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Experimental identification of the six DOF C.G.S., Algeria, shaking table system

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Abstract. Servohydraulic shaking tables are being increasingly used in the field of earthquake engineering. They play a critical role in the advancement of the research state and remain one of the valuable tools for seismic testing. Recently, the National Earthquake Engineering Research Center, CGS, has acquired a 6.1m x 6.1 m shaking table system which has a six degree-of-freedom testing capability. The maximum specimen mass that can be tested on the shaking table is 60 t. This facility is designed specially for testing a complete civil engineering structures, substructures and structural elements up to collapse or ultimate limit states. It can also be used for qualification testing of industrial equipments. The current paper presents the main findings of the experimental shake-down characterization testing of the CGS shaking table. The test program carried out in this study included random white noise and harmonic tests. These tests were performed along each of the six degrees of freedom, three translations and three rotations. This investigation provides fundamental parameters that are required and essential while elaborating a realistic model of the CGS shaking table. Also presented in this paper, is the numerical model of the shaking table that was established and validated.

Keywords: servohydralic shaking table; parameter estimation; identification; random and cyclic testing

1. Introduction

In civil engineering, understanding of structural dynamics is an important issue in the design and retrofit of structures to withstand severe dynamic loading from earthquakes. The devastating effects of the 1994 Northridge, 1995 Kobe and more specifically for Algeria the 1980 El Asnam and 2003 Boumerdes earthquakes, have highlighted the shortcomings of modern structural designs.

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In order to expand our understanding of dynamic loads and mitigating their undesirable effects, and consequently improve the future seismic design codes, recourse to the use of experimental methods is now being increasingly common. Among these methods, shaking table remains one of the most widely used experimental tools (Severn 2011). In fact, shaking table was used to investigate the seismic behavior of a wide range of civil engineering structures, such as individual elements, substructures, and complete structures including a variety of materials such as reinforced and pre-stressed concrete, wood, steel and masonry. Because it applies forces to a test structure, in the same way as the ground applies forces to a real structure, during an earthquake, shaking table still, by far, the most valuable tools for seismic testing. However, it should be noted that, in order to conduct a required research work, in an efficient manner, a thorough understanding of the shaking table dynamics and its sub-systems is necessary. Hence, it is critical to conduct an experimental system identification of the shaking table, in order to gain insight into the true system performance.

Shaking table identification has been investigated to date by several researchers. In particular, considerable amount of work can be found in Thoen and Laplace (2004) study's, in which they presented a comprehensive numerical model of a medium size shake table for off-line tuning purposes. In their study, an experimental identification of the shaking table and its sub-systems namely servovalve spool dynamics, nominal flow, table effective mass, oil column and table foundation dynamics, was performed. Subsequently, a realistic model of the shaking table using Simulink® was obtained. User Datagram Protocol (UDP), which is the communication mechanism, was used to link the real-time shaking table Controller GUI (MTS 469D) with the Simulink model. The originality of this study is that it retains the real-time controller software in the simulation, which provides the operator the same familiarities during the simulation as with the real system controller. Airouche *et al.* (2008), developed two Simulink® models of the EUCENTRE TREESLab high performance uni-axial shake table, one uses a simplified linearized servovalve actuator system while the second uses a realistic detailed nonlinear system. These numerical models include the inherent dynamic characteristics of the various components of the shake table system and their interaction, which were determined experimentally, trough a comprehensive set of random and periodic tests conducted on the EUCENTRE TREESLab shake table. Ozcelik *et al.* (2008a), developed a comprehensive mechanics- based virtual model for the large NEES-UCSD shake table under bare and loaded table conditions. The shake table model developed in this study includes a virtual replica of the actual controller, four servovalves models, two single-ended actuators, two effective accumulators, a two- dimensional mechanical subsystem model, and linear/nonlinear specimens modeled using the finite element analysis framework OpenSees. Additionally, signal tracking performance under various test conditions was performed on the NEES-UCSD table. Plummer (2008), described the modeling of a 5 m x 5 m six Degree-Of-Freedom (DOF) shaking table and the subsequent computer simulation of the system's dynamic characteristics. The simulation model is implemented using Simulink® and its multi-body mechanical simulation tool SimMechanics®. It includes a high order dynamics such as valve response and structural effects as well as significant nonlinearities associated with the hydraulic and mechanical components, such as, spool slew rate and saturation limits, valve overlap, manifold and valve body pressure losses, friction, and geometric nonlinearities. The simulated response was shown to closely match the measured response of the table. Other researchers also carried out studies on the shaking table modeling and control (Rinawi *et al.* 1991, Clark 1992, Conte and Trombetti 2000, Williams *et al.* 2001, Shortreed *et al.* 2001, Crewe and Severn 2001, Trombetti and Conte 2002, Twitchell *et al.* 2003, Plummer 2010, Shen *et al.* 2011, Gu *et al.* 2011, Ceresa *et al*. 2012).

Modeling of the shaking table requires the determination of many parameters. Some of them, mainly concerning the geometric and physical proprieties of the various system components, can be obtained from the manufacturer or through direct measurement; others need to be determined experimentally by conducting bare table tests. These parameters concern nominal flow, effective bulk modulus, servovalve spool dynamics, rigid table effective mass, friction coefficient, and foundation dynamics. The present paper presents the results of the experimental characterization of the National Earthquake Engineering Research Center, CGS, shaking table system. A comprehensive set of shake-down tests (over 100) has been conducted on the shaking table, in the six degrees of freedom: longitudinal, transversal, vertical, roll, pitch and yaw, in order to characterize its key parameters. The testing program included harmonic and random excitations in displacement and acceleration control modes. The ultimate goal of this study is to identify the different parameters that are necessary for the analytical modeling of the CGS shaking table system. The simulation model that is introduced in this paper will be used as Decision Support Tools which can offer a wide range of functionality such as feasibility and suitability of a given test arrangement by predicting the specific behaviors of the system prior to testing.

2. CGS shaking table system characteristics

The CGS laboratory is equipped with an MTS Systems Corporation servo-hydraulic shake table shown in Fig. 1. It consists of 6.1 m by 6.1 m steel platform driven by twelve servohydraulic actuators: four longitudinal, four lateral, and four vertical. The table has six active degrees of freedom: longitudinal, lateral, vertical, roll, pitch, and yaw (Airouche *et al*. 2010).

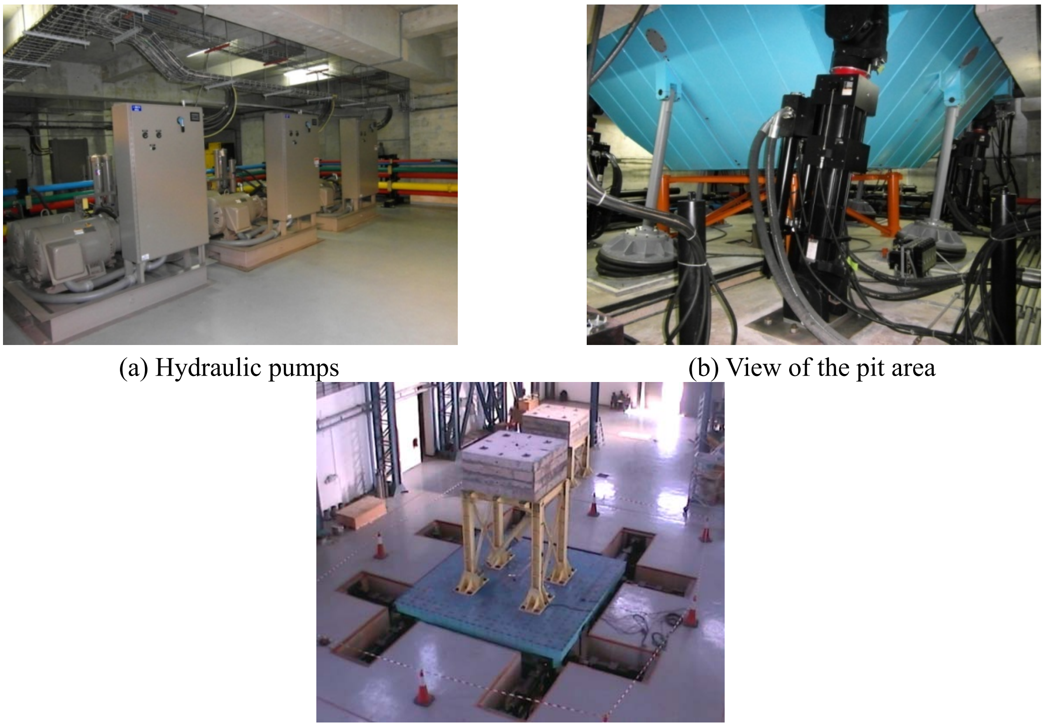
The shaking table system is capable of simulating earthquake events and other ground vibrations with maximum displacements of ±250 mm and ±150 mm in the horizontal directions. The corresponding value of displacement in the vertical direction is ±100 mm. Peak velocity of 1.1 m/s for each of the two horizontal directions and 0.8m/s in the vertical direction can be achieved. Accelerations of ±1.0 g for horizontal directions and ±0.8 g for vertical direction are possible with maximum test specimens of 60 tons. The maximum over turning moment is about 180 ton-m. The frequency range is 0-50 Hz.

The shaking table is supported by the reaction mass foundation which is designed to let the testing induced vibrations to be transmitted to the ground well below the acceptable level. It is constructed of reinforced concrete and has a total mass of 6400 tons which gives a mass ratio of 60 times the specimen-table mass.

During the test, the dead weight of the table and the specimen under the test is balanced through the use of the static support system, which consists of four separate pneumatic supports located underneath the table, with a total capacity of 140 tons.

The hydraulic power supply that supplies the shake table consists of five high pressure pumps that can deliver a total flow of 3600 liters per minute at a pressure of 21 MPa and 360 liters from accumulators for peak flow demands. Fig. 2 shows a global view of the shaking table and its components.

The table is controlled by advanced Digital Controller MTS system 469D. It provides for the shake table a high-level fixed control techniques such as Degree of Freedom Control, Differential Pressure Stabilization, Force Balance Compensation, Three-Variable Control (TVC: displacement, velocity, and acceleration), built-in filtering and adaptive compensation techniques for high fidelity and faithful reproduction of the desired table motions (Airouche *et al.* 2008). It also provides data acquisition, processing, and storage functions for data acquired during testing.



(a) Hydraulic pumps (b) View of the pit area

(c) Table with specimen

Fig. 1 View of CGS shaking table and View of CGS shaking table

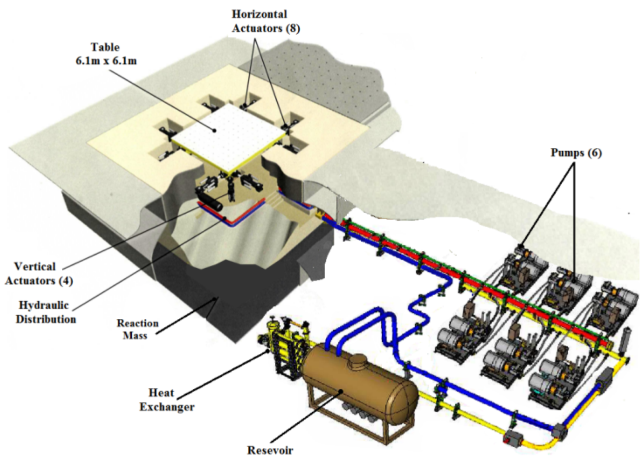


Fig. 2 Shaking table components

3. Sensors and data acquisition system

Model parameters that are measured in this study were determined experimentally from the shaking table feedbacks by conducting bare table tests. The instrumentation used in the tests consist of the permanent sensors used for controlling the shaking table. For each of the three translational degrees of freedom, the table displacement (or table force) feedback was obtained from the average of displacement feedbacks (or the algebraic summation of the force feedbacks) of the four actuators involved in the considered axis. For the three rotational degrees of freedom the rotation and moment were measured indirectly. For example, considering the illustration in Fig. 3, pitch is affected by actuators Z1 and Z2 in one direction and by Z3 and Z4 in the other direction. The pitch rotation feedback is the difference between the average displacement feedbacks of these two pairs of actuators divided by the distance between them.

Table acceleration feedbacks were measured by 11 Setra-Model 141A accelerometers with a range of ±8 g and a flat frequency response from DC to 300 Hz. As shown in Fig. 4, three accelerometers measure the longitudinal table acceleration feedback, three other accelerometers measure the lateral table acceleration feedback and the remaining five accelerometers measure the vertical table acceleration feedback. The angular accelerations corresponding to the rotational degrees of freedom were calculated from a combination of the accelerometer records.

The table velocity is another quantity measured indirectly. For a given degree of freedom, to obtain a wideband estimate of velocity, the differentiated table displacement feedback signal is combined with the integrated acceleration feedback signal via a crossover filter (Thoen 2004, Ozcelik *et al.* 2008b).

The three stage servovalve contains a spool LVDT which provides a signal that is proportional to the spool position which is needed for the servovalve dynamic investigations.

Data were acquired by the built-in data acquisition DAQ system, Data Recorder part of MTS 469D digital controller. The selected channels to be recorded during the tests were stored in user selected file with sampling rate set to 1024 Hz.

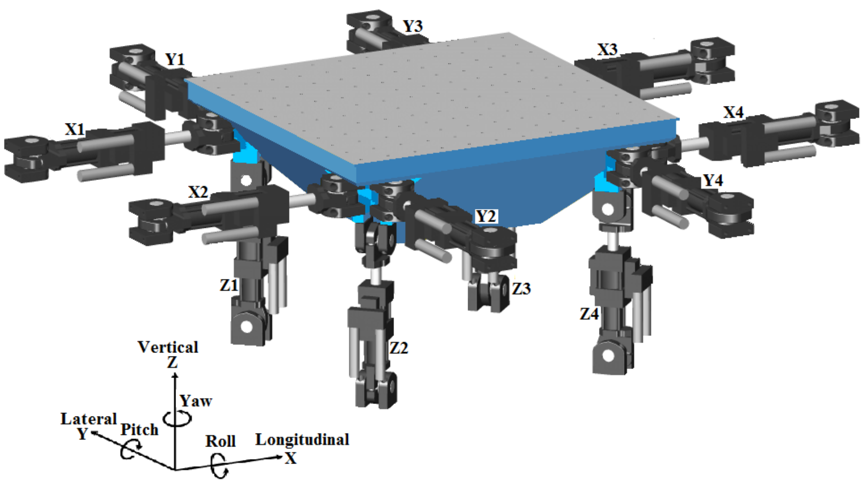


Fig. 3 Table actuator positions

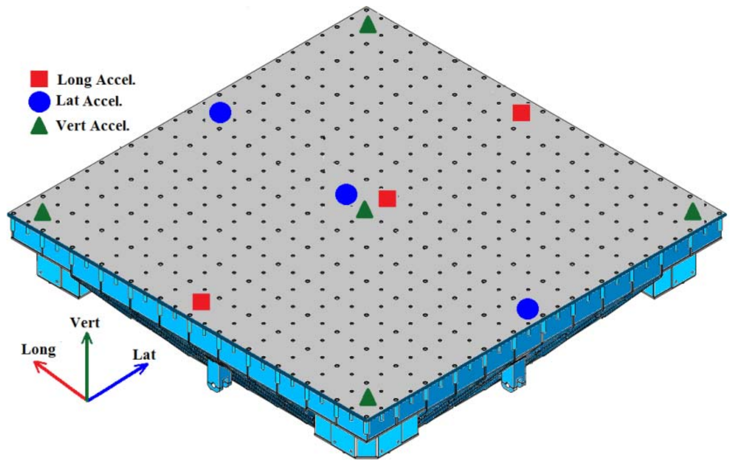


Fig. 4 Table accelerometer locations

4. Servovalve dynamics

The heart of a servohydraulic system is the servovalve, which is regarded as the most important hydraulic component, in terms of overall system performance (Kusner *et al.* 1992). It allows an electrical signal to control the rate and direction of hydraulic fluid flow from the hydraulic power supply to and from a hydraulic actuator. For the shaking table system, servovalve represents the final control element.

The large hydraulic flow needed to attain the system performance characteristics of the shaking table require the use of powerful multi-stage servovalves, which must have the critical requirements of high-flow, quick-response and low distortion. For the CGS shaking table system, the four vertical, the four lateral, and the four longitudinal actuators are equipped with an MTS 256.40A-05 three stage servovalve. It consists of a high-flow, four-way spool valve and a smaller two-stage pilot valve and has a full flow capacity of 1500 liter per minute, according to the manufacturer's specifications.

In order to gain insight into servo valve operation, experimental tests were conducted and servovalves responses in term of performance curves were measured. Indeed, for each of the three orthogonal translational degrees of freedom, the table was excited with a wideband random test program in a displacement control mode (RMS = 0.01 m, Frequency range = [0.5 -125] Hz). The conditioned commands and the third stage spool positions of the servovalves mounted on the actuators were recorded. The frequency response plots of the servovalves are shown in the Figs. 5(a) - 5(c) corresponding for longitudinal, lateral and vertical degrees of freedom respectively.

It is obvious from the Figures that the performance curves are similar, since all the actuators are equipped with the same type of servovalve, as mentioned before. The performance curves of the servovalves indicated that their frequency responses (magnitude) are constant for frequencies below 15 Hz, and rolled off as the command frequency increased.

Mathematical models of an electrohydraulic servovalve can be constructed at various levels of detail depending on the purpose of the model. However, it is often convenient to represent servovalve dynamics by a simplified equivalent linear transfer function. For example, accurate model of the servovalve dynamics can be represented by a first or second order models (Airouche *et al.* 2008, Zhao *et al.* 2005). Laplace transfer functions of these models are given below, for the first order and the second order models, respectively

$$H(s) = \frac{k_s}{\tau s + 1} \tag{4.1}$$

$$H(s) = k_s \frac{\omega_v^2}{s^2 + 2\xi\omega_v + \omega_v^2} \tag{4.2}$$

Where s is the Laplace transform variable, k_s is the valve gain and τ is the response delay, while ξ and ω_v represent the damping ratio and natural frequency of the servovalve, respectively.

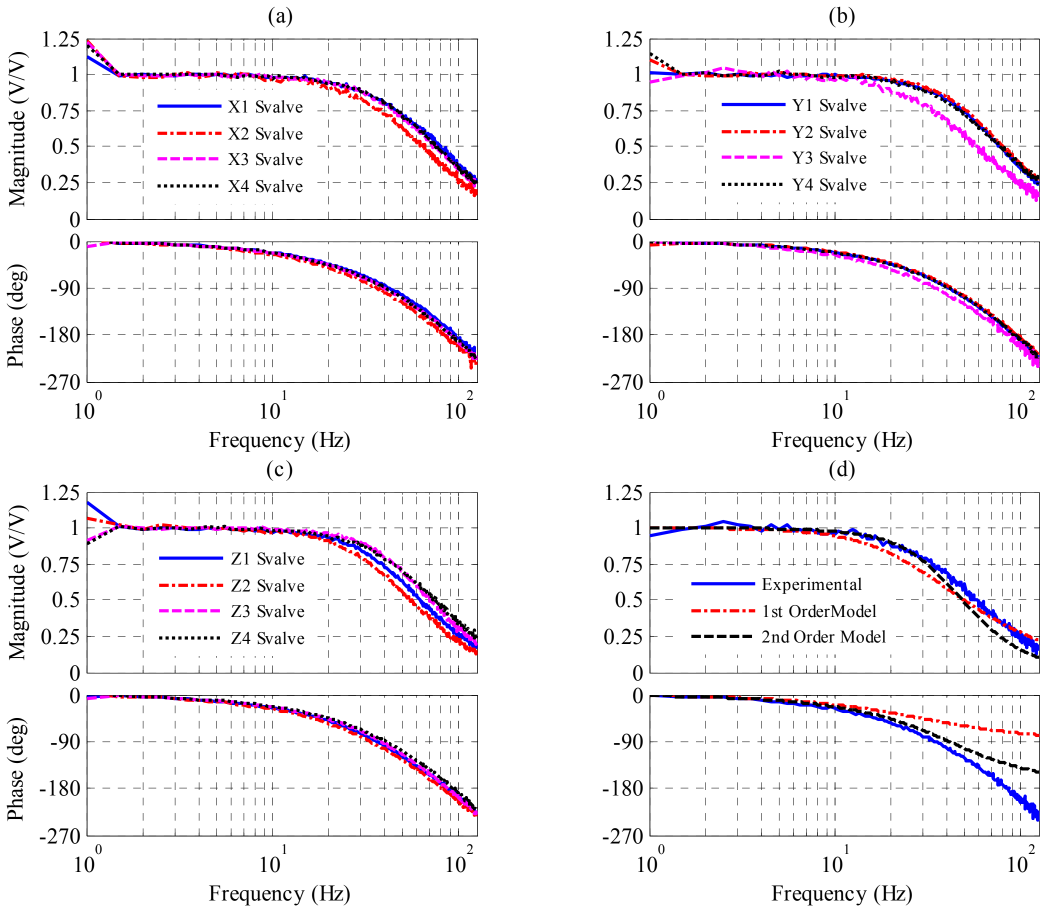


Fig. 5 Measured servovalve dynamics : (a) longitudinal, (b) lateral, (c) vertical actuator and (d) experimental vs first and second order models

The parameters in Eq. (4.1) and Eq. (4.2), were extracted from the experimental frequency response function of the servovalves using the procedure described in details by Thayer (Thayer 1965) as follow:

- The valve gain (K_s), which corresponds to the asymptotical line of the magnitude response, was found equal to 1.0.
- The time delay (τ) for the first order model was found to be 0.0056 sec.
- An equivalent natural frequency (ω_v) of 41.5 Hz and damping (ξ) of 75% were found.

The frequency responses of the first and the second order models of the servovalve are compared with the average of the experimental performance curves in Fig. 5(d). It can be seen that, the magnitude responses of both models match closely the magnitude of the experimental results. The phase response of the second order model fits perfectly the experimental phase response over the wideband frequency range (0-50 Hz), while the first order model matches the experimental phase response up to 20 Hz with reasonable accuracy.

Table 1 Test program and oil column frequencies identification results

Test			Oil- Column (Hz)
DOF	RMS	Frequency (Hz)	
Longitudinal	0.10 g	[0.5 50]	16.40
Lateral	0.1 g	[0.5 50]	17
Vertical	0.1 g	[0.5 50]	25
Roll	20 deg/sec²	[0.5 50]	29
Pitch	20 deg/sec²	[0.5 50]	27.50
Yaw	20 deg/sec²	[0.5 50]	20

5. Oil column identification

The oil column resonance frequencies represent the predominant parameters affecting the dynamics of the shaking table system and typically lie well within the operating frequency range of the system. They also play a role in system fidelity by affecting the signal tracking performances of the shaking table (Luco *et al.* 2010). By knowing the exact values of these frequencies, a more accurate interpretation of test results can be performed by separating the true model behavior from the dynamics of the shaking table. In addition, a good tuning of the table can be achieved by suggesting the correct gain adjustment factors which yield to the improvement of the system response. Hence, it is critical to determine experimentally these oil column resonant frequencies in all active degrees of freedom, for an accurate characterization of the shaking table performance.

To do so, for each degree of freedom the table was excited with a random acceleration signal control mode as illustrated in Table 1. The servovalve spool positions and the force feedbacks of the actuators involved in the considered degree of freedom, were then recorded. The oil column natural frequency corresponds to the peak of the magnitude of the transfer function between the servovalve spool position and the force feedback.

The transfer functions for the longitudinal, lateral, vertical, roll, pitch, and yaw degrees of freedom are shown in Figs. 6(a) - 6(f) respectively.

The clear peak in the magnitude transfer functions which correspond to oil column frequencies are well seen in the figures. The phase transfer functions have an inversion of phase exactly at the frequencies corresponding to those of the oil column. The oil column frequencies extracted from these figures are listed in the last column of Table 1. It should be noted that, the evaluated oil column frequencies were obtained with bare table case and these values will decrease while performing a test with a specimen.

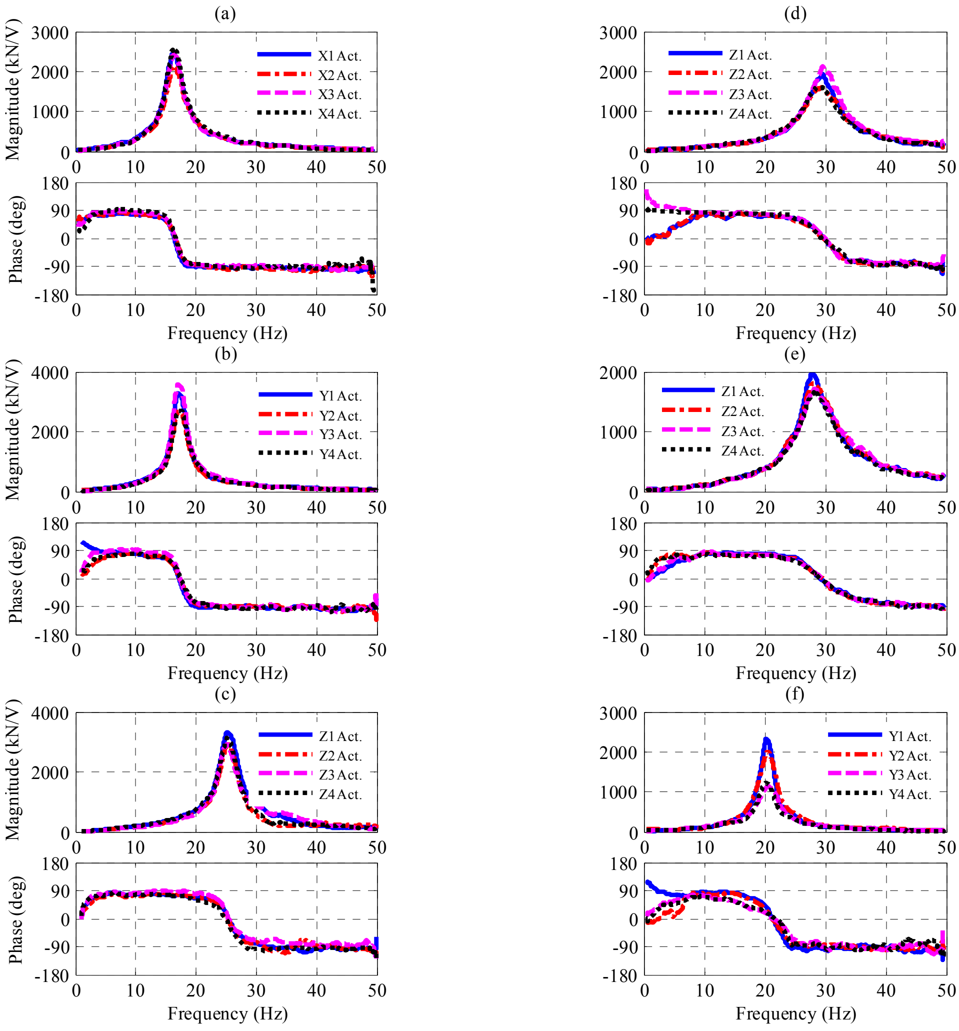


Fig. 6 Table transfer functions for the (a) longitudinal, (b) lateral, (c) vertical, (d) roll, (e) pitch and (f) yaw degrees of freedom

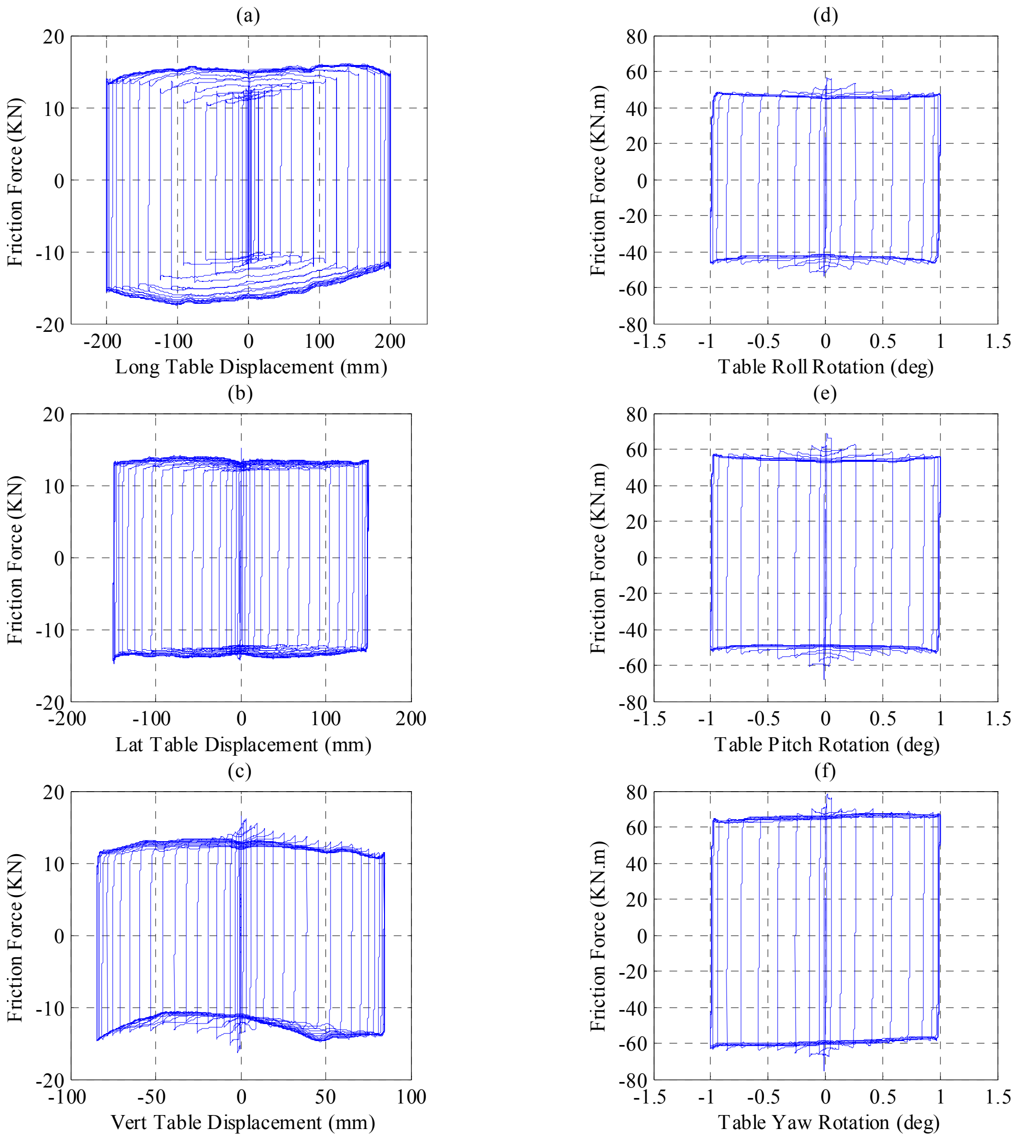


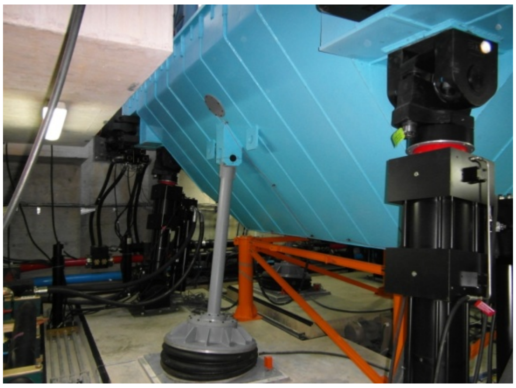
Fig. 7 Total friction forces vs table displacement for the (a) longitudinal, (b) lateral, (c) vertical, (d) roll, (e) pitch and (f) yaw degrees of freedom

6. Estimation of the effective total friction forces

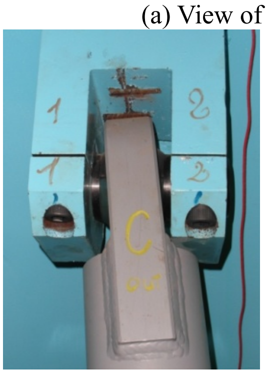
Minimizing the friction forces in actuators is a challenging topic where much analysis has to be done in order to get the optimum overall system performance. In the shaking table system, friction can arise from a number of sources (Kusner *et al.* 1992). Seals on the actuators, rod bearings, swivels, static supports and mechanical linkages constitute the predominant contributors.

Experimental investigation of friction was conducted on the CGS shaking table by commanding the table with a very low tapered sine wave with a frequency of 0.05 Hz and a maximum amplitude of 200 mm for longitudinal, 150 mm for lateral, 85 mm for vertical and 1 deg for the three rotational degrees of freedom. Several cycles of displacement and force feedbacks were recorded. Since the acceleration was very low, negligible inertial force should be involved and the overall forces measured should be purely damping and frictional dependent. The force-displacement hysteresis loops of the shaking table are plotted in Fig. 7. The rectangular hysteretic behavior is typical of Coulomb-type friction. It can be seen, from the Figs. 7(b) - 7(f), that for a given displacement/rotation amplitude the area of the hysteresis loops is constant except for the longitudinal degree of freedom, see Fig. 7(a), where a slight increasing of the frictional force is observed. The main cause is attributed to the type of pin set at the top of the static supports which is not symmetric in the two horizontal, longitudinal and lateral, directions, see Figs. 8(a) and 8(b). Another interesting behavior between vertical friction force and vertical table displacement (Fig. 7(c)) was observed. It is mainly due to the contribution of the static supports to the vertical friction forces. In fact, this behavior is typical of compression and suction effects of the compressed air that is contained in the pneumatic chamber at the base of the statistic supports see Fig. 8(c).

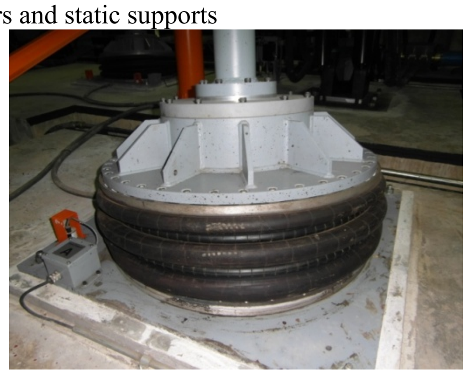
It is worth to mention here that, frictional forces account approximately for only 1.25% of the maximum system capability, and they are independent of the testing program level. We can conclude from this result that, the frictional forces have negligible effect on the performance of the CGS shaking table.



(a) View of the vertical actuators and static supports



(b) Detail of the static support-platen connection



(c) Air chamber

Fig. 8 Top and bottom connections of the static support

Table 2 Test program and estimated effective masses results

Degree Of Freedom (DOF)	Test			
	R1	R2	Frequency (Hz)	Effective Mass
	RMS	RMS		
Longitudinal	0.10 g	0.30 g	[0.5 50]	48.85 t
Lateral	0.10 g	0.30 g	[0.5 50]	49.36 t
Vertical	0.10 g	0.30 g	[0.5 50]	52.50 t
Roll	10 deg/sec²	20 deg/sec²	[0.5 50]	208 t.m²
Pitch	10 deg/sec²	20 deg/sec²	[0.5 50]	217 t.m²
Yaw	10 deg/sec²	20 deg/sec²	[0.5 50]	354 t.m²

7. Effective mass estimation

Rigid-body inertia properties (effective masses and moments of inertia) of simple structures can be easily estimated by using analytical approaches. For a complex structure however, constructed from several components and composed of different materials, such as shaking table systems, experimental approaches become necessary for precise evaluation of these inertia properties. In the case of rigid structures which have frequencies of the elastic modes high enough compared to the frequencies of the rigid body modes, there is a region in the frequency response function, where the response is dominated by the inertia properties. It is then worth to carry out a modal test on the system to generate the rigid body properties and the rigid body modes that can be used to evaluate the inertia properties. It is in the design requirement of the shaking table that, the platform must be sufficiently rigid and does not interfere during a test, so that it transmits the input motion to the test structure with a small modification as much as possible. In other word, the table platen would have a vibrational first mode well above the desired operating frequency range of the system. Thus, for the six degrees of freedom shaking table, the 6 x 6 mass matrix can be derived relating the small perturbation of the table forces to table accelerations. For each degree of freedom the table effective mass (including the mass of the moving part of the actuators, swivels, hoses, etc..) is obtained by exciting the table with a random acceleration and recording the table acceleration and force feedbacks. The transfer function between these feedbacks, acceleration as input and force as output, gives the table mass, which should be constant for a certain frequency range.

The tests were carried out with two levels of accelerations of 0.1 g and 0.3 g for translations; 10 deg/sec² and 20 deg/sec² for rotations with frequency ranging from 0.5 to 50 Hz. The test results are shown in Figs. 9(a) - 9(f). The estimated table effective mass, in the considered DOF, is the average of the values in the plots. Table 2 summarizes the average estimated effective mass.

Theoretically, the effective mass of the three orthogonal translations is assumed to have the same value. Nevertheless, it is shown from the Table 2 that the effective mass for the longitudinal and lateral are slightly different. This is due mainly to the difference (in weight) between the longitudinal and the transversal actuators. Additionally the vertical effective mass is found to be substantially larger, this is due to the contribution of the moving part of the static support and it also includes significant weight from the longitudinal, transversal and vertical actuators.

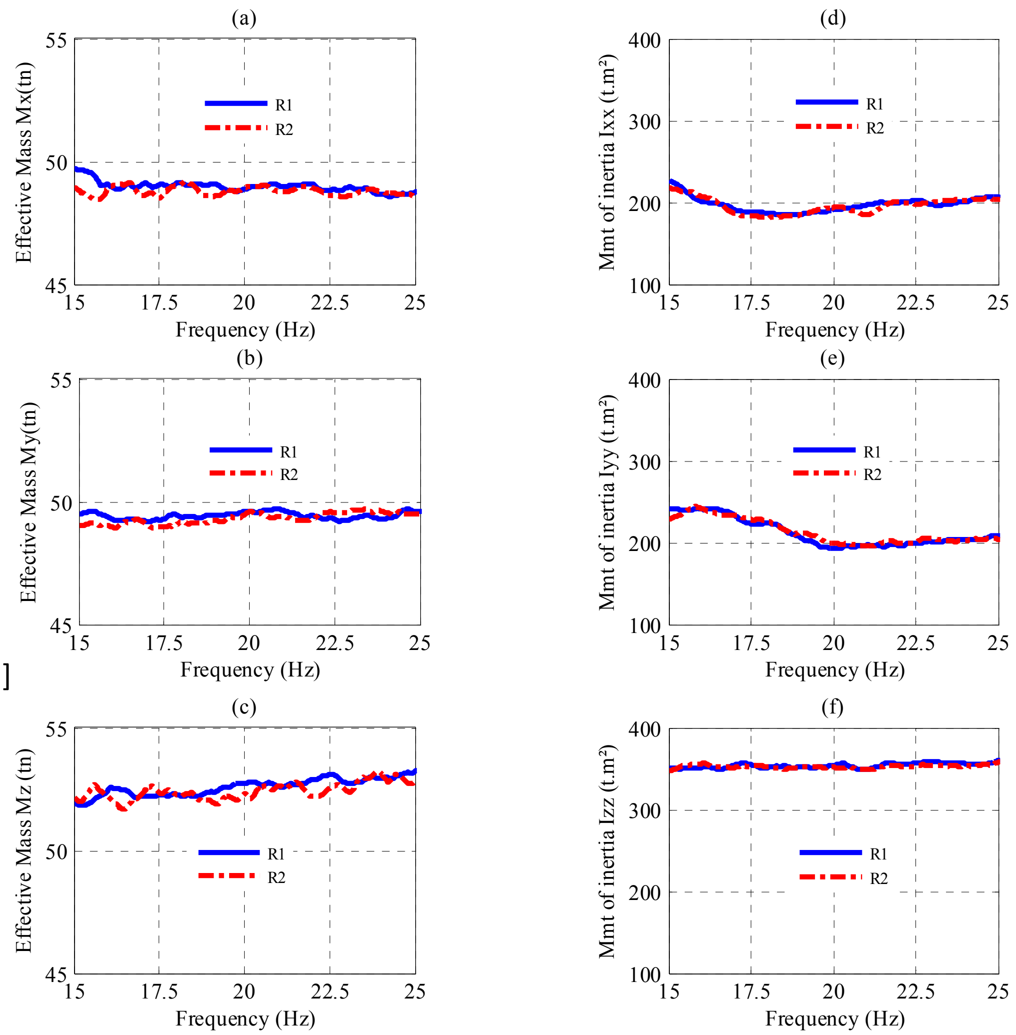


Fig. 9 Shaking table effective masses of the: (a) longitudinal, (b) lateral, (c) vertical, (d) roll, (e) pitch and (f) yaw degrees of freedom versus frequency

8. Simulation model

This section presents a mathematical model of the entire shaking table system under bare table conditions. The model includes mathematical representations of the different components of the shaking table system, such as servovalves, actuators and the table platen, and their interaction. The table parameters identified experimentally in the previous sections were incorporated in the model. Numerical values of other parameters involved in the model were taken from the manufacturer's technical specifications.

Fig. 10(a) shows the top-level view of the Simulink implementation of the complete model of the CGS shaking table. The internal view of the model is shown in Fig. 10(b).

The detailed model as well as is the equations for modeling of the different components of the system are beyond the scope of this paper and will be published in a separate manuscript.

Interaction between the components, Fig. 10(b), that comprise the model can be summarized as follow:

(1) Using valve-opening command as input, the actuators block, which includes twelve (12) individual actuator models, computes actuator force and oil flow.
(2) Actuator force is modified by static support and friction.
(3) Actuator displacement and force are transformed into table DOFs forces and moments
(4) The obtained quantities are transformed from the table origin to the system center of gravity (CG), the point in space where the rigid body dynamics are computed using Newton's law of motion in 6 DOFs, resulting in displacement, velocity, and acceleration of the CG.
(5) Displacement, velocity, and acceleration at the CG are transformed back to the table origin.
(6) Displacement, velocity, and acceleration at the table origin are transformed to actuator and accelerometer spaces.

9. Model validation

The validation of the simulation model presented in the previous section is accomplished by comparing the simulated responses with the measured ones. In fact, two types of tests were conducted in the shaking table, harmonic and earthquake tests. The input motion for the harmonic test consisted of three sinusoidal displacement records, having a frequency of 0.25 Hz and amplitudes of 100, 100 and 50 mm, conducted simultaneously on the three orthogonal DOFs, longitudinal, lateral and vertical respectively. The test was run with 90 degrees phase lags between longitudinal-lateral and lateral-vertical axes. The earthquake test consisted of the three components of El-Centro earthquake records commanded simultaneously to the three orthogonal DOFs, of the table.

Table references, table feedbacks and valve commands to the twelve (12) servovalves of the system are the quantities which were recorded during the tests. The data Recorder software part of the MTS 469D Seismic Controller was used to record the data. The DAQ system has a low-pass anti-aliasing filtering capability which was enabled during the tests in order to prevent aliasing.

It should be pointed out that the simulation model presented in the previous section models only the physical plant and does not include the controller; hence, inputs to the model are the recorded valve commands rather than the table references.

Figs. 11(a) - 11(c) show a comparison of the experimentally measured and simulated table displacement responses of the longitudinal, lateral and vertical axes respectively, for the case of harmonic test. It can be seen from these figures that the simulated and the measured responses agree very well. In the same way, Figs. 12 (a) - 12(c) compare the recorded and simulated table acceleration responses of the longitudinal, lateral and vertical axes, respectively, for the case of earthquake test. Excellent match between the responses from the model and the experimentally measured ones can be observed.

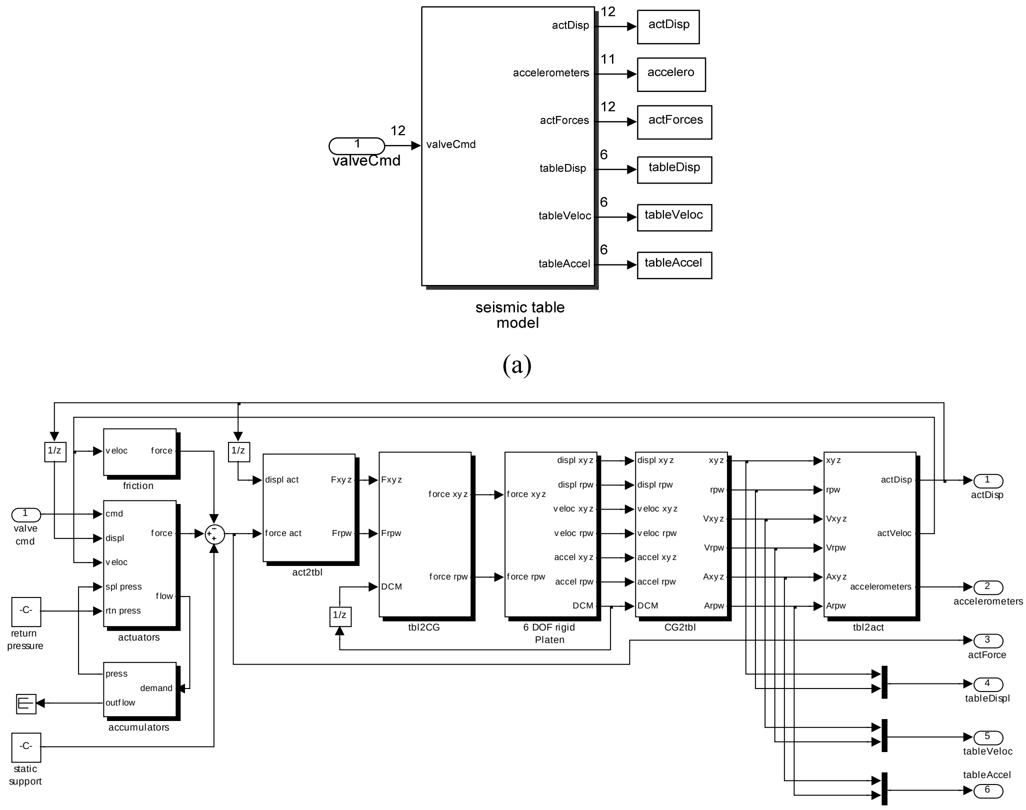


Fig. 10 Simulink model of the seismic shaking table (a) Top-level diagram and (b) Detail view of seismic table model

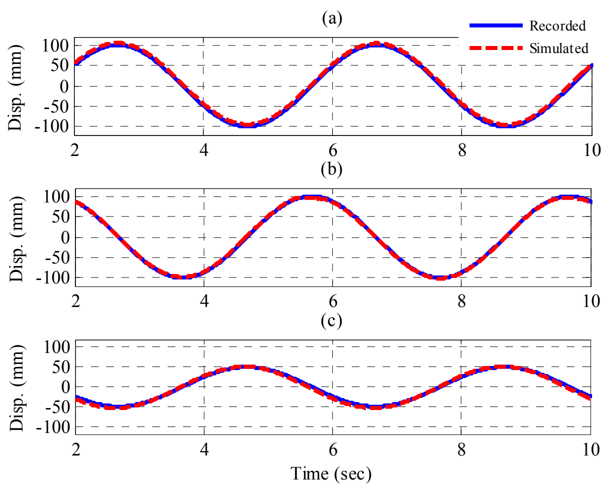


Fig. 11 Comparison between recorded and simulated table displacement responses: (a) longitudinal, (b) lateral and (c) vertical for harmonic input test

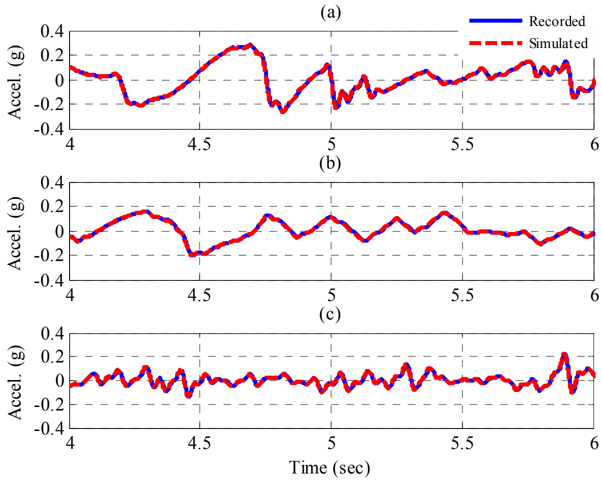


Fig. 12 Comparison between recorded and simulated table acceleration responses: (a) longitudinal, (b) lateral and (c) vertical for El Centro earthquake input test

10. Conclusions

The demand to achieve more progress in earthquake engineering has motivated many researchers and engineers to promote the need for more sophisticated seismic testing facilities, with high fidelity and high power levels. It is well known that the shaking table facility remains a very complex dynamic system, due to the various types of its components (mechanical, hydraulic and electronic components) and the complex nature of their dynamic interaction. Therefore, successful seismic testings depend on thorough characterization of the shaking table facility itself.

In this paper, the main results of the experimental characterization of the CGS shaking table were presented. Methods for estimating the key parameters from the experimental data were developed. It was found that the servovalves can be accurately modeled with either the first or the second order models. Oil column frequencies were clearly identified for the six degrees of freedom bare table case. The six by six mass matrix was estimated relating the generated random table accelerations to table forces. For all six degrees of freedom, the friction forces were nearly having a rectangular shape when plotted against displacement; this is a typical Coulomb's friction type. Globally, the frictional forces have negligible effect on the performance of the CGS shaking table, since they account only for 1.25% of the maximum system capability.

The identified parameters were used to develop the simulink model for bare table case. It includs the main components of the physical plant of the shaking table. The model was validated throught experiment tests using triaxial harmonic and earthquake signal inputs. It was found that the model was able to replicate the actual table responses with a very good accuracy. This model will be the core of more sophisticated and complete model that will be developed in the future.

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