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OFFLINE TUNING OF SHAKING TABLES

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SUMMARY

The control performance of shaking tables is greatly affected by the interaction between the table and the test specimen. This is because the dynamic characteristics of large specimens give significant force disturbances to the table. For this reason it is not appropriate to run the table with controller tunings determined using the bare table if good control is desired. The controller must be retuned with the specimen present. However, the shaking that is an unavoidable part of the tuning process can cause damage to the specimen. In this paper we present an alternative tuning method that involves tuning a dynamic model of the system. The actual digital control software is used to drive the dynamic model, thereby eliminating a large portion of modeling effort as well as the potential for error. Only the mechanical and hydraulic portions of the system need be modeled. The model is implemented in Mathworks Simulink®, and includes only those effects that characterize the dynamics to a degree sufficient for control tuning.

INTRODUCTION

Current shake table systems provide the capability of testing various structures subjected to real time input excitation, retaining real dynamic response such as inertial and damping effects. Typical high-performance shaking tables have high bandwidth capabilities provided by complex digital controllers. These controllers require extensive knowledge by the operator to achieve the desired table performance, which can vary greatly depending on the response of the system. The addition of a test specimen onto the shaking table increases the controller complexity due to the resonant force feedback during table excitation.

For a shaking table to have matched command and feedback, the transfer function between these terms must be unity gain in the bandwidth of interest for linear systems. Although typical earthquake excitation frequencies have lesser energy content above 5 Hz, matching is generally desired up to and beyond 20 Hz.

To achieve the unity gain between command and feedback, the shake table system must be "tuned" by adjusting multiple control variables, such as gains, lead terms, and notch filters. These settings cannot be determined a-priori, since the shaking table must be in motion to obtain the required feedbacks. Since any change in a controller parameter is instantaneous, any error or incorrect choice of term can result in

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damage to the shaking table, damage to the specimen, or complete destruction of the specimen. Low-level random excitation is typically used since it excites the range of desired frequencies and allows the transfer function to be computed for those frequencies. While the table is in motion, the terms are incrementally changed and the results determined from the FRF (transfer function). This “iterative” procedure is continued until the desired level of FRF matching is achieved, but is highly dependant on the skill of the operator.

Ideally, the latter procedure would be accomplished with the specimen attached to the shaking table. The FRF would then represent the system and specimen parameters, such as oil column resonance, specimen resonance, damping, friction, etc. This tuning would only be accomplished while the shaking table is in motion. As a result, the specimen is constantly being excited. This may have the undesired effect of low-level damage or fatiguing, even if the specimen is tuned while in the elastic range (Laplace, 2001, Laplace 1999).

Although the authors consistently tune the system with the specimen attached, most shaking tables are tuned using only the bare table without the specimen attached. Thus any mistakes by the operator would not be damaging since no specimen is attached, and the low level random excitation can be continued indefinitely.

Once unity gain of the bare table is achieved, the specimen is attached to the shaking table. This has the advantage of not pre-damaging the specimen due to tuning. The disadvantage is the FRF between command and feedback is no longer unity gain due to the specimen parameters. Also, if the specimen has too high of a resonant gain, once the system is brought online with the specimen attached, it may be possible for the shaking table system to go unstable.

One method presented by the authors to overcome the latter drawbacks is the development of a simulation model. This model numerically represents the shake table system and any attached specimen. Although numerical simulations of servo-hydraulic systems have been presented in the past, the uniqueness of this model is that it retains the real-time controller software in the simulation. This removes any uncertainties in controller modeling and provides the operator the same familiarity with the simulation as with the real system controllers.

If the simulation model correctly represents the real system, the operator can adjust the real-time controller settings until either unity gain is achieved or the desired FRF is obtained. Once all the controller parameters are determined, these can be directly transferred to the real system thus achieving an optimally tuned state.

SHAKING TABLE SYSTEM

The shaking table system in the Large Scale Structures Laboratory at the University of Nevada, Reno presented an ideal opportunity to develop a comprehensive simulation model. The system consists of three identical MTS Systems Corporation servo-hydraulic bi-axial shaking tables each with a nominal 50-ton payload capability and 165 kip 500gpm 24-inch stroke actuators (Figure 1). The system has been in continuous operation since 1996 and recently upgraded as part of the NEES consortium (Network for Earthquake Engineering Simulation). The system has used first generation MTS digital controllers with state of the art Three Variable Control software since installation.

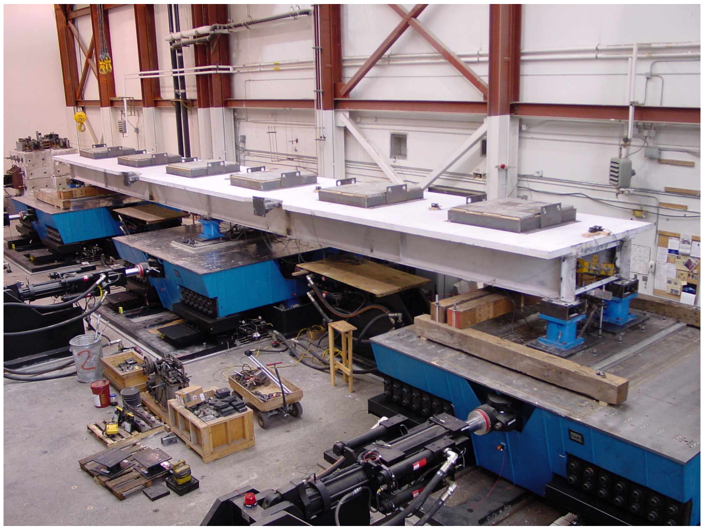

Figure 1: University of Nevada, Reno Shake Table System

Continuous operation with a multitude of research has provided significant knowledge and experience with various issues relating to servo-hydraulic control. Multiple techniques have been implemented for control of resonant specimens and command-feedback matching at high amplitudes and non-linear response. The shaking table system is mechanically "clean" with low friction, unconstrained actuators and high performance capabilities, thus providing an ideal system for numerically modeling the shake table and servo-hydraulic system.

RESONANT SPECIMEN

Most specimens tested using a shake table system have resonant force feedback and damping. The magnitude of each and their effects on the shake table system generally depend on the specimen weight and response relative to the table size. A light specimen with low force feedback relative to actuator force will not present the problems as a heavy specimen with large force feedback. Typical specimens tested at UNR fall into both ranges. For the simulation model, a resonant damped SDOF specimen was constructed with a natural frequency below oil column and a low damping ratio. This specimen provides high enough force feedback to provide difficult tuning and a realistic test of the simulation model.

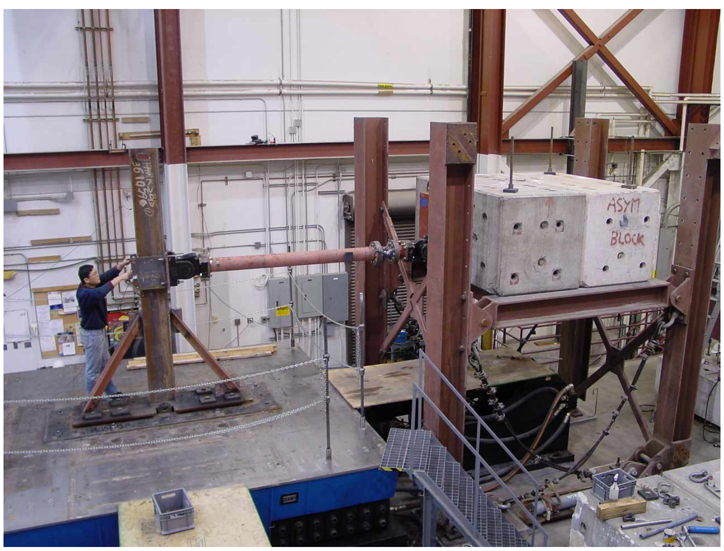

Figure 2: Resonant Specimen and Inertial System

An off-table inertial system has been used since 1996 to provide the inertial mass for SDOF specimens (Laplace, 1999). This system is linked to the specimen using ball-jointed swivels. The mass rig and link system provide low friction and no additional vibration, effectively acting as if the mass was directly on the specimen. A braced steel column was designed to rigidly attach to the shake table and connect to the inertial system (Figure 2). The column was designed with a natural frequency of 5Hz, low damping, and a force feedback of at least 50% of the desired actuator force. These parameters are typical of some of the research specimens tested in the past.

TYPICAL ONLINE TEST PROCEDURE

Specimen Characterization

Before any excitation is applied to the shake table system, the specimen parameters are measured in their untested state. This can be done with the shake table in "warm-up" mode with all gains (except displacement gain) and lead terms set to zero, which generally provides a safe start up regardless of the specimen response or table tuning. With only displacement gain, a low amplitude square wave displacement input with a long period can be applied to the system. This produces a free vibration response in the specimen. Typically the acceleration or force response waveform is used to compute natural frequency and damping (using Fourier transforms for frequency and logarithmic methods for damping). This data provides the initial state as a reference point of the specimen before tuning.

Tuning

The tuning process begins with all gain, lead, and notch filter terms at zero. A low level random motion is applied to the shaking table while monitoring both the plant FRF and all the instrumentation on the specimen. These terms are incrementally adjusted while keeping the excitation levels below set peak instrument levels. This is an iterative process since the terms interact; changing one term has a desired effect in one frequency region and an undesired effect in another. Although unity gain response can be quickly achieved when tuning a bare table, tuning a table loaded with a difficult resonant specimen may require considerable operator time and expertise, and even so the result may be far from optimum. Once the tuning process is completed, a final free vibration test is performed.

Incremental drive files until failure

Once a reasonable approximation to unity gain response is obtained, the input motion, which can be recorded ground acceleration or synthetically developed acceleration record can be used. Typical testing involves scaling the input acceleration amplitude down to a low level and running the shake table system using this scaled input. The specimen and data are analyzed and another, slightly higher amplitude motion is applied. This continues on until the desired performance is observed or specimen failure occurs.

SIMULATION MODEL

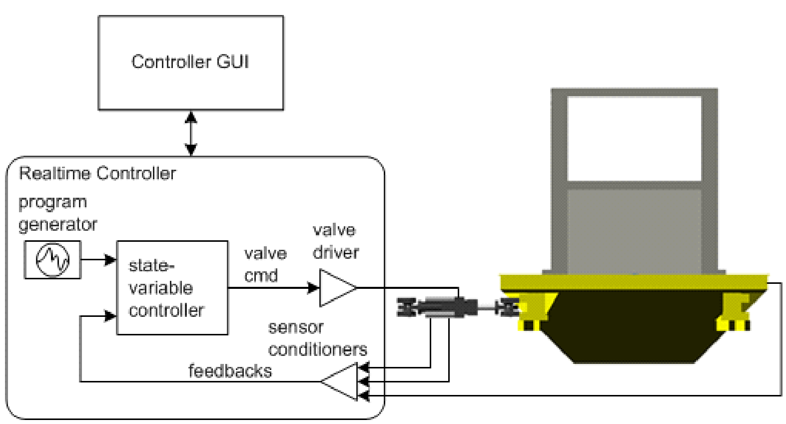

Figure 3: Actual seismic table control system

The actual seismic table control system on which the simulation is based (Figure 3) consists of a Controller Graphical User Interface (GUI), a Realtime Controller (Thoen 2004), and the mechanical system comprising a hydraulic actuator, table, and test specimen. Closed loop control is accomplished by a state-variable controller within the Realtime Controller that computes servovalve command updates on the basis of displacement, acceleration, and force sensor feedbacks.

In the simulated seismic control system shown in Figure 4, the actuator, table, and specimen is replaced with a Matlab (Mathworks 2004) Simulink model. The Controller GUI is the same as that of the actual

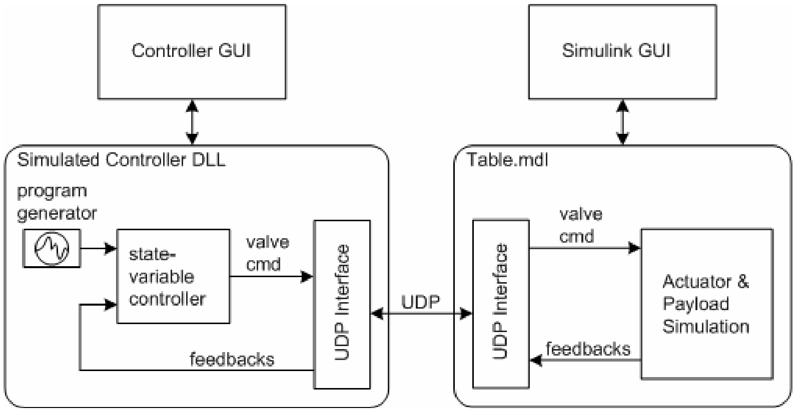

Figure 4: Simulated seismic table control system

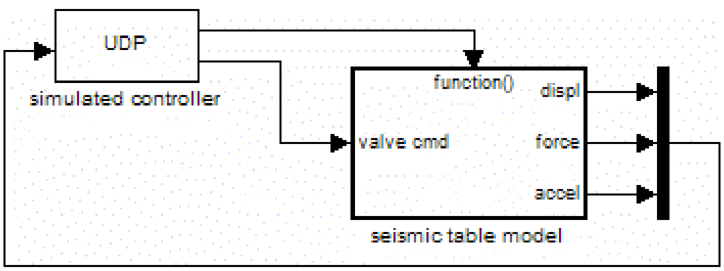

Figure 5: Top-level diagram of the Simulink seismic table model

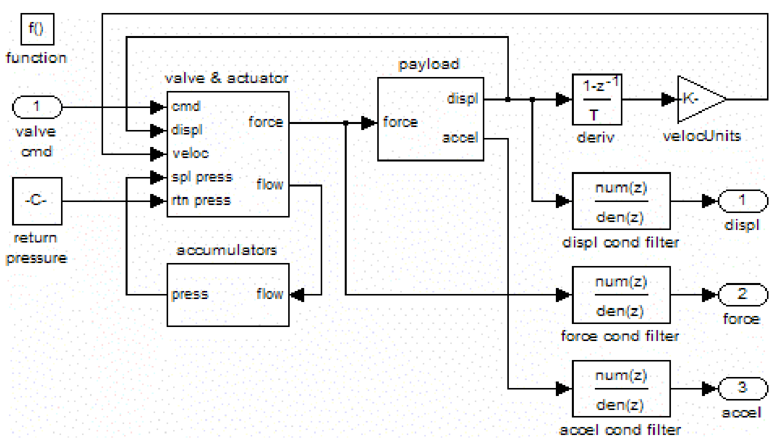

Figure 6: Seismic Table Model detail view

seismic control system, except the Realtime Controller is replaced by a Windows DLL that executes control software in non-realtime. Because sensor conditioner hardware does not exist in the Simulated Controller, the sensor feedbacks come instead from the Simulink model via User Datagram Protocol (UDP), which is the communication mechanism used to link the Simulated Controller program with the Simulink program. Likewise, the valve driver hardware does not exist in the Simulated Controller; instead the valve command is sent to Simulink via UDP.

Figure 5 shows the top-level view of the Simulink model of the seismic table. When the Simulated Controller sends a valve command update to Simulink via UDP, this value emerges from the "simulated controller" block and the "seismic table model" block is clocked once. Simulink then computes the feedbacks corresponding to one sample period of simulation time. The feedbacks are then gathered together and sent to the Simulated Controller via UDP. The Simulated Controller sets the pace of the simulation, and runs at a rate fast enough to maintain the illusion of realtime.

Looking inside the "seismic table model" block of Figure 5, Figure 6 shows the major components that comprise the model:

Valve & Actuator Model

Using valve-opening command as input, this block computes actuator force output. In addition, oil flow is also computed for the accumulator model. The detailed discussion of the internal details of the Valve & Actuator model is beyond the scope of this paper; it suffices to briefly state that it uses the flow equation (simplified version shown here)

$$X_{SPOOL} C \operatorname{sgn}(P_P - P)\sqrt{|P_P - P|} = A\dot{X} + \frac{\frac{L}{2} + L_E + X}{\beta_{OIL}}(A\dot{P} - K_D \ddot{X}_{OIL}) + K_L \Delta P$$

where

X_{SPOOL} = valve spool displacement, C = flow gain, P_P = port pressure,
P = actuator chamber pressure, A = actuator area, $\dot{X}$ = actuator velocity,
L = actuator useable stroke, L_E = actuator endcushion length , X = actuator displacement,
β_{OIL} = oil bulk modulus, K_D = oil column damping factor, X_{OIL} = oil displacement, and
K_L = piston leakage,

to compute actuator chamber pressure P; the force output of the actuator is then the difference between chamber pressures times the piston area $F = \Delta P \cdot A$.

Accumulator Model

The accumulator block utilizes the adiabatic gas law $P_1V_1^n = P_2V_2^n$ to compute hydraulic supply pressure change with net flow demand, as shown in Figure 7. Flow demand in excess of hydraulic pump flow discharges the accumulators, causing the supply pressure to drop. Conversely, an excess of hydraulic pump flow charges the accumulators, causing the supply pressure to rise.

Payload Model

The payload model, shown in Figure 8, computes the displacement and acceleration response of the table, specimen, and foundation to force applied by the actuator. With actuator force (minus friction force) applied to the input, the output of the specimen/table interaction model computes the acceleration at the table end of the actuator, whereas foundation model computes the acceleration of the base of the actuator. Summing the two accelerations yields the net acceleration seen by the actuator, which is then integrated twice to yield actuator displacement. Each of the major components of the payload model are described in more detail below.

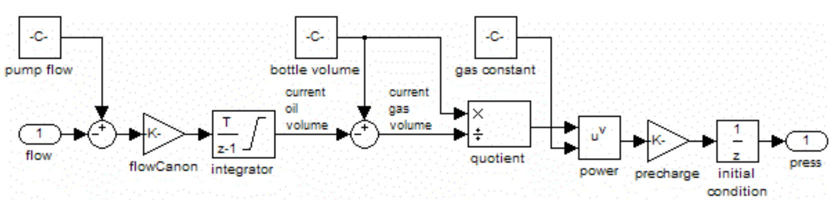

Figure 7: Accumulator model

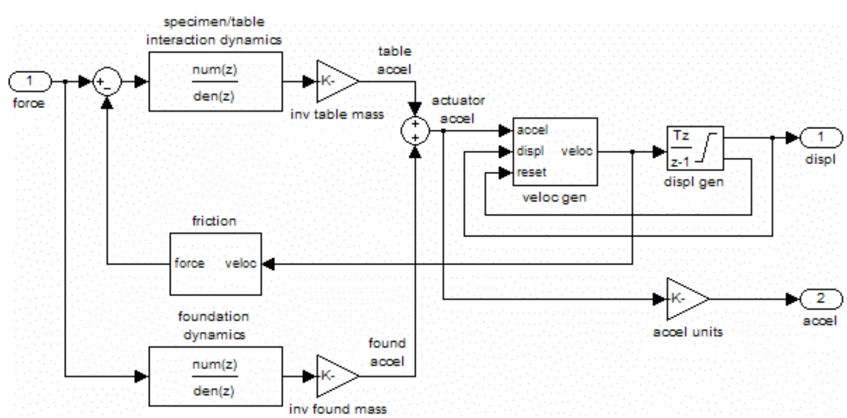

Figure 8: Payload model

Specimen/Table Interaction Dynamics

This block models the interaction of a specimen having a single dominant resonant mode with the dynamics of the table, as measured by the table accelerometer. Viewed from an effective mass viewpoint, the interaction between specimen and table occurs because the amount of mass seen by the actuator varies with frequency. At low frequencies the effective mass is the sum of table and specimen masses. As the frequency approaches the resonant frequency of the specimen ω_S, the effective mass increases to a maximum, then decreases as the resonant frequency of the table ω_T is approached. At frequencies higher than ω_T the specimen decouples from the table and the effective mass becomes that of the table alone. A Laplace transfer function that has this behavior consists of a pair of complex zeros and a pair of complex poles, written in inverted form below because the Simulink model requires an inverse mass:

$$\frac{s^2 X_T(s)}{F(s)} = \frac{s^2 + 2\zeta_S \omega_S s + \omega_S^2}{s^2 + 2\zeta_T \omega_T s + \omega_T^2} \cdot \frac{1}{M_T} \text{ where } \omega_T = \omega_S \sqrt{1 + \frac{M_S}{M_T}} \text{ and } \zeta_T = \zeta_S \sqrt{1 + \frac{M_S}{M_T}}$$

where

$s^2 X_T(s)$ = table acceleration, $F(s)$ = actuator force,

M_T = table mass, M_S = specimen modal mass,

ω_S = specimen modal frequency, and ζ_S = specimen modal damping.

Foundation Dynamics

This block models the interaction of the actuator with the foundation, modeled as a single dominant resonant mode, as measured by a hypothetical accelerometer mounted on the actuator base. Viewed from an effective mass viewpoint, at low frequencies the effective foundation mass seen by the actuator is infinite. As the frequency approaches the resonant frequency of the foundation ω_F, the effective mass decreases to a minimum. At frequencies higher than ω_F, the effective mass approaches the foundation mass. A Laplace transfer function that has this behavior is written in inverted form below because the Simulink model requires an inverse mass:

$$\frac{s^2 X_F(s)}{F(s)} = \frac{s^2}{s^2 + 2\zeta_F \omega_F s + \omega_F^2} \cdot \frac{1}{M_F}$$

where

$s^2 X_F(s)$ = foundation acceleration, $F(s)$ = actuator force,

M_F = foundation mass,

ω_F = foundation natural frequency, and ζ_F = foundation damping.

Friction Model

This block models friction using the continuous viscoplastic friction law described by Bondonet and Filiatrault (1997) as $F = \mu Z$, where μ is the coefficient of friction, Z is hysteretic dimensionless parameter that is the solution to $Y \frac{dZ}{dt} = (1 - Z^2(\beta + (1-\beta) sign(Z\dot{X}))) \frac{dX}{dt}$, Y is the equivalent yield displacement, β is a dimensionless constant, and X is displacement. The variation of coefficient of friction with velocity was assumed to not be a major factor in this system, so μ was made constant for simplicity. The internal details of the Simulink block that implements this friction law is shown in Figure 9.

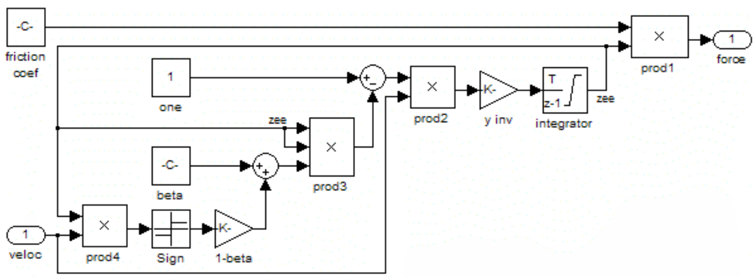

Figure 9: Friction model

Sensor Conditioner Filters
The sensor conditioners in the Realtime Controller have built-in frequency-selectable fourth-order elliptic lowpass digital filters. These are replicated exactly in the simulation model.

DETERMINING MODEL PARAMETERS

Model parameters are the numeric representations of the physical properties of the real system that are required as input to the simulation models. These parameters are both determined from product literature and from direct measurement of the system. Generally, once these parameters are determined for a particular shake table system, they would rarely if ever need to change (except for the specimen parameters). The model parameters fall into three categories:

1) Model parameters that are determined from product literature. These parameters are:

Hydraulic supply pressure	Actuator piston area
Hydraulic return pressure	Actuator useable stroke
Hydraulic pump flow	Actuator endcushion length
Servovalve overlap	Actuator piston leakage
Servovalve rated pressure	Accumulator volume
Servovalve rate flow	Accumulator precharge pressure
Sensor conditioner filter dynamics	Accumulator gas constant

2) Model parameters that are measured by exciting the table without the specimen. Because the specimen is not present, excitation can be done at high level and at length to obtain good parameter estimates. These parameters need only be estimated once because they do not change from test to test. These parameters are:

Servovalve spool dynamics	Table (+ actuator) mass
Servovalve nominal flow	Foundation mass
Oil column damping	Foundation natural frequency
Oil bulk modulus	Foundation damping
Friction force	
Friction yield displacement	

3) Specimen parameters determined either by analysis (e.g., by FEA), or by low level, short duration excitation. These parameters must be estimated on per test basis. These parameters are:

Specimen mass
Specimen natural frequency
Specimen damping

The following paragraphs describe the parameter estimation process and the results obtained.

Servovalve spool dynamics

The servovalve spool displacement control loop within the servovalve exhibits second-order lowpass dynamics. These dynamics are measured by exciting the table with a random program, recording valve driver command and valve spool displacement feedback, and using Matlab's System Identification Toolbox to fit a second-order AutoRegressive Moving Average (ARMA) dynamic model to the data. An ARMA model is a digital filter of the form

$$y_k = -a_1 y_{k-1} - a_2 y_{k-2} + b_0 x_k + b_1 x_{k-1} + b_2 x_{k-2}$$

whose current output y_k is computed from weighted past outputs $\{y_{k-1}, y_{k-2}\}$ and current and past inputs $\{x_k, x_{k-1}, x_{k-2}\}$. The System Identification Toolbox computes coefficients $\{a_1, a_2, b_0, b_1, b_2\}$ that results in the best fit between input and output in a least-squares sense. In the case of valve spool dynamics, the frequency response of the actual system compared with the ARMA model is shown in Figure 10.

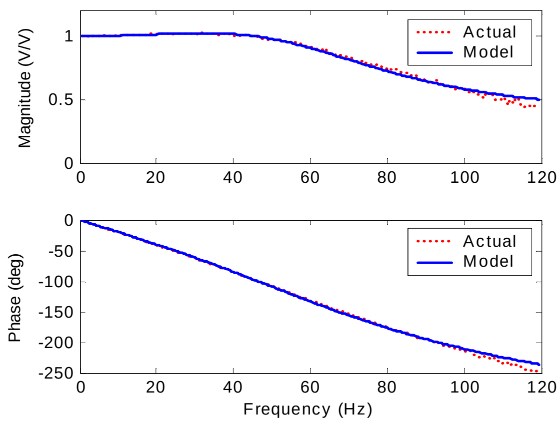

Figure 10: Valve spool frequency response

Nominal flow

Servo-valves exhibit a nonlinear flow-versus-valve opening characteristic whereby the flow at small valve opening is greater than that at full valve opening. Servo-valve product literature gives the flow at full valve opening, called "rated flow", which is measured at a "rated pressure" of 1000 psi. The flow at small valve opening, called "nominal flow", is required by the model but is not listed in the product literature and therefore must be measured. This is done by running a low frequency sine wave of amplitude sufficient to open the valve by 10% of full opening, measuring actuator displacement amplitude X and valve spool displacement amplitude X_{SPOOL}, and computing nominal flow gain Q_{NOM} as

$$Q_{NOM} = \frac{2\pi f X A}{X_{SPOOL}} \cdot \sqrt{\frac{P_{RATED}}{P_{SUPPLY}}}$$

Friction

The friction is measured by commanding the table with a very low frequency triangle wave in displacement control and recording several cycles displacement and force feedbacks. By plotting force versus displacement as in Figure 11, the friction model parameters are obtained by visual inspection.

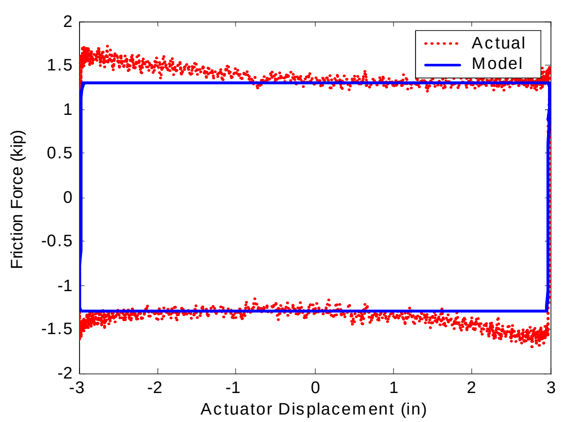

Figure 11: Friction force versus displacement

Table (+Actuator) Mass

The table mass (including the mass of the actuator and actuator swivels) is obtained by exciting the table with a random program and recording the table acceleration feedback and the actuator force feedback. The transfer function between these feedbacks is the table mass, which should be constant for all frequencies. This is

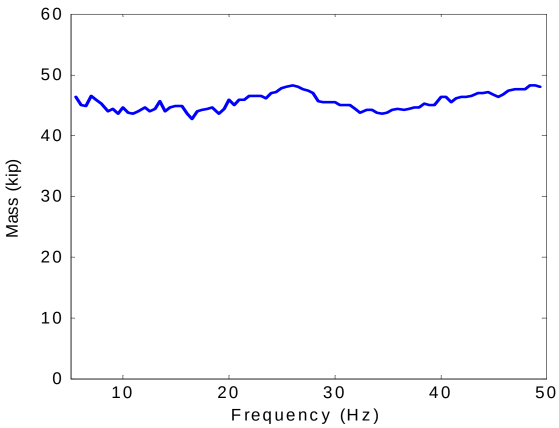

Figure 12: Table mass versus frequency

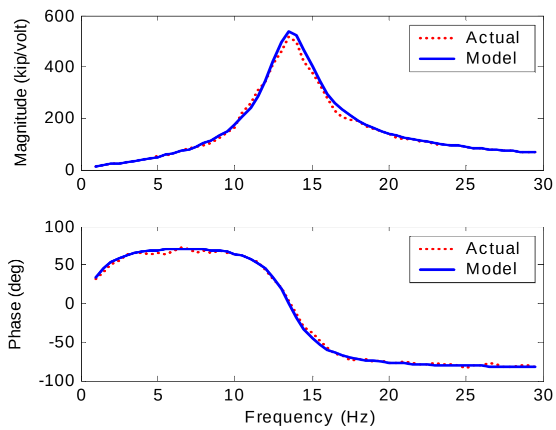

Figure 13: Oil column frequency response.

shown in Figure 12. The estimated table mass is the average of the values in the plot.

Oil Column Damping and Oil Bulk Modulus

The actuator oil spring and table mass combination exhibits second-order dynamics, which are measured by exciting the table with a random program, recording valve spool displacement feedback and actuator force feedback, and using Matlab's System Identification Toolbox to fit an ARMA model to the data. The frequency response of the actual system compared with the ARMA model is shown in Figure 13. From this model, oil column natural frequency f_{OIL} and damping ζ_{OIL} are extracted. Using f_{OIL}, table mass M_T, actuator useable stroke L, actuator endcushion length L_E, and actuator area A, oil bulk modulus β_{OIL} is computed as

$$\beta_{OIL} = (2\pi f_{OIL})^2 \frac{M_T}{2A}\left(\frac{L}{2} + L_E\right)$$

It should be noted that when β_{OIL} is computed in this fashion, it represents not only the stiffness of the oil but also other elements in series with the load train, such as actuator swivels, and additional oil volume in the servovalve manifold.

Foundation Dynamics

The foundation exhibits second-order dynamics, which are measured by exciting the table with a random program and recording the actuator force feedback and the acceleration at the base of the actuator. Using Matlab's System Identification Toolbox to fit a second-order ARMA model to the data, effective inverse foundation mass versus frequency is obtained as shown Figure 14. From this model, foundation natural frequency and damping are extracted. It was stated previously that the effective foundation mass approaches M_F at high frequencies, so the foundation mass is obtained by evaluating the inverse of magnitude response of the ARMA model at the Nyquist frequency.

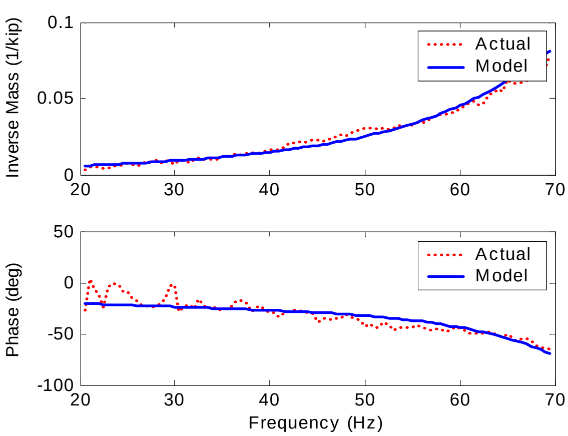

Figure 14: Effective inverse foundation mass versus frequency

SPECIMEN PARAMETER DETERMINATIONS

Specimen Dynamics

Theoretically determining specimen frequency is possible but no methods exist to directly compute damping. It is important to correctly determine both factors since a slight difference in frequency or damping can produce significant differences in the plant model.

This issue is non-problematic since the free vibration procedure discussed in the specimen characterization section can be used to directly measure specimen frequency and damping.

Another method of measuring specimen dynamics is by exciting the table with a low level, short duration random program and recording table acceleration and actuator force feedbacks. The resulting specimen/table interaction exhibits second-order dynamics, and by using Matlab's System Identification Toolbox to fit a second-order ARMA model to the data, effective specimen mass versus frequency can be obtained (Figure 15). From this model, measured specimen natural frequency and damping are extracted. It was stated previously that the effective mass approaches the sum of table and specimen mass at low frequencies. The specimen mass can be obtained by evaluating the magnitude response of the ARMA model at low frequency and subtracting the table mass estimated previously.

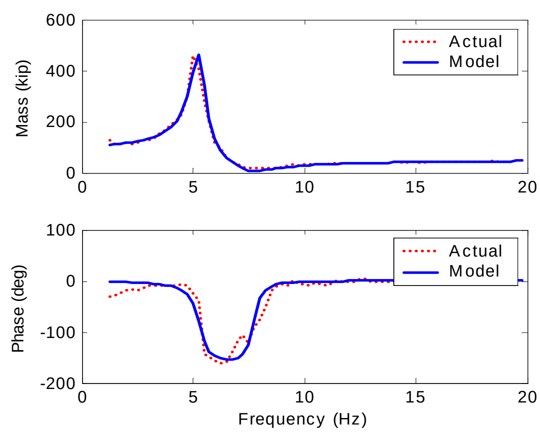

Figure 15: Effective specimen mass versus frequency.

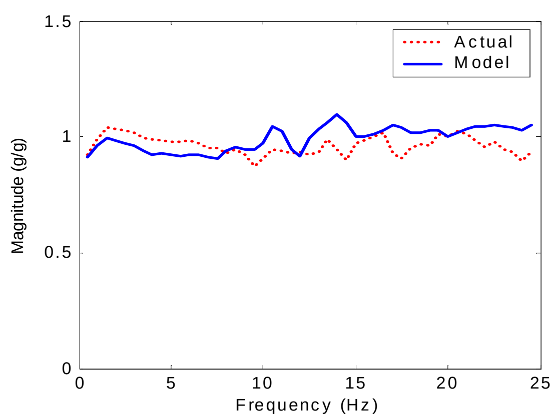

Figure 16: Acceleration frequency response for the bare table

BARE TABLE SIMULATION VERSUS ACTUAL

To validate the accuracy of the simulation model for the case of a bare table, the simulated system was tuned using the simulation model. The resulting controller settings were directly applied to the actual system. Then the acceleration frequency response of the actual system was measured and compared with that of the simulation (Figure 16). The response of simulation model matches that of the actual system within +/- 10% (+-0.9db) over a wide frequency range. Because acceleration response is very sensitive to many influences, this level of matching is considered to be quite good.

Next, the 1940 Imperial Valley Earthquake (El Centro, SMEA 1971) was run on both the simulation model and the actual system using identical controller settings. The acceleration response time histories are compared for a section of time, shown in Figure 17. The matching between the simulation and actual test is exceptional.

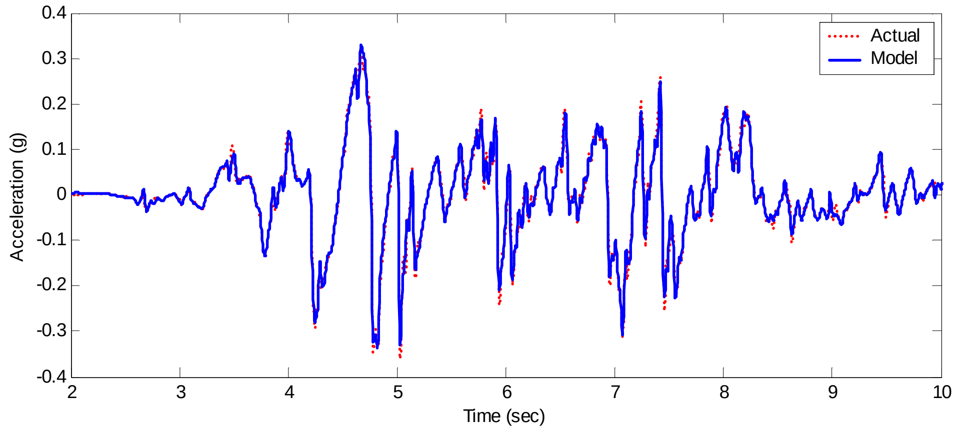

Figure 17: El Centro acceleration response time history for the bare table.

RESONANT MODEL SIMULATION VERSUS ACTUAL

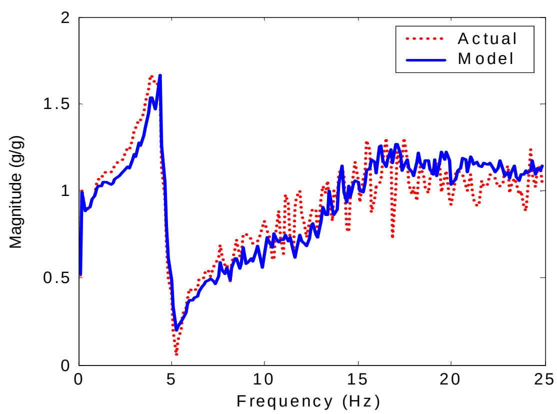

Figure 18: Acceleration frequency response for the table with a resonant specimen and bare table tuning

The more difficult task is to validate the accuracy of the model for the case of a table with a resonant specimen. Identical bare table controller settings were used for the simulation with the resonant specimen model and the actual system with the resonant specimen attached. The acceleration frequency response of the actual system was measured and compared with that of the simulation, as shown in Figure 18. The simulation model was able to capture the resonance and anti-resonant effects with exceptional accuracy.

SIMULATED CONTROLLER TUNING APPLIED TO REAL SYSTEM WITH RESONANT SPECIMEN

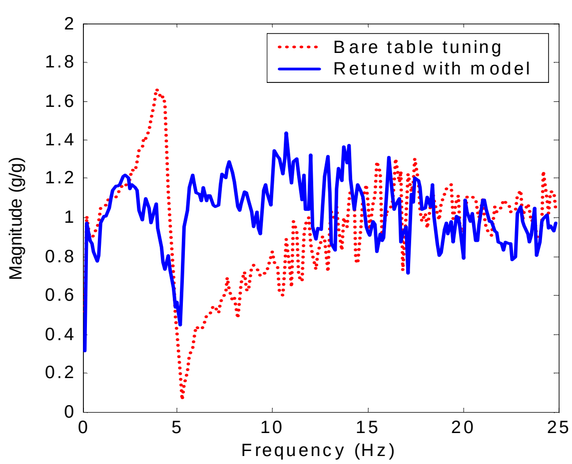

Figure 19: Acceleration frequency response for the actual table with a resonant specimen

The ultimate goal of this research was to determine whether the controller tuning obtained by tuning the simulation model could be applied to the real system with a performance benefit. The actual system's acceleration frequency response, reproduced in Figure 19 from Figure 18, has significant resonant and anti-resonant peaks when bare table tuning is used. This tuning would provide significant difficulties in matching commands and feedbacks in the subsequent testing of the specimen. After tuning the simulation model with the Simulated Controller and the resonant specimen model, these controller settings were transferred to the actual controller. The acceleration frequency

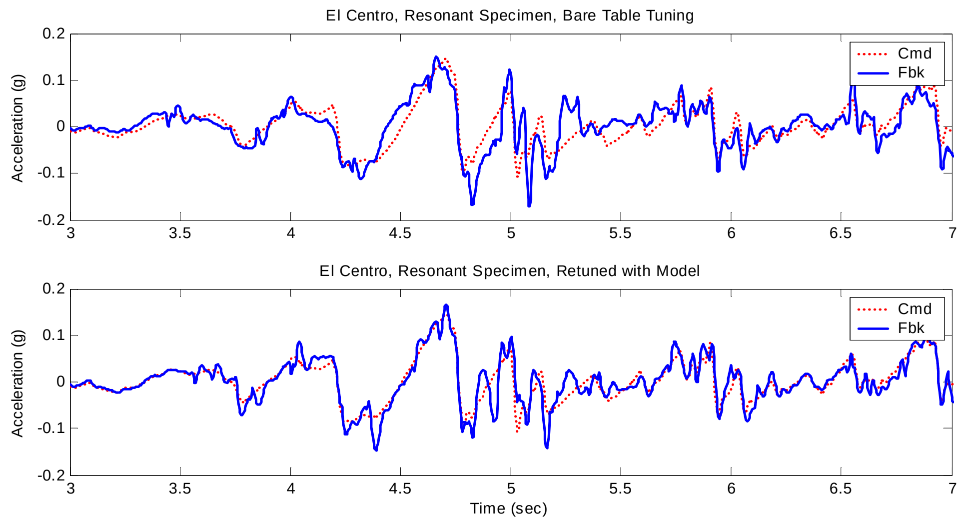

Figure 20: El Centro acceleration response time history for the actual table with a resonant specimen

response of the actual system was measured again using the simulation settings. In Figure 19, a significant improvement can be seen. The response obtained using tuning done with the simulation model, while not perfect, is considerably flatter than using the original bare table tuning.

Next, El Centro was run on the real system using both the bare table settings and the tuned simulation model settings, and acceleration response time histories were compared with the desired command (Figure 20). The time history was scaled to 30% of full scale to prevent damage to the resonant specimen. The improvement in matching is apparent. Figure 21 compares the tracking error spectral density for each case. The improvement in error is significant from 0 to 6 Hz.

CONCLUSIONS

High force-feedback resonant specimens tested on a high performance seismic shake table system present unique control problems. Unless the shake table system can achieve near unity gain between command and feedback, poor performance and test results will occur. Achieving unity gain is problematic since it generally requires exciting the test specimen before the "real" testing can occur. The purpose of developing a numerical simulation model of a seismic shake table system was two-fold; to improve the shake table performance by determining the proper controller settings without exciting the real specimen, and secondly to accurately predict system performance prior to running a test.

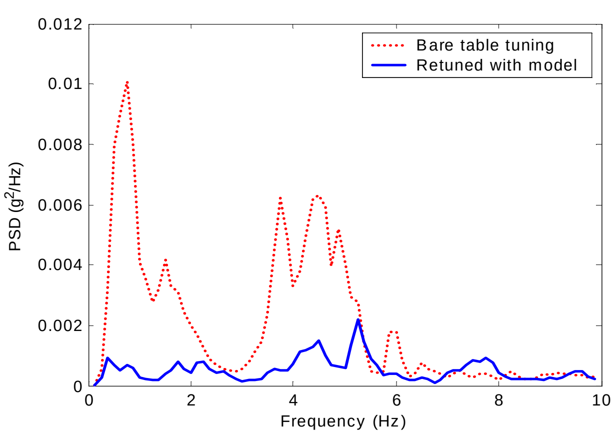

Figure 21: El Centro acceleration error PSDs for the actual table with a resonant specimen

The simulation model requires determining the shake table system parameters using only a bare table and a set of identifying procedures. These procedures were shown to be very effective at accurately identifying the system parameters. The numerical models of the system and actuators were also shown to represent the real system exceptionally well. Once the specimen parameters were determined and modeled in the simulation, it was shown that the controller tuning parameters obtained by tuning the simulation model could be applied to the real system and gain significant improvements in performance over standard operating procedures.

By using the real controller software, significant modeling effort was eliminated. The operator also maintains the illusion of the real system while in simulation mode, and thus does not have to learn “new” controller software. The operator is also free to learn new control techniques without risk of damage to the system or specimen.

REFERENCES

Laplace, Patrick N., David H. Sanders, M Saiid Saiidi, 2001, “Experimental Study and Analysis of Retrofitted Flexure and Shear Dominated Circular Reinforced Concrete Bridge Columns Subjected to Shake Table Excitation”, Center for Civil Engineering Earthquake Research, Report No. CCEER-01-06, University of Nevada, Reno.

Laplace, Patrick N., David H. Sanders, M Saiid Saiidi, Bruce M. Douglas, 1999, “Shake Table Testing of Flexure Dominated Reinforced Concrete Bridge Columns”, Center for Civil Engineering Earthquake Research, Report No. CCEER-99-13, University of Nevada, Reno.

Thoen, Bradford K., 469D Seismic Digital Control Software, MTS Systems Corporation, 2004.

Mathworks Matlab Version 6.5 Release 13, www.mathworks.com, 2004.

Bondonet & Filiatrault, "Frictional Response of PTFE Sliding Bearings at High Frequencies", Journal of Bridge Engineering, Nov. 1997, pp. 139-148.

Strong Motion Earthquake Accelerograms, Digitized and Plotted Data”, Volume II, Part A, EERL 71- 50, Earthquake Engineering Research Laboratory, California Institute of Technology, Pasadena, California, 1971.