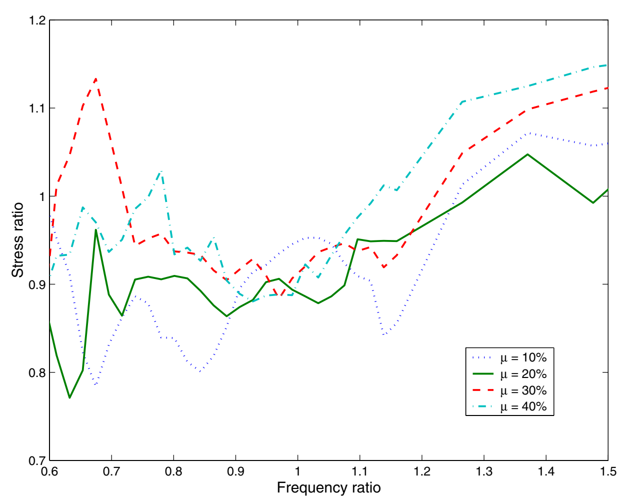


Figure 6. Ratio of maximum stress in the rail for different frequency ratios of the vibration absorber for El Centro earthquake

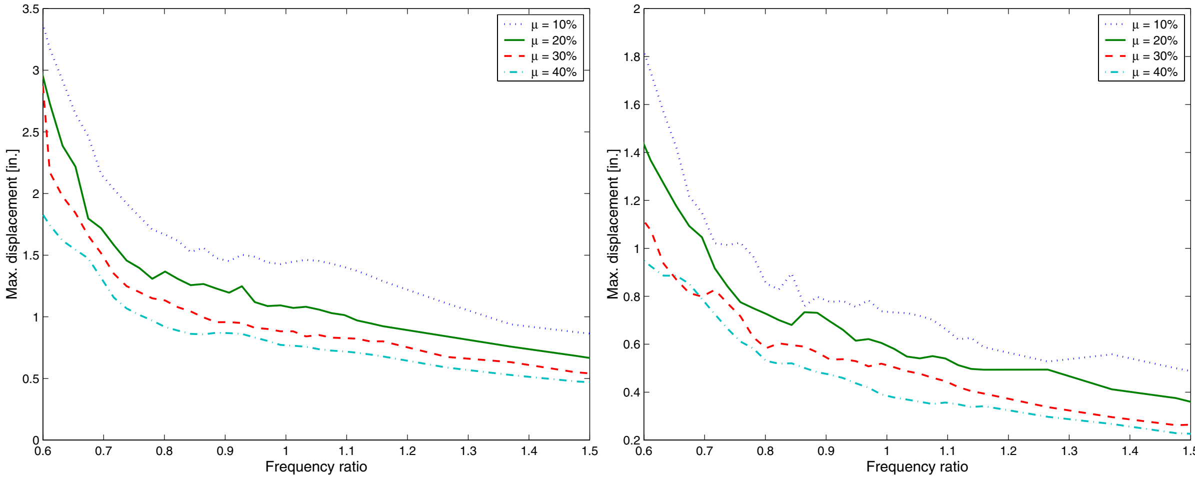


Figure 7. Maximum displacement of the tuned vibration absorber for different frequency ratios for Northridge and El Centro earthquakes

In Figures 8 and 9 we compare the response reduction effects of the passive and active vibration absorbers for the mass ratio of 20%. It can be seen that active vibration absorber is able to reduce the maximum stress in the rail further while the maximum displacements are about the same as in the passive case.

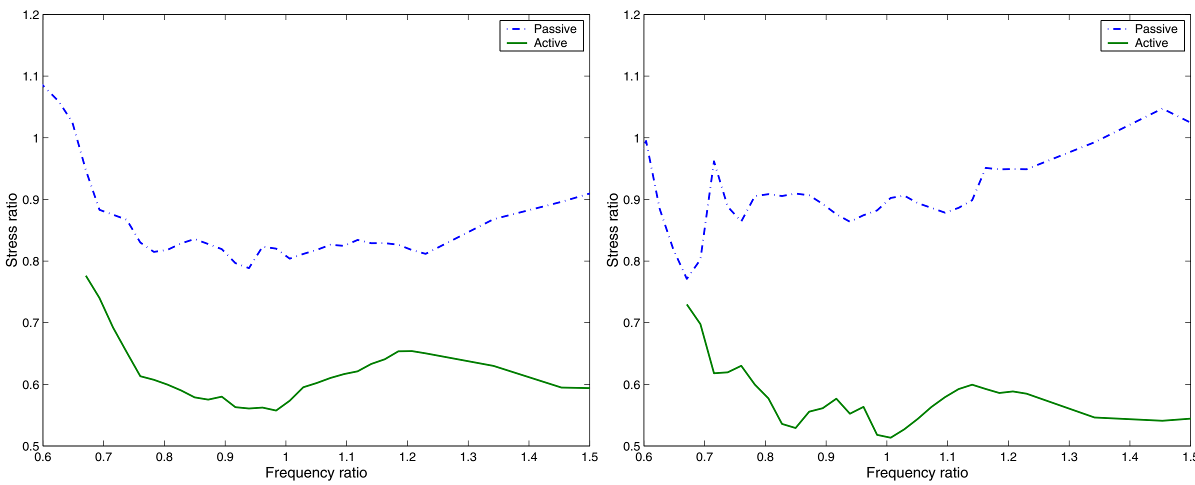


Figure 8. Ratio of maximum stress in the rail for passive and active vibration absorbers with μ = 20% under Northridge and El Centro earthquakes

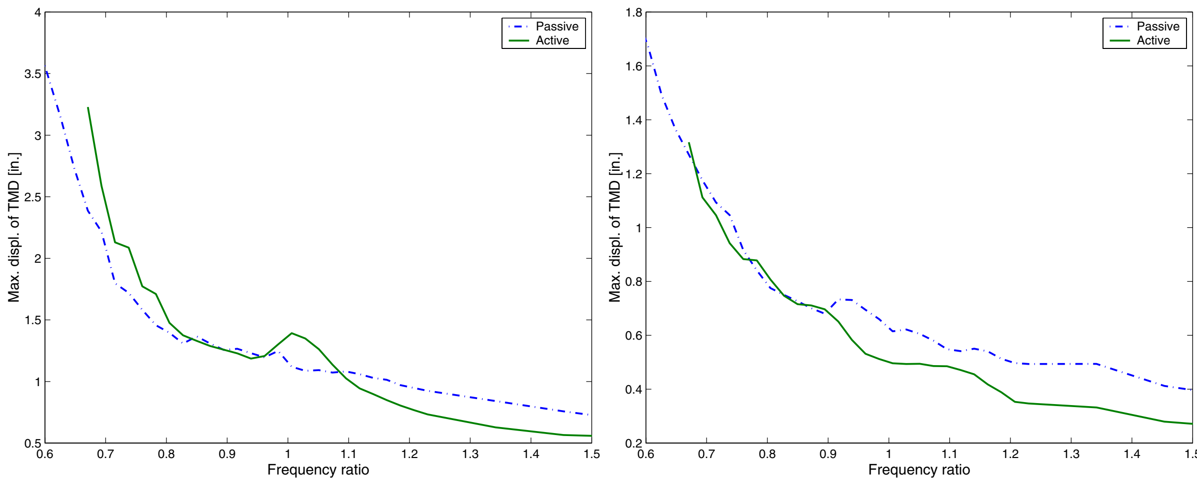


Figure 9. Maximum displacement of the vibration absorber with μ = 20% under Northridge and El Centro earthquakes

In Figure 10, stress ratios (indicating reduction in the response) achieved by the active vibration absorbers with different mass ratios are shown. Interesting, the system with smaller mass ratio of 20% seem to be more effective that the systems with higher mass ratios. The maximum control force required in the active method is shown in Figure 11. Again the system with smaller mass ratio seems to perform better than the system with higher mass ratios as its force requirements are smaller. It is noted that the maximum values of the required actuator forces are less than 10% of the total weight of counterweight which can be easily applied by an small size actuator.

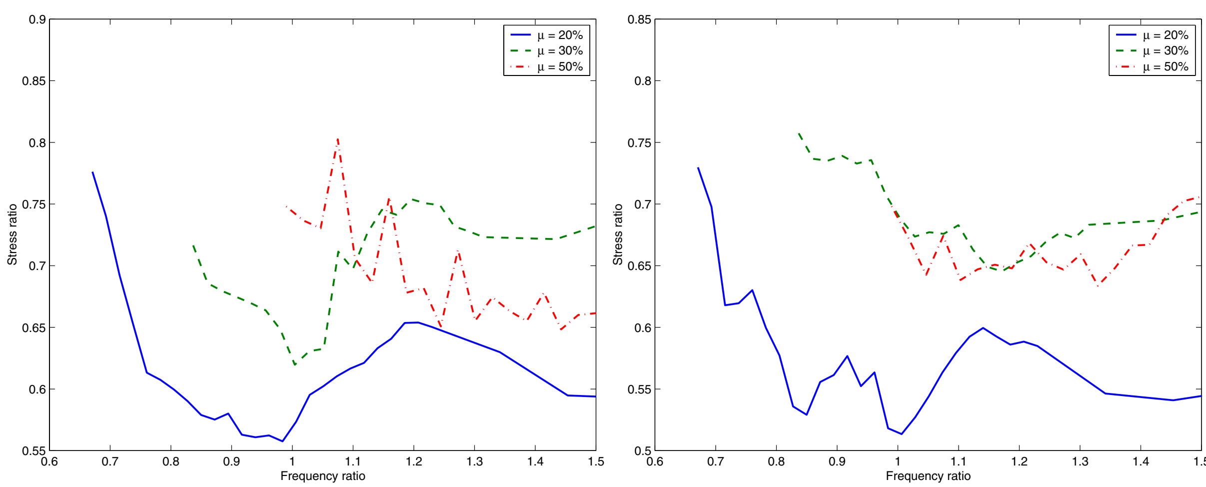


Figure 10. Ratio of maximum stress in the rail for active vibration absorber with different mass ratios under Northridge and El Centro earthquakes

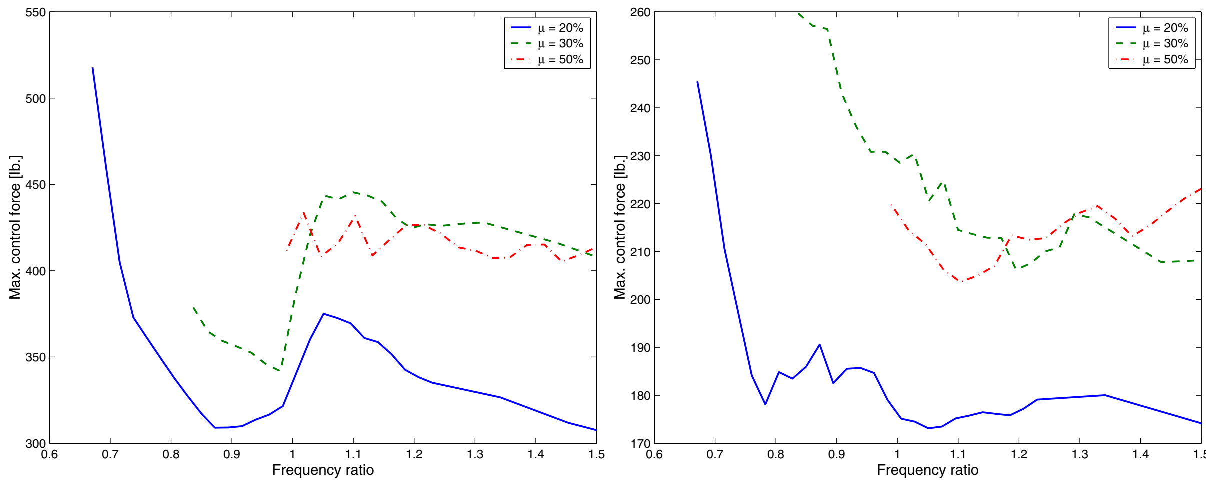


Figure 11. Maximum control force for active vibration absorber with different mass ratio under Northridge and El Centro earthquakes

SYSTEM FRAGILITY REDUCTIONS BY THE PROTECTIVE SYSTEMS

In the previous sections, the effectiveness of the tuned vibration absorbers, for both the active and passive systems, was evaluated in terms of the level of reduction in the maximum stresses in the rails they cause. Another global and a more comprehensive measure of the effectiveness of a protective device can be made in terms of the improvements it can bring in the system reliability or reduction in the system fragility. Herein we use the fragility curves to show the level of improvement that can be achieved by the use of the passive and active tuned vibration absorbers.

The fragility of a system is defined as the conditional probability of failure, give a level of seismic intensity such as maximum ground acceleration. For our rail-counterweight system, the failure is defined as the event of induced stress in the system exceeding the allowable stress value. Both the induced stresses and allowable stresses are random variables. To calculate the probability of failure due to overstressing of the rails for a given level of maximum ground acceleration, an ensemble of earthquake ground motions was generated to match a frequency content described by a well-known form of the spectral density function that has been used in the past to represent earthquake motions. The system was then analyzed for this ensemble to obtain the maximum stresses in the rails at several critical locations for different positions of the counterweight along the height. These maximum stress values were then used to obtain the mean and coefficient of variation values of the maximum stresses at these critical locations. Assuming that the maximum induced stresses and the allowable stress can be represented by lognormal distributions, the probabilities of failures were calculated at each critical location for each position of the counterweight. Assuming that the failures at these critical locations are correlated, the maximum value of their probabilities of failure then defined failure probability for that particular counterweight location. Assuming that the counterweight can be at any location along the building height with equal probability, the system probability of failure can then be calculated as the average of all the probabilities calculated for all equally spaced counterweight positions.

Figure 12 shows the fragility curves obtained for the rail-counterweight considered in this study for the system for three situations: (a) with no protective device (uncontrolled), (b) with a passive tuned vibration absorber with mass ratio of 20% and tuning frequency ratio of 0.9, and (c) with an active tuned vibration absorber again with a mass ratio of 20% and the frequency ratio of 0.9. The figure clearly indicates the effectiveness of the tuned vibration absorber, which can be further enhanced by making it active with an actuator. The similar conclusions about the effectiveness of these protective systems can also be drawn from Figure 13 where we compare the mean plus one standard deviation values of the maximum stresses for the three cases. Also shown are the maximum values of the stresses for the three cases. These are the maximum values not at any one point, but the maximum of values at all critical positions for all counterweight locations. The maximum values of the relative displacement of the absorber mass and also the actuator forces, as shown in Figure 14, are also within the reasonable limits thus establishing the feasibility of using these protective schemes on the counterweights.

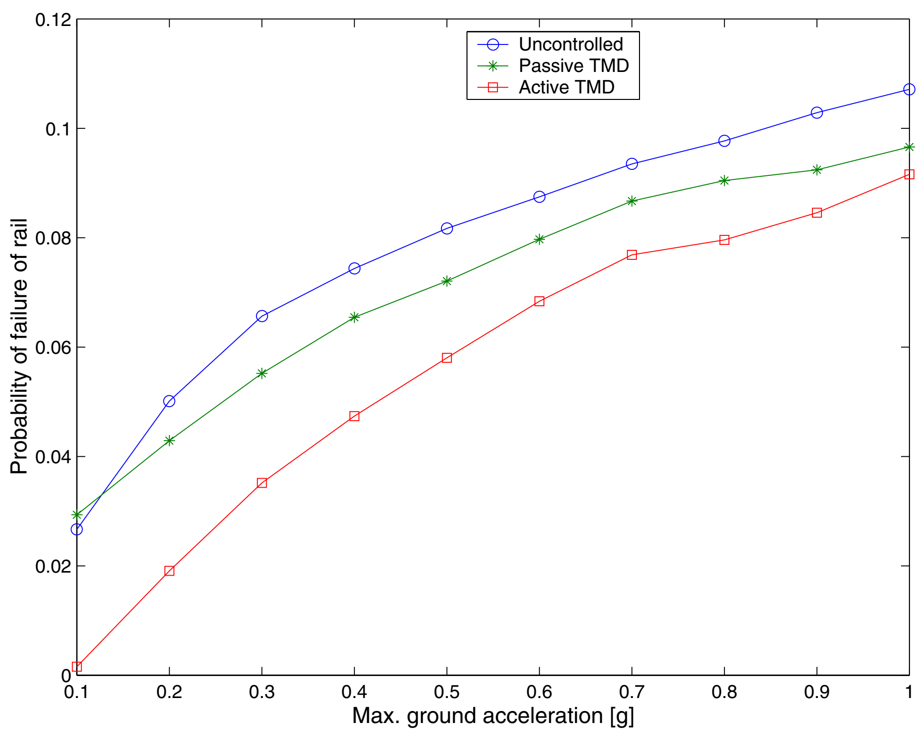


Figure 12. Fragility curves for rail-counterweight system with and without vibration absorber

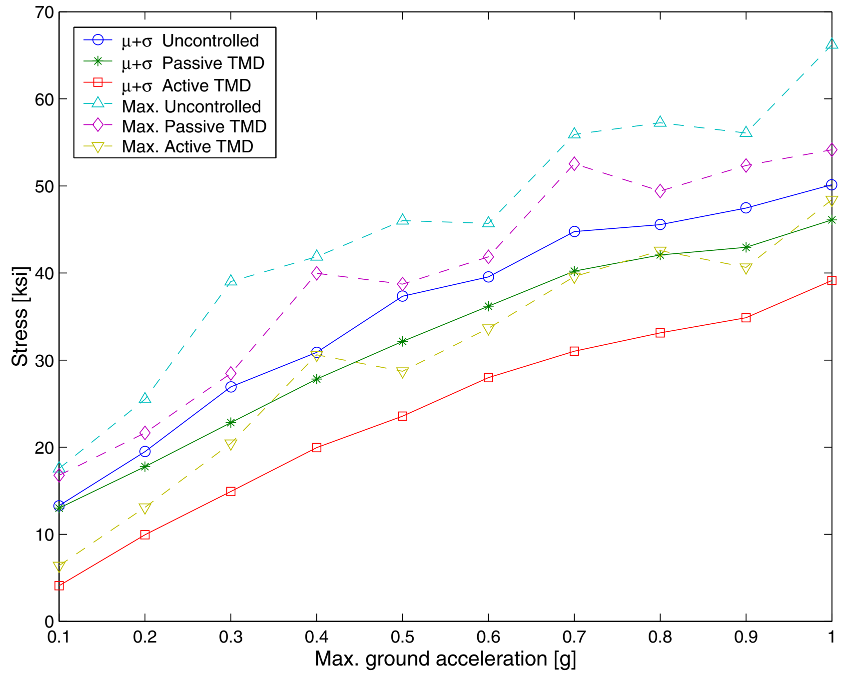
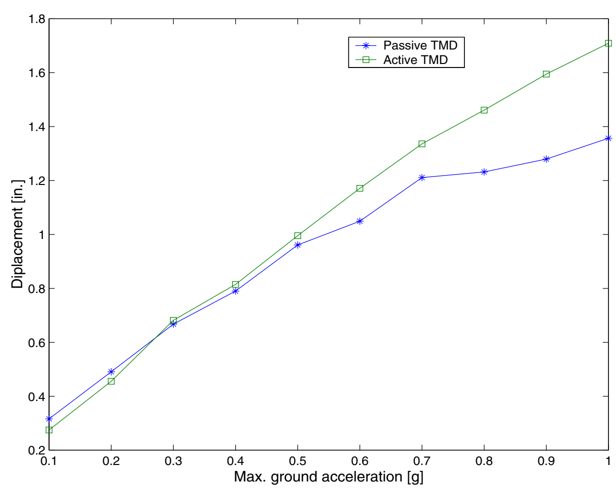
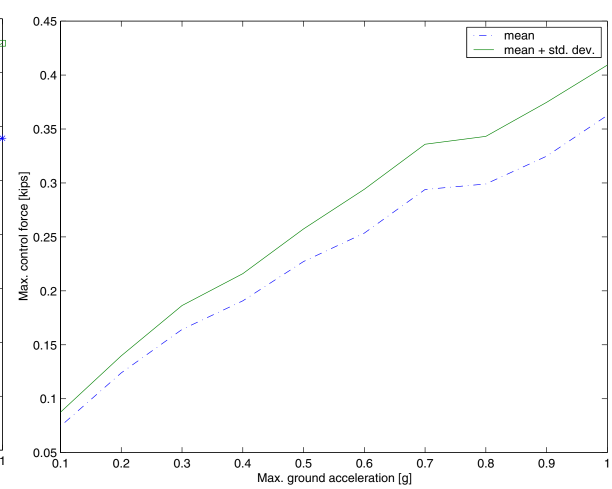


Figure 13. Absolute maximum and mean plus standard deviation of the maximum stress in the rail



(a)



(b)

Figure 14. (a) Mean plus standard deviation of the maximum displacement of mass damper; (b) Mean and mean plus standard deviation of maximum control forces

CONCLUDING REMARKS

The study describes three seismic protective schemes that can be used with rail-counterweight systems to reduce their seismic response. The protective schemes are the: (a) use of discrete dampers, (b) use of passive tuned vibration absorbers, and (c) use of active tuned vibration absorbers. The discrete dampers are difficult to use because there is not much space to install them on the system. A tuned vibration absorber, on the other hand, can be easily configured in the existing system by utilizing a part of the counterweight mass as the mass of the absorber. The study shows that a tuned vibration absorber can be quite effective in reducing the seismic response. Furthermore, its effectiveness can be further enhanced by operating it in an active mode by installing an actuator which can be operated by an appropriate control algorithm. There is enough clearance available on the counterweight to accommodate the motion of the absorber or the stroke of the actuator, and the level of the control force needed is also reasonably small to make this active scheme practically feasible.

ACKNOWLEDGEMENTS

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