DEVELOPMENT OF EFT FOR NONLINEAR SDOF SYSTEMS

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ABSTRACT

This paper presents the development and implementation of the Effective Force Testing (EFT) method for nonlinear elastic single-degree-of-freedom (SDOF) systems. Linearized velocity feedback compensation was found inadequate for negating the effect of natural velocity feedback when the test structure behaves nonlinearly and its velocity/displacement responses are significantly large. Detailed simulation models were developed to study the influence of system nonlinearities. A nonlinear compensation scheme was derived and tested. Experimental results indicated that the EFT method with proper velocity feedback compensation is viable for real-time seismic simulations.

Introduction

Real-time dynamic testing is necessary for the study of seismic response of structures, which exhibit velocity dependent behavior (e.g. structures incorporating active or passive control devices). Effective force testing (EFT) is an experimental technique, which can be used to apply real-time earthquake simulations to large-scale structures. EFT is applicable to lumped-mass systems anchored to a fixed base, of which the dynamic response can be obtained by applying "effective forces" through the center of mass of the structure. The effective forces are equal to the mass multiplied by the earthquake ground acceleration, thus known a priori.

The concept of EFT was described in papers discussing the pseudodynamic test method in 1980's. Among them is the one by Mahin et al. in 1989, in which they noted that with a pseudodynamic test setup, it is possible to perform real-time tests based on a force control method. While this method eliminates the need of solving equations for required displacements during tests, the implementation difficulties should be further studied.

The experimental investigation of EFT method has been underway at the University of Minnesota (Murcek 1996 and Timm 1999). Hydraulic actuation was used to apply the effective forces. It was found that a feedback path intrinsic to the servo-hydraulic system, which is called "natural velocity feedback," combined with the required force stabilizing loop, limits the ability of actuators to apply forces accurately near the natural frequency of the test structure. A solution

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to the problem was further proposed: to create an additional feedback loop to negate the effect of the natural velocity feedback. Timm (1999) implemented the velocity feedback compensation using a measured velocity, and found that the expected resonance at the natural frequency was still not able to be excited due to an inevitable time delay in the response of the servo system. Having noted that this delay does not influence the test with precorrected command signals, he proposed and implemented a phase-lead network to the velocity feedback compensation loop to compensate the time delay.

EFT with an additional feedforward compensation loop has been successful in negating the effect of natural velocity feedback in past studies conducted on a linear elastic SDOF system, for which the structural responses were relatively small. The performance of the compensated system relies on accurate knowledge of the servo-hydraulic system, which shows significant nonlinearites when large fluid/pressure is required during the tests. To investigate such situations, a nonlinear elastic SDOF system has been built and tested using EFT. This paper presents the study and a nonlinear compensation scheme.

Description of Test Structure and Experimental Setup

The nonlinear elastic test structure consists of a 15.8 kip concrete mass sitting atop four caster wheels with two springs on either side as shown in Fig. 1. The springs, which have a stiffness of 1 kip/in., are compressed by 1 inch before tests so that the initial system stiffness is 4 kip/in. The stiffness is reduced to 2 kip/in. when springs on one side of the mass lose contact with the mass due to a large mass movement (>1 inch). A suspension strut for cars serves as a damper. The measured system damping is 1.5% critical damping plus 7.5 lb Coulomb friction force. The initial natural frequency of the system is 1.56 Hz.

Effective forces were applied to the structure using a 35 kip servohydraulic actuator (model 244.23) with a three-stage 90 gpm servovalve (model 256.09), both made by MTS Systems Corp. The displacement response of the SDOF system was measured using the actuator LVDT. A velocity transducer was used to measure the velocity response, and fed back the signal to a controller implementing the velocity feedback compensation. The controller was implemented using a dSpace DS1102 DSP controller with a TI TMS320C31 floating-point digital signal processor with a 2k Hz sampling rate.

Dynamic System Model and Linearized Feedforward Compensation

In order to better understand the natural velocity feedback problem and derive control algorithms, system models developed by Zhao et al. were adopted. Fig. 2 shows the major components of the test system and the linearized feedforward compensation scheme. A PC sends the effective force command signal to a servovalve controller. The controller compares the command signal to a feedback signal and sends a current proportional to the difference between these two signals to a servovalve to drive the valve spool. The spool controls the hydraulic fluid flow into/out of the chambers of the actuator to control the fluid pressure on both sides of the actuator piston. The pressure difference between these two multiplied by the actuator piston area produces the force applied to the test structure. The force measured by a load cell on one side of the actuator is finally fed back to the controller to close the control loop.

In a force-controlled test, the movement of the actuator piston affects the performance of the servovalve controller so that required forces are difficult to apply. When the command forces are in phase with the piston movement, the amplitude of the applied forces can be greatly reduced. The physical phenomenon is called "natural velocity feedback" and is identified in Fig. 2. The velocity feedback compensation loop tries to solve the problem by altering the command signal to negate the effect of the natural velocity feedback.

In the linear compensation scheme, the servovalve is assumed to perform near its null position, thus behaves linearly. The current piston/structure velocity is predicted based on velocity signals measured in the past. The velocity signal is multiplied by a factor and added to the effective force command signal. The compensation factor is assumed constant and determined by

$$K_f = \frac{A}{K_v K_s G_p}, \tag{1}$$

where A is the actuator piston area, K_v is the nominal flow gain of the servovalve, K_s is the inner loop feedback gain of the servovalve, and G_p is the proportional gain of the servovalve controller. Both factors describing the servovalve properties are assumed constant. The phase adjustment of the velocity feedback compensation signal required to compensate for time delays in the system dynamics is provided by a phase-lead network, for which the transfer function in the frequency domain is

$$G = \frac{output}{input} = \frac{T_{ld}s + 1}{aT_{ld}s + 1} \tag{2}$$

where *s* is the complex variable and T_{ld} and *a* are two constants used to determine the lead time. A discrete equivalent of the phase-lead network by numerical integration is used in the controller implementation. By applying the trapezoid-rule substitution for the frequency variable *s* (Franklin et al. 1990), Eq. 2 can be rewritten in time domain as

$$x(t) = \frac{1}{T + 2aT_{ld}}\left((T + 2T_{ld})u(t) + (T - 2T_{ld})u(t - T) - (T - 2aT_{ld})x(t - T)\right), \tag{3}$$

where *u* is input signal, *x* is output signal, *t* is current time, and *T* is sampling period.

Nonlinear Feedforward Compensation Design

The linearized controller was successfully implemented by Dimig et al. and Shield et al. Tests with linear velocity compensation herein repeated by the authors also indicated good system performance under limited conditions. However, experiments with large amplitude excitations, which can bring the structure to its nonlinear range of performance, showed that the compensation loop based on a linearized servo system model could not compensate for the natural velocity feedback completely in these cases. Fig. 3 shows the performance difference of the linearized controller in a test with 0.5 kip sine sweep excitation and another with 2 kip sine sweep excitation. Computer simulation with the linearized servo system model and linearized feedforward compensation loop cannot explain the sharp drop around the natural frequency of the system when large amplitude excitation is applied to the structure. With large structural velocity responses, the servovalve spool moves away from its null position, and system nonlinearities become significant. This indicates that both the system model and hence the controller design need to include some nonlinearities.

System nonlinearities

Two types of system nonlinearities have been identified; both are related to the flow property of the servovalve. The first one describes the state of flow going through an orifice formed by the movement of the servovalve spool (Merritt 1967),

$$Q = K_v x_v \sqrt{1 - \frac{x_v}{|x_v|}\frac{p}{p_s}}, \tag{4}$$

where Q is the load flow, x_v is the spool opening, p_s is the hydraulic pressure supply, p is the load pressure or the pressure drop across the piston, and K_v is flow gain, which is the slope of the flow-spool opening curve. This relationship is called the load pressure influence. Computer simulations with this system nonlinearity and linearized velocity compensation are able to explain the shallow dip in Fig. 3 (a).

The second nonlinearity is related to the flow gain K_v, which is taken as the initial slope of the curve in the linearized system model and controller design and assumed constant through the operating range of the servovalve. Fig. 4 shows a typical flow-spool opening relation for a servovalve under 1 ksi constant pressure drop across the piston. As can be seen, the flow gain decreases as the spool opening increases though it is fairly constant up to 12% of total spool opening. Further simulations with both nonlinearities included in the system model while keeping the linearized compensation matched experimental results well as shown in Fig. 3 (b). This indicates that more advanced compensation schemes are necessary when a large amount of flow is required, and an accurate inverse of the nonlinearities in the forward path should be included in the velocity compensation loop.

Nonlinear Controller Design

Computer simulations were conducted first to test the nonlinear control methodology. The flow-spool opening relation shown in Fig. 4 was used to represent the nonlinear property of the servovalve. An accurate inverse of that relation obtained through the method described below was used in the velocity compensation loop. Fig. 5 shows the comparison of its performance along with that of a linearized compensation scheme. In general both controllers behave well over the entire range of the excitation, while with the nonlinear velocity compensation, the actuator is able to reach the force peak around 2 seconds. The corresponding displacement response is also improved

The flow-spool opening curve varies from one servovalve to another. The relation for the servovalve used in this study was determined using existing experimental data. Spool opening can be measured directly while the flow can be calculated based on the conservation of mass:

$$Q = K_a \dot{p} + C_l p + A\dot{x}, \tag{5}$$

where $K_a = V_t/4\beta_e$ is the hydraulic fluid compressibility coefficient, V_t is the total volume of the actuator chambers, β_e is the effective bulk modulus of the hydraulic fluid, C_l is the leakage coefficient assumed constant, $\dot{p}$ is the time derivative of the pressure drop across the piston, which is determined by dividing the derivative of the applied load by the piston area, and $\dot{x}$ is the piston/structure velocity.

Fig. 6 shows some experimental data for the servovalve. A polynomial curve fit was made to best represent the existing data with one equation, then axes of the above curve were exchanged and another curve fit was made to obtain the inverse of the relation. Although the resulting curves best fit the existing data in statistical sense, they may not be able to represent the behavior of the servovalve around the null position. In addition, the required inverse relationship may not be guaranteed with two curve fits.

A piecewise linear curve with 21 control points and 5% spool opening interval was used to describe the flow-spool opening relation over the positive spool opening range, and the servovalve was assumed to behave symmetrically. The flow value of each control points is the average of the flow corresponding to several data points around the control points. A lookup table was coded into the nonlinear compensation scheme to determine the spool opening needed to generate a given amount of flow. Because the existing data (up to 30% spool opening) cannot cover the full operating range of the servovalve, the inverse of the typical curve shown in Fig. 4 was used beyond the 30% of the spool opening. A short transition is created by the controller to make the curve smooth. It was anticipated that more experimental data with larger spool opening would be possible during tests based on this curve, and new data would in turn calibrate the current curve. The load pressure influence was not included in the current compensation scheme because it was assumed that this part was relatively insignificant compared with the effect of nonlinear flow gain.

Experimental Implementation Incorporating the Nonlinear Compensation

Tests implementing the nonlinear velocity compensation were conducted with the first 10 seconds of the 1940 El Centro, N-S record at half scale with a peak ground acceleration of 0.17g. This earthquake segment contains a demanding portion of the El Centro record. Its frequency content is similar to that of the entire record while reducing the amount of data to be collected. The natural frequency of the test structure is in the demanding frequency range of the earthquake record, thus large displacement and velocity responses are expected.

The system response to the effective force input with implementation of the nonlinear velocity compensation is shown in Fig. 7 (a). For comparison purpose, Fig. 7 (b) illustrates the system response with the linear compensation and without any compensation. With implementation of either the linear or proposed nonlinear velocity compensation, the FFTs show reasonable results over the entire frequency range. The reduction of the Fourier amplitude of the applied force at the natural frequency with the nonlinear compensation is less than that with the linear controller. Thus the corresponding measured displacement in general followed closely the expected response while the structural response with linear controller showed significant reduction. The expected response was determined using Newmark's method with Newton-Raphson iteration. On the other hand, the applied force still cannot reach the peak at 2 seconds, thus the displacement around 3 seconds showed some reduction. The incomplete compensation was believed due to other nonlinearities not accounted for in the proposed compensation scheme.

The force time histories shown in Fig.7 (a) appear to contain components of high frequencies. This was in part attributed to noise signal picked by the actuator load cell. The overall system dynamics also contributed to the noisy force measurement because the system damping is small, thus the actuator tended to vibrate before settling at a target force. In addition, the compensated system could have a narrow stability margin if the damping is small. System uncertainties, such as the fluctuation in the hydraulic pressure supply, may influence the actual flow properties of the servovalve during a test, thus causing instantly incomplete compensation or over-compensation with the current velocity compensation scheme. System variability is shown in Fig.6 where experimental data points are scattered while the controller design is based on an idealized curve.

To further improve the system performance, the load pressure influence should be included in controller design by taking additional pressure input signals. In addition, an adaptive compensation scheme seems necessary to account for the above uncertainties in the system, especially when structural damping is small. A standard calibration procedure should be developed to determine the flow property of a given servovalve.

Conclusions

Effective Force Testing (EFT) is a real-time earthquake simulation method for testing large-scale lumped mass structural system. EFT uses existing test equipment in a typical structural laboratory; however, it requires large flow capacity servovalves. A nonlinear velocity feedback compensation scheme is necessary when large velocity response is expected which causes large flow demands to the servovalve. The system performance relies on an accurate knowledge and model of the servo-hydraulic system. The stability margin of the overall system can be narrow if the system damping is small.

Experimental studies on a nonlinear elastic SDOF test structure demonstrated that real-time dynamic tests could be performed using EFT though further efforts are needed to refine the nonlinear controller. The implementation of EFT in this study validated that the test method is independent of the properties of the test structure. As the EFT method becomes available to researchers, the testing capability of existing laboratory equipment will be expanded.

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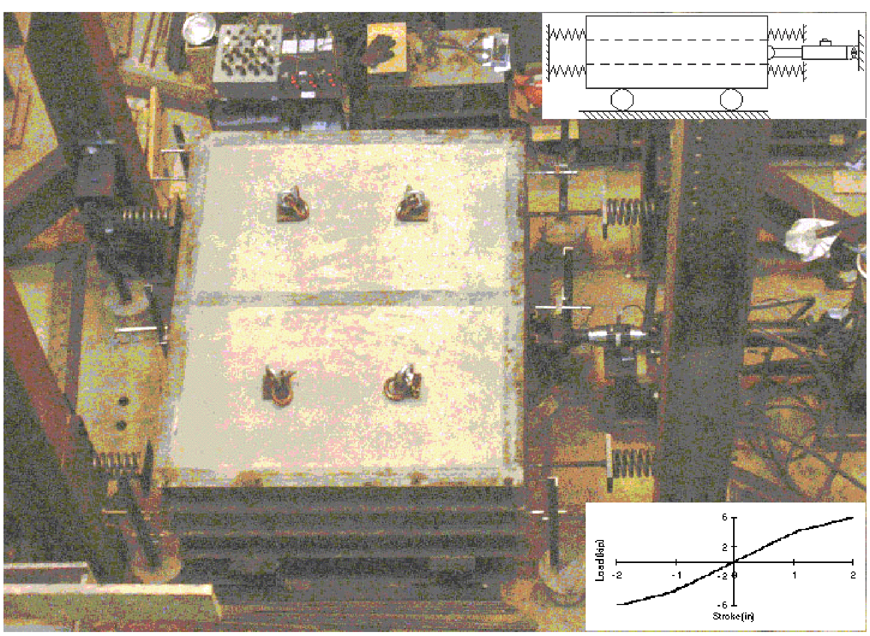


Figure 1 The testing structure with the servo-actuator.

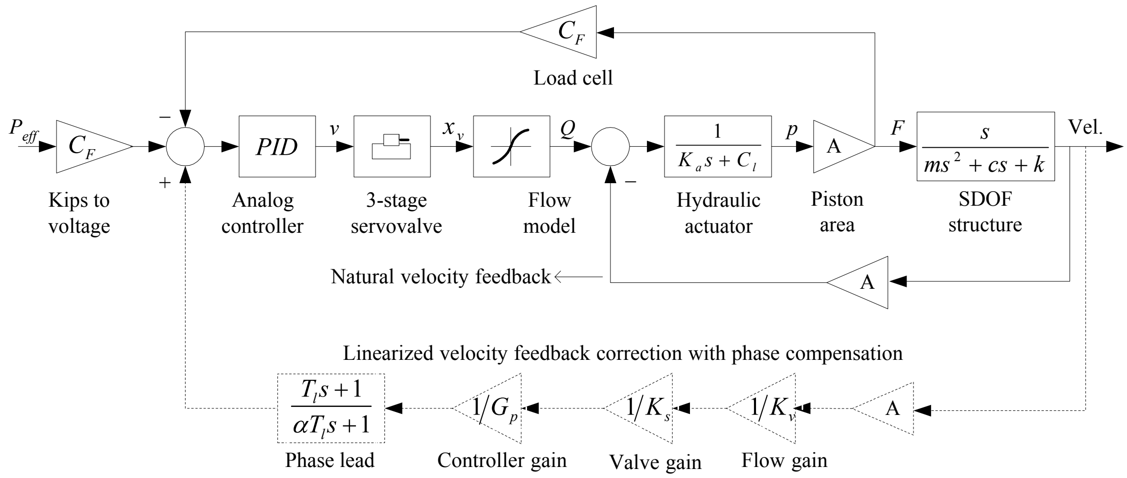


Figure 2 Model of dynamic system incorporating EFT with linearized velocity feedback correction (dashed lines)

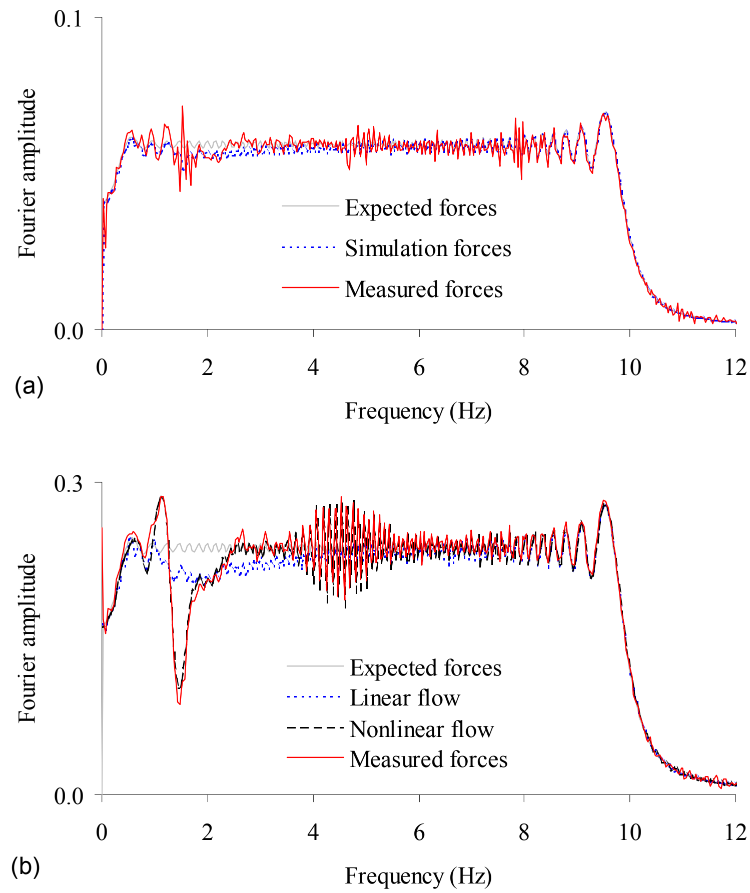


Figure 3 Fast Fourier Transforms of the command forces, measured forces and simulation forces. (a) Tests for a 0.5kip sine sweep; (b) Tests for a 2kip sine sweep.

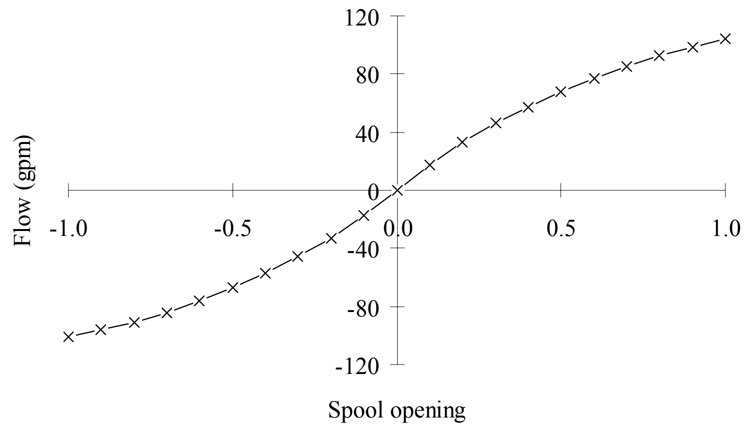


Figure 4 A typical flow-spool opening relation.

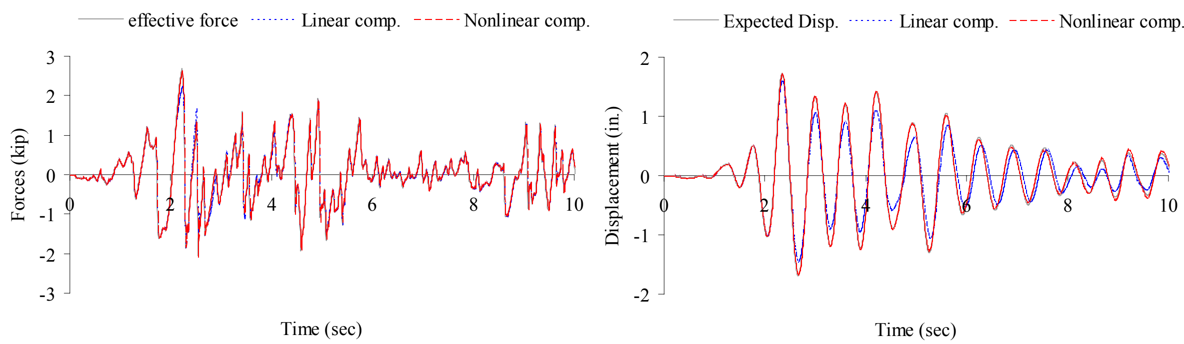


Figure 5 Comparison of the simulated performance of a linear and a nonlinear controller.

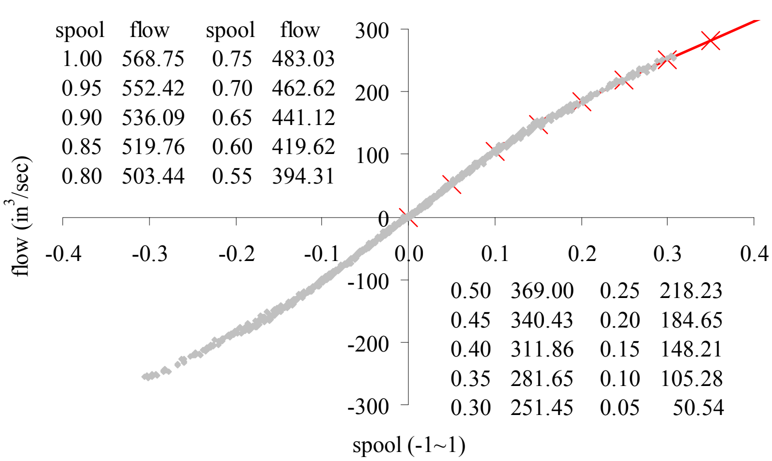
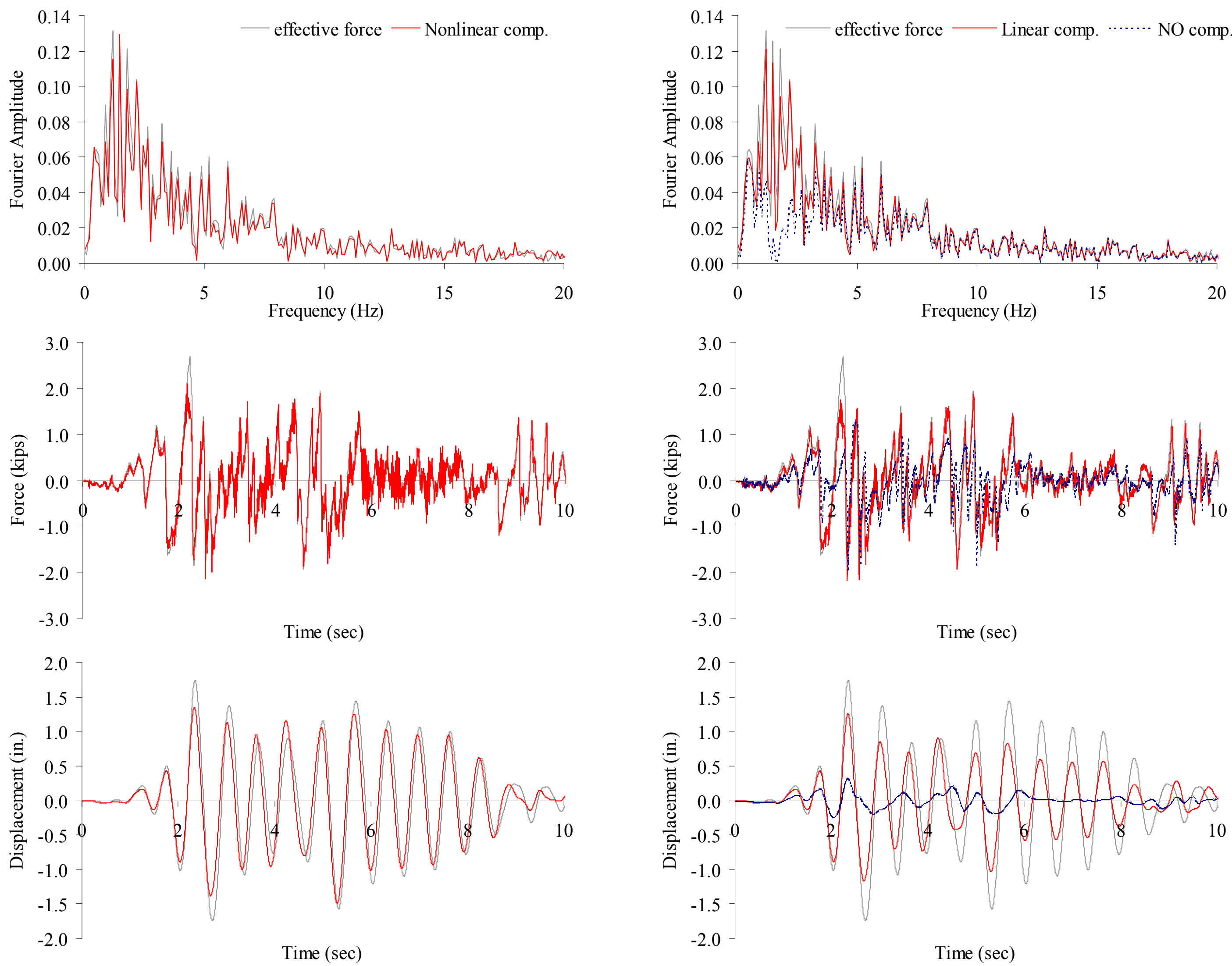


Figure 6 A measured flow-spool opening curve with a piecewise linear approximation.



(a) (b)

Figure 7 Comparison of the expected vs. measured response for the first 10 second of El Centro earthquake segment (0.17g) with nonlinear velocity compensation, with linear velocity compensation, and without velocity compensation.