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EFFECTIVE FORCE TESTING WITH NONLINEAR VELOCITY FEEDBACK COMPENSATION

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SUMMARY

Effective force testing (EFT) is a test procedure that can be used to apply real-time earthquake simulations to large-scale structures. Servo-system nonlinearities are significant for testing structures with large hydraulic flow demands, which can be caused by large structure velocities and/or large effective forces. This paper first reviews the nonlinear servo-system modeling and the design of nonlinear velocity feedback compensation, and then explains a procedure for the lab implementation of the EFT method. The feasibility of EFT with the nonlinear velocity feedback compensation was evaluated by comparing response of a single-story steel structure incorporating two fluid dampers tested on a shake table and subsequently tested using the EFT method. Global responses (e.g., displacement, velocity, and acceleration) and local responses (e.g., damper force and column shear) were compared. The test results indicate that with the nonlinear velocity compensation, dynamic forces and real-time seismic simulation can be successfully applied to the test structure.

INTRODUCTION

Passive or semi-active damping devices are used worldwide to mitigate seismic damage to buildings and bridges. Design and control algorithms for these energy dissipation devices are typically evaluated using computer simulation. However, the accuracy of the simulation results depends on the characterization of the damping devices. Real-time dynamic testing is likely to yield more accurate information regarding the behavior of structures employing velocity dependent devices under seismic loading. A shake table is often used to simulate the dynamic effects of earthquakes on structural models. However, structures tested on shake tables typically have to be scaled down due to limited table capacities. At smaller scales, structural details such as connections cannot be represented realistically, and energy dissipation of structural control devices may not be demonstrated accurately.

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Effective force testing (EFT) is a dynamic testing procedure to apply real-time earthquake loads to large-scale structures that can be simplified as lumped mass systems. As schematically shown in Figure 1, the test structure in an EFT test is anchored to a stationary base, and dynamic forces are applied by hydraulic actuators to the center of each story mass of the structure. The force to be imposed (effective force) is the product of the structural mass and the ground acceleration record, and thus is independent of the structural properties such as stiffness and damping, and their changes during the test. Motions measured relative to the ground are equivalent to the response that a structure can develop relative to a moving base as in a shake table test or an earthquake event.

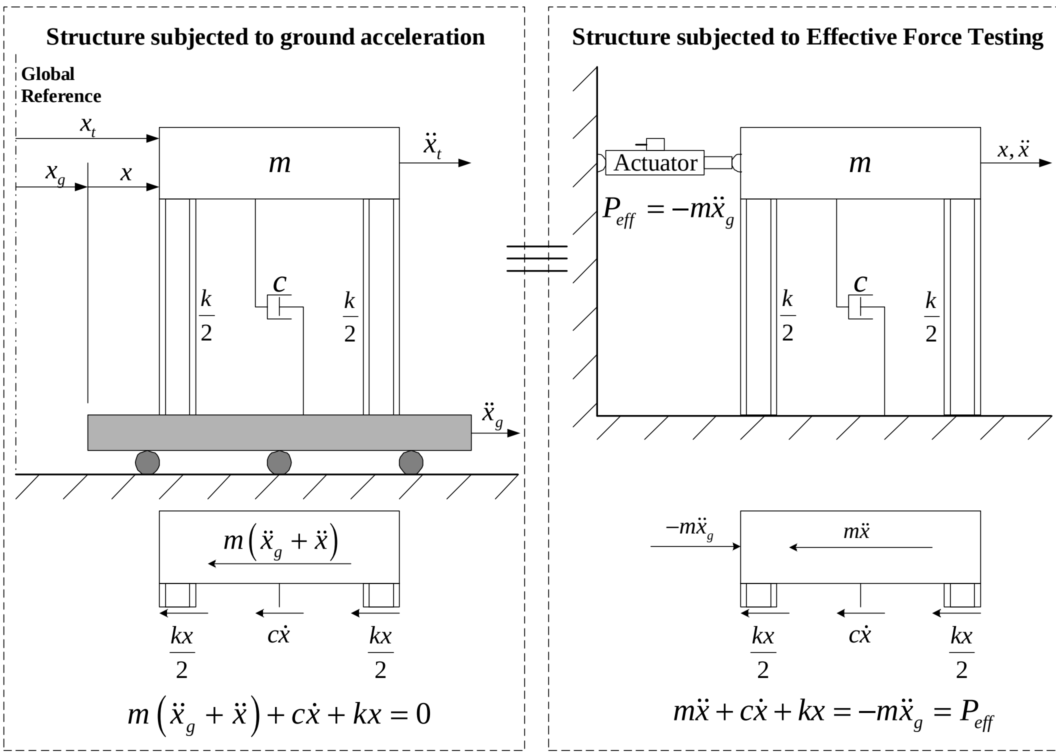

Figure 1 Concept of Effective Force Testing

Early investigation of the EFT method indicated that in the direct implementation of EFT, the actuator was not able to apply forces accurately near the natural frequency of the test structure (Dimig et al. 1999), which was attributed to the natural velocity feedback in the servo-hydraulic system (i.e., the interaction between the actuator piston velocity and the actuator control) identified by Dyke et al. (1995) in a study of active structural control. The effect of the natural velocity feedback can be explained as follows: the actuator is controlled by a servovalve through hydraulic flow under pressure, and the differential pressure inside the actuator chambers causes the force applied to the test structure. The actuator chamber volumes change due to the actuator piston moving with the structure, resulting in unwanted chamber pressure variation. A standard Proportional-Integral-Derivative (PID) controller is unable to compensate for the pressure variation near resonance, thus causing force-tracking errors.

The natural velocity feedback needs to be compensated to successfully implement EFT. In the velocity feedback correction proposed by Dimig et al. (1999) and verified by Shield et al. (2001), the effect of the natural velocity feedback was compensated by modifying the command to the servovalve (i.e., a compensation signal was added to the servovalve command signal). The compensation signal was determined as the product of the measured piston/structure velocity and the piston area, and multiplied by the inverse of the forward system dynamics before being added to the original command signal. The compensation algorithm is independent of the structural properties (i.e., damping and stiffness) and their changes during a test.

Characterization of the forward system dynamics is critical because the modified command signal compensates for the effect of the piston motion after going through the forward dynamics. Nonlinearities of the servo-system can have significant impact on the performance of the velocity feedback compensation (Zhao et al. 2003a; Zhao et al. 2003b), especially when a test involves large effective forces and/or large velocities compared to the actuator/servovalve capacities. The large forces and/or velocities usually cause large hydraulic flow demands. Hence, nonlinear velocity feedback compensation is necessary to extend the EFT method to fully utilize the equipment.

This paper first presents a review of nonlinear servo-system modeling and nonlinear velocity feedback compensation followed by a procedure for implementing the EFT method. The application of the EFT method with the nonlinear velocity feedback compensation to a simple one-story steel structure with viscous dampers is presented. The test results are compared with those of a shake table study to validate the EFT method and the nonlinear velocity feedback compensation scheme.

NONLINEAR SYSTEM MODELING

Figure 2 presents a block diagram for the EFT system, which shows the relations between system components (blocks with inputs and outputs labeled), including the interaction between the actuator piston velocity and the actuator control (natural velocity feedback). During a test, the servovalve controller compares the command signal to a feedback signal (both converted from force to voltage signals by C_F) and sends an amplified error signal (v) to the servovalve (H_s) to drive the valve spool. The spool regulates the hydraulic flow entering the actuator (Q_L), causing differential pressure across the actuator piston (P_L). The pressure difference multiplied by the piston area (A) produces the force (F) applied to the test structure. The force measured by a load cell on the actuator piston is finally fed back to the controller to close the control loop.

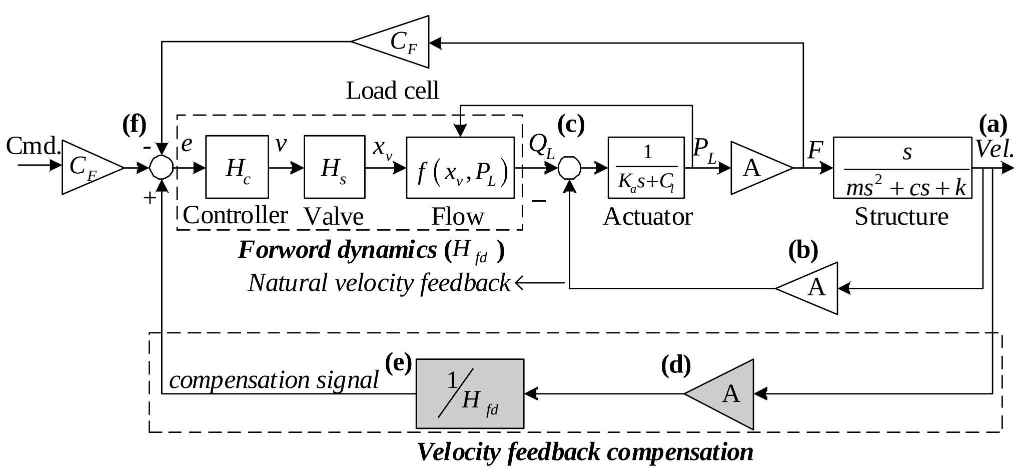

Figure 2 A block diagram model of an EFT system

The natural velocity feedback is shown by the loop (a)-(b)-(c) from the structural velocity to the summing point (c), which represents the law of conservation of mass: the hydraulic flow into the actuator needs to counteract the fluid compressibility(K_a), system leakage(C_l), and chamber volume change ($A\dot{x}$). The effect of the natural velocity feedback loop (a)-(b)-(c) is compensated by a positive feedback loop (a)-

(d)-(e)-(f). In order to cancel the effect of the natural velocity feedback at point (c), the compensation loop needs to incorporate the inverse of the dynamics between (f) and (c) (forward dynamics H_{fd}), which represents the behavior of the servovalve and its controller. Therefore, the servovalve and its controller need to be characterized in detail.

The mathematical models of the servo-system have been derived by Zhao et al. (2003) based on the formulations by Merritt (1967). The dynamics of the three-stage servovalve (H_{fd}) contain three major components: (1) the proportional-integral-derivative (PID) control with zero I gain,

$$H_c = G_p + G_d s \, , \tag{1}$$

where G_p and G_d are the proportional and derivative gain of controller, respectively; (2) the second-order servovalve dynamics,

$$H_s = \frac{K_{vp}}{\tau A_v s^2 + A_v s + K_3 K_{vp}} \frac{1}{x_{v\max}} \, . \tag{2}$$

where τ is the equivalent time constant of the pilot-stage valve, K_{vp} is the pilot-stage valve flow gain, A_v is the main-stage spool area, K_3 is the sensitivity factor of the internal LVDT, and x_{vmax} is the maximum spool stroke; and (3) the nonlinear servovalve flow characteristic stated by

$$Q_L = K_v x_v \sqrt{1 - \frac{x_v}{|x_v|}\frac{P_L}{P_s}} \tag{3}$$

where x_v is the spool opening of the servovalve (-1 to 1), K_v is the no-load flow gain of the servovalve, which is a function of spool opening, P_L is the load pressure ($P_L A$ is approximately the force applied to the structure, and A is the actuator piston area), and P_s is the supply pressure. The servovalve flow relation includes two types of nonlinearity: the **load pressure influence** expressed by the square root term and the **nonlinear no-load flow gain** (K_v) (Zhao 2003).

The forward dynamic components are shown in detail in Figure 3. In order to implement the velocity feedback compensation, the inverse of the component dynamics is needed.

NONLINEAR VELOCITY FEEDBACK COMPENSATION

To inverse the nonlinear servovalve flow characteristic, the chamber volume variation to be compensated ($A\dot{x}$) was first multiplied by $1 \Big/ \sqrt{1 - \frac{x_v}{|x_v|}\frac{P_L}{P_s}}$ to consider the effect of large forces applied to the structure (load pressure influence). This process required two more inputs, the spool opening (x_v) and the load pressure (P_L). The spool opening was obtained directly from the servovalve controller while the load pressure was approximated by the applied force divided by the piston area. Secondly, the nonlinear no-load flow gain (K_v) was represented by a piecewise linear curve (servovalve flow vs. spool opening); hence, the inverse relation was simple once the flow curve was identified: A look-up table based on the piece-wise linear flow curve was used to find the required spool opening to compensate the flow to the actuator.

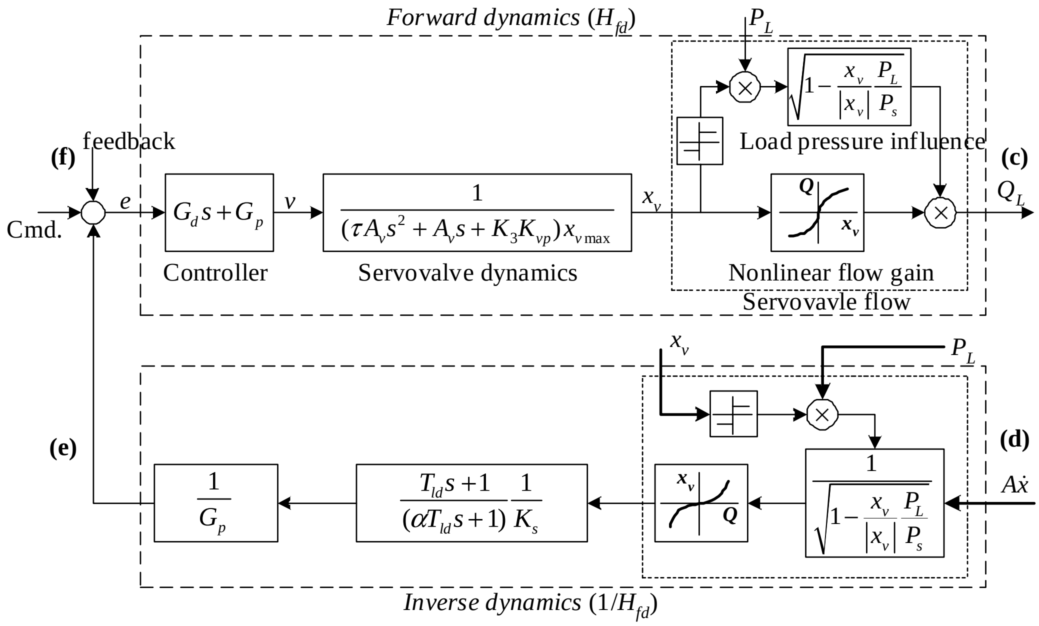

Figure 3 Nonlinear velocity feedback compensation design

The direct inverse of the servovalve dynamics results in a transfer function with a second-order term in the numerator, corresponding to an inherently unstable system because it can greatly amplify signals with high frequencies, such as electrical noise. For the frequency range of interest (i.e., 0-10 Hz in this study), the servovalve dynamics shown in Eq (2) was represented with reasonable accuracy by a first-order delay ($K_s/(T_d s+1)$, where K_s is the valve gain) with a time constant (T_d); hence, a first-order phase-lead network was used to invert the valve dynamics.

$$H(s)=\frac{1}{K_s}\frac{T_{ld}s+1}{\alpha T_{ld}s+1} \text{ and } T_{ld}=\frac{T_d}{1-\alpha}. \tag{4}$$

where the constant α was taken as 0.1 because it could provide both good phase-lead performance (the performance would be reduced if α were too large) and acceptable noise amplification (noise would be greatly amplified if α were too small).

PID control with a zero I gain introduces some phase lead into the DC error signal if the derivative gain (controller D gain) is not zero. Because the D gain was usually set very small (e.g., 0.2 ms) in this study (systems with large D gain would amplify noise signals), the controller dynamics were simplified as a pure gain, and the related phase lead was lumped into the dynamics of the servovalve: the lead-time (G_d/G_p) was considered by reducing the servovalve response delay (T_d). Hence, the inverse dynamics of the servovalve controller was simply $1/G_p$.

The inverse of the forward dynamics and the design of the nonlinear velocity feedback compensation are presented in Figure 3. The lab implementation of the EFT method with the nonlinear velocity feedback compensation is discussed in the next section.

LAB IMPLEMENTATION OF EFT

In addition to a servo-hydraulic controlled actuator and a data acquisition system, the implementation of the EFT method requires the following hardware: a velocity transducer to measure the piston/structure velocity ($\dot{x}$) and determine the actuator chamber volume variation ($A\dot{x}$), a Digital Signal Processor (DSP) and a host computer. This section presents a typical procedure for testing structures using the EFT method. Note that the discussion shown below includes some empirical quantities obtained in this study.

Equipment Capacity

Equipment capacity includes the load capacity of the actuator and the flow capacity of its servovalve. The load capacity of an actuator can be found in its product specification (e.g., 156 kN (35 kips) for an MTS 244.23 actuator), or estimated by 90% of the supply pressure times the actuator piston area, which can be found in the actuator specification. The servovalve flow capacity was estimated as $Q_{\mathrm{rated}}\sqrt{0.9P_s/P_{\mathrm{srated}}}$, where Q_{rated} is the rated flow of the servovalve at a pressure drop of P_{srated} across the servovalve (e.g., 341 lpm (90 gpm) for an MTS 256.09 three-stage servovalve under 6.9 MPa (1000 psi)) and P_s is the supply pressure (roughly 21 MPa (3000 psi) in this study). The calculated flow capacity was limited by other factors in the hydraulic system, such as the capacity of the pump and service manifold, and the diameter of hydraulic supply hoses. Accurate flow capacity of a servovalve was obtained as presented in a later section.

During an EFT test, the maximum structural velocity should be smaller than 80% of the servovalve flow capacity divided by the actuator piston area, and the maximum effective force should be smaller than 50% of the actuator load capacity. If the maximum force likely happens at the same moment as the maximum velocity (i.e., the effective force input has significant content near the resonant frequency of the test structure), the maximum spool opening should be smaller than 60%. Refer to Spink (2002) for an actuator/servovalve sizing technique.

Structural Properties Identification

Static loading tests and free vibration tests were used for structure identification. Note that the effective force command is directly related to the structure mass, and errors in the structural mass estimation would affect the force applied to the structure and potentially the nonlinear structural behavior. In addition, the stability of the test system is related to the structural damping. Testing of a structure with a minimum of 2% critical damping using EFT can be conducted with reasonable confidence (Zhao 2003). The identified structural properties were used to estimate the peak structural responses for the capacity check of equipment and sensors.

Servovalve Dynamics Identification

The inverse servovalve dynamics shown in Eq. (4) requires the valve gain (K_s) and the response delay (T_d), which may be determined using the second-order servovalve model shown by Eq. (2). However, equation (2) requires many valve parameters such as valve spool area and the maximum spool stroke. If the valve parameters are not available, a measured frequency response can be used to estimate the parameters for an equivalent second-order model, from which the valve gain and the response delay can be estimated.

$$H_s = \frac{K_s}{\frac{1}{\omega_v^2}s^2 + \frac{2\zeta_v}{\omega_v}s + 1}. \tag{5}$$

A test was conducted to generate a frequency response plot, in which the actuator was in displacement control, the actuator piston was kept in its neutral position, and the hydraulic supply to the main-stage valve was turned off. The proportional gain of the servovalve controller was set to unity and the derivative gain set to zero, such that the valve command signal could be controlled without additional equipment. A sine wave sweep (0-100 Hz in 100 seconds) with amplitude equivalent to 20% spool opening was chosen as the input signal. The spool opening was obtained as an output from the servovalve controller. The obtained frequency responses are shown in Figure 4, in which the magnitude response is the Fourier magnitude ratio of the output to the input signal, while the phase response is the phase difference between the two signals.

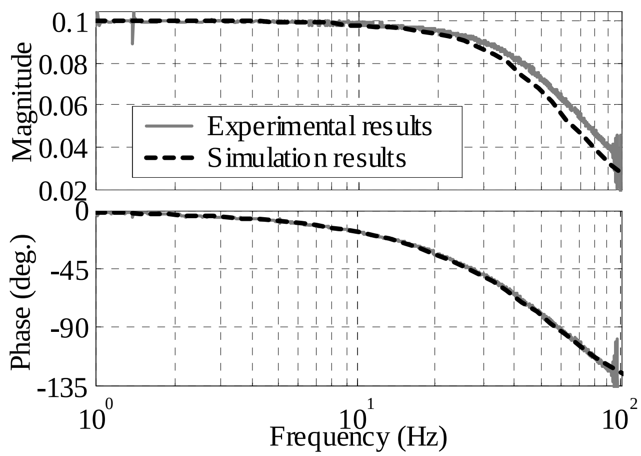

Figure 4 Measured servovalve dynamics

The amplitude corresponding to the asymptotical line of the magnitude response gave a valve gain (K_s) of 0.1. An equivalent natural frequency of 57.5 Hz and a damping of 80% fit the phase response well, from which the response delay was found to be 4.4 ms for low frequencies. It should be noted that the servovalve dynamics can be affected by system nonlinearities, and the response delay increases with an increase in the hydraulic demand. To account for this increase, a response delay of 5 ms was found to be appropriate for this study through small amplitude tests as stated in a later section (Tests with Small Amplitude Sinesweep Input).

Servovalve Flow Property Identification

Dynamically measuring the servovalve flow using a flow meter was deemed inappropriate because flow meters typically do not respond quickly. Hence, a flow curve (i.e., the flow controlled by servovalve Q_L vs. the main-stage spool opening x_v) was constructed as follows. The actuator was put into displacement control with a unity controller P-gain and zero D-gain. Tests were conducted under no load condition (the structure was disconnected) such that the pressure difference across the actuator piston (load pressure P_L) was negligible. The spool opening was obtained directly by measuring the main-stage spool position while the corresponding flow was calculated as the piston velocity multiplied by the piston area ($A\dot{x}$). The piston velocity was calculated using the central difference method from the measured piston displacement. Note that the leakage flow was neglected in the above calculation because the leakage was typically small (less than 0.5% of valve capacity).

Sinusoidal inputs with a frequency of 3 Hz were used to control the spool opening. Refer to Zhao (2003) for a procedure to determine the input frequency. Tests with 90% (4.5 in.), 80% (4 in.), 60% (3 in.), 40% (2 in.), and 20% (1 in.) full stroke command were conducted. A piecewise linear curve that connected 21 control points at intervals of 10% of the spool opening was constructed to represent the flow property of the servovalve. The flow values at the control points were calculated as the average value of all test results. The resulting piecewise linear flow curve is compared to the test result with the 90% full stroke command in Figure 5. It is noted that the real flow property of the servovalve (as shown in grey dots) scatters, indicating some instantaneous over-or under- compensation for the natural velocity feedback when the compensation is based on the flow curve.

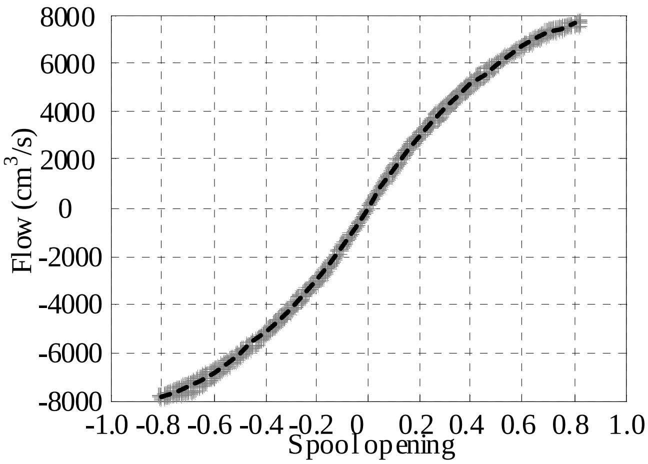

Figure 5 Servovalve no-load flow property

Determination of Maximum Controller P Gain

Relatively large P gains should be used because they usually improve the overall performance of a stable system. On the other hand, larger controller P gains may cause instability, and result in a high-frequency vibration of the actuator. The maximum controller P gain can be obtained through trial and error; however, a stability analysis of the test system as shown in Zhao (2003) could be used to provide a guideline. A unity P gain was used in this study.

Tests with Small Amplitude Sinesweep Input

Small amplitude tests with sinesweep inputs were conducted before the "real" tests to investigate if the identified parameters (i.e., the response delay and flow property of the servovalve, and the controller gains) were suitable for use in "real" tests. The following observations were made:

- When the P gain was too large, a high frequency vibration was excited even with zero command.
- When the flow curve underestimated the real flow property of the servovalve, a large amplitude spike resulted at the natural frequency of the test structure or the system became unstable with vibration at the resonant frequency of the structure. On the other hand, tests based on an underestimated flow curve had a sharp amplitude drop at the natural frequency of the structure.
- With insufficient delay compensation, a peak before a valley appeared in the FFT of the measured force while a peak after a valley appeared in the frequency domain when the delay was overcompensated.

Refer to Zhao (2003) for detailed analysis and experimental verification of the effect of the various parameters on the performance of the EFT system.

Following this procedure, the EFT method with the nonlinear velocity feedback compensation was applied to a one-story steel structure, and the results were compared to a companion shake table study.

EXPERIMENTAL VALIDATION OF EFT

Experimental Program

A simple one-story structure was selected for the study. The structure consisted of a rigid diaphragm (a rectangular steel frame filled with reinforced concrete) supported on four replaceable steel columns at its corners as shown in Figure 6. The shake table study was conducted at the University of Illinois at Urbana-Champaign. The concrete mass weighed about 44.5kN (10.1 kips) to fit the load capacity of the table, and the column spacing was 1.52×1.83 meters (60×72 inches) to fit the hole-pattern of the base plate of the table. Four plates with tapped holes were welded on the steel frame of the diaphragm to provide connections for the columns.

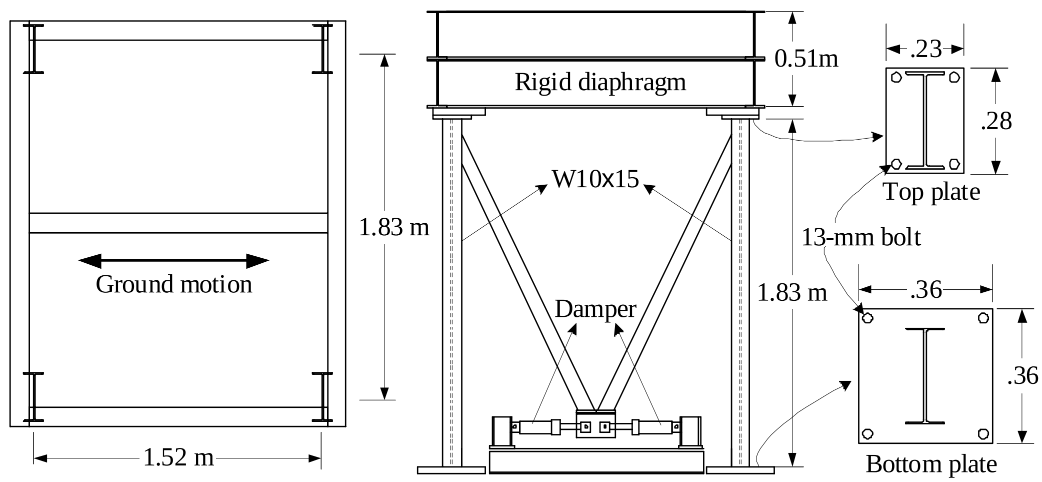

Figure 6 One-story test specimen

The columns were made of W10x15 sections (A572 grade 50 steel) with a reported yield stress of 431.1 MPa (62.5 ksi). The columns were 1.83 m (72 inch) high and oriented in weak-axis bending such that the resonant frequency of the structure was low enough to be excited by most earthquake ground acceleration records. The structural stiffness in the orthogonal direction was much (20 times) larger than that in the motion direction such that out-of-plane motion was prevented without additional diagonal braces. In order to minimize the effects of the connections on the comparison of the dynamic response of the structure, the columns were welded to a 38-mm (1.5-inch) plate at the bottom and a 25-mm (1-inch) plate at the top. The end plates were bolted to the diaphragm and base using four 13-mm (½-inch) diameter A490 bolts.

The middle chevron brace connected two Taylor Devices® fluid dampers to limit the structural responses such that tests could be repeated and results compared. The behavior of the dampers was found to be nonlinear as shown in Figure 7; hence, the structural behavior was difficult to predict even when the columns were in their linear range of behavior.

In the shake table study, the columns were bolted to a 13-mm (½-inch) thick base plate, which was bolted to the diaphragm of the table; while in the EFT study, the columns were bolted to a 19-mm (¾-inch) thick base plate, which was anchored to the strong floor. Due the difference in the column boundary conditions, the structural stiffness during the EFT tests was 1% greater than that in the shake table study. In addition, the structural mass increased by 2% during the EFT tests, which was in part due to the addition of a thick plate for connecting the actuator. With the above structural properties, the natural frequency of the structure in the two studies changed by approximately 1% (from 2.89 Hz in the shake table study to 2.87 Hz in the EFT study). In addition, the structural damping during the EFT tests decreased by 10% (from 9.6% in the shake table study to 8.2% in the EFT study), which was attributed to an unknown change in the dampers and a change in test environment. With reduced damping, it was anticipated that the displacement and velocity of the structure in the EFT study would be slightly greater than those in the shake table study.

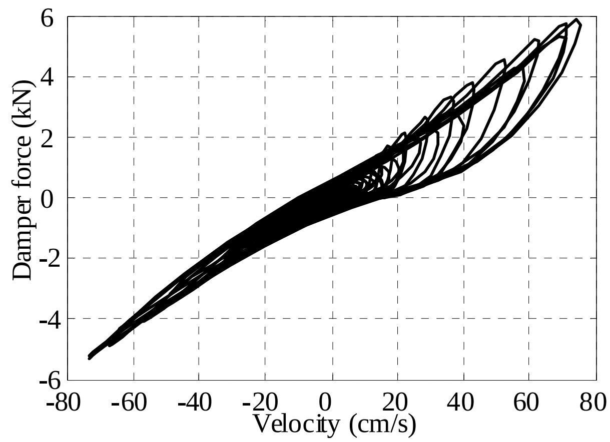

Figure 7 Force-velocity characteristic of the viscous dampers

The effective forces were applied to the structure by a 156-kN (35-kip) MTS 244.52 actuator controlled by a 341 lpm (90-gpm) MTS 256.09 servovalve, which was in turn controlled by an MTS 407 analog controller. The velocity feedback compensation schemes were implemented using a dSpace® DS1102 DSP controller with a TI TMS320C31 floating-point digital signal processor with a 2 kHz sampling rate. A tachometer type velocity transducer by Unimeasure® (V series linear velocity transducer) was used to measure the structure/piston velocity. In addition, strain gages and load cells were used to monitor the local structural responses such as column base shear and damper forces.

Test Results

With the identified parameters, a series of tests were conducted using the EFT method and results were compared to those of the shake table study. In the following force comparison, "Shake table test" represents the measured table acceleration times the estimated structural mass, which is also the effective force command for EFT tests, while "EFT test" represents the force applied to the structure measured by the actuator load cell.

Test results with a 0.55 g Northridge earthquake input are presented in Figures 8 and 9. Only 11 seconds of response (from 6 to 16 sec) are shown to make the graphs more readable. The force comparison in Figure 8 shows that the effective force command was followed by the actuator closely in the EFT test. The Fourier amplitude of the force applied to the structure by the actuator was slightly greater than the force command in the frequency domain, indicating a slight overcompensation of the natural velocity feedback due to uncertainties in the estimation of the servovalve flow properties. Both the global responses (displacement, velocity, and acceleration) and local response (column base shear) of the EFT test matched well with those of the shake table test as shown in Figure 9. The structural responses in the EFT test were slightly greater than those obtained in the shake table tests, which was attributed to the slight overcompensation of the natural velocity feedback and the decrease in structural damping. In addition, the after-shock free vibration, which began at 15 sec, was accurately captured.

After-test inspection indicated that the column ends partially yielded during the test. The maximum spool opening in the test was about 25%, which is beyond the linear range (10%) of the servovalve performance (refer to Figure 5). These observations indicate that with nonlinear velocity feedback compensation, the EFT method can be used to test nonlinear structures with large hydraulic demands.

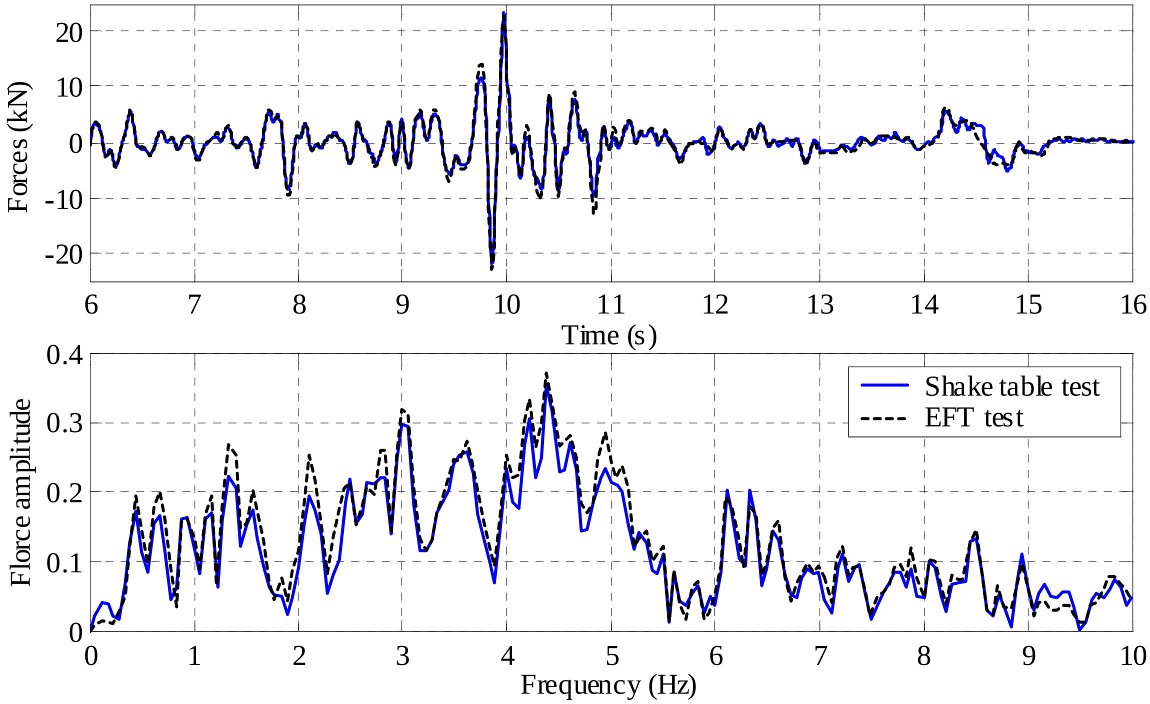

Figure 8 Comparison of effective forces and the force output of the test with 0.55g Northridge earthquake

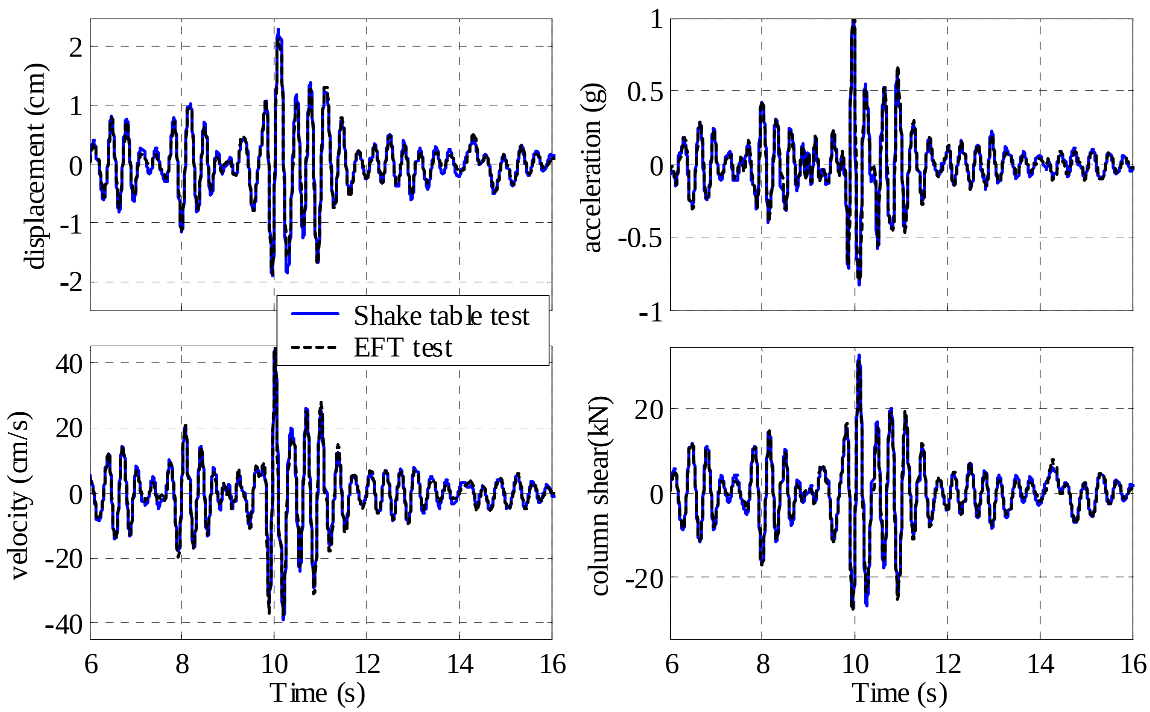

Figure 9 Comparison of structural responses of the test with 0.55g Northridge earthquake

CONCLUSIONS

Effective Force Testing (EFT) is a real-time earthquake simulation method for testing large-scale lumped mass structural systems. The effect of natural velocity feedback must be compensated in the laboratory implementation of the EFT method. The velocity feedback compensation requires an accurate characterization of the servovalve and its controller. The high-order servovalve dynamics can be accurately represented by a first-order delay with a valve gain for the frequency range of interest (0-10 Hz); hence, a first-order phase-lead network can be used in the velocity feedback compensation. The servovalve behaved nonlinearly when large flow demands were required during testing associated with large structural velocity responses (represented by the nonlinear no-load flow gain) and/or large effective forces (represented by the load pressure influence). To fully utilize the servovalve capacity, the nonlinearities were considered in the velocity feedback compensation with an experimentally determined servovalve flow curve and measured spool opening and load pressure.

Detailed procedures were discussed in this paper to determine the parameters required in the system modeling and the laboratory implementation of the EFT method. The feasibility of EFT with the nonlinear velocity feedback compensation was demonstrated by testing a one-story steel structure using a shake table and the EFT method. The comparison of the test results with the two test techniques showed that with proper velocity feedback compensation, the EFT method can be used to apply real-time seismic simulation to a structure that has complex damping properties and hysteretic behavior. As the EFT method becomes available to researchers, the testing capability of existing laboratory equipment will expand from static testing to real-time dynamic testing.

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