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REAL-TIME DYNAMIC HYBRID TESTING OF STRUCTURAL SYSTEMS

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SUMMARY

This paper presents the development and implementation of a novel structural testing method involving the combined use of shake tables, actuators and computational engines for the seismic simulation of structures. The structure to be simulated is divided into one or more experimental and computational substructures. The interface forces between the experimental and computational substructures are imposed by actuators and resulting displacements and velocities are fed back to the computational engine. The earthquake ground motion is applied to the experimental substructures by shake tables. The unique aspect of the above hybrid system is force-based substructuring. Since the shake tables induce inertia forces in the experimental substructures, the actuators have to be operated in dynamic force control as well, since either the force or the displacement, but not both can be controlled at a given point. The resulting testing method is more versatile that existing seismic testing methods. First the substructuring strategy and the numerical integration algorithms associated with the computational substructures are discussed along with the implementation of the computational engine. Then the theory of dynamic force control and two different force control algorithms, one based on real-time convolution and the other on series elasticity, are described. Issues related to time-delay compensation are also discussed. The real-time hybrid system is implemented using a distributed architecture based on the replicated shared memory concept. The architecture allows for flexibility in the design of the system and in the components used. This real-time architecture is discussed next. An example of a real-time hybrid test is presented next.

INTRODUCTION

Several experimental procedures are used to simulate and test the behavior of structural systems and components under earthquake loads. These include (1) Quasi-static testing (2) Shake-table testing (Nagarajaiah [1]) (3) Effective force testing (Shield [2]) and (4) Pseudo-dynamic testing (Shing [3]) (see

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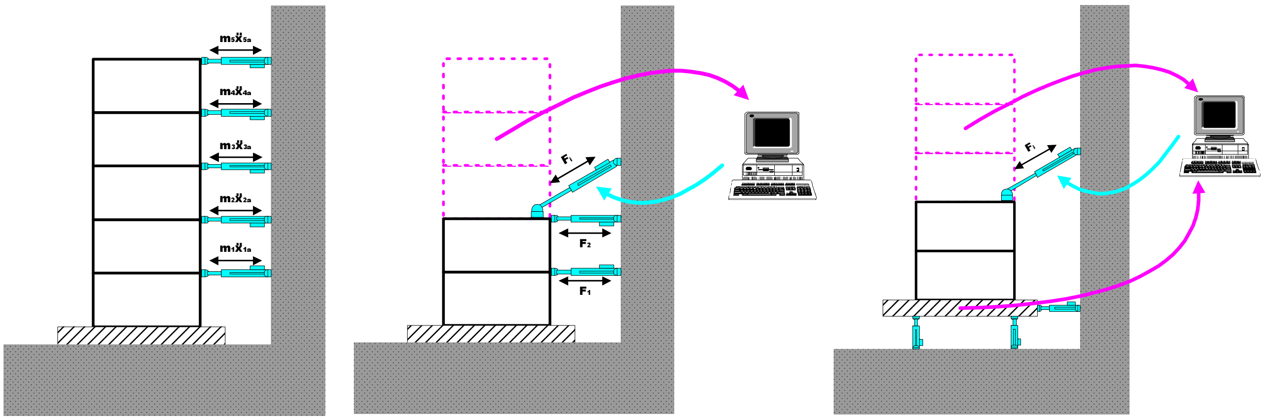
Figure 1). Real-time dynamic hybrid testing extends these methods by allowing for testing substructures under realistic dynamic loads and for representing rate-dependent and distributed inertia effects accurately. In a recent development a fast pseudo-dynamic method was developed to account for rate dependency. While the fast pseudo-dynamic and the real-time dynamic hybrid testing use sub-structures for physical testing and online computations to simulate the global system in real-time, the latter technique includes the inertia effects are part of the physical system testing.



(a) Effective Force (b) Pseudodynamic – Substructure (c) Real-Time Dynamic - Substructure

Figure 1. Modern Methods for "dynamic" testing in earthquake engineering.

Real-time Dynamic Hybrid Testing (RTDHT) shown in Figure 1(c) is a novel structural testing method involving the combined use of shake tables, actuators and computational engines for the seismic simulation of structures. The structure to be simulated is divided into a physical substructure and one or more computational substructures. The interface forces between the physical and computational substructures are imposed by actuators and resulting displacements and velocities are fed back to the computational engine. The earthquake ground motion, or motion of other computational substructures, is applied to the experimental substructure by shake tables. A schematic of the RTDHT system is shown in Figure 2. The right side in Figure 2 shows the physical computational infrastructure required for the implementation of the forces and motions at the interface of the physical and computational substructures.

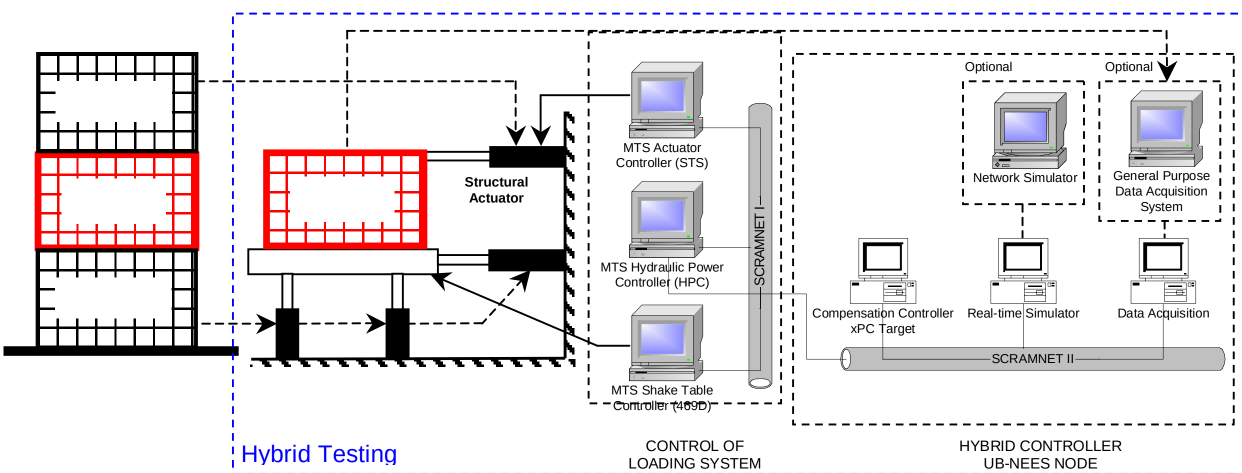


Figure 2. Schematic of Real-time Dynamic Hybrid Test System

The theoretical basis and the implementation of real time dynamic hybrid testing (RTDHT) is presented in the next sections.

SUBSTRUCTURING

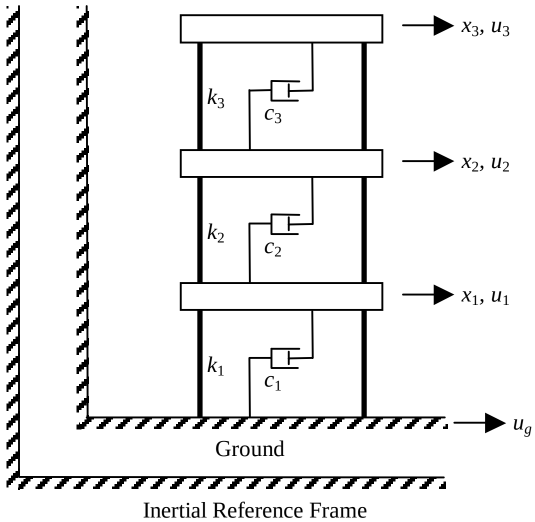


Figure 3. Three story model

The RTDHT implies first determining the model of the physical substructure being tested within the whole structural model identifying the interface parameters. The computational model must be simple in order to be executed in real time. For the sake of simplicity, derivations are presented here only for a structural configuration shown in Figure 3 A three-story model is shown in Figure 3 with its parameters. If u_g is the motion of the ground with respect to the inertial reference frame. u_i and x_i are the motions of the i^{th} story with respect to the fixed reference frame and with respect to the ground respectively, then $u_i = u_g + x_i$. The damping is assumed to be of the form shown in Figure 3, in order to preserve a simple formulation for the computational model Defining the fist and third floor in Figure 3 as *computational substructures* and the second floor as the *experimental substructure* as shown also in Figure 2, the equations of motion in the inertial reference frame are then given by:

$$\begin{aligned}
&m_1\ddot{u}_1 + (c_1+c_2)\dot{x}_1 - c_2\dot{x}_2 + (k_1+k_2)x_1 - k_2x_2 = 0 \rightarrow \text{Computational Substructure 2} \\
&m_2\ddot{u}_2 - c_2\dot{x}_1 + (c_2+c_3)\dot{x}_2 - c_3\dot{x}_3 - k_2x_1 + (k_2+k_3)x_2 - k_3x_3 = 0 \rightarrow \text{Experimental Substructure} \\
&m_2\ddot{u}_2 - c_3\dot{x}_2 + c_3\dot{x}_3 - k_3x_2 + k_3x_3 = 0 \rightarrow \text{Computational Substructure 1}
\end{aligned} \tag{1}$$

By considering the influence of the *experimental substructure* as external disturbance, the equations of the *computational substructures* may be written as:

$$\begin{aligned}
m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 &= -m_1\ddot{u}_g + \underbrace{k_2(x_2-x_1) + c_2(\dot{x}_2-\dot{x}_1)}_{\substack{\text{Force measured at the base}\\ \text{of experimental substructure}}} \\
m_3\ddot{x}_3 + c_3\dot{x}_3 + k_3x_3 &= -m_3\ddot{u}_g + \underbrace{k_3x_2 + c_3\dot{x}_2}_{\substack{(k_3\text{*displacement} + c_3\text{*velocity) of}\\ \text{experimental substructure}}}
\end{aligned} \tag{2}$$

Similarly the equation governing the *experimental substructure* may be arranged as follows:

$$m_2\left(\underbrace{\ddot{u}_g}_{\text{External Load}} + \ddot{x}_2\right) + c_2\left(\dot{x}_2 - \underbrace{\dot{x}_1}_{\text{Computed}}\right) + k_2\left(x_2 - \underbrace{x_1}_{\text{Computed}}\right) = k_3\left(\underbrace{x_3}_{\text{Computed}} - \underbrace{x_2}_{\text{Measured (Feedback)}}\right) \tag{3}$$

It is useful to introduce the relative displacement $x_{21} = x_2 - x_1$. Then equation (3) becomes:

$$m_2(\ddot{u}_1 + \ddot{x}_{21}) + c_2\dot{x}_{21} + k_2x_{21} = k_3(x_3 - x_2) \tag{4}$$

Being able to use both a shake table and an actuator to excite the experimental substructure introduces several possibilities. A number of alternatives exist for the application of the first floor acceleration $\ddot{u}_1$ and the thirds story force $k_3(x_3 - x_2)$: (a) One possibility is to apply the acceleration using the shake table and the force using the actuator. (b) Another option is to apply the ground acceleration using the actuator as well (as in the Effective Force method). (c) Yet another alternative is obtained by rearranging equation (4) as follows:

$$m_2\left[\underbrace{\ddot{u}_1-\frac{k_3}{m_2}\left(x_3-x_2\right)}_{\text{Equivalent acceleration}}+\ddot{x}_{21}\right]+c_2\dot{x}_{21}+k_2x_{21}=0 \tag{5}$$

The equivalent acceleration can be applied using the shake table only. However, the first story acceleration and the third story force can each be divided into two components, one to be applied by the shake table, and the other by the actuator. The actuator is assumed fixed in the inertial reference frame, while the structure is in a non-interial frame attached to the shake table. The actions are shown below:

$$\begin{aligned}
&\text{Shake table acceleration, } \ddot{u}_t = \underbrace{\alpha_1(s)\ddot{u}_1}_{\substack{\text{First story contribution}\\ \text{to shake table acceleration}}} \underbrace{-\alpha_3(s)\frac{k_3}{m_2}\left(x_3-x_2\right)}_{\substack{\text{Third story contribution}\\ \text{to shake table acceleration}}}\\
&\text{Actuator Force, } F_a = \underbrace{-\left[1-\alpha_1(s)\right]m_2\ddot{u}_1}_{\substack{\text{First story contribution}\\ \text{to actuator force}}}+\underbrace{\left[1-\alpha_3(s)\right]k_3\left(x_3-x_2\right)}_{\substack{\text{Third story contribution}\\ \text{to actuator force}}}
\end{aligned} \tag{6}$$

where $\alpha_1(s)$ and $\alpha_3(s)$ are frequency dependent splitting function such as for example band-pass filters. Such a splitting has several advantages. For instance, the oil column frequency of shake tables is relatively low due to their heavy mass and limits their bandwidth. Moreover, shake tables that are restrained by bearings do not perform well at very low frequencies. These frequency components can then be transferred to the actuator. The splitting can also be based on instantaneous or average power minimization for the test. These issues are discussed by Kausel [4].

The above substructuring and force splitting strategies are shown schematically in Figure 4. If $\alpha_1(s)\neq 0$ and $\alpha_3(s)\neq 0$, then the control requires a shake table and an actuator to implement the substructure testing. The actuator therefore has to operate in force control. However, if $\alpha_1(s)=0$ and $\alpha_3(s)=0$, however, two possibilities exist. Let the force input to the actuator in Figure 4 be F. Then the two possibilities are shown in Figure 5. In dynamic testing, the inertia is part of the experimental system, whereas in pseudo-dynamic testing, inertia effects are computed. Thus for hybrid testing ($\alpha_1(s)\neq 0$ or $\alpha_3(s)\neq 0$) or dynamic hybrid testing, the actuator should operate in force control. This is addressed in the next section.

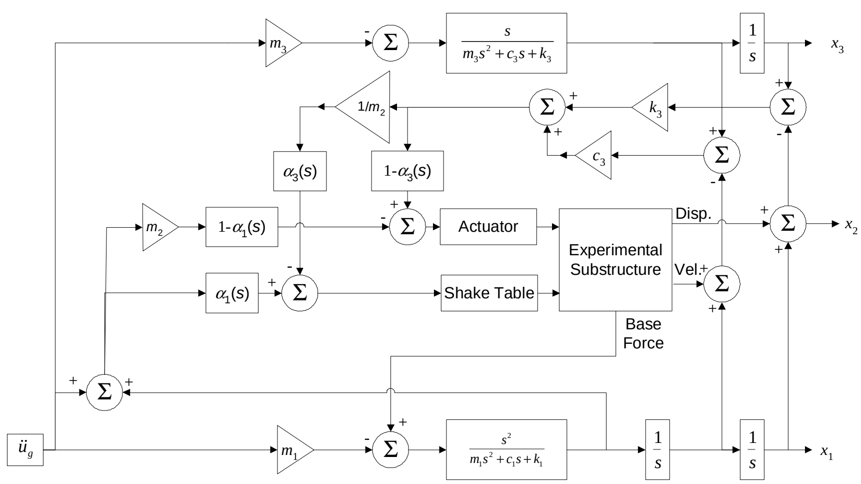
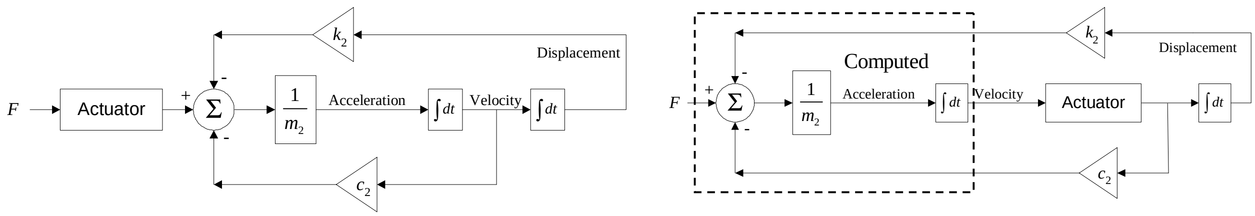


Figure 4. Schematic of hybrid substructuring



(a) Dynamic (b) Pseudo-dynamic

Figure 5. Dynamic and pseudo-dynamic testing

Such a unified view of hybrid simultaneous computation and experimentation testing systems provides a better perspective to develop algorithms and software. The similarities between compensation strategies in force and displacement control are reviewed for instance by Zhao [5].

DYNAMIC FORCE CONTROL

The implementation of the RTDHT requires therefore implementation of force control in the hydraulic actuators. This control is sensitive to the acceleration and force measurements, to the modeling of the compressibility of fluid, to the nonlinearities of the servo control system (servovalves) and other stiffness issues as indicated by Dimig [6] and Shield [2]. The authors developed two approaches for dynamic force control:

Approach 1: Convolution Method

A small-scale pilot setup was devised using an electromagnetic actuator to test force control. The setup is shown in Figure 6. The left side show the model of 2-dof structure – schematic and physical – while the right side shows the substructure testing set-up using bottom d.o.f. and an actuator.

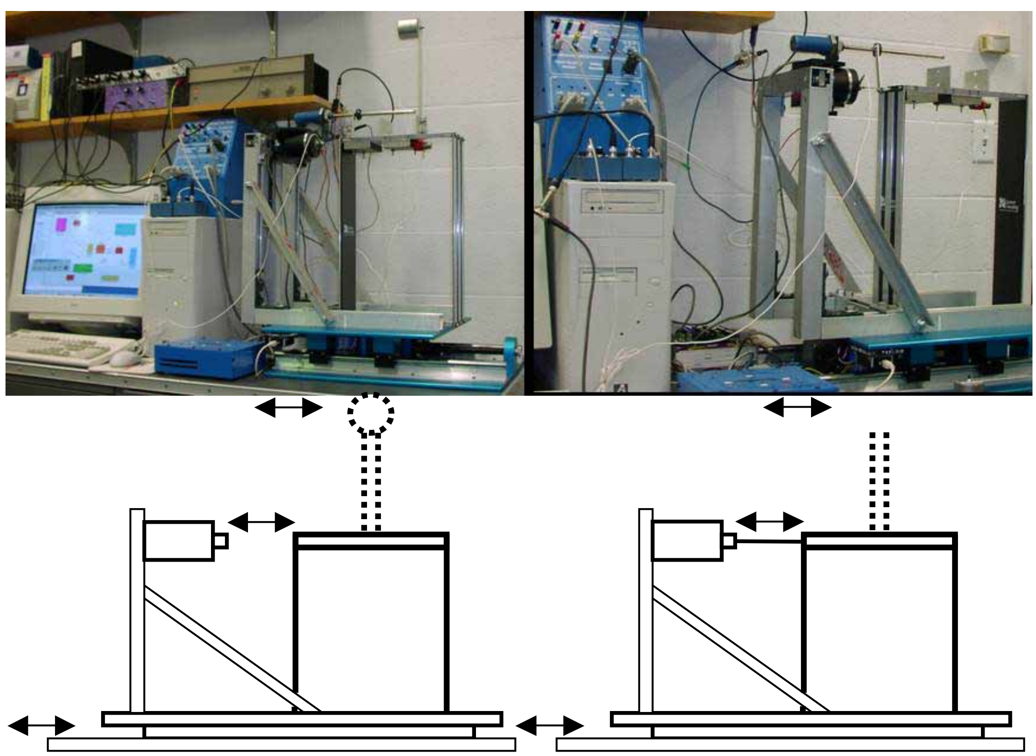


Figure 6. Small scale pilot testing of convolution method

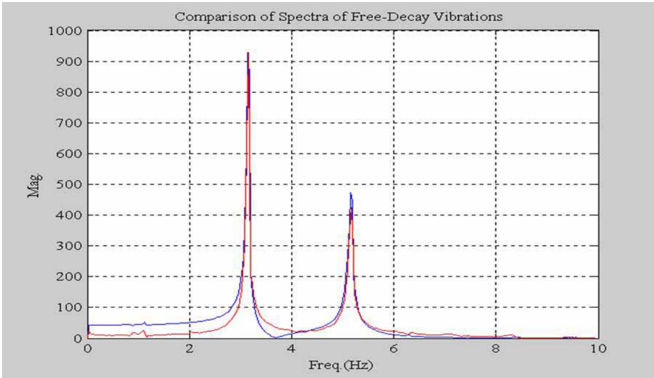


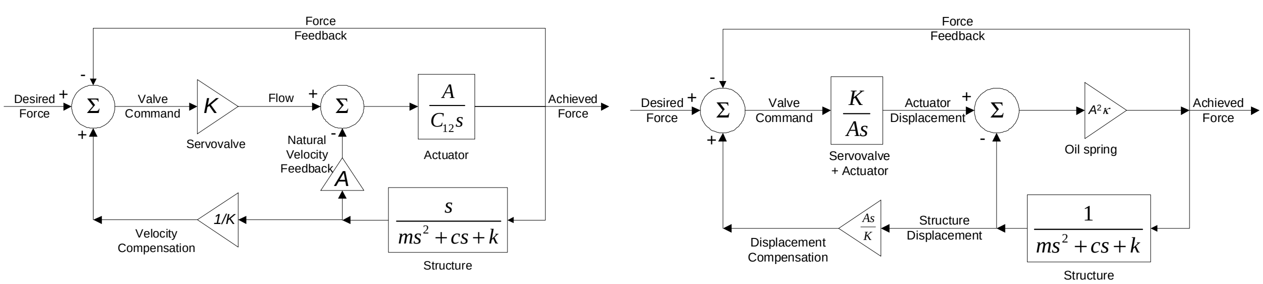
Figure 7 Pilot test performance for white noise

The convolution method is using a compensation technique that is based on identification of the frequency response function (FRF) of the system and modifying the force input by the inverse of the FRF. The operation is done in the time domain by evaluating the convolution integral. The forces are calculated based on Equation 6 with $\alpha_1(s)=1$ and $\alpha_3(s)=0$. Without the compensation the implementation is not feasible. The system was tested for free vibrations, and base motion –white noise and earthquakes.

The performance for the white noise of the 2-dof and the 1-dof hybrid set-up is shown in Figure 7. The hybrid system simulates the 2-dof over the entire frequency range except for the very low frequencies with errors of up to 5%. This approach requires a very careful identification of the actuator and structure as tested and an off-line computation which is not feasible in real time. A second approach was developed and tested as described below.

Approach 2: Series Elasticity with Displacement Compensation

By its physical nature, a hydraulic actuator is a rate-type device or velocity source, i.e., a given controlled flow rate into the actuator results in a certain velocity. Moreover, hydraulic actuators are typically designed for good position control, i.e., to move heavy loads quickly and accurately. They are therefore by construction high impedance (mechanically stiff) systems (Nachtigal [7]). In contrast for force control, a force source is required. Such a system logically would have to be a low-impedance (mechanically compliant) system. Thus force control using hydraulic actuators is an inherently difficult problem. It was recognized by Conrad [8] that closed-loop control with force feedback is ineffective without velocity feedforward or full state feedback. This is further supported by Dimig [6] and Shield [2] in their work on the effective force method. Figure 8(a) shows the force control with velocity compensation proposed by Dimig [6]. Rearranging terms by multiplying by *A* and dividing by *s*, the control loop in Figure 8(b) is obtained. It can be seen that the oil behaves as a spring providing the flexibility required for force control. Relative deformation occurs across this spring and force is applied through it.



(a) Control loop (b) Rearranged

Figure 8. Force control of Dimig [6].

Actuators designed for position control have stiff oil columns, making force control very sensitive to control parameters and often leading to instabilities. Moreover friction, stick-slip, breakaway forces on seals, backlash etc. cause force noise, making force a difficult variable to control. Several strategies have been introduced to work around this problem. For instance the dual compensation scheme of MTS [9] uses a primary displacement feedback loop with force as a secondary tracking variable. It also supports features such as acceleration compensation to compensate for some of the effects that distort the control force. In robotics, and impedance control strategy has been employed wherein the force-displacement relationship is controlled at the actuator interface (Mason [10] and Whitney [11]). Pratt [12] has used the idea of series elastic actuators where a flexible mechanism is intentionally introduced between the actuator and the point of application of force, along with force feedback.

Motivated by these observations and by the fact that causality requires a flexible component in order to apply a force (Paynter [13]), in the force control scheme described here, a spring is introduced between the actuator and the structure as shown in Figure 9. Notice that the scheme is similar to that of Figure 8(b) except that (1) the intentionally introduced series spring, K_{LC}, assumes the role of the oil spring and (2) there is no force feedback loop.

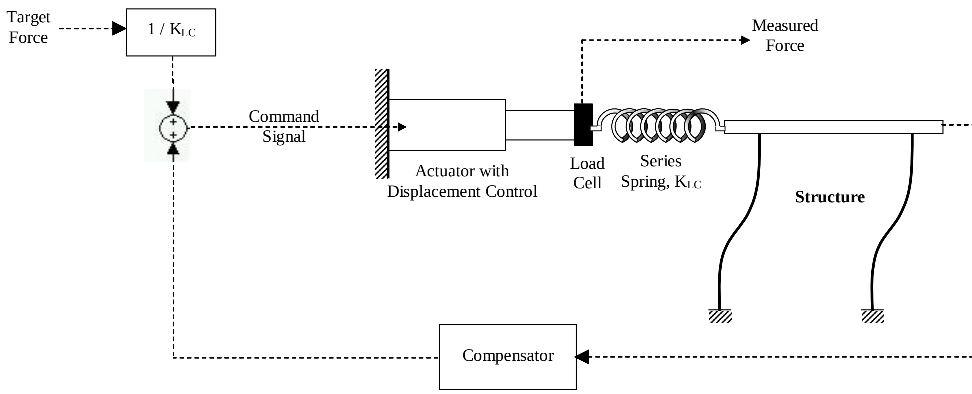


Figure 9. Proposed force control scheme

In the schemes of both Figure 8(b) and Figure 9, the actuator behaves as a displacement device. The tuning of the hydraulic actuator in displacement control can be done relatively easily, very precisely and independent of the structure and leads to a well behaved closed-loop transfer function with good force disturbance rejection. Hence the actuator in the control scheme of Figure 9 is operated in closed-loop displacement control with a PIDF controller. The resulting block diagram is shown in Figure 10.

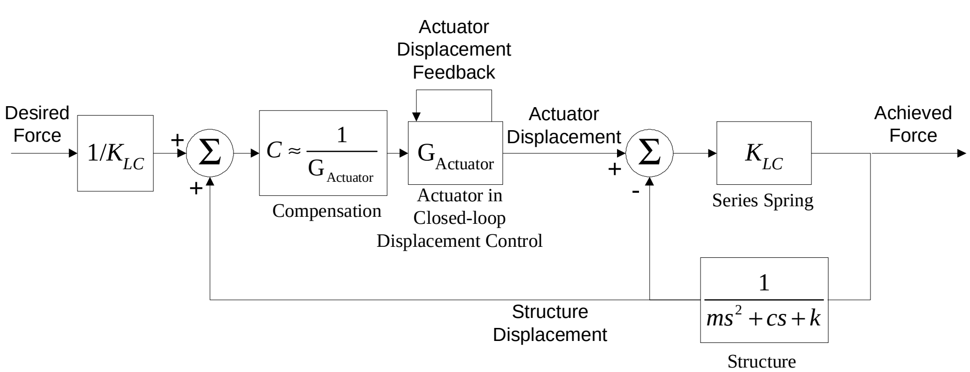


Figure 10. Block diagram of proposed force control

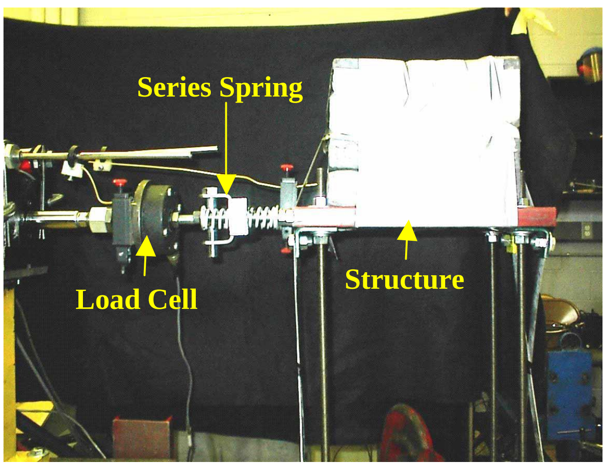
Although the system as a whole controls force function, *internally the actuator operates in closed-loop displacement control*. Hence, there is no need for an additional force feedback loop to ensure stability. Besides, this would prevent the actuator responding to spurious force errors caused by the sources listed above. The force transfer function is given by:

$$\frac{\text{Achieved Force}}{\text{Desired Force}} = \text{CG}\frac{ms^2 + cs + k}{ms^2 + cs + k + K_{LC}\left(1 - \text{CG}\right)} \tag{7}$$

In the ideal case where C = 1/G, the transfer function has the value of one. The advantages of using the series spring are also now apparent: (1) the actuator can be well tuned and operated in displacement control, (2) it provides for one more parameter than can be altered in the control design (the oil stiffness cannot be) and (3) the term K_{LC}(1-CG) in the transfer function indicates that the smaller the value of KLC the less sensitive is the transfer function to deviations of C from 1/G. The following paragraphs present results from applying the above control procedure. Experiments were performed using a small-scale test setup and a medium-scale test setup.

Small-scale Test

The test setup is shown in Figure 11. An MTS 252.22 two-stage servo-valve and an MTS actuator were used. The controller used was the MTS FlexTest GT. The structure was designed to have a resonant frequency of 3.8 Hz. The force control system consisting of the compensator C and the displacement feedback was implemented as a cascade controller around the MTS controller using *Simulink* and *xPC Target* (Mathworks [14]).



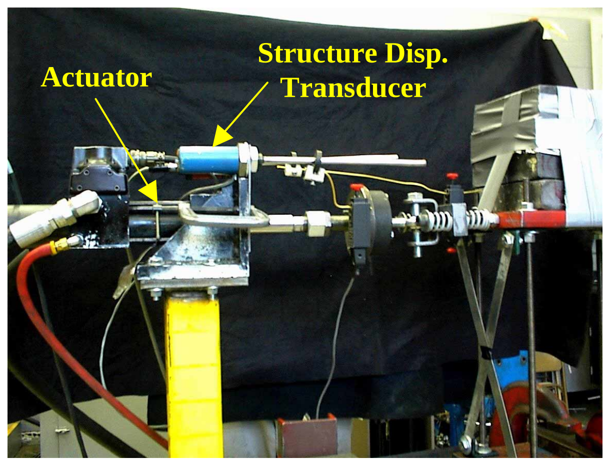
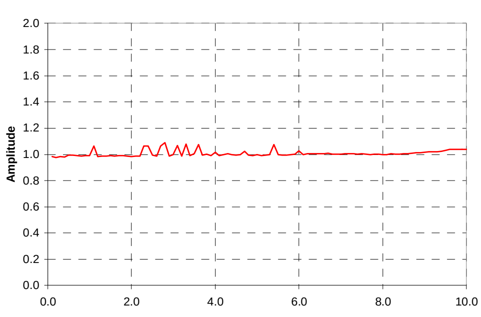
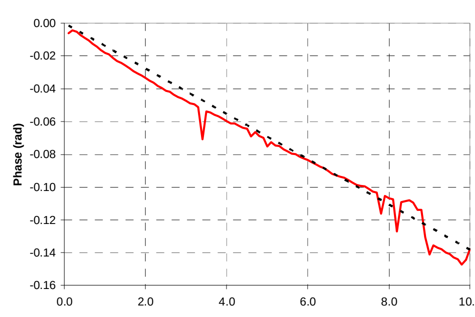


Figure 11. Small scale force control setup

The actuator was tuned first in displacement control and the closed-loop displacement transfer function was measured. This is shown in Figure 12. For the frequency range of interest, it is seen that the actuator can be modeled as a pure time-delay system with a delay of 2 ms.



(a) Amplitude



(b) Phase

Figure 12. Actuator Transfer Function

One approach to compensating for this delay was by using a lead-lag compensator (see for example Shield [2]). The other approach was to use a Smith-type predictor (Marshall [15]) as shown in Figure 13.

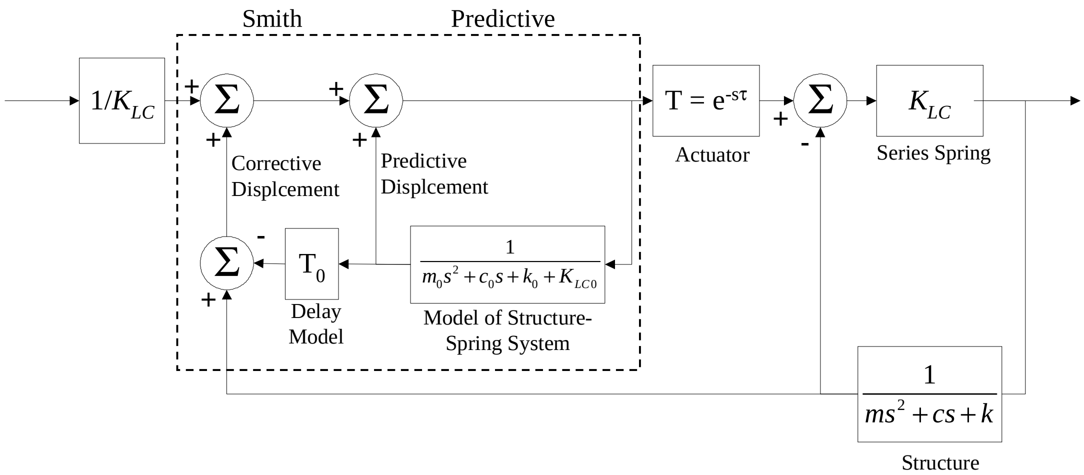
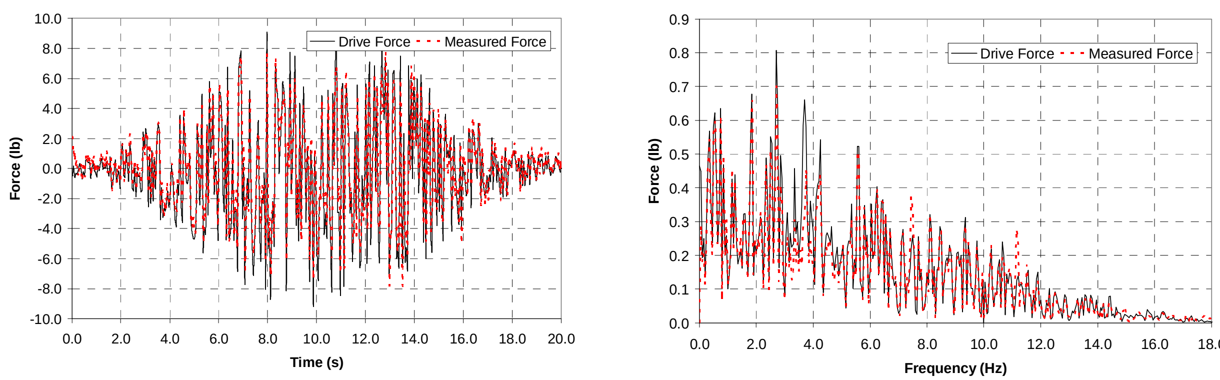


Figure 13. Smith predictive time-delay compensator

A typical result from a force control test using a random excitation is shown in Figure 14.



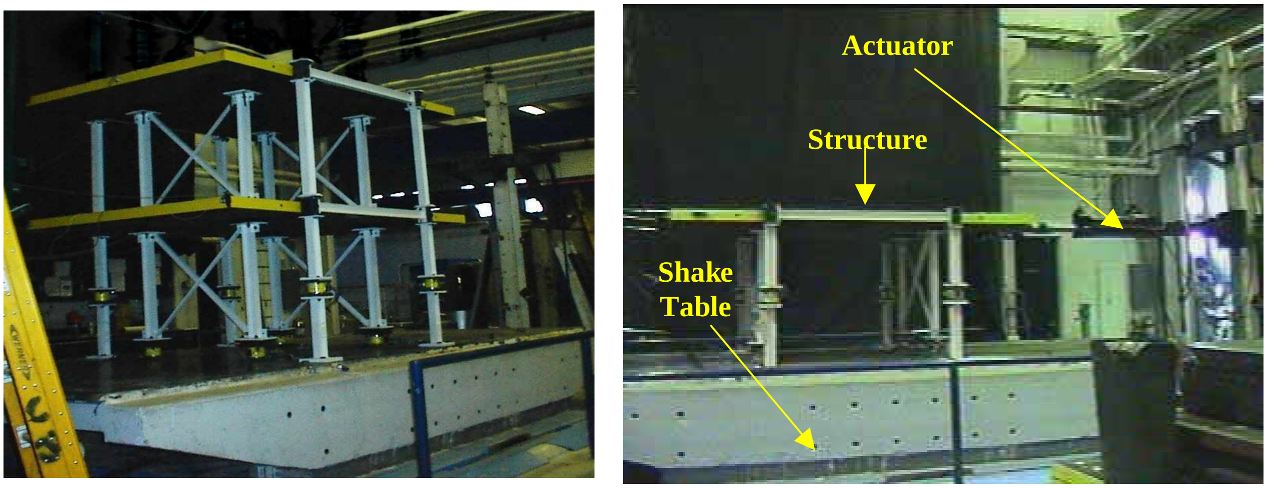
(a) Time Domain

(b) Frequency Domain

Figure 14. Target vs. achieved force in small-scale test

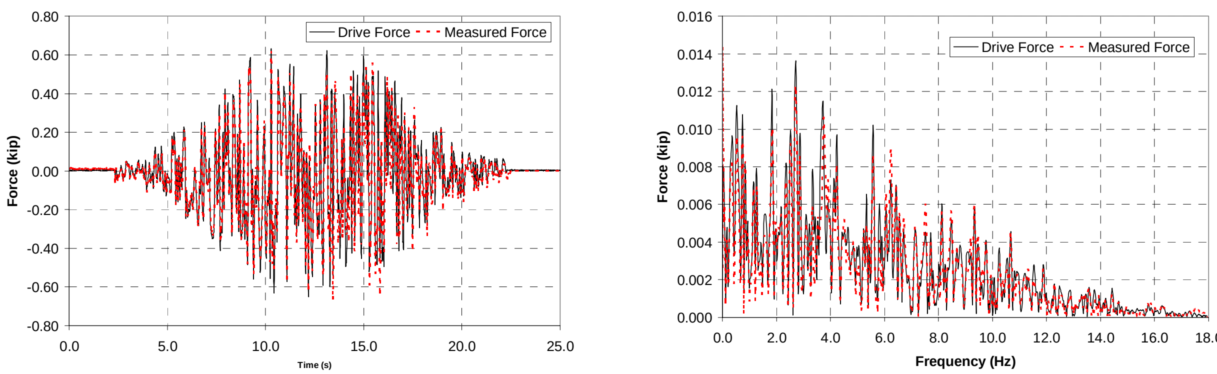
Medium-scale test

The medium-scale test is setup for hybrid testing. The results from force control tests are presented in this section and those from real-time hybrid tests are presented in the next section. A 24 inch stroke actuator with a 30 gpm two-stage servo valve was used. The controllers were the same as for the small-scale test. Figure 15 shows the one story substructure. Figure 16 shows the results from a force control test with random force excitation in time and frequency domains. Figure 17 shows the comparison between shake table test and an effective force test applying the same ground acceleration as equivalent force. Figure 17(a) shows the ground acceleration as applied by the shake table vs. that applied by the actuator. Figure 17(b) shows the respective structure displacements.



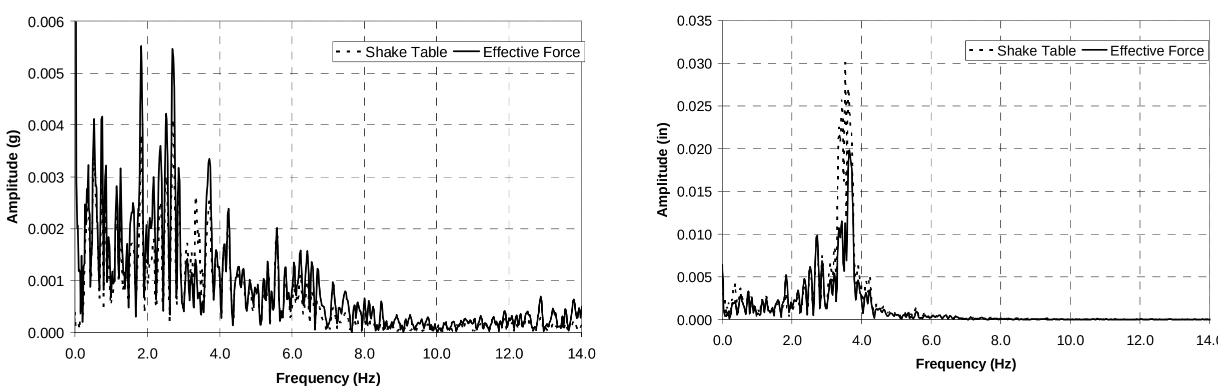
(a) Two stories structure (b) Hybrid system –shake table and actuator

Figure 15. Medium-scale test setup



(a) Time Domain (b) Frequency Domain

Figure 16. Target vs. achieved force in medium-scale test



(a) Applied force (b) Structure displacement

Figure 17. Effective force test

REAL TIME HYBRID TESTING

A hybrid test was performed on a two-story structure with the first story built on the shake table and the second story simulated as shown in Figure 15. In relation to Figure 4, $u_1 = u_g$, $\alpha_1(s)=1$ and $\alpha_3(s)=0$. The control scheme is shown in Figure 18. A sample result from a sine-sweep test in shown in the frequency domain in Figure 19. The result is compare with a computational simulation of the two story model. The result shows a small discrepancy in the damping representation. This is believed to stem from unmodelled damping in the system and from some latency. This is the subject of current work.

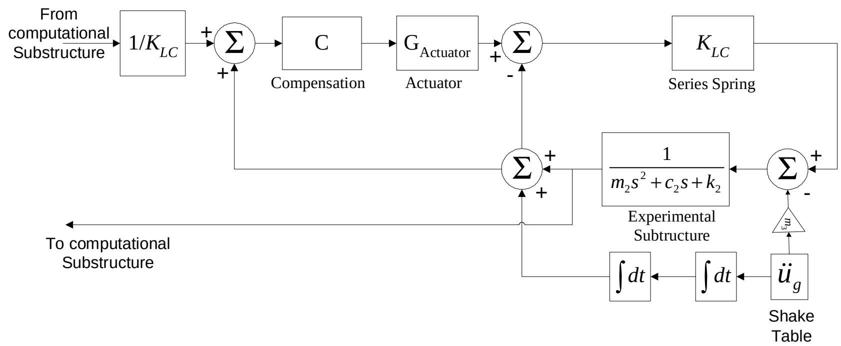


Figure 18. Control scheme for two story hybrid test

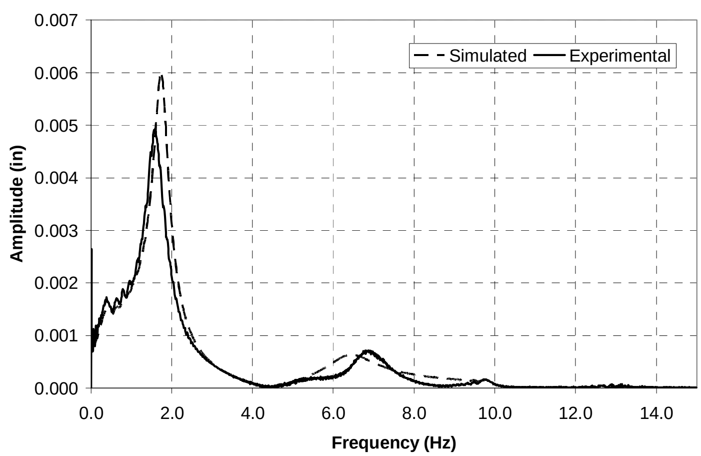


Figure 19. Two story hybrid sine-sweep test result

DISTRIBUTED REAL-TIME ARCHITECTURE

The real-time hybrid system is implemented using a distributed architecture that uses Shared Random Access Memory Network (SCRAMNET^{TM}), a very low-latency replicated shared memory fiber optic network. The architecture of hardware-software controller (see right side of Figure 2) allows for flexibility in the design of the real-time operating system and in the implementation of the components used. There are three units as shown in the Hybrid Controller.

1. **The Compensation Controller** which contains the cascade control loop for force control presented above. This controller also compensates for time-delays that are inherent in the physical system and those that are introduced by filtering, computation, network communication, etc.

2. **The Real-time Simulator** which simulates the computational substructures. The architecture has been designed so that this simulator could be seamlessly replaced by one that is at a remote location. With the current state-of-the-art, however, network speed would become the bottleneck to real-time operation. This is accommodated using an additional interrupt mechanism.
3. **The Data Acquisition System** (DAQ) that is used for feedback from the experimental substructure as well as for archiving information during the test.

The controller operates in a synchronous-asynchronous manner. The controller was developed to allow parallel operations of each of the three units while sharing only essential information through a "pool" memory provided by the 1μsec update rate SCRANNET. Each individual component / unit operates at each own time rate, accessing the shared memory when needed, without delaying other units. The compensation controller, which determines the control signals send to the actuators, is designed to compensate also for all other latencies in communications, computing and hydraulic operations. The current implementation at University at Buffalo uses dedicated units for each of the three components; however, the architecture shown above allows to substitute the Simulation Component with any computational device - such as a supercomputer operating in a Grid – or to substitute the dedicated data acquisition unit with a multipurpose laboratory DAQ connected to the SCRAMNET. The controller design allows for a seamless substitution which can be used to scale the RTDHT to larger structural systems.

CONCLUDING REMARKS

The Real Time Dynamic Hybrid Testing System is implementing combined physical testing and computational simulations to enable dynamic testing of sub-structures including the rate and inertial effects while considering the whole system. The paper presents a new force control scheme with a predictive compensation procedure which enabled the real-time implementation. The new system was tested through bench tests and medium scale pilot testing successfully. The procedures are implemented in the full / large scale University at Buffalo NEES node which includes two six degree of freedom shake tables and three high speed dynamic actuators and a structural testing system controller (STS) capable to implement the control algorithms presented above.

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