# IMPLEMENTATION OF EFFECTIVE FORCE TESTING: METHOD OF SEISMIC SIMULATION FOR STRUCTURAL TESTING

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## SUMMARY

This paper describes experimental implementation of a real-time earthquake simulation test method for large-scale structures. The method, Effective Force Testing (EFT), is based on a force control algorithm. For systems which can be modeled as a series of lumped masses (e.g. frame structures where masses are assumed lumped at the floor levels) the EFT forces are known a priori for any acceleration record. As opposed to the pseudodynamic test method (a displacement-based control procedure), there is no computational time required for the EFT method in determining the required loading to be applied to the structure.

Research has been conducted on a single-degree-of-freedom system at the University of Minnesota to investigate the potential of the EFT method. A direct application of the EFT method was found ineffective because the actuator was unable to apply force at the natural frequency of the structure due to actuator/control/structure interaction. A detailed model of the control, hydraulic and structural systems was developed to study the interaction problem and other nonlinear responses in the system. The implementation of an additional feedback loop using the measured velocity of the test structure was shown to be successful at overcoming the problems associated with actuator/control/structure interaction, indicating EFT is a viable real-time method for seismic simulation studies.

## INTRODUCTION

Effective Force Testing is a real-time method of seismic simulation under development at the University of Minnesota that complements existing types of earthquake testing. In the Effective Force Testing (EFT) method, the dynamic structural response that would be produced by base excitation due to earthquake ground motion ($\ddot{x}_g$) is produced by applying lateral effective forces ($p_{eff_i}(t) = -m_i \ddot{x}_g$) to each degree of freedom (DOF) of the test structure attached to a stationary base. The equation of motion of a single-degree-of-freedom (SDOF) system subjected to an earthquake excitation can be used to illustrate the concept,

$$m\ddot{x} + c\dot{x} + kx = -m\ddot{x}_g = p_{eff}(t)\ , \tag{1}$$

where $m$, $c$, and $k$ are the mass, damping coefficient and stiffness of the SDOF, respectively. The product of the mass and the ground acceleration ($-m\ddot{x}_g$) is equivalent to a lateral effective force ($p_{eff}(t)$) applied directly to the mass [Clough et al., 1975]. Because the mass of the structure and the earthquake ground acceleration are known before a test, the effective forces are known a priori. The effective forces are also independent of the structural properties (other than mass), and thus do not change even as the structure is driven into the inelastic range of behavior. For a lumped mass multi-degree-of-freedom (MDOF) system, the effective forces at each level would just be the product of the mass at the respective level multiplied by the ground acceleration.

Implementation of the EFT concept was explored by Murcek [1996] at the University of Minnesota on an SDOF

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test structure.  The research showed that the actuator was unable to apply the effective force near the natural frequency of the system. The results of Murcek's research [1996] verified the conclusions drawn by Dyke et al. [1995] regarding effects of control-structure interaction; that is, for lightly damped structures, the actuator has a greatly limited ability to apply forces near the natural frequency of the structure.  This inability to apply the correct force at the natural frequency of the structure is due to “natural” velocity feedback of the actuator. Dimig et al. [1999] proposed a solution to the natural velocity feedback problem that comprised incorporation of an additional feedback loop to negate the natural velocity feedback of the actuator.

The current paper describes improvements to the control/actuator/structural model and an experimental implementation of the natural velocity feedback correction suggested by Dimig.

## DYNAMIC SYSTEM MODEL FOR SDOF WITH A THREE-STAGE SERVOVALVE

Equations describing the system dynamics for an SDOF structure excited by a hydraulic actuator have been proposed by Merrit [1967].  These equations predict that the actuator will be unable to apply force to the SDOF at the natural frequency because of the interaction between the velocity of the actuator piston and the velocity of the structure.  If the increased flow needed by the actuator is accounted for, then it should be possible for the actuator to apply force to the system at its natural frequency.  Figure 1 shows the dynamic system model for an SDOF structure excited by a hydraulic actuator with a three stage servovalve in force control with the proposed correction for the natural velocity feedback.  A detailed derivation of the precursor to this model can be found in Dimig et al. [1999].

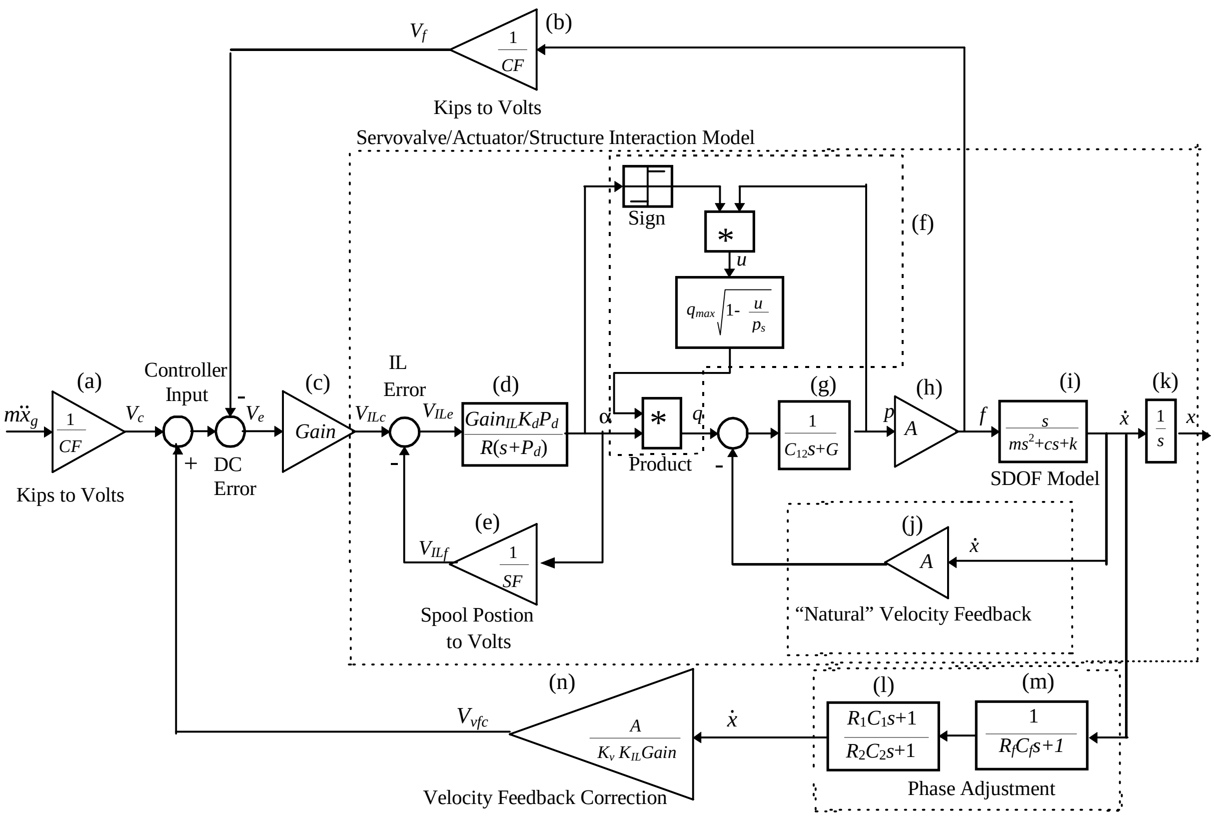



**Figure 1  Model of the dynamic system incorporating the nonlinear relationship describing servovalve flow with the linearized model for velocity feedback.**

The blocks in Fig. 1 are labeled with letters a - n for the purpose of describing the model. The earthquake effective force $(-m\ddot{x}_g)$ is the model input.  Blocks (a) and (b) represent the conversions of the effective force and actuator load cell feedback signals from kips to volts, respectively. The gain factor in block (c) represents the proportional gain, which was set within the controller.

Block (d) represents the three-stage valve driver, showing the conversion from voltage to the servovalve main stage spool opening, α. Included in this transfer function is a model for a single pole valve driver, with parameter $P_d$ which approximates the valve dynamics by representing the time delay associated with movement of the spool [Bailey, 1987]. The mechanical gain, $K_d$, represents the dc gain of the valve driver. The oil flow through the servovalve is a nonlinear function of the servovalve spool opening and the pressure across the piston described by the following flow-pressure relationship:

$$\frac{q}{q_{\max}} = \alpha\sqrt{1 - \frac{\alpha}{|\alpha|}\frac{p_l}{p_s}}\ , \tag{2}$$

where $q$ is the flow through the actuator, $q_{max}$ is the maximum flow with full effective supply pressure drop across the servovalve, $p_l$ is the difference in pressure across the actuator piston, and $p_s$ is the hydraulic supply pressure. Block (f) is a representation of this relationship.

Blocks (g) and (h) represent the actuator, where $C_{12}$ is the oil compressibility constant, $G$ is the sum of the actuator leakage constant and the valve leakage constant, and $A$ is the cross-sectional area of the actuator piston [Bailey, 1987 and Merrit, 1967]. The transfer function for the linear elastic test structure is shown in block (i). The velocity response of the SDOF system is integrated in block (k) to obtain the displacement response. The "natural" velocity feedback is represented by block (j).

The natural velocity feedback correction is shown in blocks (l-n). The measured velocity multiplied by the piston area is the flow which needs to be added to the actuator. Because the command signal is modified rather than the flow, all of the model parameters which convert the command voltage, $V_c$, to flow, $q$, must be used to convert the "correction flow" to the correction voltage, $V_{vfc}$. Hence the measured velocity multiplied by the piston area is multiplied by an inverse of all the model parameters in blocks (c-f) to produce the proper correction voltage. This voltage is then summed with the effective force voltage signal to produce the "corrected" command signal which is input to the controller. For experimental simplification, a **linearized equivalent** of Eqn. (2) is used in the feedback correction, where the flow in the servovalve is assumed to be proportional to the spool opening, independent of the pressure across the actuator piston:

$$q = K_v \alpha\ , \tag{3}$$

where $K_v$ is the null flow gain.

In the velocity feedback correction block (n), the factor $K_{IL}$ is introduced. This factor represents the overall gain applied by the inner loop parameters in block (d) which converts the inner loop command signal, $V_{ILc}$, to the valve spool opening, α,

$$K_{IL} = \frac{\left(Gain_{IL}\right)\left(\frac{K_d}{R}\right)}{1 + \left(Gain_{IL}\right)\left(\frac{K_d}{R}\right)\left(\frac{1}{SF}\right)}\ . \tag{4}$$

The phase adjustment incorporated into the velocity feedback correction, shown in blocks (l-m), was needed to offset time delays in the servovalve dynamics [block (d)]. The lead/lag compensation network in block (l) was the primary means of adjusting the phase of the velocity feedback correction. The low-pass filter [block (m)] was used only to provide a simple means of making small, "fine tune" adjustments to the phase of the velocity signal. The gain factor and phase adjustment factors were independent of the properties of the SDOF structure. The servovalve dynamics in block (d) and the phase adjustment in blocks (l-m) described further in Timm (1999) represent refinements over the model originally developed by Dimig (1999). Computer simulations of the SDOF system response using this model were conducted using SIMULINK® dynamic system simulation software available within MATLAB® version 5.1.

## DESCRIPTION OF TEST STRUCTURE AND EXPERIMENTAL SETUP

The model described above was experimentally implemented using a structure that was representative of a classic mass-spring-dashpot SDOF model shown in Fig. 2. The system consisted of a 17.5 kip cart, which served as the mass, and a 23 ft. long, 1 in. nominal diameter Grade 150 ksi Dywidag threadbar, which acted as a spring when pretensioned. A pretension of 15 kips was chosen to avoid potential buckling problems. The size and length of the bar were chosen to enable a cart displacement that would provide adequate resolution with the measuring devices while the bar remained linear elastic with an applied force within the capacity of the actuator. The measured system stiffness and damping ratio were 67.1 kip/in and 0.0191, respectively. The natural frequency of the system was 6.13 Hz. A linear variable differential transformer (LVDT) and velocity transducer were used to monitor the displacement and velocity response of the SDOF test structure, respectively.

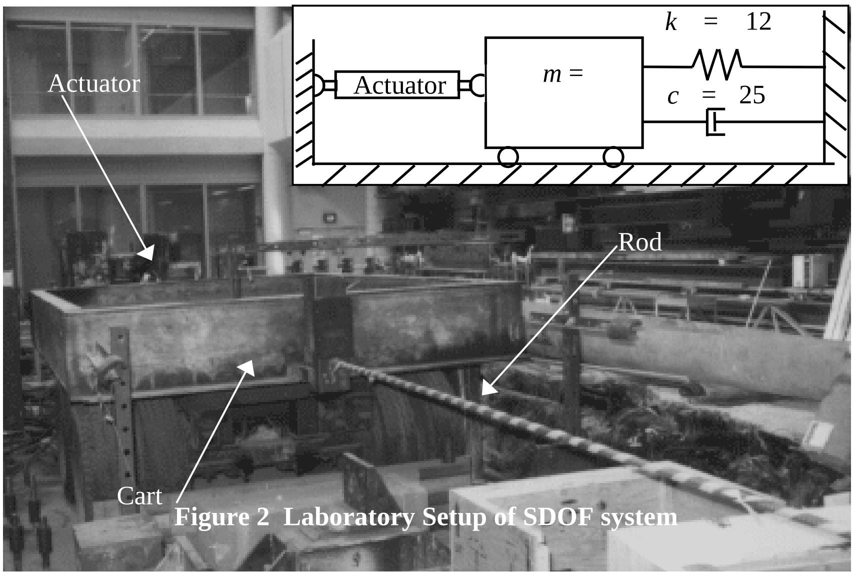



**Figure 2 Laboratory Setup of SDOF system**

Force was applied to the SDOF structure using an MTS Model 244.40 servohydraulic actuator aligned with the center of mass of the test structure. The actuator was controlled with an MTS 407 controller using proportional gain only. Hydraulic pressure was supplied to the actuator through a hydraulic service manifold connected to a pump with a flow capacity of 150 gpm at a supply pressure of 3000 psi. The actuator was equipped with an MTS 256.09 three-stage servovalve that had a flow rating of 90 gpm for a 1000 psi pressure drop across the valve. Force-velocity analyses of the hydraulic system indicated that the power capacity of the system was adequate for the proposed tests (Timm, 1999).

The velocity feedback correction was implemented by applying the appropriate correction factor to the measured velocity signal and summing this signal with the effective force input command signal using an op-amp summing amplifier. The magnitude of the feedback correction gain factor was slightly underestimated because an overestimate of the gain factor produced potentially unstable system response, and the response of the system was sensitive to small changes in the actuator supply pressure caused by other tests in the laboratory using the hydraulics. A separate op-amp circuit was built to provide phase adjustment of the velocity feedback correction signal required to compensate for time delays in the servovalve dynamics. Circuit components for the lead compensation network were chosen to optimize the applied force. Values for the parameters shown in Fig. 1 for the experimental system are given in Table 1. More detail on the evaluation of these parameters can be found in Timm (1999).

**Table 1 System Parameters**

| Parameter | block | Value |
|---|---|---|
| $CF$ | a, b | 8.0 kips/volt |
| $Gain$ | c, n | 1.05 volts/volt |
| $SF$ | e | 10volts/1000% |
| $Gain_{IL}$ | d | 2.0 volts/volt |
| $R$ | d | 200 ohms |
| $K_d$ | d | 2000%/Amp |
| $K_v$ | n | 548in.$^3$/sec |
| $C_{12}$ | g | 0.81 in.$^5$/kip |
| $G$ | g | 0 |
| $A$ | h, j, n | 27 in.$^2$ |
| $m$ | i | 17.5 kips |
| $c$ | i | 2% |
| $k$ | i | 67.1 kip/in |
| $P_d$ | d | 352 sec$^{-1}$ |
| $R_f$ | m | 12 kΩ |
| $C_f$ | m | 0.047 μF |
| $R_1$=$R_4$ | l | 39 kΩ |
| $R_2$=$R_3$ | l | 10 kΩ |
| $C_1$ | l | 0.1 μF |
| $C_2$ | l | 0.047 μF |

The earthquake records used for the experimental program were the 1940 El Centro, N-S, and the 1968 Hachinohe, N-S, ground acceleration records as well as sine wave sweeps over 0-10 Hz.

## SYSTEM RESPONSE WITHOUT VELOCITY FEEDBACK CORRECTION

The EFT concept was explored by Murcek [1996] at the University of Minnesota using experiments and computer simulations performed on the SDOF test structure. The research showed that the actuator was unable to apply components of the effective force which were near the natural frequency of the SDOF system. An example of the system response without velocity feedback correction is given in Fig. 3, which shows a comparison of the fast Fourier transforms (FFTs) of the command signal, measured applied force, and simulation, that were obtained for a 3 kip sine wave sweep input with frequencies ranging from 0 to 10 Hz. The point at which the applied force FFT is at a minimum is the natural frequency of the system (6.13 Hz). The simulation model correctly predicted the response of the system with the exception of portions of the signal around 4, 9, and 18 Hz. The sharp drop in the applied force FFT at 4 Hz was indicative of an additional mode of vibration of the cart believed to be associated with a bouncing or rocking motion. The additional discontinuity in the applied force at 9 and 18 Hz was attributed to transverse vibrational modes of the bar.

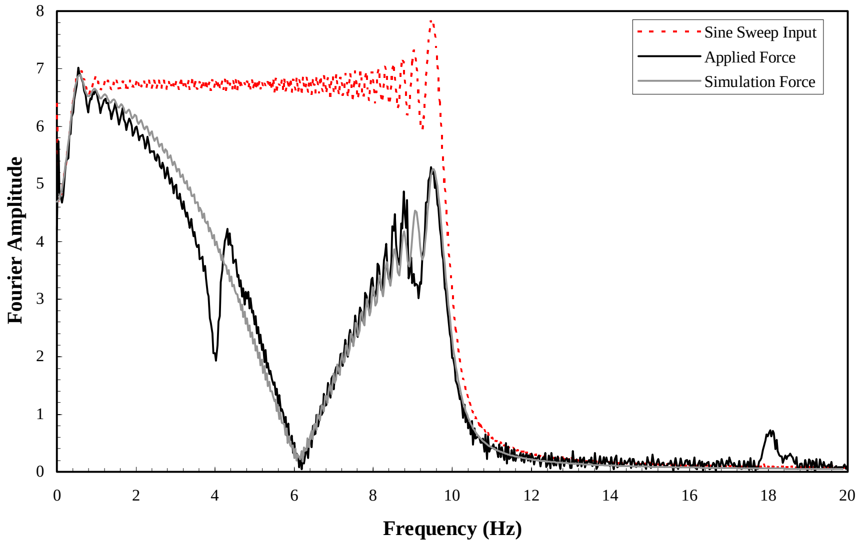



**Figure 3. Fast Fourier Transforms of the input signal, experimental applied force, and simulation force for a 3 kip sine wave sweep without velocity feedback correction.**

## EXPERIMENTAL IMPLEMENTATION OF THE VELOCITY FEEDBACK CORRECTION

### Sine sweep tests

Tests implementing the velocity feedback correction were conducted with a 0.5 kip sine sweep input function. FFT's of the sine sweep input, applied force, and simulation force for this test are shown in Fig. 4. The FFT of the applied and simulation forces in Fig. 4 show a large spike around 12 Hz (approximately twice the natural frequency) and a sharp drop at the natural frequency. The discontinuity at 4 Hz, due to the additional vibration mode of the system, was not predicted by the simulation model because it was not included in the transfer function assumed for the SDOF.

The spike around 12 Hz (twice the natural frequency) was an indication of nonlinearity in the flow-velocity relationship that was not accounted for in the simplified linearized velocity feedback correction. Simulation studies considering the nonlinear pressure-flow relationship in the velocity feedback correction loop (block n of Fig. 1) did not have a spike at 12 Hz. The 15 kip pretension in the bar (to avoid bar buckling) was thought to be the cause of the inadequacy of the linearized pressure-flow relation in the velocity feedback correction. Simulation studies of the system without a pretension force in the spring also did not have a spike in the applied force FFT near 12 Hz.

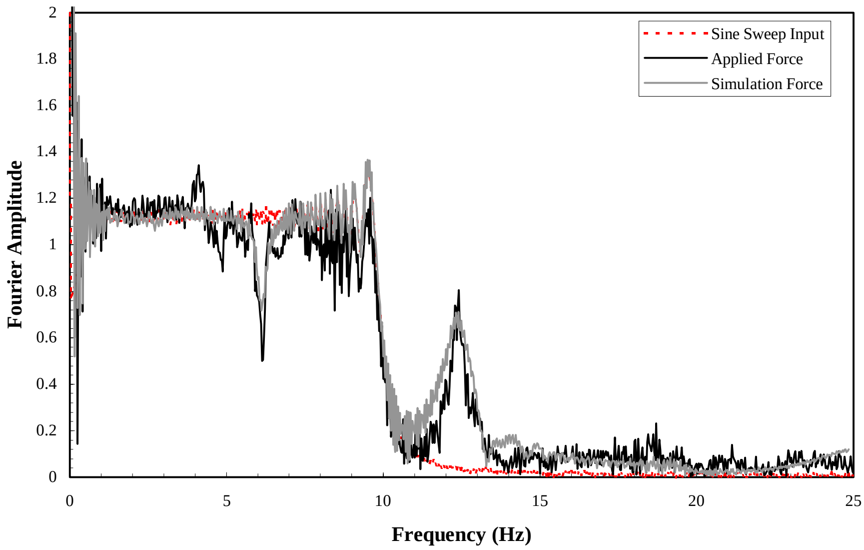


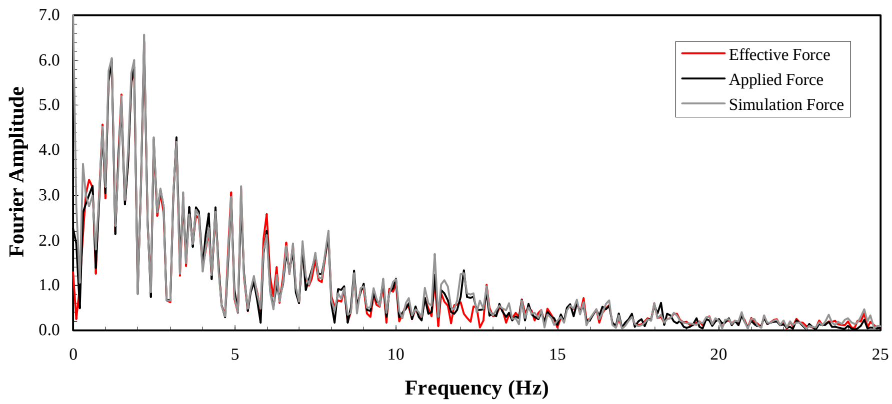

**Figure 4. Fast Fourier Transforms of the input signal, experimental applied force, and simulation force with the nonlinear servovalve flow model for a 0.5 kip sine sweep with velocity feedback correction.**

Simulation tests with the sine sweep input using the model in Fig. 1 indicated the drop in the applied force FFT at the natural frequency was due to the magnitude of the gain factor in the velocity feedback correction being slightly lower than the proper magnitude. In simulation tests, a reduction of the velocity feedback correction gain factor by approximately 3% from the correct value produced results which corresponded to the drop in the applied force FFT in Fig. 4. Experimental tests in which the correction factor was increased were not successful in eliminating the drop in the applied force FFT at the natural frequency. Only a small reduction in the drop was obtained while the applied force FFT at frequencies slightly less than the natural frequency increased. The reason for the inability to eliminate the drop in the applied force FFT may have been due in part to the nonlinearity in the hydraulic system or may have been the possible result of a small discrepancy in the phase adjustment from the precise adjustment which would provide the correct phase shift for the time delay in the system.

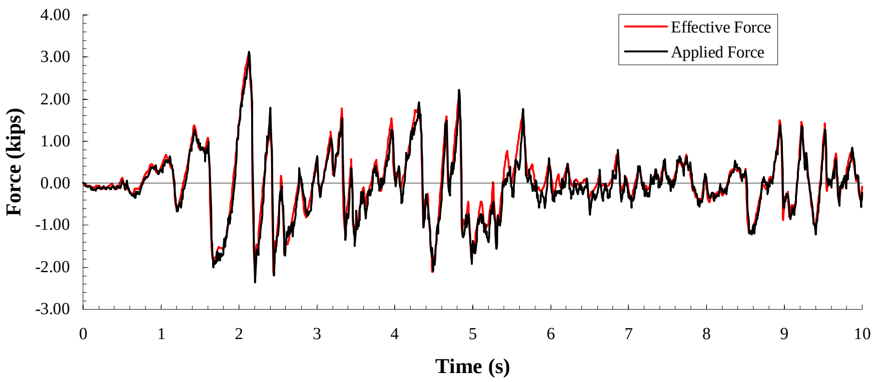
### Earthquake simulation tests

The primary earthquake ground acceleration used in the experimental investigations was the first 10 seconds of the El Centro ground acceleration (Elcn10) at half scale with a peak ground acceleration of 0.17g. This earthquake segment contained a demanding portion of the El Centro record that had frequency content similar to the entire record while reducing the amount of data to be collected.

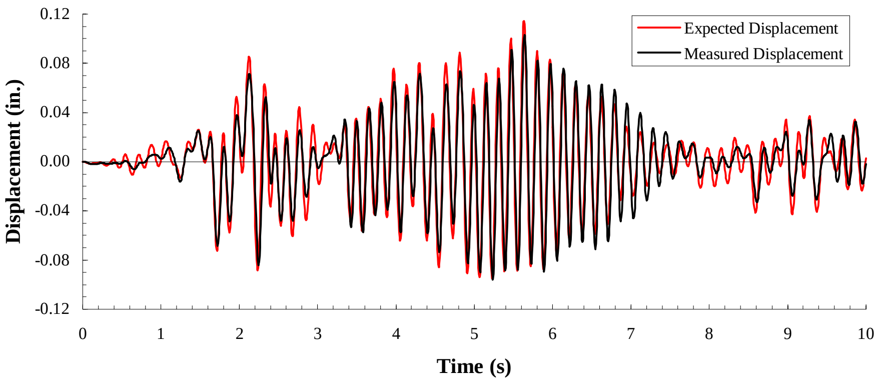
The system response to the Elcn10 (0.17g) effective force input function is shown in Fig. 5. In general, the FFTs show reasonable results over the entire frequency range. A slight reduction of the FFT of the applied force relative to the effective force input can be seen at the natural frequency of the system (6.13 Hz). The reduction in force was not as noticeable as in the case of the sine sweep input, which may be due to lower demands on the hydraulic system for the Elcn10 input signal.




**a) FFT**




**b) Force**




**c) Displacement**

**Figure 5. Comparison of the expected vs. measured response for Elcn10 (0.17g) earthquake segment with the velocity feedback correction.**

The force history, plotted in Fig. 5b, shows that the applied force response for Elcn10 (0.17g) closely matched the effective force input; however, portions of the applied force were shifted down in relation to the effective force. The corresponding measured displacement response, plotted in Fig. 5c, shows a measured response that generally followed and was in phase with the expected response as determined by a piecewise linear integration of the equation of motion for the SDOF. The measured response tended to be less than the expected response except for a portion of the Elcn10 response between 6 and 7.5 seconds where the measured response was slightly greater than the expected response.

The SDOF system tended to have difficulty achieving the expected response in the low amplitude portions of the test shown in Fig. 5b between 0 and 1 seconds and around 8 seconds. The reason for the increased difficulty in achieving the expected response at low amplitudes was attributed to the amplitude of the effective force input function and the resolution of the electronics. The resonant response of the system at its natural frequency is driven by the frequency content of the effective force input function at the natural frequency. For sinusoidal input at the natural frequency of the SDOF system, an input function of approximately 0.055 kips is all that is theoretically required to produce a 0.02 in. displacement response. The 0.055 kip input function which would produce this amplitude of displacement represents a voltage of less than 7 mV (0.07% of full scale). At this small voltage, good resolution of the input signal was difficult to achieve. An improved response at lower amplitudes would be expected with a lower force capacity actuator or recalibration of the actuator load cell to a smaller range.

## CONCLUSION

Effective Force Testing (EFT) is a method of earthquake simulation for testing large-scale lumped mass structural systems. In EFT, earthquake effective forces are applied to the test structure through the center of mass at each degree-of-freedom. EFT uses the same laboratory test setup as pseudodynamic testing; however, EFT is conducted in real-time with an effective forcing function which is known a priori and an actuator control signal which is modified using the measured real-time velocity response of the test structure. Because testing is conducted in real-time, the structure develops real inertial and damping forces as in shake table testing.

In previous research conducted at the University of Minnesota by Murcek [1996], a direct application of EFT was studied to investigate how accurately the actuator was able to follow the command signal in applying real-time effective forces to a SDOF model. In that research, it was demonstrated that it was not possible to achieve the correct effective force near the natural frequency of the test structure. Interaction of the actuator and test structure through "natural" velocity feedback was determined to be the reason that force was not applied near the natural frequency of the structure. Through simulations of the system response to effective force input, a velocity feedback "correction" loop was proposed by Murcek that negated the "natural" velocity feedback interaction, allowing the effective force to be accurately applied near the natural frequency of the structure.

Experimental tests using sine sweep and earthquake record inputs with velocity feedback correction demonstrated the viability of EFT and that real-time velocity feedback correction could negate the effects of natural velocity feedback. FFT's of the forces for these tests showed that the actuator was able to apply forces that were approximately equal to the desired effective forces near the natural frequency of the system . Because orce was applied to the system at the natural frequency, the resonant response of the SDOF system was excited which was approximately equal to the expected response. The response of the SDOF system with the velocity feedback correction demonstrated that a real-time dynamic test could be performed using the Effective Force Testing method and implementation of this method is independent on the properties of the test structure.

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