〈Original Papers〉

Semiactive Control for Structural Vibration Mitigation

구조물 진동 저감을 위한 반능동 제어

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ABSTRACT

Past research has repeatedly demonstrated the fact that hydraulic semiactive systems, if operated properly, can provide levels of control authority in structural vibration control systems that are comparable to a fully active hydraulic damper. The performance of the semiactive system when used to provide vibration mitigation for a laboratory test structure is described in this paper. Numerical and experimental verification of the effectiveness of the proposed bistate controller which relies on a Lyapunov approach that seeks to dissipate the energy of the system is also presented. The results based on the bistate control are compared with those of two different control strategies. The work indicates that hydraulic semiactive actuator provides a reliable, and inexpensive means of achieving structural control.

요 약

지금까지의 여러 연구들에서 유압식 반능동 시스템은 적절하게 동작을 한다면 전능동 유압식 댐퍼 만큼의 구조물 진동저감 능력이 있음을 끊임없이 보여주고 있다. 이 논문에서는 축소 구조물에 설치된 반능동 시스템의 진동저감 성능을 기술하고 있다. 본 논문에서 제안한 시스템의 에너지를 소산시키기 위해 리아푸노브방법을 적용한 바이스테이트 제어의 효과를 수치적 및 실험적으로 입증한 결과들을 먼저 제시한다. 또한 바이스테이트 제어 성능을 다른 두 제어기와 비교 평가하였다. 이 연구결과를 통해 반능동 시스템은 구조물 진동저감에 저렴하면서도 효과적임을 보여준다.

1. Introduction

The lexicon of control engineering has expanded during the past two decades to include various techniques and approaches to system control that do not fit classical definitions. One relatively recent modification is that systems, previously treated as either active or passive, are now examined in terms of an intermediate possibility: semiactive. The introduction of a new descriptor has been adopted to make clear the nature of the power needed to achieve the control objective. At one extreme, a passive control design is comprised of a collection of components with non-time varying characteristics. Properly selected passive components can very often achieve a best solution. This is especially true if the dynamics of disturbances are known a priori. Examples of passive control design include hydraulic shock absorbers on automobiles, and elastomeric bearings that are routinely used to mitigate the dynamic response of structures to a seismic event.

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A purely passive design requires no external power source to achieve vibration mitigation.

Active control systems represent the other limited control design. Active controllers typically rely on the availability of an external energy source to power an actuator, which is in turn regulated to achieve prescribed objectives. The power required to operate an active control system is (in general) assumed to be of the order of the power dissipated from a vibrating system. Active systems are often plagued by features that make them less effective than might have been desired. Realities such as saturation, backlash friction, and actuator dynamics can severely compromise the sought for performance of an active control design.

The middle ground between a passive motion management design and active control designs for a system has emerged that presents today's control system engineer with a much-expanded family of control solutions. While no hard and fast definition can be pointed to, this middle ground is typically referred to as semiactive(SA), or parameter adaptive control. The SA label was first introduced by control engineers in the automotive industry. The most common qualifier of a SA control system is a prescription for the extent of external power utilized by the actuator, relative to the energy (power) managed (or dissipated) by the actuator; it must be small . In general, a SA actuator provides judiciously selected levels of compliance during a dynamic event. The varied compliance, in its simplest form, might be linear damping and/or linear stiffness. Indeed, almost all of the articles that treat SA control design that appear in the open literature prior to 1990 discuss the SA actuator as a linear damper with selectable levels of damping[1~3]. Continuously variable linear stiffeners have also received attention[4~5]. The articles assumed an ideal actuator or component in which the proposed linear compliance characteristic could be designed into the hardware.

The nature of the SA device employed in an actual application governs the dynamics of the actuator which is typically complex, nonlinear, and generally coupled to the dynamics of the system that is to be controlled[6~7]. Those considerations must be addressed by the engineer before a system design is finalized. While a simplified analysis assumes that the actuator can afford automatically selectable level of linear damping or stiffness, the actual hardware dynamics should be relied on at some point early in the design to confirm that the control does achieve desired levels of performance[8~9].

Leitmann[5] demonstrated the control strategy for SA systems that is based on a Lyapunov approach. That control scheme has different switching conditions compared to a conventional on-off controller(10). Patten et al.[11] have also proposed the SA control algorithm based on a Lyapunov stability theorem in which the controller represents on-off control. The performance of that control scheme when applied to a SAVA system was verified in the areas of structure both analytically and experimentally[11~12]. Hitada and Smith[13] presented a nonlinear controller using variable damping devices for civil structure under earthquake excitations based on Lyapunov stability theorem. That proposed controller takes the form of filtered bang-bang control.

The paper will demonstrate that a bistate control using Lyapunov approach provides a rigorous technique for discovering the best possible control action at each point in time. Numerical and experimental verification of the effectiveness of the bistate control for SA system is presented.

2. A Bistate Control

2.1 Control Law[14~15]

The effectiveness of the proposed SA device is examined first. A vibration test assembly (Fig. 1) was constructed, which consisted of a two degree of freedom assemblage of masses and springs. The masses were mounted on linear bearings. An active hydraulic cylinder was attached through a spring to one of the masses. The system represents the essential dynamics of a two-story structure with base excitation.

The experiment was conducted to provide the effectiveness of the proposed controller and comparisons of the performance for two cases which depend on the location of the SA damper.

1) Case 1: SA damper is on between the 1^{st} and the 2^{nd} floor

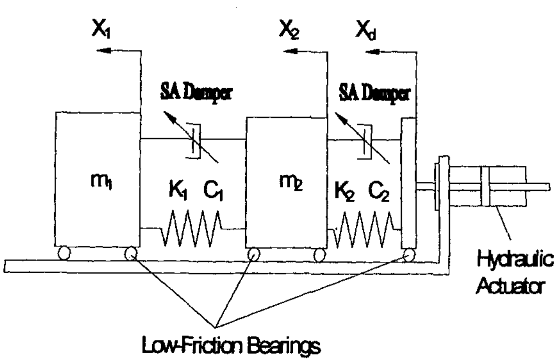


Fig. 1 A Vibrating stand

2) Case 2: SA damper is on between the base and the 1st floor

The state space form of the system for Case 1 (with the hydraulic SA damper and the external disturbance) is:

$$\dot{X} = AX + Bg(X)u + D \tag{1}$$

where

$X = [x_1, x_2, \dot{x}_1, \dot{x}_2, \triangle P]$,

$g(X)\, sgn(X_5)\sqrt{\dfrac{2\,|X_5|}{\rho}}$, $u = A_v$, and

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ \dfrac{k_1}{m_1} & \dfrac{k_1}{m_1} & 0 & 0 & \dfrac{A_p}{m_1} \\ \dfrac{k_1}{m_2} & \dfrac{(k_1+k_2)}{m_2} & 0 & 0 & \dfrac{A_p}{m_2} \\ 0 & 0 & \alpha A_p & -\alpha A_p & 0 \end{bmatrix}$$

$$B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ -\alpha C_d \end{bmatrix} \quad D = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \dfrac{k_2}{m_2} x_d \\ 0 \end{bmatrix}$$

where, ρ: density of the fluid, X_5 is the fifth state variable, $\triangle P$: differential pressure, $\alpha = \beta\,(V_1 + V_2)/(V_1 V_2)$, β : bulk modulus, V_1, V_2: volume of each chamber, A_p: effective face area of piston, C_d: valve loss coefficient, A_v: valve opening area

A bistate controller is developed here that provides a method for the automatic regulation of the SA system. The procedure relies on a Lyapunov approach, which is initiated by selecting a candidate function or functional that is a mapping into a scalar. The following function is typically used:

$$V = \frac{1}{2} X^T Q^* X,\ Q^* > 0 \tag{2}$$

The first time derivative of V is:

$$\dot{V} = \frac{1}{2} X^T [A^T Q^* + Q^* A] X + X^T Q^* Bg(X)u + X^T Q^* D \tag{3}$$

The following form is next adopted.

$$A^T Q^* + Q^* A = -P,\ P > 0 \tag{4}$$

Let $Q^* = [\,\hat{q}_1, \hat{q}_2, \hat{q}_3, \hat{q}_4, \hat{q}_5\,]$ be a column matrix and recalling the definition of $g(X)$, then equation (3) can be rewritten as:

$$\dot{V} = -\frac{1}{2} X^T P X - \alpha C_d \sqrt{\frac{2\,|X_5|}{\rho}}\, X^T \hat{q}_5\, sgn(X_5) A_v + X^T \hat{q}_4 \frac{k_2}{m_2} x_d \tag{5}$$

The objective of the proposed controller is to drive the state to the origin rapidly by maximizing $-\dot{V}$ for given V. If a matrix pair (P, Q^*) can be found which satisfies equation (4) such that both P and Q^* are positive definite, then the first term on the right hand side of equation (5) is guaranteed to be negative definite. The last term in equation (5) is problematic. The disturbance, in the work considered here is not known a priori. The work here ignores the expression because there is no control variable involved. Next, noting that $\alpha C_d \sqrt{\dfrac{2\,|X_5|}{\rho}}$ is always positive, then the most obvious way to make the second term on the right hand side of equation (5) negative semidefinite is to impose the following bistate control logic:

$$\begin{cases} A_v = A_{v\min} & \text{if} \quad X^T \hat{q}_5\ sgn(X_5) < 0 \\ A_v = A_{v\max} & \text{if} \quad X^T \hat{q}_5\ sgn(X_5) \geq 0 \end{cases} \tag{6}$$

The vector $\hat{q}_5$ provides a means of weighting the different states to emphasize a particular control objective.

Similarly, it is straightforward to find the switching law with the modified state space form of equation (1) for Case 2.

2.2 Experimental Results and Analysis

The work presented here utilized a simple hydraulic architecture to implement SA technology[16]. A DC servomotor-controlled single-stage valve was selected to regulate the flow from one chamber to another. The orifice area of the valve was selectable between the fully open and a second position near or at the closed position. A PC based ADA conversion board was used to interface between the PC and the analog channels(inputs and outputs). The parameter values obtained by rigorous identification procedure of a mock structure for each case are listed in Table 1. Accelerometers and LVDT's were mounted to measure absolute acceleration of each mass and relative displacement between floors.

Table 1 Parameter values for experiment and simulation

Symbols	m_1	m_2	k_1	k_2
Unit	kg	kg	N/m	N/m
Case 1	131	136	45746	28420
Case 2	144	193	28420	48470
Symbols	c_1	c_2	ω_1	ω_2
Unit	N/m/sec	N/m/sec	Hz	Hz
Case 1	100	645	4.50	1.52
Case 2	645	100	1.63	3.46

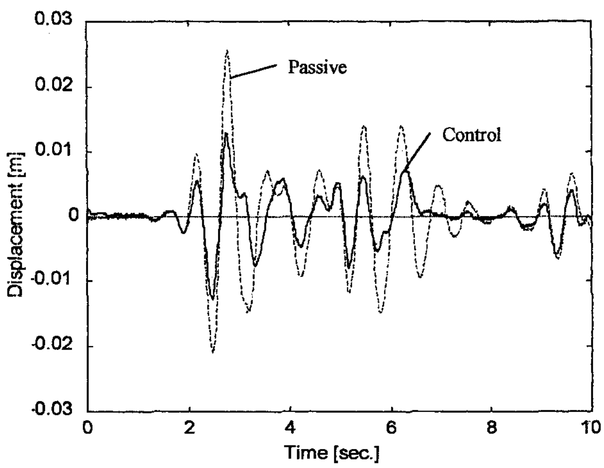


Fig. 2 Comparison of relative displacement ($x_1 - x_2$) response for the second floor: Case 1

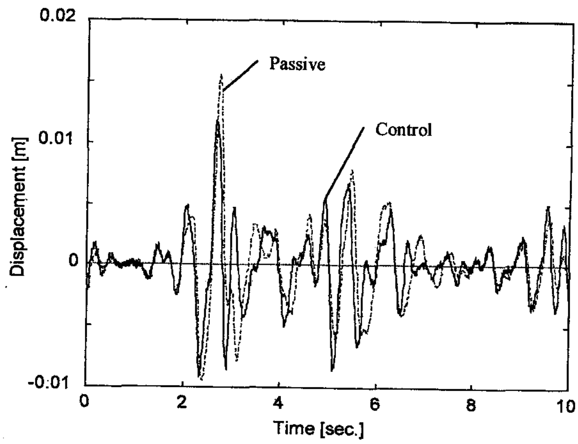
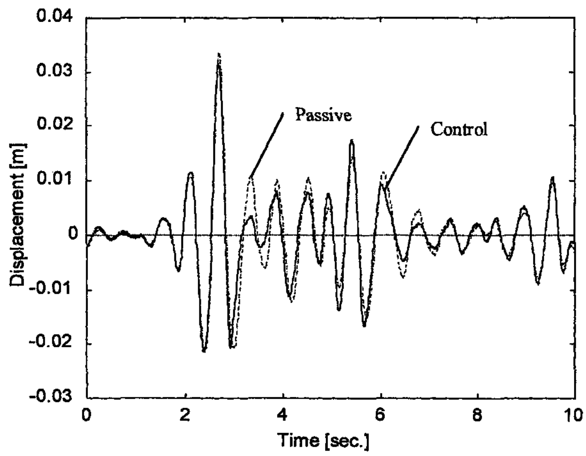
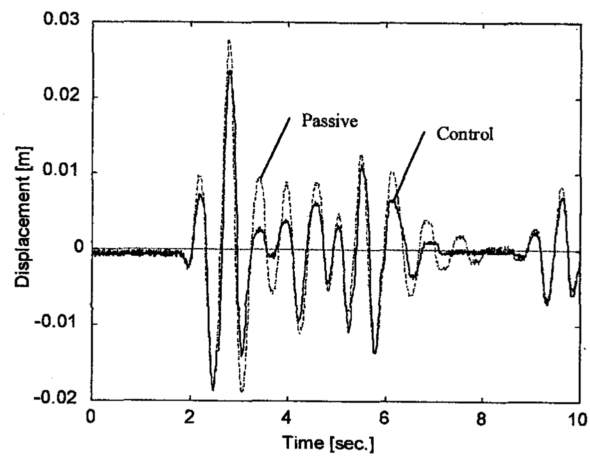


Fig. 3 Comparison of relative displacement ($x_2 - x_d$) response for the first floor: Case 1



Time[sec.]

Fig. 4 Comparison of relative displacement ($x_1 - x_2$) response for the second floor: Case 2



Time[sec.]

Fig. 5 Comparison of relative displacement ($x_2 - x_d$) response for the first floor: Case 2

Table 2 Reduction of relative displacement response by the bistate control[%]

	Peak		RMs	
	1st floor	2nd floor	1st floor	2nd floor
Case 1	24.4	49.3	9.3	47.7
Case 2	14.7	6.5	15.9	3.9

An open loop control experiment was conducted first. An earthquake time history (Elcentro 1940) was used to excite the structure. The SA control valve was fixed during the test, thus providing a fixed passive damper. While the damping characteristic is nonlinear, it can be demonstrated that the particular configuration produced about 9.2%/5% added damping to the first mode and 8%/16.5% to the second mode of the structure for the case 1/case 2. Next the system was subjected to same excitation, but the valve orifice area of the SA damper was adjusted in accordance with the control law (equation (6)). Figure 2 and Fig. 3 depict the open-loop and the closed-loop relative displacement responses for Case 1. The open-loop and the closed-loop relative displacement responses for Case 2 are shown in Fig. 4 and Fig. 5. Table 2 lists the peak and RMS (root mean square) reduction of the relative displacement for both Case 1 and Case 2. The open-loop maximum peak responses are reduced by 49.3% (the second floor) and 22.4% (the first floor) for Case 1, while 6.5% (the second floor) and 14.7% (the first floor) reductions for Case 2 are achieved.

The results provide the importance of the location of the SA damper. It can be noted that the SA system provides only marginal performance on the floor where the SA damper is not installed.

3. Comparison of Different SA Control Strategies

The purpose of this section is to provide a comparison of the performance of various control laws that are often suggested for the regulation of the SA system. The control strategies include a heuristic rule, the clipped optimal control, and the bistate control.

The comparisons were made relative to the performance of the proposed bistate control. In order to examine the performance of each design, a simulation of a two story laboratory test structure was conducted. Two configurations depended on the location of SA damper were considered as in the previous section.

The effectiveness of each control design was established by comparing the RMS and maximum amplitude reduction of the relative displacements and accelerations. The parameter values listed in Table 1 were used for the simulation.

3.1 A Heuristic Rule(HR)

A generic control algorithm that has been proposed previously for application to SA automotive suspensions[3,17] and structural vibration suppression[8,16] is considered here first. The control law is defined as follows:

$$\begin{cases} c = c_{\min} & \text{if } (x_1 - x_2)(\dot{x}_1 - \dot{x}_2) \le 0 \\ c = c_{\max} & \text{if } (x_1 - x_2)(\dot{x}_1 - \dot{x}_2) > 0 \end{cases} \tag{7}$$

This rule was proposed by Rakheja and Sankar[3] as an alternative to the sky hook damper[1] to avoid the need to estimate (or measure) the absolute velocity $\dot{x}_1$. It can be realized by a single sensor like a LVDT and a software filter to establish the relative velocity

3.2 Clipped Optimal (CO)

This approach to the control of the SA system has been suggested previously to provide a suboptimal means of regulating automobile dampers[18~19]. The control law can be found by minimizing a performance index. According to the desired performance, weighting factors for the absolute accelerations of the second ($\ddot{x}_1$) and first ($\ddot{x}_2$) floors, and for the relative displacements of the second ($x_1 - x_2$) and first ($x_2 - x_d$) floors can be selected. Unlikely to the active LQR design, the implementation of the control is constrained by the fact that the SA actuator is only capable of dissipating energy. If the dissipative rule is not satisfied, then the control force is set as close as possible to 0. That is accomplished by adjusting the valve to its maximum opening area.

Table 3 RMS reduction for Case 1

Control Algorithms	Relative displacement [m]		
	1^{st} floor	2^{nd} floor	$1^{st}+2^{nd}$
HR	1.2%	-22.5%	-21.3%
CO	11.8%	-9.4%	2.4%
BC	0.0067	0.9683	
Control Algorithms	Absolute acceleration [m/s^2]		
	1^{st} floor	2^{nd} floor	$1^{st}+2^{nd}$
HR	-3.3%	-12.5%	-13.8%
CO	14.8%	17.0%	31.8%
BC	0.00217	1.11	

Negative values indicate larger values relative to the BC.
Positive values indicate smaller values relative to the BC.

Table 4 Maximum-peak reduction for Case 1

Control Algorithms	Relative displacement [m]		
	1^{st} floor (x_2-x_d)	2^{nd} floor (x_1-x_2)	$1^{st}+2^{nd}$
HR	-2.0%	-38.3%	-40.3%
CO	7.3%	-71.0%	-63.7%
BC	0.0473	0.0108	
Control Algorithms	Absolute acceleration [m/s^2]		
	1^{st} floor ($\ddot{x}_2$)	2^{nd} floor ($\ddot{x}_1$)	$1^{st}+2^{nd}$
HR	-21.4%	-38.0%	-59.4%
CO	8.7%	48.1%	56.8%
BC	6.656	13.1	

Negative values indicate larger values relative to the BC.
Positive values indicate smaller values relative to the BC.

Table 5 RMS reduction for Case 2

Control Algorithms	Relative displacement [m]		
	1^{st} floor (x_2-x_d)	2^{nd} floor (x_1-x_2)	$1^{st}+2^{nd}$
HR	-22.1%	-22.4%	-44.5%
CO	16.4%	-17.6%	-1.2%
BC	0.0017	0.0015	
Control Algorithms	Absolute acceleration [m/s^2]		
	1^{st} floor ($\ddot{x}_2$)	2^{nd} floor ($\ddot{x}_1$)	$1^{st}+2^{nd}$
HR	-38.8%	-22.4%	-61.2%
CO	-33.5%	-17.6%	-51.1%
BC	0.537	0.512	

Negative values indicate larger values relative to the BC.
Positive values indicate smaller values relative to the BC.

Table 6 Maximum-peak reduction for Case 2

Control Algorithms	Relative displacement [m]		
	1^{st} floor (x_2-x_d)	2^{nd} floor (x_1-x_2)	$1^{st}+2^{nd}$
HR	-31.2%	-46.4%	-77.6%
CO	27.2%	-24.0%	3.2%
BC	0.0097	0.0097	
Control Algorithms	Absolute acceleration [m/s^2]		
	1^{st} floor ($\ddot{x}_2$)	2^{nd} floor ($\ddot{x}_1$)	$1^{st}+2^{nd}$
HR	-95.3%	-46.4%	-141.7%
CO	-1.5%	-23.9%	-25.4%
BC	8.059	3.389	

Negative values indicate larger values relative to the BC.
Positive values indicate smaller values relative to the BC.

3.3 Simulation Results and Analysis

The simulation was first conducted to examine the performance of each of the three control designs when a SA damper is on between the 1^{st} and the 2^{nd} floor. Table 3 lists the RMS and maximum peak values of the response of the relative displacements and absolute accelerations. The comparisons are offered relative to the performance of the bistate control(BC). Negative values in Table 3 indicate that the response is larger relative to the BC. It is noted that larger relative displacements and accelerations are less desirable. The results in Table 3 indicate that there is no one "best" controller. The RMS relative displacement of the HR is 30.8% better at the first floor than the BC, yet the relative displacement between the first and second floor is -35.9%(larger). The acceleration responses (RMS) using the HR are reduced (22.2% and 34.4%) from those that result from the BC.

In order to make a decision on which design is best, the increase (or decrease) of motion (displacement or acceleration) for each floor was added as listed in Table 3 and Table 4. The nature of the problem (seismic protection) indicates that the maximum peak measurements are the most important criteria (Table 4) and that peak displacements are more critical than peak accelerations. Given that prioritization, then one can conclude that BC provides significantly greater seismic protection than do the HR and the CO.

The work next examines the performance of each of the three control designs when a SA actuator is on between the base and the 1^{st} floor.

The results of the simulations are listed in Table 5 (RMS) and Table 6 (maximum peak values). The results in this case also suggest that the BC design provides the best performance.

4. Conclusion

The performance of the semiactive system when used to provide vibration mitigation for a laboratory test structure is described. The effectiveness of the proposed bistate controller which relies on a Lyapunov approach that seeks to dissipate the energy of the system was verified experimentally. The results also provide the importance of the location of the SA damper. It can be noted that the SA system provides only marginal performance on the floor where the SA damper is not installed.

The results based on the bistate control are then compared with those of two different control strategies. The algorithms provide comparable results, indicating that the bistate control is an adequate means of providing orifice area regulation for the semiactive design in terms of seismic protection.

The work presented indicates that a hydraulic semiactive actuator provides a reliable, and inexpensive means of achieving structural vibration mitigation.

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