Semiactive Tuned Liquid Column Dampers: Experimental Study

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Abstract: A tuned liquid column damper (TLCD) is a special type of auxiliary damping device which relies on the inertia of a liquid column in a U-tube to counteract the forces acting on the structure. Damping in the TLCD is introduced as a result of headloss experienced by the liquid column moving through an orifice. The primary objective of this paper was to examine the performance of a prototype semiactive TLCD. Experiments were conducted to determine the dynamic characteristics of a coupled structure-TLCD system. The experimental setup included a prototype TLCD attached to a model of a single-degree-of-freedom structure which was mounted on a shaking table. The prototype TLCD was equipped with an electropneumatic valve to provide optimal damping at a wide range of structural motion amplitudes. The optimum absorber parameters, i.e., the optimal tuning ratio and damping ratio, were determined experimentally and compared to the analytical results obtained previously reported by the writers. A control strategy based on gain scheduling was utilized, which was designed to maintain the optimal damping based on a prescribed *look-up* table. This scheme was experimentally validated. It was noted that the semiactive system provided an additional 15–25% reduction in response over a passive system. Finally, a design example was presented to demonstrate the application of semiactive TLCDs to tall buildings under wind loads. The response of uncontrolled structure, braced structure, and structure with a passive damper, and semiactive damping system using numerical simulations were compared. The semiactive TLCD reduced the RMS acceleration at the building top by 45% at all wind speeds.

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Introduction

The current trend toward structures of ever increasing heights and the use of lightweight and high-strength materials have led to very flexible and lightly damped structures. Understandably, these structures are very sensitive to environmental excitations such as wind and earthquakes. Under the action of one or a combination of these loads, a structure may experience dynamic load effects which may lead to structural failure, fatigue, occupant discomfort, and difficulty in the elevator operation and related equipment. Among other solutions to address these serious concerns, motion control devices have been introduced in buildings to reduce structural response (Soong and Dargush 1997; Kareem et al. 1999).

A tuned liquid damper (TLD) is a special class of tuned mass dampers (TMD) where the mass is replaced by liquid (usually water). The sloshing of the liquid mimics the motion of the TMD mass. Tuned liquid column dampers (TLCDs) are a special type of TLDs relying on the motion of the liquid column in a U-tube to counteract the forces acting on the structure, with damping introduced in the oscillating liquid column through an orifice (Sakai and Takaeda 1989; Kareem 1994). However, the potential of liquid dampers in their passive state is not fully realized due to the dependence of their damping on the motion amplitudes (or the level of excitation) and their inability to respond quickly to suddenly applied loads such as earthquakes and fast moving weather fronts. Therefore, semiactive systems were proposed in order to correct some of the problems inherent in TLCDs (Haroun and Pires 1994; Kareem 1994; Abe et al. 1996; Yalla et al. 1998). Semiactive studies for increasing the effective range of application of TLDs were suggested in Lou et al. (1994) in which the natural period of the water tank is regulated by controlling the orientation of a set of rotatable baffles in the tank, thereby transforming the system from a passive damper to a variable-stiffness damper.

The full-scale installation of a bidirectional passive liquid column vibration absorber (LCVA) on a 67 m steel communications tower has been reported by Hitchcock et al. (1999). This device does not include an orifice/valve in the U-tube and hence, it is not possible to control damping in the LCVA. The writers also acknowledge that due to the absence of an orifice, the damping ratio of the LCVA was not expected to be optimum. The writers also observed that the LCVA did not perform optimally at all wind speeds. Maximum response reduction of almost 50% was noted, however, nonoptimal performance of the damper was observed above and below the design wind speed. This observation reaffirms the fact that passive systems are inadequate in performing optimally at all levels of excitation (Kareem 1994). Similar observations were made concerning tuned sloshing dampers both involving scaled experiments and full-scale studies (Tamura et al. 1995).

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This paper presents a semiactive system which has the potential to overcome the aforementioned shortcomings of a passive TLCD system. Although researchers have studied theoretically semiactive versions of TLCDs, there have been no reported experimental verification of such a system. The main objectives of this paper are the following: (1) to study the dynamic characteristics of a coupled structure-damper system and compare the experimentally determined optimum absorber parameters to previously obtained analytical results (Yalla and Kareem 2000); (2) experimental verification of a gain-scheduled control law to deliver optimum damping based on a prescribed look-up table; and (3) to offer a design example for the implementation of semiactive TLCDs in a tall building under wind loading.

Analytical Model

The optimal TLCD parameters for a variety of excitations are derived in Yalla and Kareem (2000). For the coupled structure-TLCD system subjected to ground motion, the equations of motion are given by

$$\begin{bmatrix} M_s+m_d & \alpha m_d \\ \alpha m_d & m_d \end{bmatrix}\begin{bmatrix} \ddot{X}_s \\ \ddot{x}_f \end{bmatrix}+\begin{bmatrix} C_s & 0 \\ 0 & c_{\mathrm{eq}} \end{bmatrix}\begin{bmatrix} \dot{X}_s \\ \dot{x}_f \end{bmatrix}+\begin{bmatrix} K_s & 0 \\ 0 & k_f \end{bmatrix}\begin{bmatrix} X_s \\ x_f \end{bmatrix}=-\begin{bmatrix} M_s \\ m_d \end{bmatrix}\ddot{x}_g, \quad |x_f|\leqslant\frac{(l-b)}{2} \tag{1}$$

In these equations, an equivalent linear damping coefficient c_{eq} has been used in lieu of the nonlinear damping term $c_{\mathrm{nonlinear}}=1/2\rho A\xi|\dot{x}_f(t)|$. Based on these equations, the transfer functions that relate the ground motion to structural acceleration and liquid velocity are given by

$$H_{\ddot{X}_s\ddot{x}_g}(\omega)=\frac{-\omega^4\mu\alpha+\omega^4-2\zeta_d\omega_d(i\omega)-\omega_d^2\omega^2}{[-(1+\mu)\omega^2+2\zeta_s\omega_s(i\omega)+\omega_s^2][-\omega^2+2\zeta_d\omega_d(i\omega)+\omega_d^2]-\omega^4\alpha^2\mu}$$

$$H_{\dot{x}_f\ddot{x}_g}(\omega)=\frac{\alpha i\omega^3+i\omega}{[-(1+\mu)\omega^2+2\zeta_s\omega_s(i\omega)+\omega_s^2][-\omega^2+2\zeta_d\omega_d(i\omega)+\omega_d^2]-\omega^4\alpha^2\mu}$$

Symbols in the preceding equations are defined in the nomenclature. The variance of the acceleration of the primary system and the variance of liquid velocity in the TLCD are obtained accordingly

$$\sigma_{\ddot{X}_s}^2=\int_{-\infty}^{\infty}|H_{\ddot{X}_s\ddot{x}_g}(\omega)|^2S_0(\omega)d\omega \tag{2}$$

$$\sigma_{\dot{x}_f}^2=\int_{-\infty}^{\infty}|H_{\dot{x}_f\ddot{x}_g}(\omega)|^2S_0(\omega)d\omega \tag{3}$$

where $S_0(\omega)$=spectral density function of the ground motion. For a white noise type disturbance, this is approximated by a constant value across all the frequencies as equal to S_0. The optimal parameters, i.e., damping and tuning ratio, are obtained by solving the following equations:

$$\frac{\partial\sigma_{\ddot{X}_s}^2}{\partial\zeta_d}=0; \quad \frac{\partial\sigma_{\ddot{X}_s}^2}{\partial\gamma}=0 \tag{4}$$

A closed-form solution of these equations is obtained for the case of an undamped primary system, i.e., $\zeta_d=0$. The optimal values for this case are

$$\zeta_{\mathrm{opt}}=\frac{\alpha}{2}\sqrt{\frac{\mu\left(1+\mu-\alpha^2\frac{\mu}{4}\right)}{(1+\mu)\left(1+\mu-\frac{\alpha^2\mu}{2}\right)}};$$

$$\gamma_{\mathrm{opt}}=\frac{\sqrt{1+\mu\left(1-\frac{\alpha^2}{2}\right)}}{1+\mu} \tag{5}$$

Yalla and Kareem (2000) also obtained similar optimal absorber parameters for damped primary system cases. These values will be used later to compare the optimum absorber parameters determined experimentally with the analytical values.

Experimental Studies

A schematic diagram of the experimental setup is shown in Fig. 1(a). It consists of a model of a single story structure fixed on a shaking table. A TLCD consisting of a U-shaped tube made of PVC material with an electropneumatically actuated ball valve at the center of the tube is attached to the model [Fig. 1(b)]. The U-tube has a circular cross section with an inner diameter of 3.8 cm and a horizontal length of 35.5 cm and a total length of 81 cm. The valve used in this study is a ball valve of 3.8 cm (1.5 in.) diameter. A command signal (4–20 mA) changes the valve opening angle θ, which effectively changes the orifice area of the valve. A pneumatic air-line supplies the necessary 80 psi of air pressure for the valve actuator. A position transmitter inside the valve measures the valve position and transmits as a 4–20 mA signal. Signal conditioning units were used to convert the signals from ADC (analog to digital) and DAC (digital to analog) channels of the data acquisition board to convert the 0–5 V to 4–20 mA signals. The shaking table control is accomplished by using proportional-integral-derivative (PID) control using position feedback from the motor encoder. Details of the valve characteristics are presented in the Appendix, where the valve opening angle is related to the headloss coefficient (ξ).

The system transfer functions were obtained by exciting the shaking table with a band-limited random white noise (cutoff frequency $f_c=2$ Hz), at different levels of excitation amplitudes and the acceleration was measured at the top of the structure. The excitation amplitude in these experimental studies is referred to as S_0 and it represents RMS value of excitation (in volts). The RMS

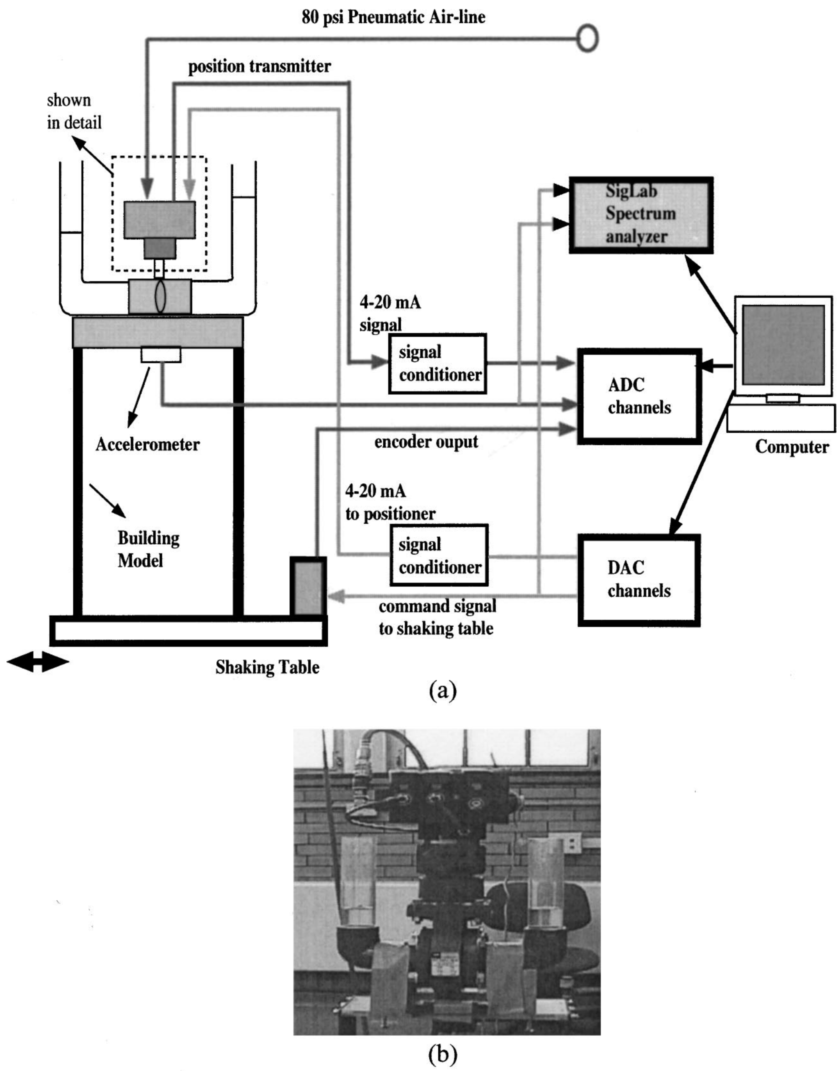

Fig. 1. (a) Schematic diagram of experimental setup and (b) photograph of electropneumatic actuator

excitation displacement amplitude of the shaking table was varied between 0.05–0.3 V to avoid spilling of water out of the U-tube.

The model structure without the damper is a linear system, which was confirmed through identification of the transfer function at different amplitudes of excitation. The effect of the pneumatic actuator used to drive the TLCD valve on the dynamics of the model structure was found to be negligible. This was accomplished by comparing the transfer functions with and without the air-supply to the pneumatic actuator. All transfer functions were obtained using SigLab™ spectrum analyzer by averaging a set of 15 measurements. From the transfer function and free vibration decay curves, the natural frequency and damping ratio of the uncontrolled building was determined to be 0.92 Hz and 0.6%, respectively. The mass ratio (ratio of the liquid mass in the damper to the first modal mass of the structure) is kept approximately 10% of the total mass of the structure. The nature of building model construction and unavailability of a light-weight pneumatic valve precluded a lower-mass ratio value normally desired in full-scale applications. However, a lower-mass ratio can be easily obtained in actual applications as the prototype valve mass will be much lower than the building mass.

Effect of Tuning Ratio

The tuning ratio γ is defined as the ratio of the natural frequency of the damper $(=\sqrt{2g/l})$ to the natural frequency of the structure. In order to determine the optimum tuning ratio, liquid columns of different lengths were considered. Fig. 2(a) shows the transfer function with different tuning ratios. The $\|H\|_2$ norm is used as a performance measure for evaluating the performance of each tuning ratio, which is defined as

$$\|H\|_2 \approx \int_{\omega_a}^{\omega_b} |H_{\ddot{X}_s \ddot{x}_g}(\omega)|^2 d\omega \tag{6}$$

where $\ddot{X}_s$=acceleration of the structure; $\ddot{x}_g$=shaking table acceleration; $\omega_a = 0.5\,\mathrm{Hz}$ and $\omega_b = 1.5\,\mathrm{Hz}$. The range of frequencies were limited to 0.5–1.5 Hz because below 0.5 Hz there was considerable noise in the system and above 1.5 Hz, there was negligible change in each transfer function. Fig. 2(b) shows the variation of the H_2 norm as a function of the tuning ratio. A fourth-order polynomial was fitted to the data to determine the optimum tuning ratio equal to 0.953, which corresponds to a liquid length of 25 in. (63.5 cm). In the present experiment, the damper parameters were set such that $\mu=0.1$, $\alpha=0.56$, and $\zeta_s=0.006$. Accord-

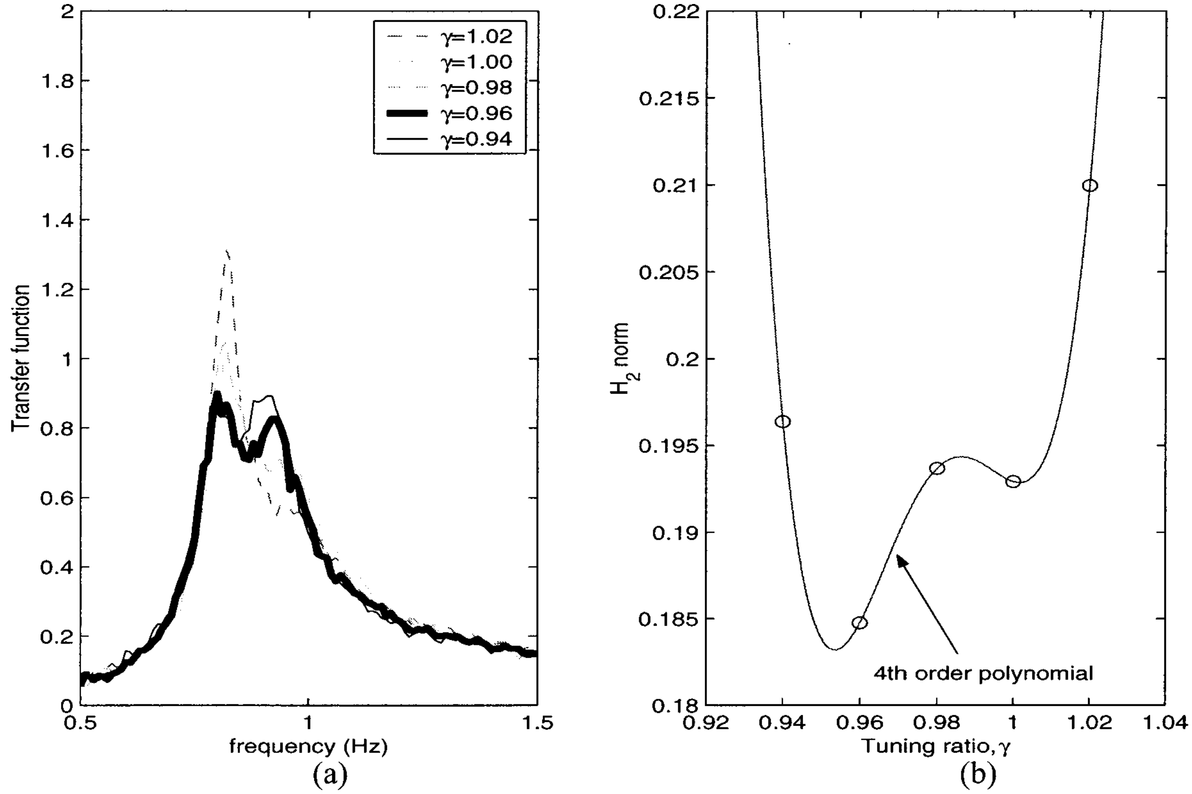

Fig. 2. (a) Transfer functions for different tuning ratios and (b) variation of H_2 norm with tuning ratio

ingly, optimum value of the tuning ratio γ_{opt} using expressions in Yalla and Kareem (2000) was found to be equal to 0.95. It is noteworthy that the experimental value matches well with the analytical model. In the locale of the optimum tuning the two peaks in the transfer function are almost equal in height, which is consistent with the analytical considerations (Den Hartog 1956).

Effect of Damping

The effective damping in the TLCD is obtained by changing the orifice opening of the valve. As noted in the section, ''Analytical Model'' the effective damping of the TLCD is an important parameter for optimum absorber performance. The semiactive TLCD system relies heavily on the ability of the damper to supply

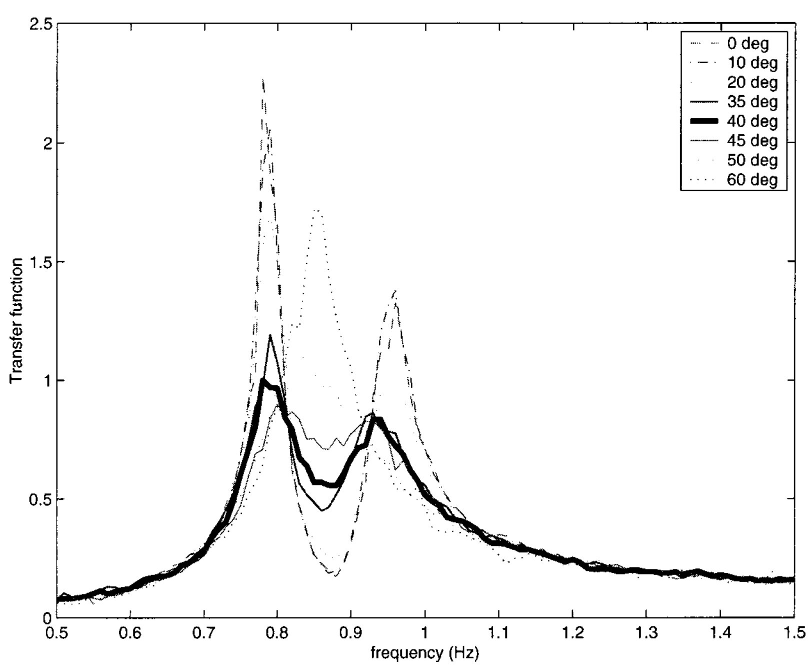

Fig. 3. Transfer functions for different valve opening angles

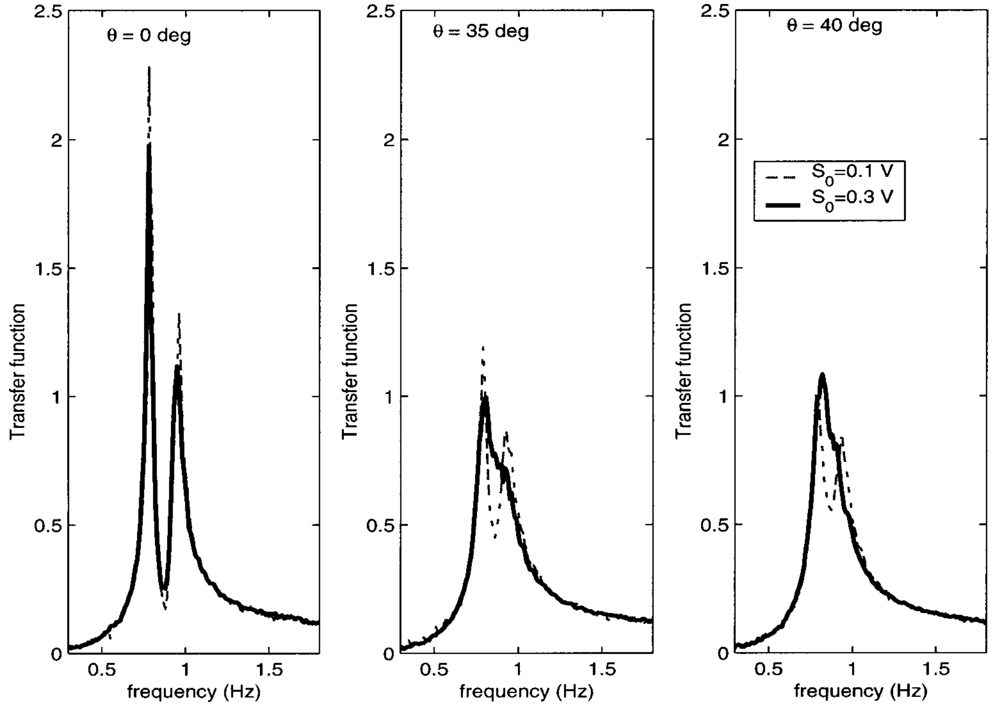

Fig. 4. Variation of transfer functions for different amplitudes of excitation

the desired effective damping. This damping is varied by changing the valve angle, where $\theta=0$ and $\theta=90°$ correspond to fully open and fully closed valve positions, respectively. In the fully closed position, no liquid oscillations take place and the system becomes a single-degree-of-freedom (SDOF) system. An upper limit of $\theta=60°$ is used in this study. At this position, the valve is almost fully closed. Fig. 3 shows how the transfer function of the coupled system changes as the valve opening angle is increased.

Effect of Excitation Amplitude

It is well known that the damping introduced in an oscillating liquid column through valves and orifices is quadratic in nature. This aspect has been studied experimentally for passive TLCDs (Sakai and Takaeda 1989; Balendra et al. 1995). The damping force is dependent on the liquid velocity in the tube

$$F_d=c|\dot{x}_f|\dot{x}_f \tag{7}$$

This implies that the damping introduced by the valve is nonlinear and changes as a function of the amplitude of excitation. Fig. 4 shows the transfer functions of the combined system at two different excitation levels, i.e., $S_0=0.1$ and 0.3 V with different valve angles. The transfer functions at $\theta=0°$ (fully open) with the exception of peak amplitudes are virtually identical as no nonlinearity is introduced due to the valve. At other valve openings, however, the nonlinearity introduced by the valve can be clearly noted.

From Fig. 4, it can be noted that the change in effective damping as the excitation amplitude is varied. Therefore, for the damper to perform optimally at all levels, one needs to determine the optimum damping required at each amplitude of excitation and to organize in the form of a *look-up* table. The main idea of a look-up table is to determine the angle of opening which minimizes the $\|H\|_2$ norm of the structural response. This corresponds to the optimal valve opening for a particular amplitude of excitation, as shown in Fig. 5(a) for $S_0=0.1$ V and $S_0=0.3$ V. This procedure is repeated for a range of amplitudes of excitation. Using these optimal values, one can construct a look-up table as shown graphically in Fig. 5(b).

In most tall buildings, building acceleration and/or liquid velocity measurements could be related to wind velocities. Accordingly, a similar look-up table, which relates the optimum headloss coefficient to wind velocities, can be easily established for semiactive control. The look-up table is discussed further in the section "Control Strategy" in a gain-scheduling framework.

Equivalent Damping

The equivalent damping of the TLCD is a function of the excitation amplitude and the valve opening. The experimental transfer functions were curve fitted by minimizing the norm of the error function. The equivalent damping was found to range from 2% (for fully open, $\theta=0°$) to 30% (for almost closed, $\theta=60°$). The optimal damping ratio was obtained as 9% ($\theta=40°$ at $S_0=0.1$ V) as seen in Fig. 6(a). Fig. 6(b) shows the transfer function with a nonoptimal damping (about 30%) which is realized at $\theta=60°$.

The optimum value of the damping ratio ζ_{opt} based on analytical considerations in Yalla and Kareem (2000) was equal to 8.9% which is in good agreement with the experimentally obtained value of $\zeta_{\mathrm{opt}}=9.0\%$. It should be noted that this optimum damping ratio is high due to the large mass ratio of 0.1, as compared to actual implementations where mass ratios are typically smaller.

Fig. 7(a) shows a three-dimensional (3D) plot of the magnitude of the experimental transfer function as a function of the valve opening angle/effective damping and the frequency at $S_0=0.1$ V. It is noteworthy that the double peaked transfer function changes to a single peak as the valve opening angle is increased. Fig. 7(b) shows the simulated 3D transfer function as a function of frequency and equivalent damping ratio. A similar curve is obtained by solving the actual nonlinear equations of the TLCD and plotting the dynamic magnification ratio as a function of fre-

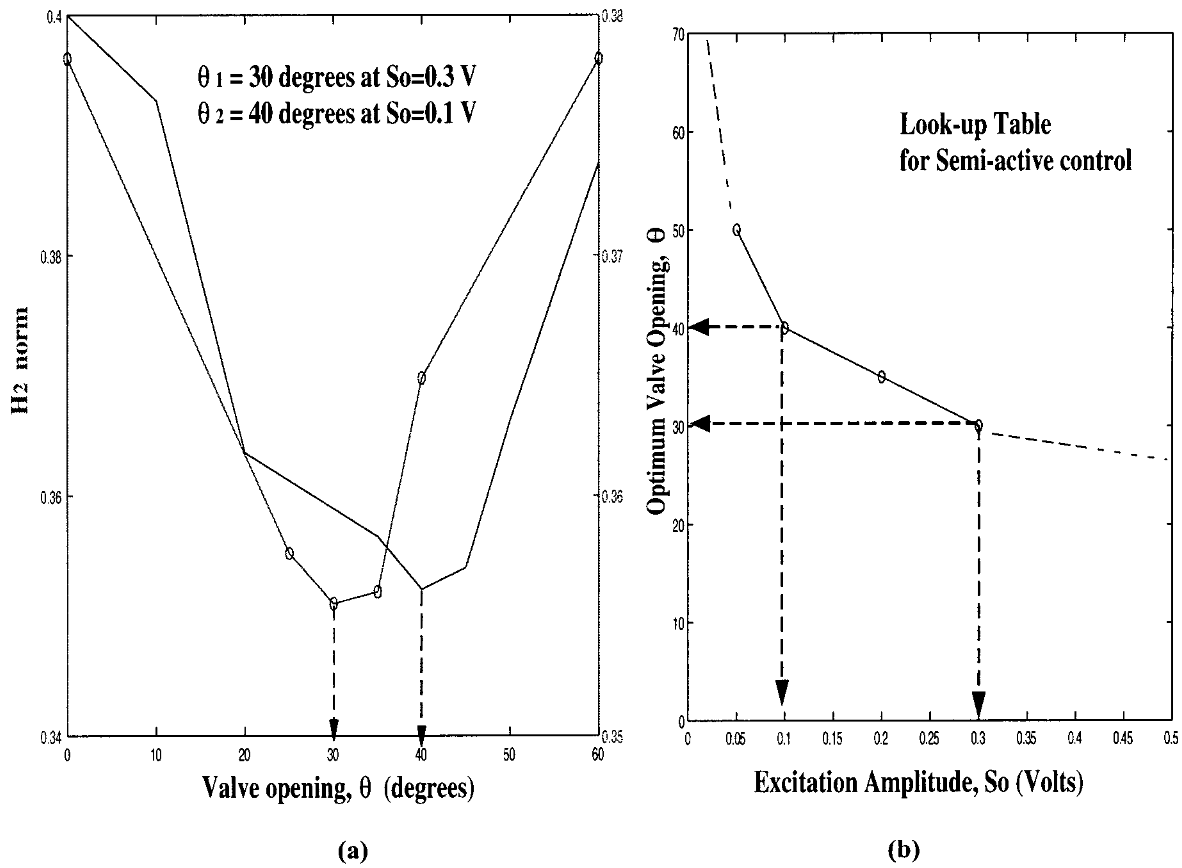

Fig. 5. (a) Optimization of H_2 norm and (b) look-up table for semiactive control

quency and the headloss coefficient (e.g., see Haroun and Pires 1994). The effect of coalescing of the modal frequencies, from a double peaked transfer function to a single peaked is also discussed in Yalla and Kareem (2001a) in which the *beat phenomenon* of the combined structure-TLCD system was examined.

The experimental results show that the damping depends on the amplitude of excitation and the valve angle opening, i.e.

$$\zeta_d = f(S_0, \theta) \tag{8}$$

In theory, the equivalent damping coefficient can be obtained by minimizing the mean square value of the error between the nonlinear and equivalent linear system, which results in the following expression for the equivalent damping:

$$\zeta_d = \frac{\xi \sigma_{\dot{x}_f}}{2\sqrt{\pi g l}} \equiv f(\sigma_{\dot{x}_f}, \xi) \tag{9}$$

By referring to the Appendix, one can note that the headloss coefficient is a function of the valve opening angle, i.e.

$$\xi = f(\theta) \tag{10}$$

while the standard deviation of liquid velocity is related to the amplitude of excitation in Eq. (3):

$$\sigma_{\dot{x}_f} = f(S_0) \tag{11}$$

Therefore, it follows that

$$\zeta_d = f(S_0, \theta) \equiv f(\xi, \sigma_{\dot{x}_f}) \tag{12}$$

Note that the damping is dependent on $\sigma_{\dot{x}_f}$ which in turn is dependent on ζ_d. This, therefore, implies that the relationship in Eq. (12) is a nonlinear function.

Control Strategy

Gain scheduling is defined as a special type of control scheme with a nonlinear regulator whose parameters are changed as a function of the operating conditions in a preprogrammed way (Astrom and Wittenmark 1989). As shown in Fig. 8(a), the regulator is optimized for each operating condition. Though gain scheduling is an open-loop compensation technique that may be time consuming to design, its regulator parameters can be changed very quickly in response to process dynamics. This kind of gain-scheduled/adaptive control is commonly used in aerospace and process control applications.

Comparing Figs. 8(a and b), the *look-up table* is the gain scheduler, the regulator is the controllable valve of the TLCD, and the headloss coefficient is the parameter being changed. The external environment is the wind acting on the structure and the process is the combined structure-TLCD system. Gain scheduling is an ideal control policy for maintaining optimal damping in TLCDs. The sensors on the building (e.g., anemometers) estimate the excitation level and the headloss coefficient is changed in accordance with the look-up table.

In Fig. 5(b), it was noted that the look-up table was generated using a white noise excitation. In order to extend it to wind excited structures, one needs to find a relationship between the wind force spectra $S_{FF}(\omega)$ and an "equivalent" white noise excitation.

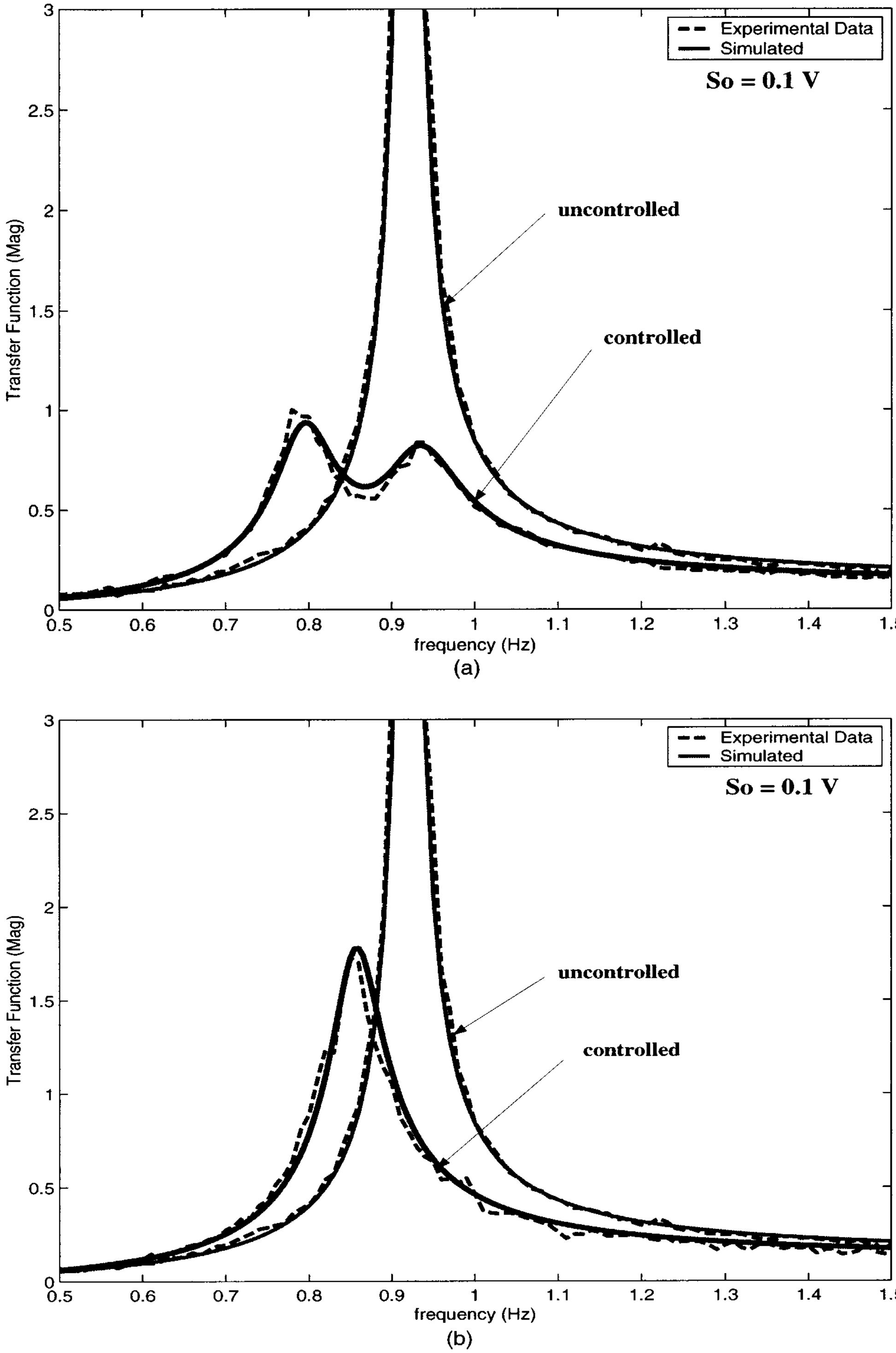

Fig. 6. (a) Comparison of transfer functions: (a) $\theta=40°$, $\zeta_d=9\%$ (optimal damping) and (b) $\theta=60°$, $\zeta_d=30\%$ (nonoptimal damping)

This can be done for small values of ζ_s, by approximating $S_{FF}(\omega)$ by an equivalent white noise level S_0, which is equal to $S_{FF}(\omega)$ evaluated at the natural frequency of the structure (e.g., Lutes and Sarkani 1997).

Experimental Verification

The next step involves experimental verification of the control strategy outlined in the previous section. The main idea was to benchmark the semiactive system performance to a passive system. In the case of a passive system the orifice is kept open, whereas in the semiactive case, the valve opening is changed to adjust damping according to the look-up table developed in Fig. 5(b).

For this experiment, two different loading time histories were selected. The first time history, referred to as Case 1, comprised of two segments each 20 s in length with a RMS excitation level of 0.1 and 0.3 V. The second time history (Case 2) comprised of two segments each 40 s in length with a RMS excitation level of 0.1

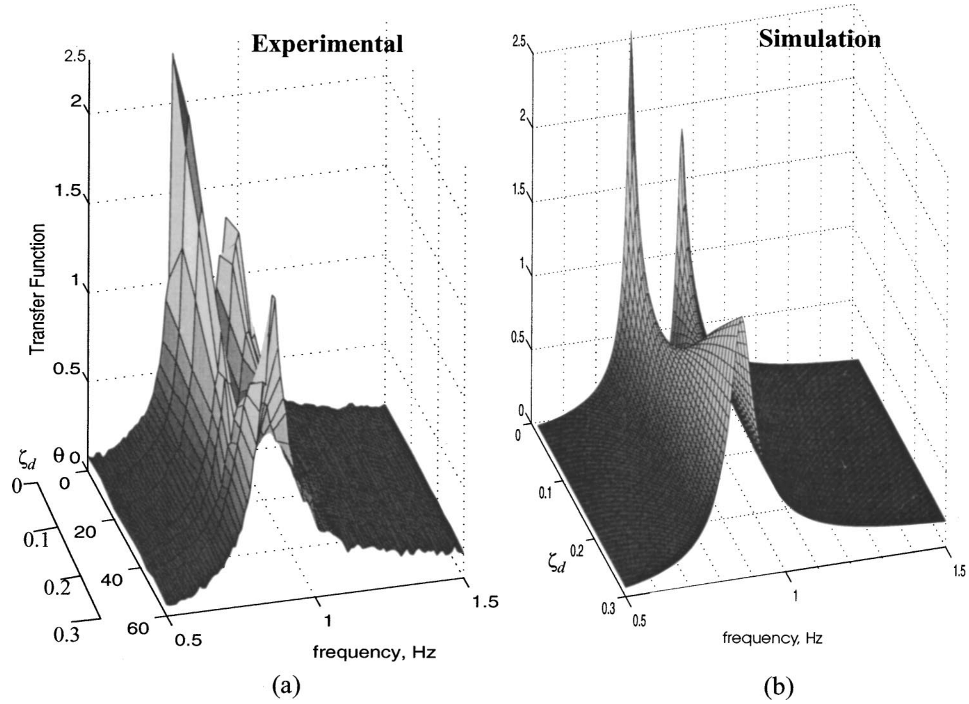

Fig. 7. 3D plot of transfer function as function of effective damping and frequency (a) experimental results and (b) simulation results

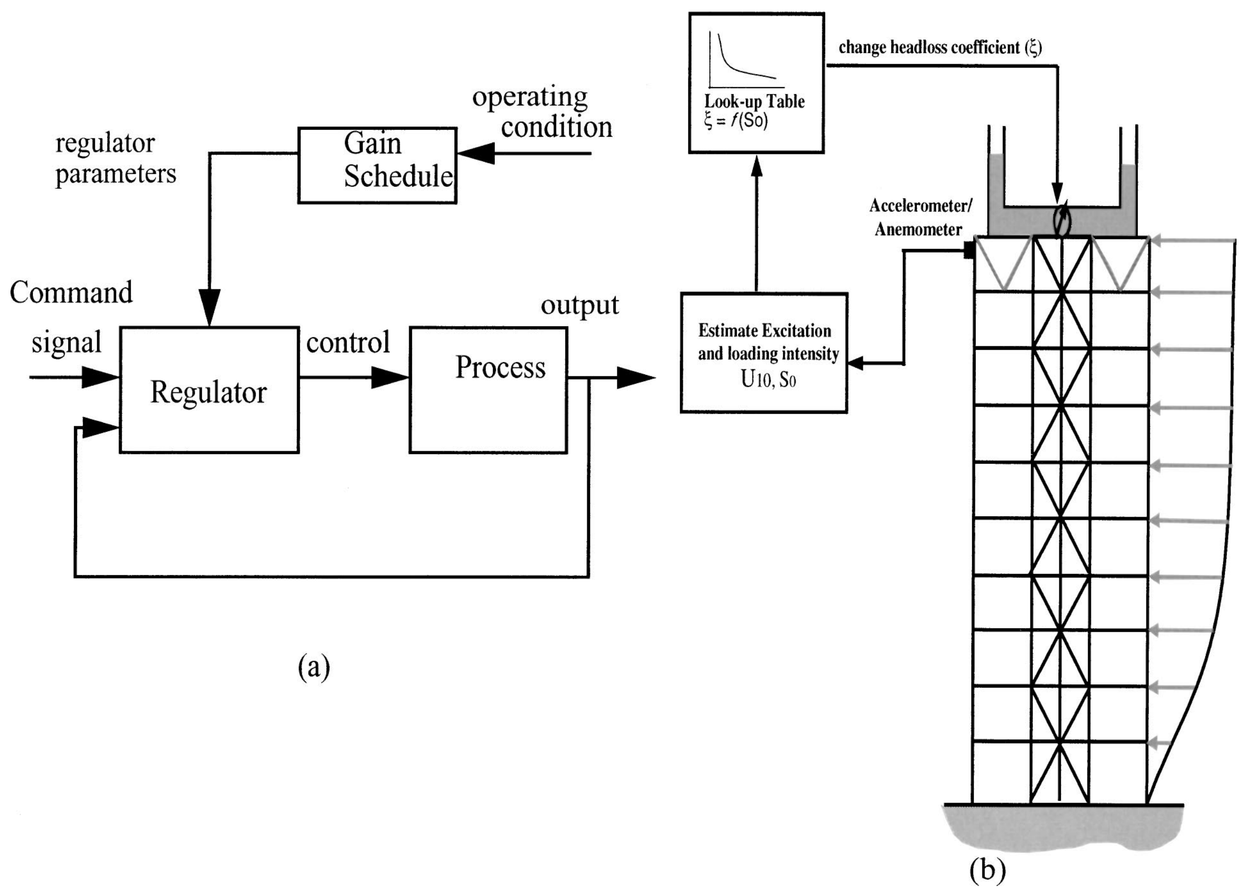

Fig. 8. (a) Gain scheduling concept and (b) Semiactive control strategy in tall buildings

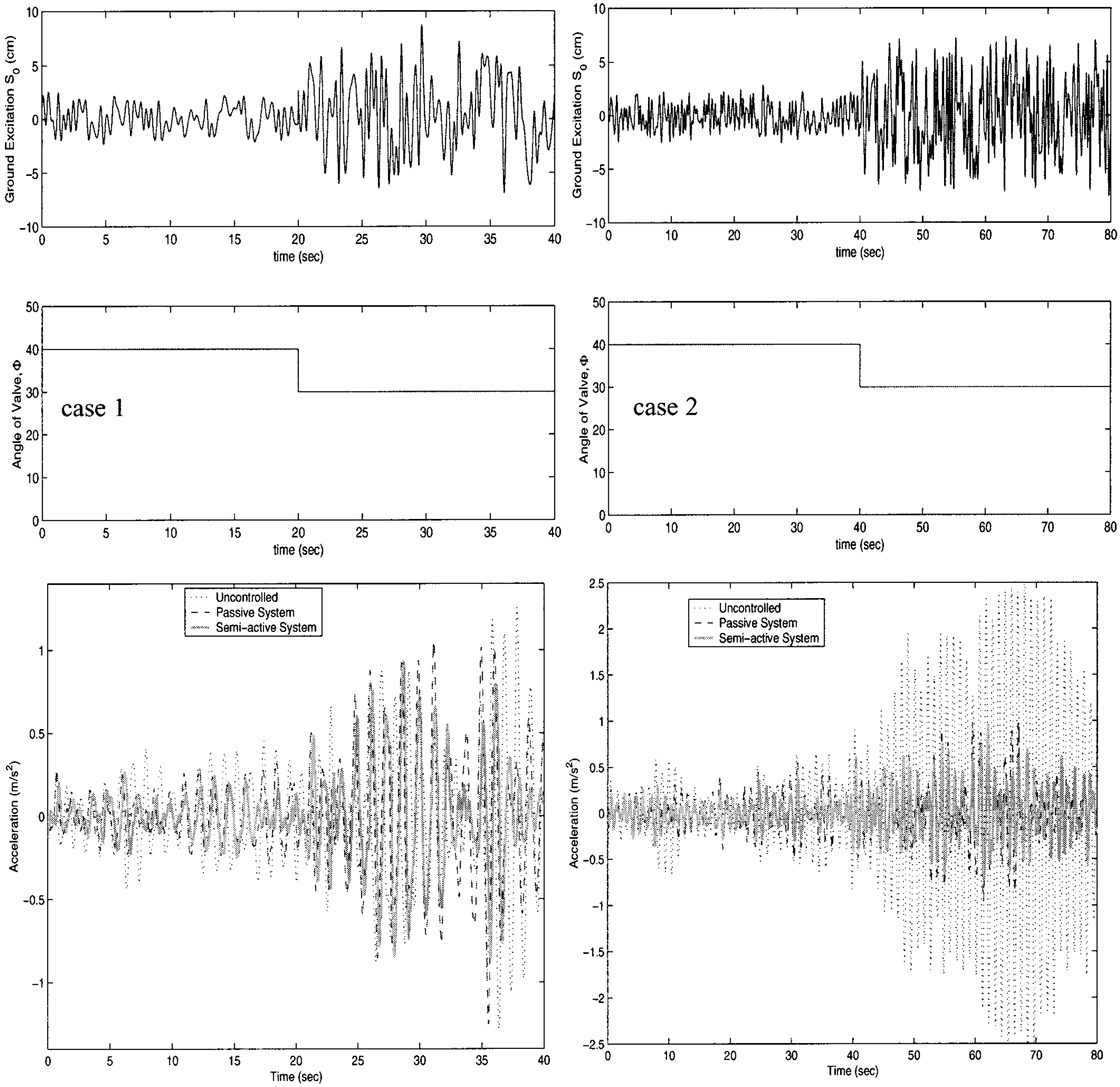

Fig. 9. Excitation time histories, valve opening angle, and acceleration response for uncontrolled, passive, and semiactive systems for loading Cases 1 and 2

and 0.3 V. The underlying objective was to assess the effectiveness of semiactive TLCD in which the headloss coefficient was adjusted in response to the changes in external excitation.

From Fig. 9 and Table 1, it was noted that for 0.3 V excitation, there was hardly any response reduction in Case 1, while there was a 76% reduction for Case 2. This is because the Case 2 record was of a longer duration and hence the excitation reached a steady state. This increased the liquid damper effectiveness as the liquid column was fully mobilized. At higher levels of excitation, a fully open valve was closer to the optimum damping, so the improvement of a semiactive system was not so substantial (about 13% improvement over the passive system). On the other hand, for lower levels of excitation, the improvement was more significant (about 27% improvement over the passive system). The overall RMS response reduction of the semiactive system over the passive system was 23% for Case 1 and 15% for Case 2. It is worth mentioning that the response reduction of 76% is very significant. As alluded to earlier, this is because the mass ratio of the damper considered in the scaled-down experiment was 10%.

Application to Wind-Excited Tall Building

Serviceability is an extremely important issue in the design of tall buildings under wind loading. There are primarily two types of serviceability problems caused by winds. The first concerns large deflections causing architectural damage to nonstructural members like glass panes, cladding, etc., and fatigue damage to struc-

Table 1. Performance of Semiactive System as Compared to Uncontrolled and Passive System

Time history	Root Mean Square (cm/s^2)	Peak (cm/s^2)	Root Mean Square (cm/s^2)	Peak (cm/s^2)	Root Mean Square (cm/s^2)	Peak (cm/s^2)
1	Portion 1: First 20 s		Portion 2: Next 20 s		Total 40 s	
Uncontrolled	20.17	45.08	46.65	125.57	35.94	125.57
Passive	13.69	32.0	45.30	105.25	33.46	105.25
	(32.1%)	(29%)	(2.8%)	(16.2%)	(6.9%)	(16.2%)
Semiactive	10.09	26.34	34.95	92.76	25.73	92.76
	(50.0%)	(41.6%)	(25.08%)	(26.1%)	(28.4%)	(26.1%)
Time history 2	Portion 1: First 40 s		Portion 2: Next 40 s		Total 80 s	
Uncontrolled	27.69	64.49	125.72	262.67	91.03	262.67
Passive	17.04	55.34	34.73	100.12	27.35	100.12
	(38.5%)	(14.2%)	(72.3%)	(61.8%)	(69.95%)	(61.8%)
Semiactive	12.56	40.86	30.2	95.02	23.15	95.02
	(54.6%)	(36.64%)	(75.97%)	(63.8%)	(74.56%)	(63.8%)

tural elements. The other is the oscillatory motion which may cause discomfort or even panic to the occupants. It is now widely known that acceleration and the rate of change of acceleration (commonly known as *jerk*) are the main causes of human discomfort (Kareem et al. 1999). Usually, risks of serviceability problems (i.e., excessive deflections or accelerations) are calculated assuming that failure occurs when the deflections or accelerations exceed a certain specified value.

The example considered in this paper was a 60 story, 183 m tall building with a square base of 31×31 m. The first five natural frequencies were 0.2, 0.58, 0.92, 1.18, and 1.34 Hz. The spectral characteristics of wind loads were defined in Li and Kareem (1990). Two types of TLCD systems were considered for the along-wind motion control. The first was a passive system with the TLCD tuned to the first mode frequency of the building while the damping ratio was not optimal. The second was a semiactive system, in which the damping was maintained at the optimal value at all levels of building motion. In the case of TLCDs, for the passive case, the damping was assumed to be arising due to the wall friction in the tube. The headloss coefficient for this case was assumed to be equal to 1, which is typical of such a system. In the case of the semiactive system, the optimal damping ratio of 4.5% was maintained at all levels of excitation by means of a controllable orifice using a gain-scheduled law as outlined in the previous section. The mass ratio is 1% and the tuning ratio was 0.99 which corresponded to a total mass of 280 tons and a length of 12 m of liquid. Multiple units of TLCDs of 1 m diameter can be used to accommodate the weight of the damper and these can be spatially distributed on the building roof or on a mechanical floor. Slightly detuned battery of multiple units of TLCD offer

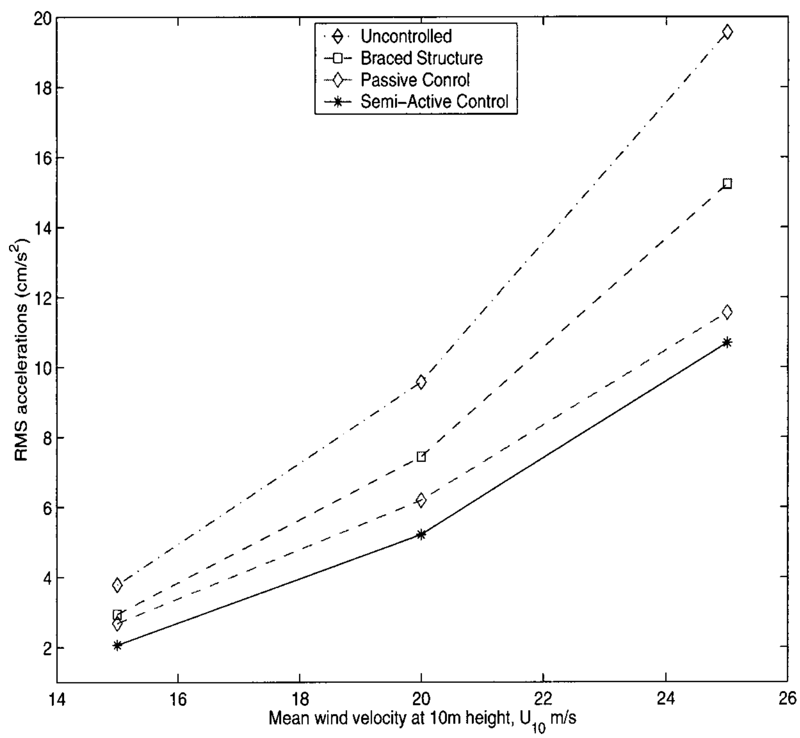

Fig. 10. RMS top floor accelerations versus wind speed

Table 2. Comparison of Different Systems for Varying Wind Conditions

	Root mean square displacement $U_{10}=15$ m/s (cm)	Root mean square displacement $U_{10}=20$ m/s (cm)	Root mean square displacement $U_{10}=25$ m/s (cm)	Root mean square acceleration $U_{10}=15$ m/s (cm/s^2)	Root mean square acceleration $U_{10}=20$ m/s (cm/s^2)	Root mean square acceleration $U_{10}=25$ m/s (cm/s^2)
Uncontrolled	2.37	5.97	12.19	3.79	9.57	19.56
Stiffened structure	1.54 (30.4%)	3.87 (35.1%)	7.92 (35%)	2.95 (22.1%)	7.44 (22.2%)	15.23 (22.1%)
Passive system	1.73 (23.4%)	3.93 (34.1%)	7.17(41.2%)	2.69 (29%)	6.20 (35.2%)	11.56 (40.9%)
Semiactive system	1.26 (40.6%)	3.18 (46.7%)	6.49(46.7%)	2.07 (45.4%)	5.22 (45.4%)	10.69 (45.3%)

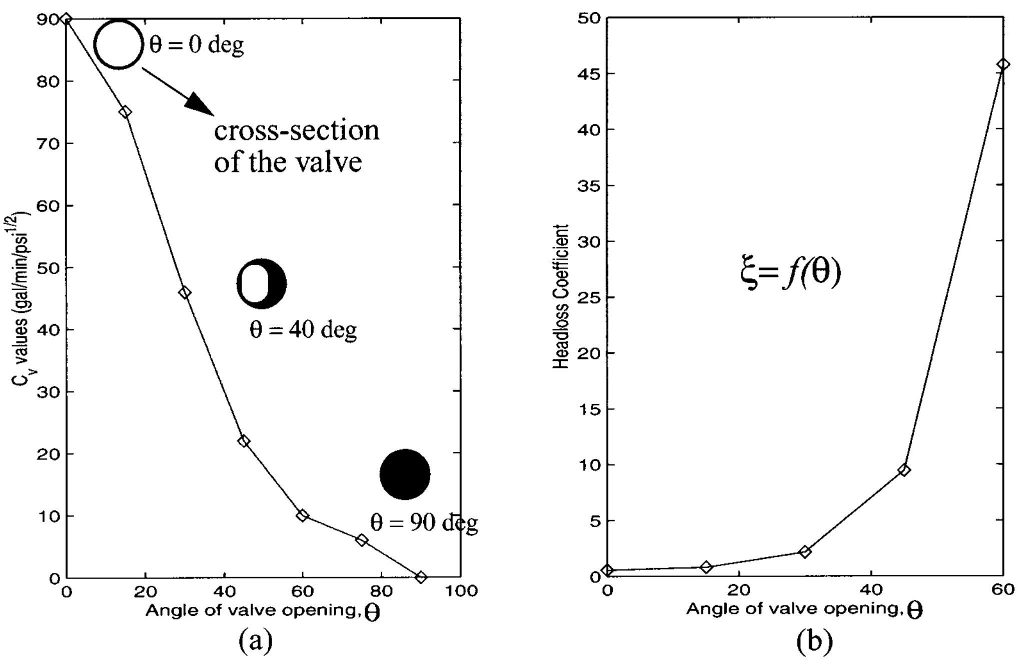

Fig. 11. (a) Variation of valve conductance and (b) variation of headloss coefficient with angle of valve opening

improved performance and robustness (Kareem and Kline 1995).

The RMS acceleration of the uncontrolled, passive and semiactive system are plotted as a function of the mean wind velocity, U_{10} (Fig. 10). Both Fig. 10 and Table 2 point at the effectiveness of the dampers in reducing structural motion. In this analysis, the effect of bracing the structure was also examined. It was assumed that the super-structure stiffness increased by the addition of different bracing systems by 20%. Table 2 shows, however, that the bracing system was effective in reducing displacements but not so effective in reducing accelerations. Moreover, bracing can increase overall building costs significantly due to the amount of steel required.

Table 2 results suggest that by using a semiactive system there was an additional improvement of 10–25% over the passive systems in terms of RMS acceleration response for the range of wind velocities considered. The semiactive system offered a 45% improvement over the uncontrolled system. This improvement justifies the additional costs associated with the semiactive system, e.g., sensors, controllable valves, etc. The main advantages of TLCDs are their low initial and maintenance costs and the fact that most tall buildings need water tanks for the building water supply for occupant's usage and fire-fighting purposes, making them a viable and attractive choice over other mechanical vibration absorbers.

Concluding Remarks

The performance of a structure-TLCD system was investigated experimentally. The experimental results were compared to the previously obtained analytical results and were found to be in good agreement. A control strategy based on a gain-scheduled look-up table was presented and verified experimentally. It was observed that at lower amplitudes of excitation, higher damping was achieved by constricting the liquid flow through the orifice and at higher amplitudes opening of the orifice and higher liquid velocity contributed to the appropriate level of damping. The semiactive TLCD can enhance performance of the passive TLCD with fixed orifice by 15–25%. This justifies the additional cost of using sensors and controllable valves in the system. Finally, an example of a tall building under along-wind loading was used to demonstrate the advantage of using semiactive TLCDs in comparison with a typical bracing.

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Notation

The following symbols are used in this paper:

A = cross-sectional area of tube;
b = horizontal length of column;
C_s = damping in primary system $= 2M_S\zeta_s\omega_s$;
C_V = valve conductance;
c_{eq} = equivalent damping of TLCD $= 2m_d\omega_d\zeta_d$;
D = diameter of valve;
$\|H\|_2$ = H_2 norm of transfer function;
K_s = stiffness of primary system;
k_f = stiffness of liquid column $= 2\rho Ag$;
l = length of liquid column;
M_s = mass of primary system;
m_d = mass of liquid in tube $= \rho A$;
S_0 = RMS of white noise excitation;
$S_{FF}(\omega)$ = spectra of wind force excitation;
U_{10} = mean wind velocity at 10 m height;
X_s = response of primary system (structure);
x_f = liquid displacement in damper (TLCD);
$\ddot{x}_g$ = ground excitation;
α = length ratio $= b/l$;
γ = frequency ratio $= \omega_d/\omega_s$;
ζ_s = damping ratio of primary system;
ζ_d = damping ratio of TLCD;
θ = angle of valve opening;
μ = mass ratio;
ξ = coefficient of headloss;
ρ = liquid density;
$\sigma_{\dot{x}_f}$ = RMS value of liquid velocity in TLCD;
$\sigma_{\ddot{X}_s}$ = RMS value of acceleration of structure;
ω_d = natural frequency of liquid damper; and
ω_s = natural frequency of primary system.

Appendix

A 1.5 in. ball valve has been used for the experimental study described in this paper. The valve manufacturer provided the valve conductance values as a function of the valve opening angle [Fig. 11(a)]. According to the derivation given in Yalla and Kareem (2001b),

$$\xi = \frac{\pi^2 D^4}{8C_V^2} \tag{13}$$

can be used for generating Fig. 11(b).

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