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Real-time dynamic testing system for structures with delay compensation

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ABSTRACT

The paper describes the modelling and implementation of a new method for testing structures under dynamic loading. The method uses sub-structuring to link a physical test specimen with a numerical model of a surrounding structure. Displacements at the interface between the two are calculated in real time using an explicit time-stepping routine. The displacements are applied using hydraulic actuators, with measured restoring forces fed back to the numerical model. Time delays, inherent in the actuator, are compensated for, to prevent instability. The complete system is modelled in order that a good estimate of the test parameters may be ascertained prior to the physical test. Special attention is paid to the servo-valve and actuator dynamics. A simple portal frame system is used to assess the performance of the system and very good agreement is demonstrated between theoretical, experimental and simulated responses for a range of excitation forces.

INTRODUCTION

Although seismic codes and practices have been established and improved in recent years, the high levels of damage sustained by structures in the recent Northridge and Kobe earthquakes have demonstrated that further improvements could be made. In order to develop better design guidelines a great deal of experimental, analytical and numerical research has been carried out.

Current facilities for testing structures under dynamic loading fall into two categories; pseudo-dynamic test rigs and shaking tables. Pseudo-dynamic test rigs (e.g. Mahin et al. 1989) are used for work on large-scale structures or sub-structures that contain large discrete masses. Actuators supply the inertial forces that these masses are subjected to under dynamic loading. However, tests are usually conducted over a greatly expanded time scale, with the result that dynamic effects within the test specimen itself are lost.

Shaking tables are used for testing models of structures (e.g. Dritsos et al. 1998). The model is mounted on the table, which is driven by actuators so as to impart a prescribed base motion to the model. Shaking tables can provide loading at the correct rate, but require the use of small-scale models. The problems involved in extrapolating the results to prototype scale are considerable, particularly for a heterogeneous material such as reinforced concrete.

This paper describes the development of a new testing system that attempts to overcome the limitations of both methods, by allowing tests to be performed on critical components of structures at full or large scale, and in real-time. These critical components may be energy absorbing structural elements, which are therefore subject to complex non-linear behaviour, which is difficult to quantify satisfactorily, either analytically or experimentally, by other means. The rest of the structure, which may remain essentially linear, is modelled numerically. The numerical model and the test component interact in real-time. The method is referred to as real-time substructure testing. The resulting experiments more accurately represent the critical dynamic behaviour of real structures. However, preliminary tests indicated that problems arise due to inevitable delays between command signal and motion of the actuators. This has the effect of introducing negative damping into the system, resulting in instability. In order to counteract the delay a predictor has been incorporated so that the actuator moves to the correct position at the correct time.

The paper, in particular, focuses upon the development of a computer simulation model of the complete substructure testing facility. This is used to validate the test before it is performed, in order to ensure that the mechanical limits of the system are not exceeded. The system dynamics for elements of the complete system are described and analysed, with special attention being paid to the servo valves and actuators. The system dynamics have been used to create a computer simulation of the complete feedback system. Tests have produced encouraging results. Comparisons between the simulation and experimental results for both open-loop and closed-loop feedback tests for a simple portal frame sub-structure under sinusoidal loading, have shown that the simulation matches the experimental results very well.

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ACTUATOR DELAY COMPENSATION

Due to the inherent dynamics of the actuator, coupled with the test specimen, there is an inevitable delay between the actuator command signal and the actual change in displacement. The delay is of the order of 10ms, which is sufficient, in some cases, to cause instability of the feedback system. It can be shown that a delay has the effect of introducing negative damping into the system. Therefore instability will occur if the negative damping is greater than the inherent positive damping of the complete system, and in all cases there will be a net change in damping.

Clearly, this time-delay must be compensated for. The proposed method, following Horiuchi et al. 1996, allows the desired output to be predicted 10ms ahead, so that the delayed response approximately achieves the desired response. This is achieved by using a polynomial fit of previous command displacements and, hence, extrapolating the displacement, 10ms after the current desired displacement. The prediction is most easily formulated when the time-delay is equal to the time-step of the numerical model (although this is not a requirement). A fourth order polynomial has been found to work well, with good stability properties. The prediction is given by the following equation, where u_n is the desired displacement at the n^{th} time-step:

$$u_{n+1}=5u_n-10u_{n-1}+10u_{n-2}-5u_{n-3}+u_{n-4} \qquad (1)$$

The method has the advantage that there is no need to have any knowledge of the system dynamics prior to the test and it is, therefore, independent of any particular test and non-linear physical changes during a test. In addition, it produces a minimal computing overhead, which is clearly important for real-time systems in which there is a finite length of time in which to perform all the calculations for each time-step.

TEST SETUP

In order to compare the computer simulation with the actual system in the laboratory, a simple test system was developed. The physical test element was a 1.2m long 100mm×100mm×5.0mm thick square hollow section. The element was considered to be one column from a simple portal frame structure, consisting of two columns and a beam, with the actuator acting horizontally at the pin-jointed beam/column interface (Figure 1). An imposed load of 4kN/m was applied to the beam in order to add additional mass to the system. The column base was fixed rigidly to the floor, via a stiff bracket, with the actuator attached to the other end via a swivel joint, so that no moment would be applied to the top of the column.

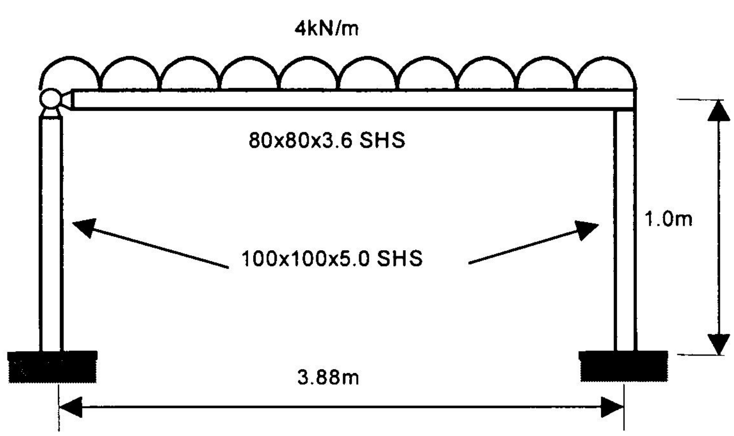

Figure 1. Portal Frame Test System.

For the test, the computational sub-system (the beam and the right hand side column) was modelled using two beam finite elements. The distributed masses of both the structural members and the imposed loading were approximated by lumped masses, applied at the two beam/column joints. The vertical degree of freedom at the beam/test-column connection was restrained, with both rotation and horizontal translation free. The resulting natural frequencies and mode-shapes were calculated and incorporated into a time-stepping discrete form of the equation of motion (for further details, see Darby et al. 1999). The first natural frequency of the complete structure was approximately 5Hz.

In general, the test specimen can be fully non-linear, in order to reflect the true system as accurately as possible. However, in order to allow a direct comparison between experimental results and the computer simulation (which, at present, only represents linear structures) the physical specimen was required to remain linear. Therefore, all sources of non-linearity, such as backlash, were minimised and the input excitation scaled such that the test specimen would remain in the linear elastic range.

In order to carry out the physical test, a program was written to implement the discrete form of the equation of motion in a time-stepping routine. The system was excited via base accelerations, given by a predetermined waveform (such as an earthquake acceleration record). These manifest themselves in the model as inertial forces acting at the lumped masses at the two beam/column joints. At each time step the desired displacement of the beam/column joint at the end of the time step is calculated, in real time. Next, the displacement after 10ms is predicted using the delay compensation method. A displacement command signal is then sent to the actuator attached to the column specimen via a digital-analogue converter. The restoring force is then measured by a load cell on the end of the actuator piston and is fed back to the numerical model via an analogue to digital converter. Hence, the excitation force for the numerical sub-structure model is a combination of the input base accelerations and the fed-back restoring force. These are then used to calculate the displacement at the next time-step. This cycle continues until the end of the input waveform.

APPARATUS MODELLING

The dynamic response of the testing apparatus is of prime importance in assessing the performance of the overall testing procedure. In order to take account of these dynamics in any such assessment, mathematical models of the testing equipment are useful tools. As part of the ongoing work at Oxford, computational models of the hydraulic actuators used in the testing process have been developed and implemented using the Matlab™ system modelling program Simulink™. These allow prior simulation and analysis of test algorithms and configurations. Displacements are imposed upon the physical substructure using Instron hydraulic actuators. An Instron 8800 controller controls the actuators via Moog E760 electrohydraulic servo valves. The controller sends a drive signal to the servovalve that determines the position of a spool within the valve body. Depending upon its position, the spool connects one side of the actuator to a high pressure oil supply and the other to an exhaust and vice versa. In this way the actuator piston may be driven back and forth. A schematic of the overall experimental set-up is given in Figure 2.

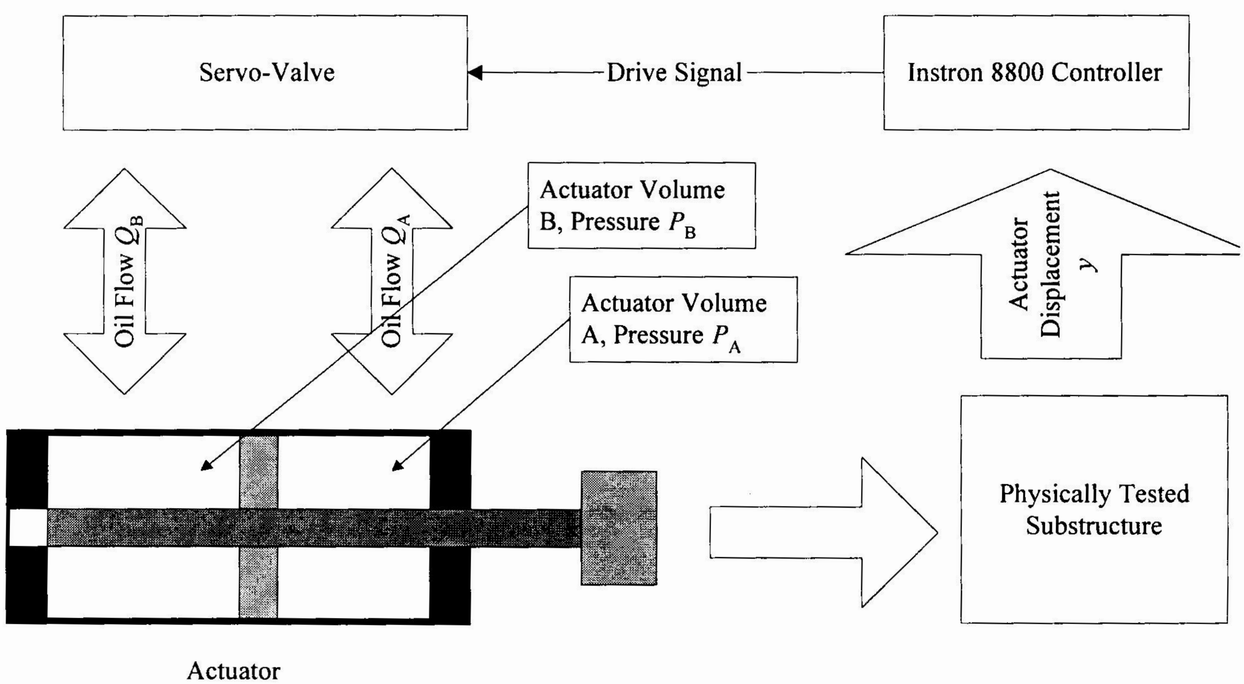

Figure 2. Experimental Schematic.

The controller is a proportional, integral and derivative type (Clarke 1984) and may be modelled as shown in Figure 3. The servo valve controls the flow to and from both sides of the actuator. Flow through the valve is dependent upon a square root relationship between the orifice areas and the pressure drop across it (Merrit 1967). The areas are given by the terms $A_{SA}(x)$, $A_{AR}(x)$, $A_{SB}(x)$ and $A_{BR}(x)$, where the subscripts SA refers to the area between supply pressure P_s and actuator side A, the subscript AR refers to the area between actuator side A and return pressure P_R, and so on. The flows to sides A and B of the actuator may be evaluated from equations 2 and 3.

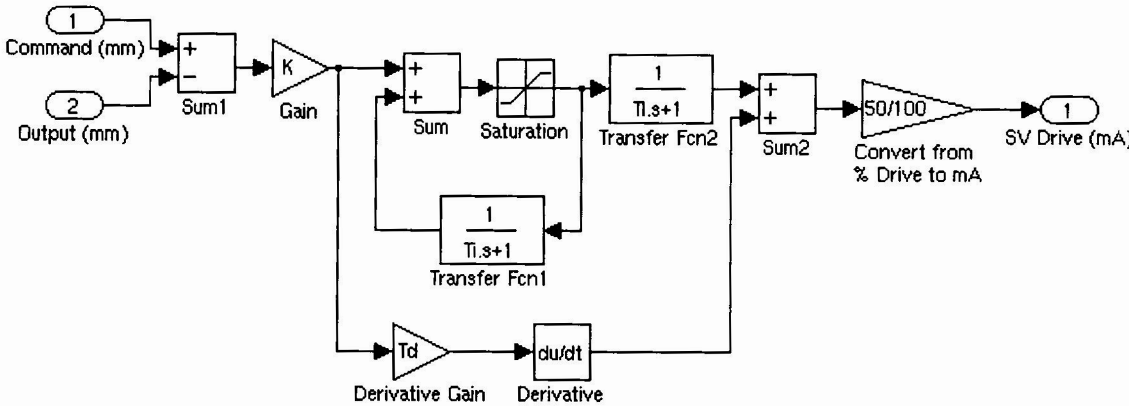

Figure 3. Block Diagram of the Controller Model.

$$Q_A = K_V A_{SA}(x)\sqrt{P_S - P_A} - K_V A_{AR}(x)\sqrt{P_A - P_R} \quad (2)$$

$$Q_A = K_V A_{SA}(x)\sqrt{P_S - P_B} - K_V A_{AR}(x)\sqrt{P_B - P_R} \quad (3)$$

The orifice areas are non-linear functions of the valve spool displacement x. This relationship must take account of leakage effects, which are significant at low values of x, as illustrated in Figure 4.

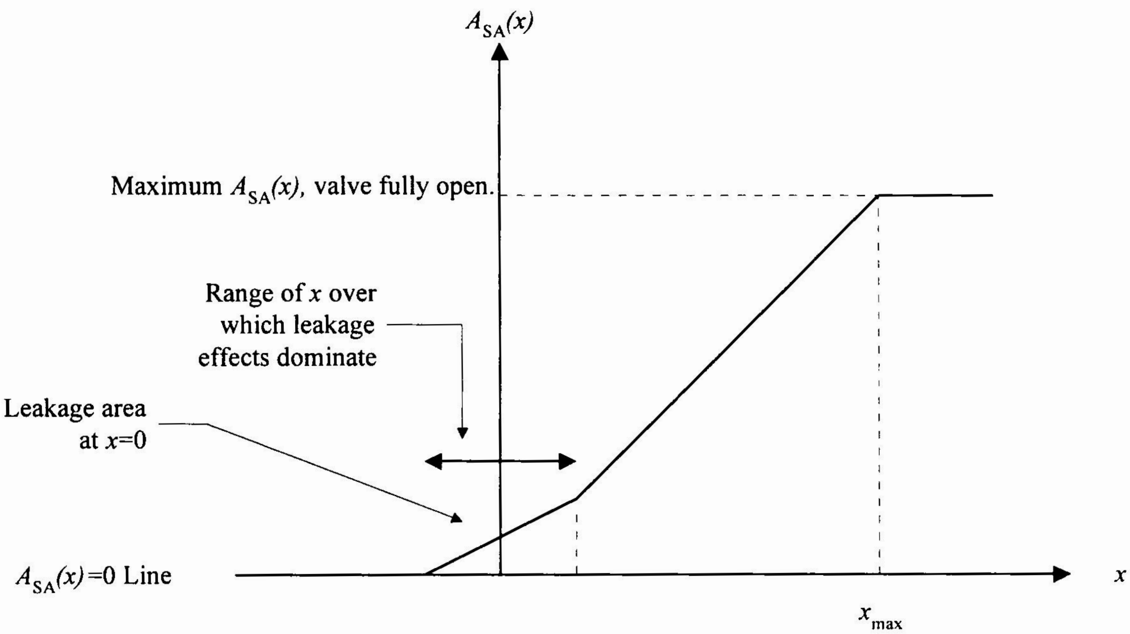

Figure 4. The Variation of $A_{SA}(x)$ with x.

Application of the principal of continuity to volumes A and B of the actuator yields equations 4 and 5, where A_p is the piston area, V_T the total actuator volume, s the actuator stroke, y the displacement of the actuator and β_E the effective oil bulk modulus.

$$\frac{dP_A}{dt} = \frac{2\beta_E (Q_A - A_P\, dy/dt)}{V_T(s+y)} \quad (4)$$

$$\frac{dP_B}{dt} = \frac{2\beta_E (Q_B + A_P\, dy/dt)}{V_T(s-y)} \quad (5)$$

Equations 4 and 5 may be subsequently integrated to give the pressures of oil in volumes A and B. The product of piston area, A_p, and pressure differential, P_A minus P_B, gives the net force acting on the actuator load, in this case the physically tested substructure. An overall model of the laboratory test is shown in Figure 5.

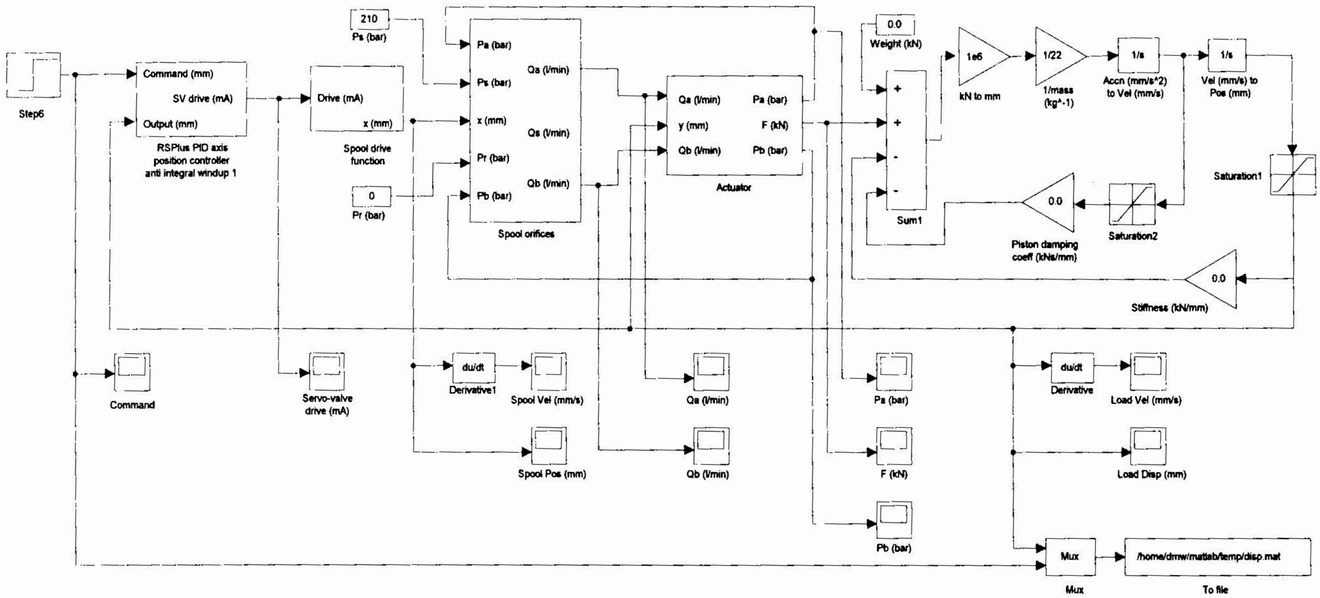

Figure 5. Overall Experimental Model.

TEST RESULTS

A series of tests were performed, using the portal frame, in order to compare the computer simulation of the complete system, the actual test and the theoretical desired response of the structure. Figure 6 shows the response of the system for a sinusoidal base acceleration of 2.5Hz. Figure 7 shows the response of the structure to an earthquake record (Northridge

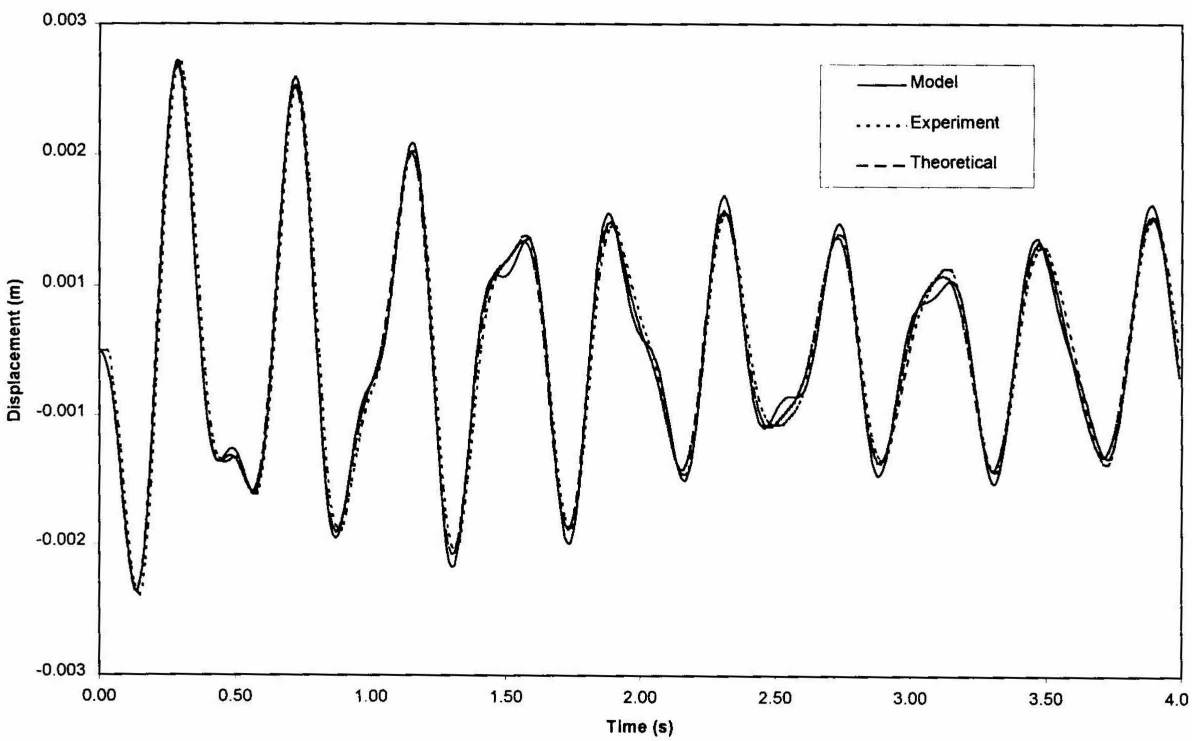

Figure 6. Response of Column Tip to $2.5ms^{-2}$ Amplitude, 2.5 Hz Sinusoidal Ground Acceleration.

earthquake, Tarzana station, North-South direction). The three responses for each excitation case are in very good agreement, particularly between the theoretical and experimental results. Hence, this indicates the accurate performance of the time-stepping method and the delay compensation method. The computer simulation of the system is in good agreement with the test, with discrepancies thought to be due to difficulties in modelling the damping properties of the actuator itself.

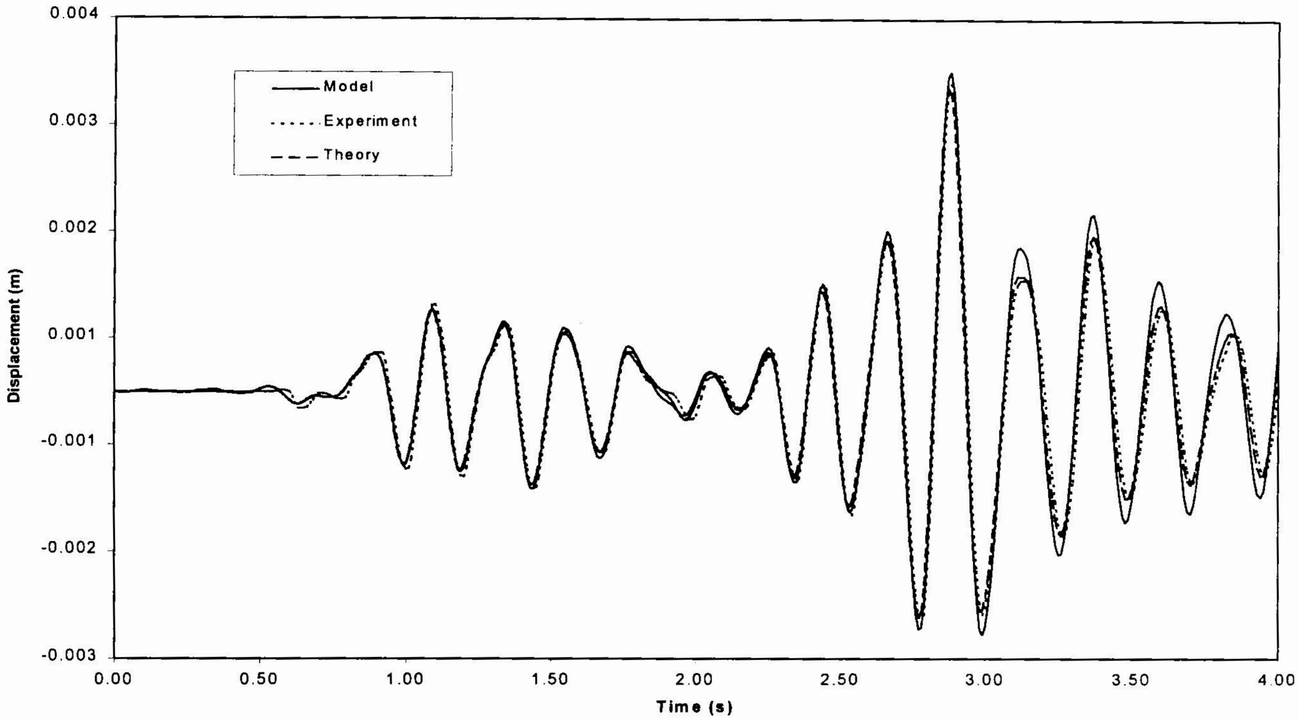

Figure 7. Response of Column Tip to Northridge Ground Acceleration Record.

CONCLUSIONS

A new type of sub-structure method for testing structures under dynamic loading has been described. The method uses sub-structuring to link a physical test specimen with a numerical model of a surrounding structure. Displacements at the interface between the two are calculated in real time using an explicit time-stepping routine. The displacements are applied using hydraulic actuators, with measured restoring forces fed back to the numerical model. Time delays, inherent in the actuator, are compensated for, to prevent instability. The compensation method involves extrapolating the desired position ahead by a time equal to the time-delay using a polynomial fit of previous data. Modelling of the complete system, incorporating the physical specimen, the numerical sub-structure, the delay compensation and, most importantly, the servo-valve hydraulic actuators, has been described in detail. This allows the response of the system to be estimated prior to a test, to ensure that the physical parameters of the system will not be exceeded during the actual test. A simple portal frame system has been used to assess the performance of the system. Very good agreement is demonstrated between theoretical, experimental and simulated responses for a range of base acceleration inputs.

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