Behaviour of equipment isolation systems under seismic conditions

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ABSTRACT: Analytical and experimental studies of the behaviour of equipment vibration isolators under seismic conditions are presented. A model of a prototype air handling unit mounted on vibration isolators was constructed for use in the experimental studies. Two types of vibration isolators - elastomeric isolators (EI) and open spring isolators with uni-directional restraint (OSI) - were tested under static and dynamic conditions. The experimental results indicate that the vibration isolators have non-linear stiffness characteristics and high damping values; they also define the mode shapes and natural frequencies of the systems.

These model identification test results were used to formulate analytical models of the vibration isolated systems. A numerical procedure, utilizing time series analysis, was then applied to solve the equations of motion of the systems. Good agreement was observed between the model response parameters measured experimentally during tests on a shaking table and the corresponding values calculated by this analytical approach. This study indicates that the analytical procedure can be used to predict the response characteristics of vibration isolated equipment systems subjected to known base excitations.

1. INTRODUCTION

The protection of equipment from seismic hazards is of wide engineering interest. Past earthquakes have shown that failure of equipment systems which lead to the breakdown of emergency services can produce loss of human life and extensive property damage. Vibration isolated equipment is particularly vulnerable to seismic forces. Typically, this type of equipment is mounted on special spring units, or other resilient supports, in order to reduce significantly the transmission of shock or periodic forces into the primary system (structure) as a result of normal equipment operation. Motion restraining devices are sometimes used to prevent the development of excessive response in the resiliently mounted equipment. Experience from past earthquakes has demonstrated that resiliently mounted equipment without motion restraining devices may be totally destroyed even though the structures housing such equipment may suffer relatively minor damage. In the case of open spring isolators, failure may result from the dislodgement of the isolator spring units, which can lead also to damage in the vicinity of the equipment.

Some resilient mountings with motion restraining devices have not been extensively tested either in actual earthquakes or in a simulated seismic environment. A reliable performance evaluation program, including a consistent analysis and design procedure, is needed if resiliently mounted equipment is to be protected from earthquake damage. The investigation reported herein is intended as a contribution toward such a program.

During the past decade a great deal of research has led to a considerable advance in our understanding of the response of equipment systems to earthquake excitation. Newmark (1972), Nakahata et al. (1973), Sackman and Kelly (1978), Villaverde and Newmark (1980), Robinson and Ruzicka (1980), Der Kiureghian et al. (1983,1984) have investigated different analytical methods designed to take into account the important effects of dynamic interaction between the primary and the secondary

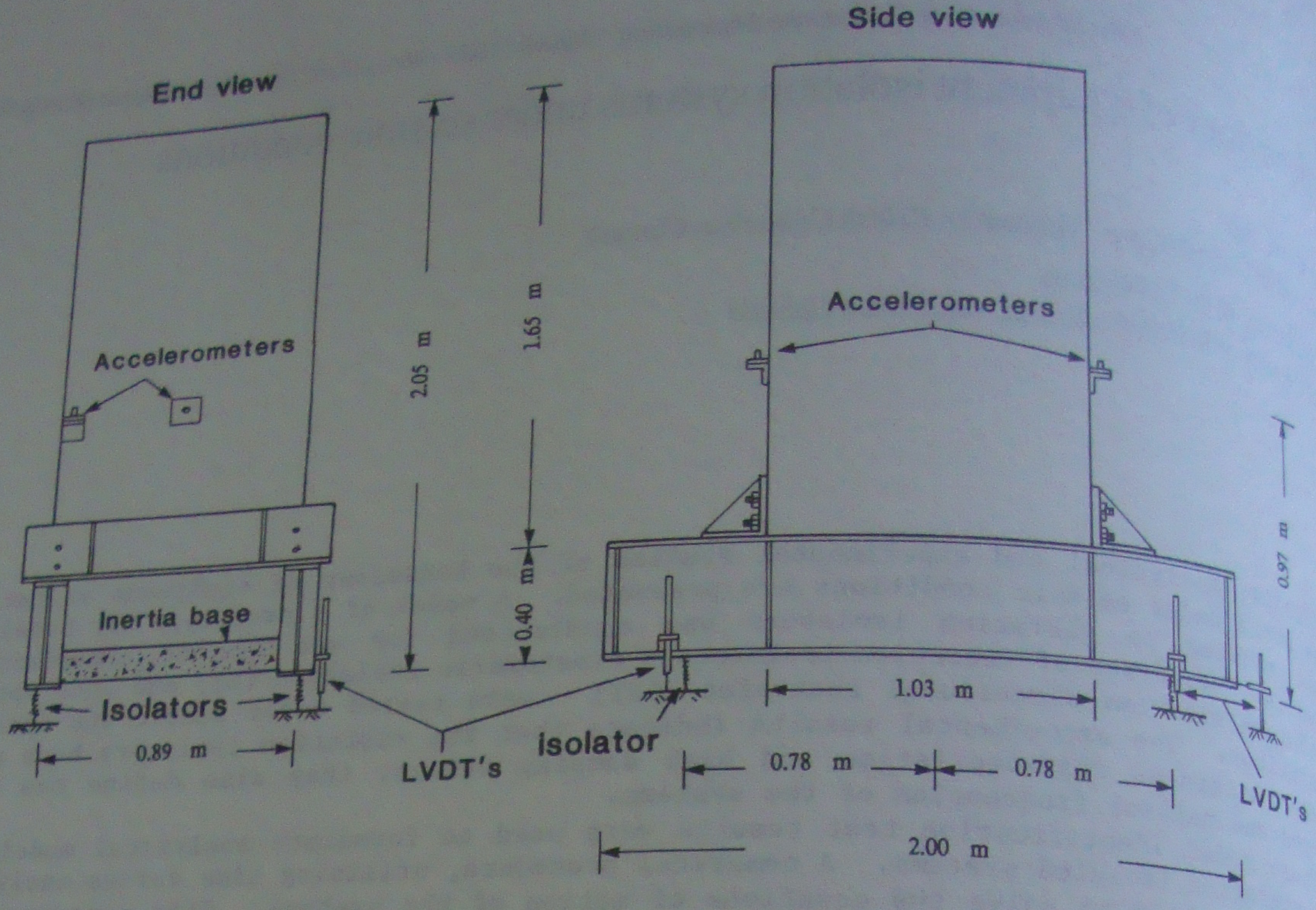

Fig. 1 Experimental model showing instrument locations

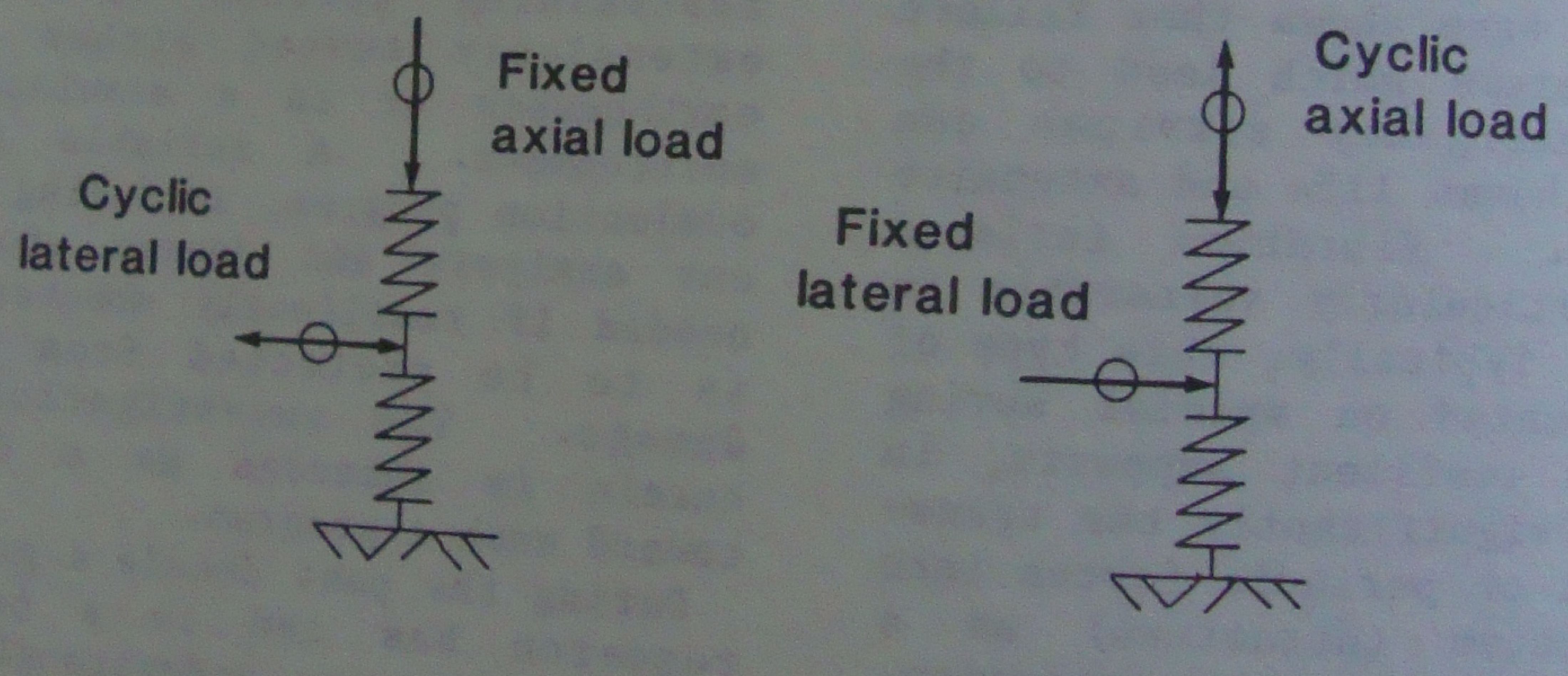

Fig. 2 Schematic representation of bi-directional static test

(equipment) systems. In these methods, the combined equations of motion of both the primary and the secondary systems must be considered. Good analytical solutions to the problem of linear equipment-structure interaction were obtained by these researchers.

The seismic response of a nonlinear equipment isolation system is frequently determined by a numerical time-step analysis, which allows for the proper modelling of the equipment nonlinearities. Resiliently supported equipment with motion limiting constraints represents one example of a nonlinear isolation system; the nonlinear characteristics of such units are developed when the support system comes into contact with stiff, motion limiting devices. Iwan (1977,1978) has studied this problem analytically. He replaced the set of nonlinear springs comprising the isolators by a set of equivalent linear springs whose stiffness depends on the displacement amplitude. Using a conventional floor response spectrum, and an iterative procedure, Iwan developed a method to evaluate the seismic response of equivalent linear isolated equipment systems with motion restraint devices.

The main objective of the research reported in this paper is to develop and verify a general analytical procedure for investigating the seismic response of vibration isolated equipment systems. To meet this goal, the dynamic characteristics (natural frequencies, mode shapes, and damping values) of a model of a prototype industrial ventilating fan mounted on commercially available vibration isolator units were first determined experimentally. Based on the test observations, an analytical model of the vibration isolated equipment units was then developed and a solution to the response problem was obtained using a time-step analysis method. The analytical procedure was then verified by comparing the predicted results with those obtained experimentally from shake table tests.

2. EXPERIMENTAL STUDY

The experimental program was divided into two parts involving: i) static tests of the vibration isolators and ii) dynamic tests of the isolator mounted model. Fig. 1 illustrates the experimental model constructed for use in the tests. The model represents an air handling unit mounted on vibration isolators - the prototype is a Chicago Vane Axial Fan (Design 34, size 54 1/4, arrangement 9) with a 405 US 100 bhp motor, frame and inertia base. The model parameters are based on the mass, mass moment of inertia, dimensions, centre of mass location, and vibration isolator locations of the equipment prototype. Two types of isolators were selected for testing: an open spring isolator with uni-directional restraint (OSI) and an elastomeric isolator (EI). The isolators are commercially available. Their sizes were chosen in accordance with the manufacturer's recommendation for the model weight, fan speed and geometry; their respective rated vertical stiffnesses were 350 kN/m and 630 kN/m.

2.1 Static tests

The static testing program was conducted to determine the stiffness characteristics of the isolators. Only the isolator stiffness in the axial direction was available from the manufacturers; however, the stiffness characteristics in both the axial and the lateral directions are needed to define these elements when modelling the prototype.

In these tests, isolators were mounted horizontally in pairs and arranged in series as shown schematically in Fig. 2. This set-up was needed in order to utilize existing test facilities. The isolator assemblage was initially loaded to a specific axial load and then laterally displaced. This procedure was repeated for a range of axial preloads so that the lateral load-deflection relationships for each pair of isolators, at various levels of axial preload, were obtained. The lateral stiffness and the effect of axial load on the lateral stiffness was determined from these lateral load-deflection tests.

Axial load-deflection curves were obtained in a similar manner using the same experimental set-up. In this case the lateral preload was maintained at a fixed value while the isolators were displaced axially. The axial stiffness characteristics of the isolators were determined and compared with the commercially rated values. The influence of lateral load on the axial stiffness characterisitcs was evaluated by repeating the above procedure for a range of lateral preloads.

The mass and the location of the mass center of the model were also determined experimentally. The weight of the model

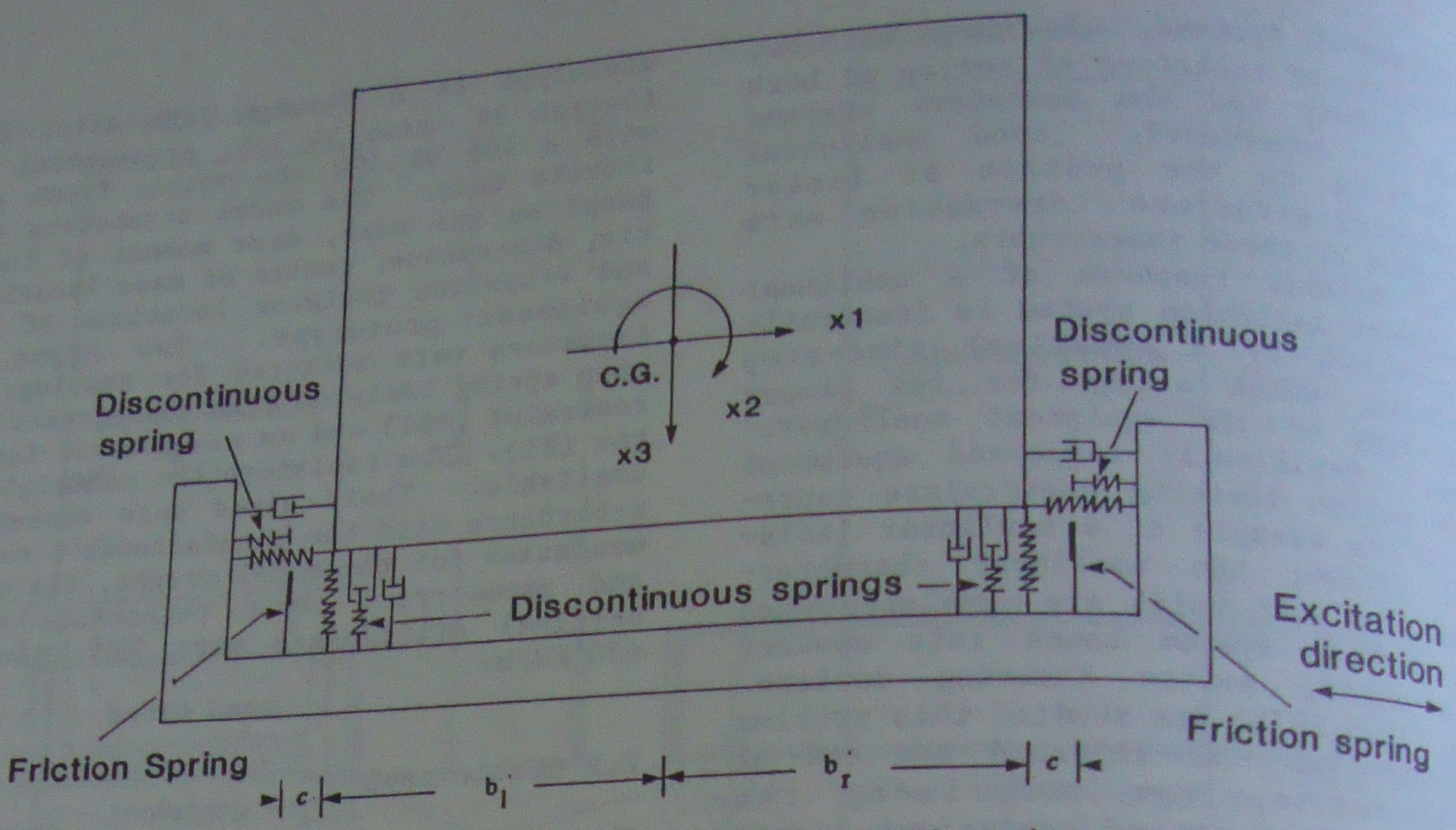

Fig. 3 Idealized model of the vibration isolated equipment system

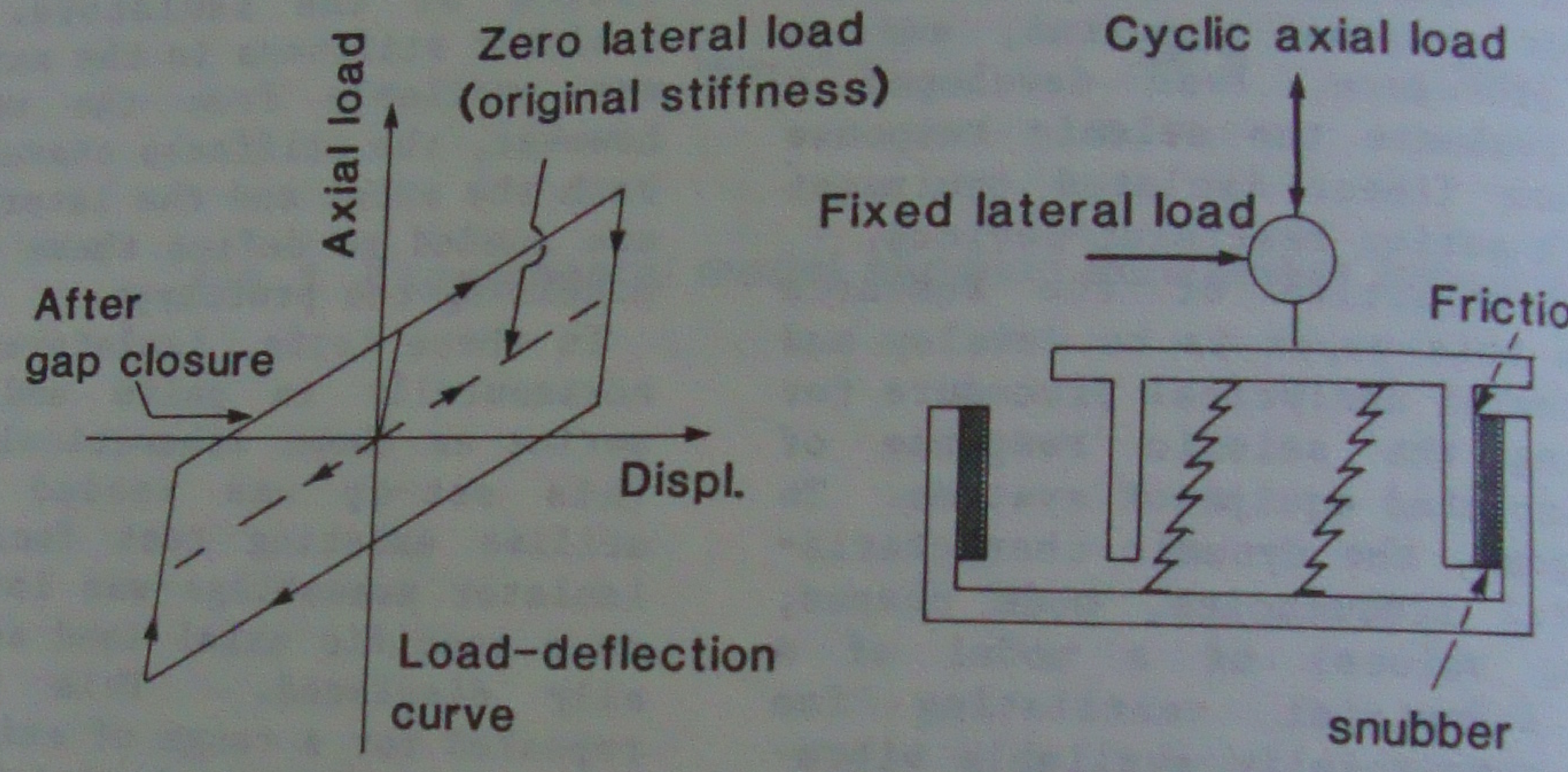

Fig. 4 Schematic representation of friction force between Isolator (OSI type) and snubber device

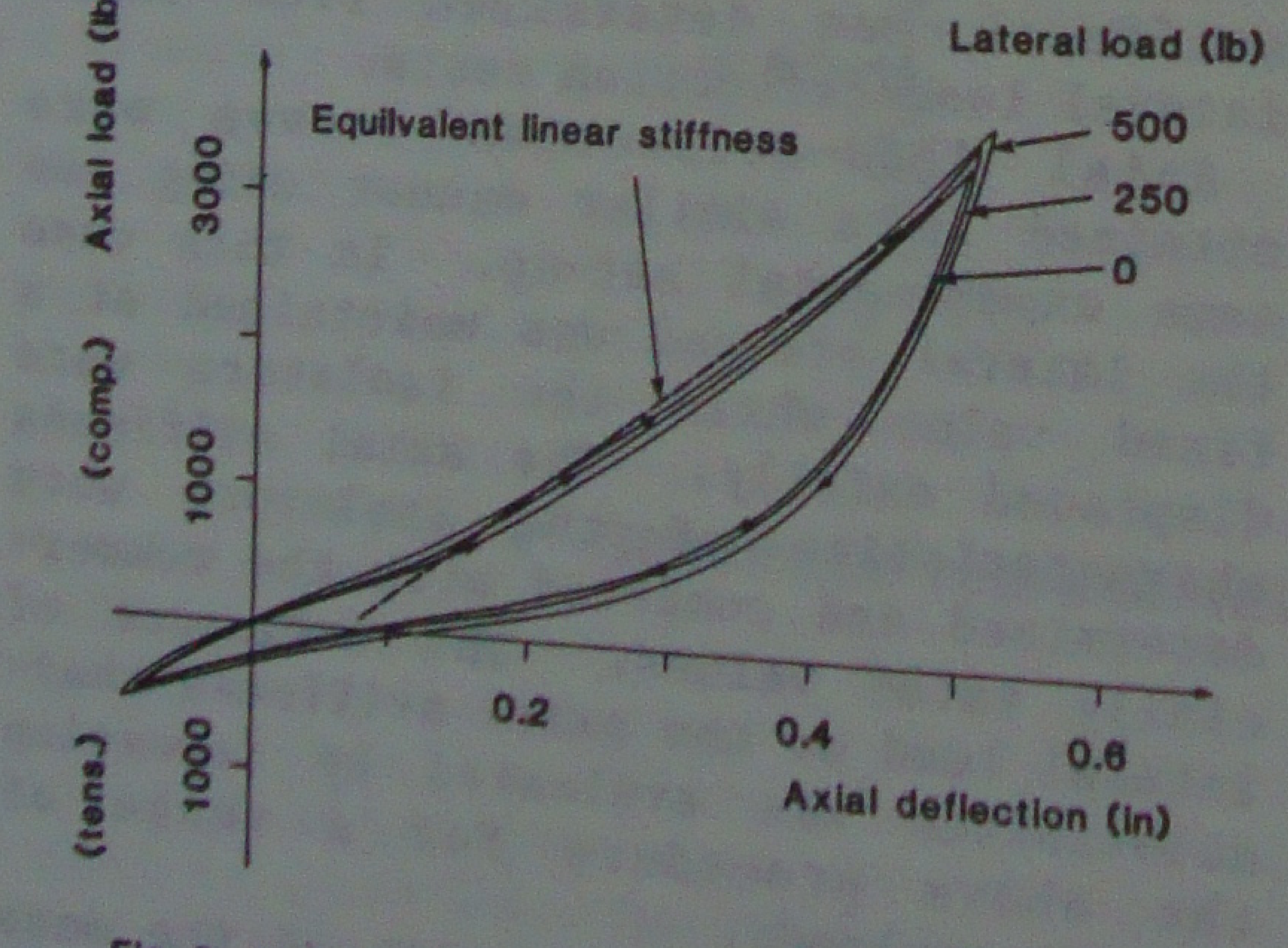

Fig. 5 Axial load-deflection curves of an elastomeric isolator

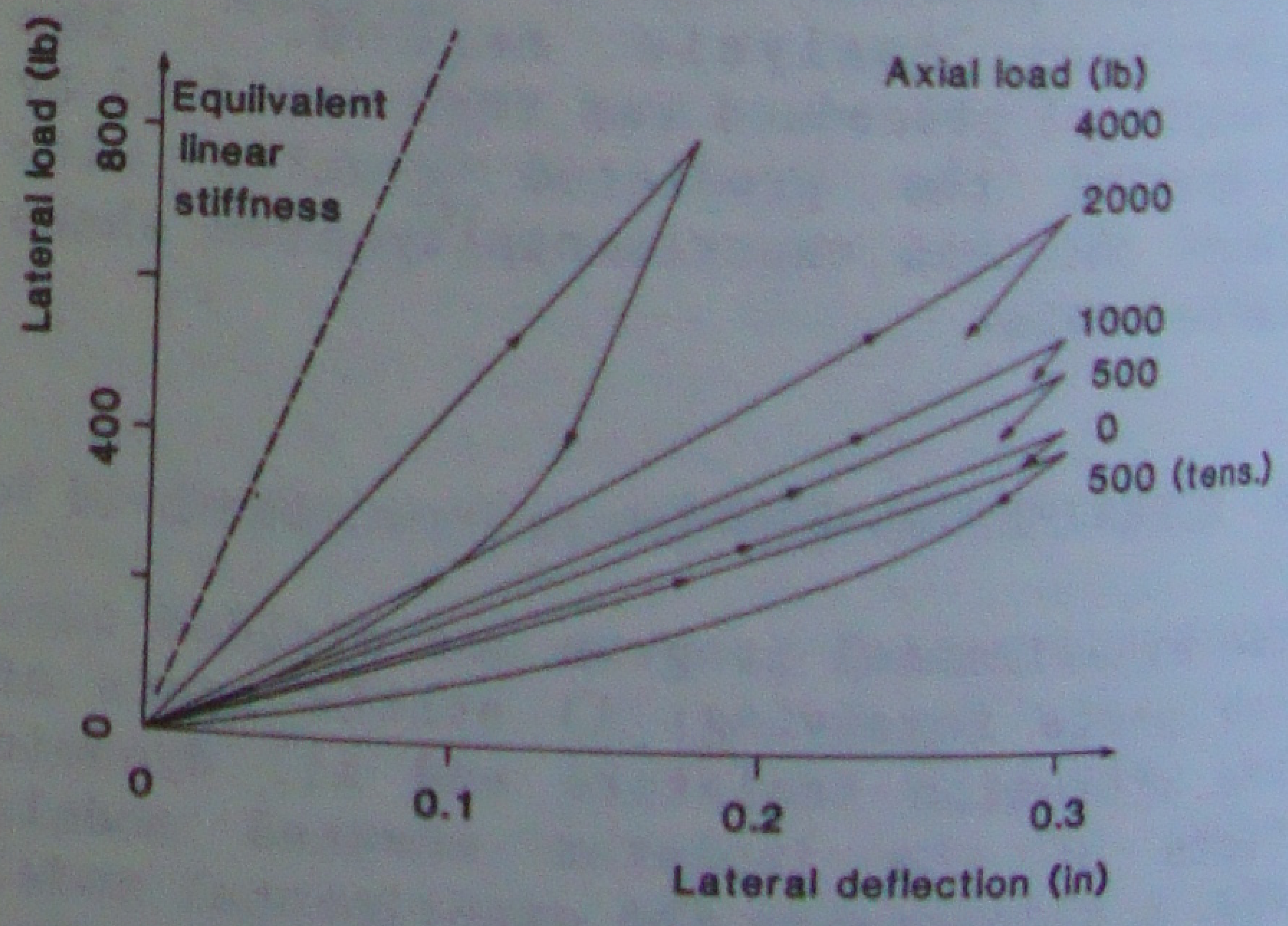

Fig. 6 Lateral load-deflection curve of an elastomeric isolator

was measured by means of a load cell. The mass centre was located by suspending the system from two different support points and locating the intersection of the plumb lines attached from the same points. These quantities and the measured dimensions of the model were then used to calculate its mass moment of inertia.

2.2 Dynamic tests

The dynamic testing program was conducted on a 3.3 m × 3.3 m shaking table. The first series of tests was performed to define the dynamic characteristics of the resiliently mounted systems and involved the application of sinusoidal base motions to the specimens. The base displacement amplitudes were kept relatively small to avoid the possibility of damage to the prototype models at system resonance; for the EI and the OSI tests, the peak-to-peak amplitudes of the table motion were 3.6 mm and 2.6 mm respectively. Based on a preliminary estimate of the natural frequencies of the models, a range of forcing frequencies (0-15 Hz) was chosen to excite the systems. Approximately 25 frequencies were used within this range.

The frequency response curve for each model was obtained by plotting the ratio of the measured steady state response amplitude for each degree of freedom of the model to the amplitude of the forcing motion of the table, versus the forcing frequency. The natural frequencies of the system were obtained from these results.

The modal damping ratios of the two isolator systems were determined from the results of sinusoidal decay tests. In these experiments, the model was excited into steady state motion at each of its natural frequencies in turn, so that the system responded by vibrating in the corresponding natural mode. When the table motion was stopped suddenly, the system responded at its natural frequency in a mode of free damped virbation. The resulting decaying response was recorded and the modal damping ratio was determined by the well known logarithmic decrement method.

The system identification studies were followed by a series of tests to measure the response of the model due to seismic disturbances. For this purpose, acceleration records corresponding to real earthquake ground motions were used as input motions to the shaking table. The lateral and the rotational accelerations at the model mass centre were monitored. Care was taken to ensure that the frequency content of each of the input records bracketed the natural frequencies of the model. It was for this reason that floor response records were not used although it was recognized that at a particular resonance the response would be more severe; the purpose of the test was to verify the mathematical model, not to seek maximum response. The chosen records were: i) El Centro N-S (1940), and ii) San Fernando N21E (1971). The peak input base motion acceleration was scaled to appropriate levels (0.1-0.2 g), so that strong model responses were recorded without damaging the model.

After the completion of the seismic ground motion tests, the measured response time-histories were compared with the model response time-histories predicted from an analytical study; the method of analysis is discussed below.

3. ANALYTICAL STUDY

The formulation of the mathematical model of the vibration isolated equipment involves some approximations. First consider the idealized vibration isolated equipment system shown in Fig. 3. The system consists of a rigid mass with its inertia properties lumped at the centre of gravity of the model. The rigid mass is supported by four sets of springs, which provide the system with its lateral, rotational, and vertical stiffness.

Each set of springs consists of a linear elastic spring and a discontinuous linear spring. The springs provide restoring forces to the system when the rigid mass is displaced from its equilibrium position. In real equipment systems, the development of excessive seismic motion is prevented by the provision of snubbers, which act as motion restraint devices. Such devices provide an abrupt increase in the restoring force of the isolator when the displacement of the unit exceeds a prescribed value. In order to model the characteristics of these devices, a discontinuous linear spring is added to the assemblage, which provides the required additional stiffness (see Fig. 3). As a result, the simple spring assemblage actually models a bilinear hardening system.

The static testing program demonstrated that when the horizontal snubber units in the OSI are engaged at some fixed lateral

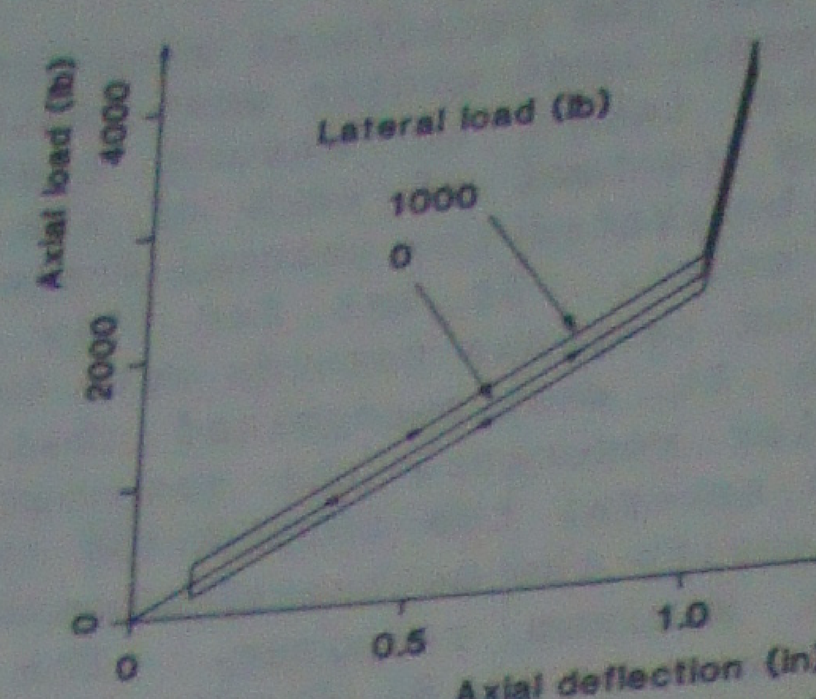

Fig. 7 Axial load-deflection curves of an open spring isolator

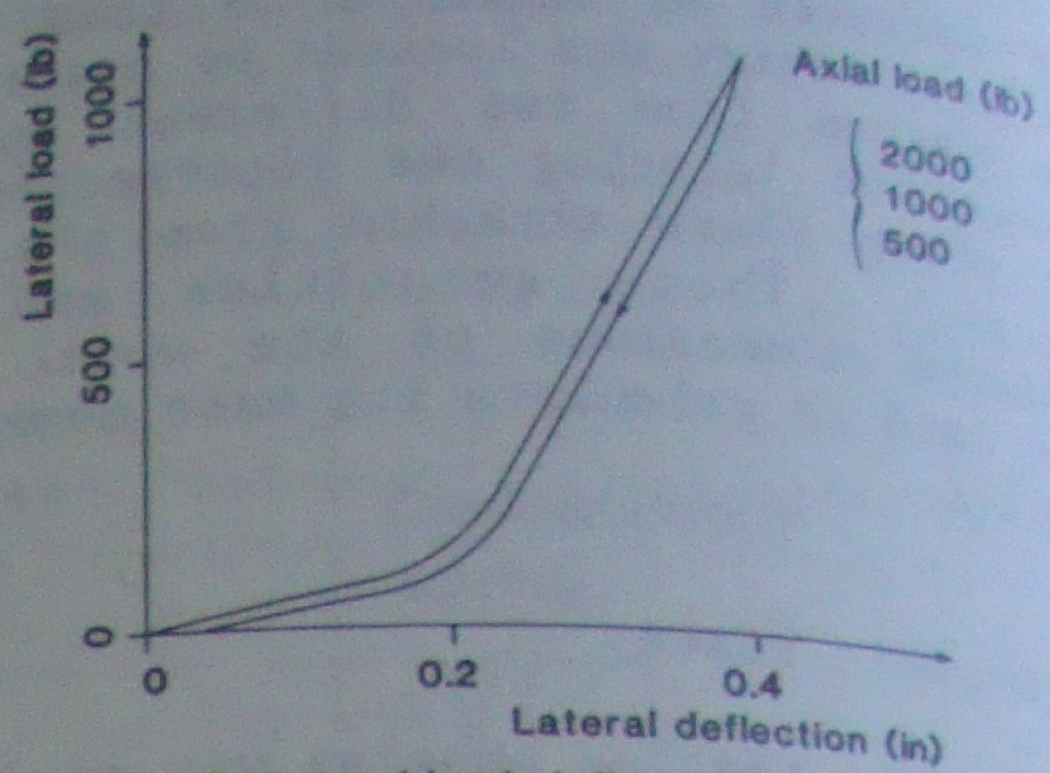

Fig. 8 Lateral load-deflection curves of an open spring isolator

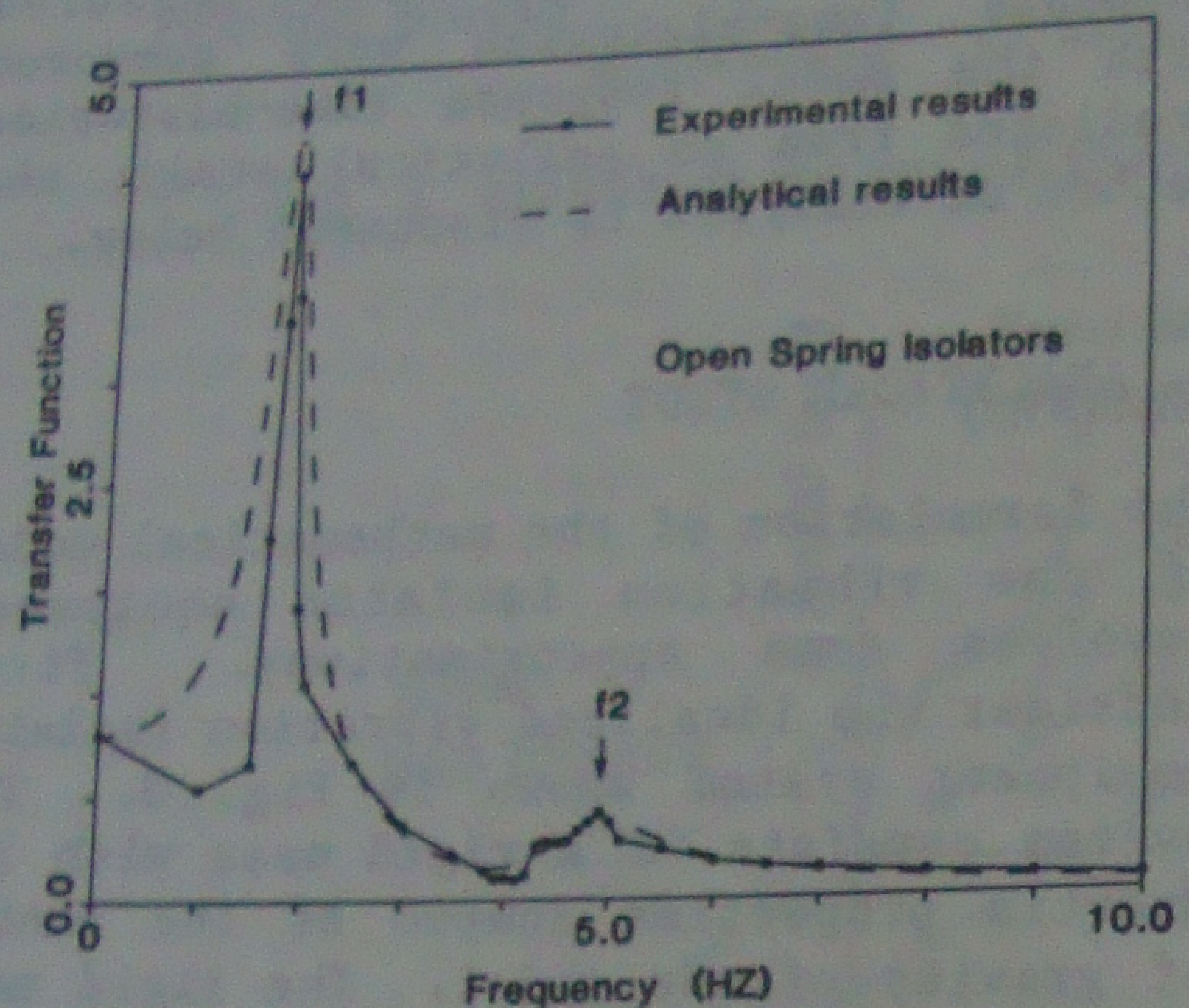

Fig. 9 Horizontal acceleration frequency response function

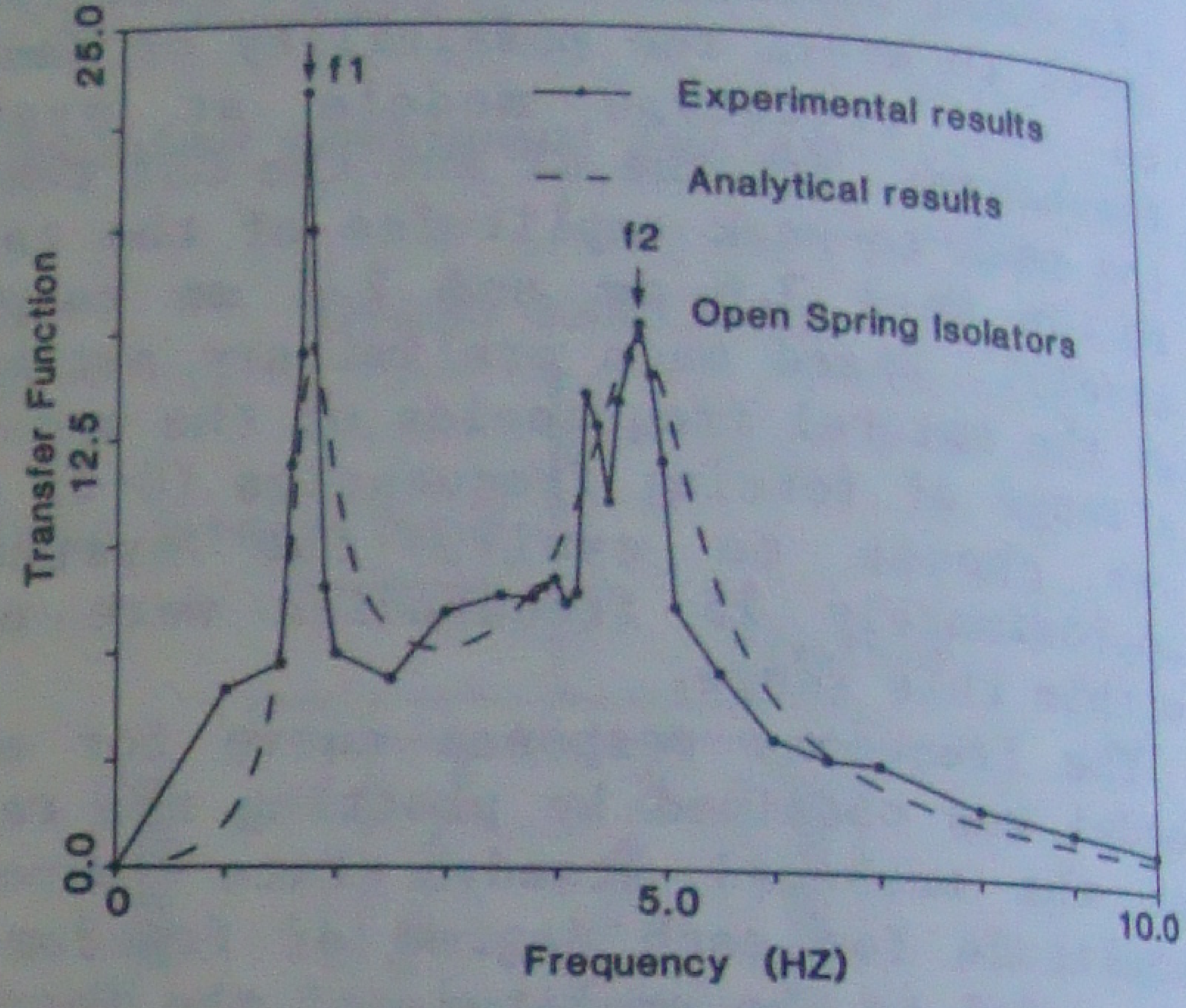

Fig. 10 Rotational acceleration frequency response function

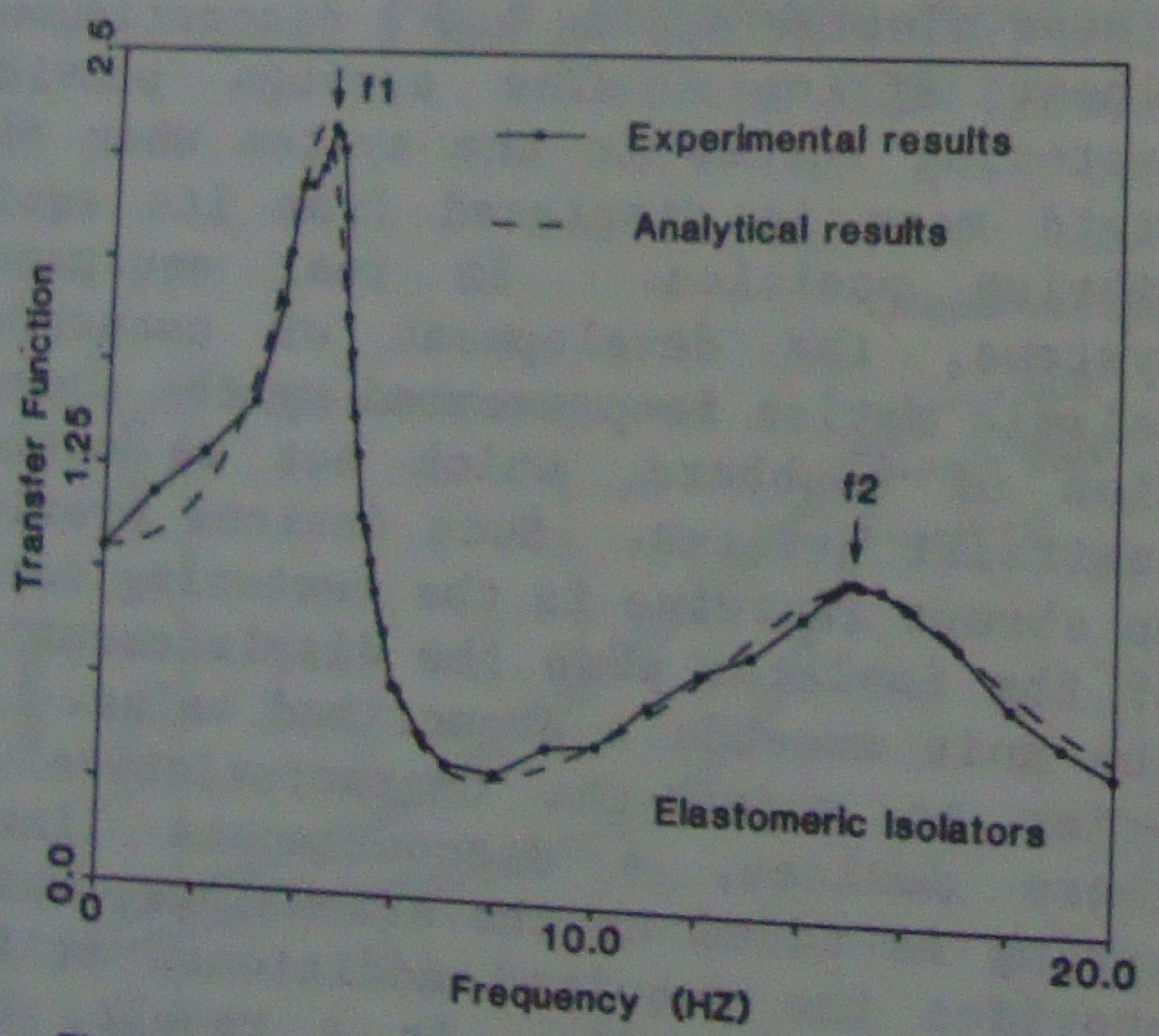

Fig. 11 Horizontal acceleration frequency response function

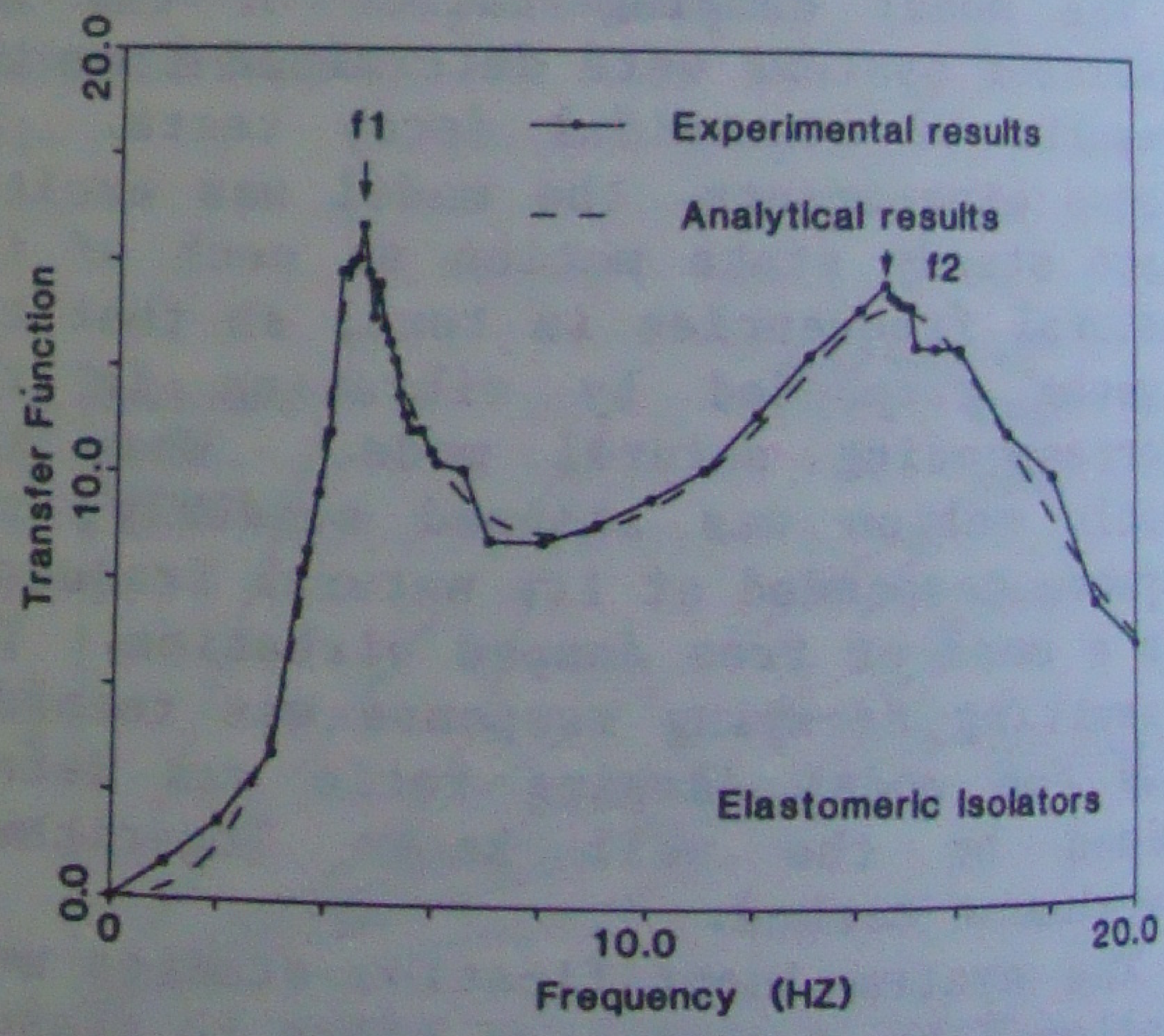

Fig. 12 Rotational acceleration frequency response function

load, the axial resistance of the springs increases substantially. However, the stiffness, i.e. the rate of increase of load with deflection, is unaffected, as shown in Fig. 4. This phenomenon is the result of the development of a friction force. With the horizontal snubbers engaged, the springs are held axially by friction between the snubber and the spring assemblage. An increased axial load is needed to overcome this friction force before slippage between the snubber and spring assemblage can occur in the axial direction. To model this effect, a friction spring is included in each set of vertically mounted springs in the mathematical model; the friction spring becomes active only when the horizontal snubber units are engaged. Finally, in addition to the effect of the friction force, the effect of material damping of the vibration isolators also needs to be considered. In the mathematical model, material damping is modelled by the provision of linear viscous dashpots.

Assuming small displacements, the equations of motion of the model with reference to the static equilibrium position of the system, when subjected to a horizontal base excitation $\ddot{U}_g(t)$ at the point of equipment support and formulated about the mass centre of the model, are given by:

$$M\ddot{X} + C\dot{X} + K(X)X = -M\ I\ \ddot{U}_g(t) \qquad (1)$$

where $X = [x_1, x_2, x_3]$ is the relative displacement vector, $\dot{X} = [\dot{x}_1, \dot{x}_2, \dot{x}_3]$ is the relative velocity vector, and $\ddot{X} = [\ddot{x}_1, \ddot{x}_2, \ddot{x}_3]$ is the relative acceleration vector of the equipment, all measured with respect to the moving base, whose acceleration is defined by $\ddot{U}_g(t)$. The three degrees of freedom x_1, x_2 and x_3 are respectively the lateral motion, the rotation, and the vertical movement of the mass centre of the equipment system (see Fig. 3). Positive displacements are defined by horizontal motion to the right, clockwise rotation, and downward motion. The unit column vector I is the influence vector coupling the base input motion to the individual degrees of freedom of the equipment. M, C, and K(X), are the equipment mass, damping, and stiffness matrices respectively.

The stiffness matrix, K(X), may be nonlinear because of the presence of the discontinuous springs and friction springs. K(X) depends on the equipment displacement, X, which defines whether these springs are activated. The equipment mass matrix, M, is determined from the mass and mass moment of inertia of the model. Also, by assuming orthogonal modal damping, the equipment damping matrix C is derived from the equipment mass matrix M, the modal damping values, and the equipment mode shapes and natural frequencies. The equipment natural frequencies and the mode shapes are obtained from an eigenvalue solution of the frequency equation for the undamped free vibrating system. Knowing the equipment mass, stiffness and damping matrix, Equation (1) can be solved by the well-known Newmark-Beta method.

4. EXPERIMENTAL AND ANALTYICAL RESULTS

4.1 Static test results

The static test results were used to define the inertia and stiffness characteristics of the analytical models representing the vibration isolated equipment systems.

Table 1 lists the measured mass and mass centre of the model and the manufacturer's specification of these parameters for the prototype system. Also shown are the calculated rotatory inertia properties of the prototype and the model based on an assumed uniform distribution of mass over the known geometry of the systems. A fairly close match between the model and prototype inertia properties was achieved. An exact match of inertia properties proved to be difficult; the model actually represents a vibration isolated system which is similar to the prototype system but possesses the measured inertia properties summarized in Table 1.

The lateral and axial stiffness characteristics of the EI and the OSI were obtained from the static testing program; these properties are shown in Figs. 5-8. For the system containing the OSI, the measured axial and lateral stiffness variations were replaced by idealized bi-linear springs having the stiffnesses shown in Table 2; in addition, the analytical model also incorporated friction springs whose friction coefficient measured 0.08. The spring stiffness increases when the spacings between the spring coils are eliminated under a compressive load or when the spacing between the snubber and the spring unit is reduced to zero. The snubber gap and the spring coil spacing of the OSI tested were both 50 mm (0.2 in).

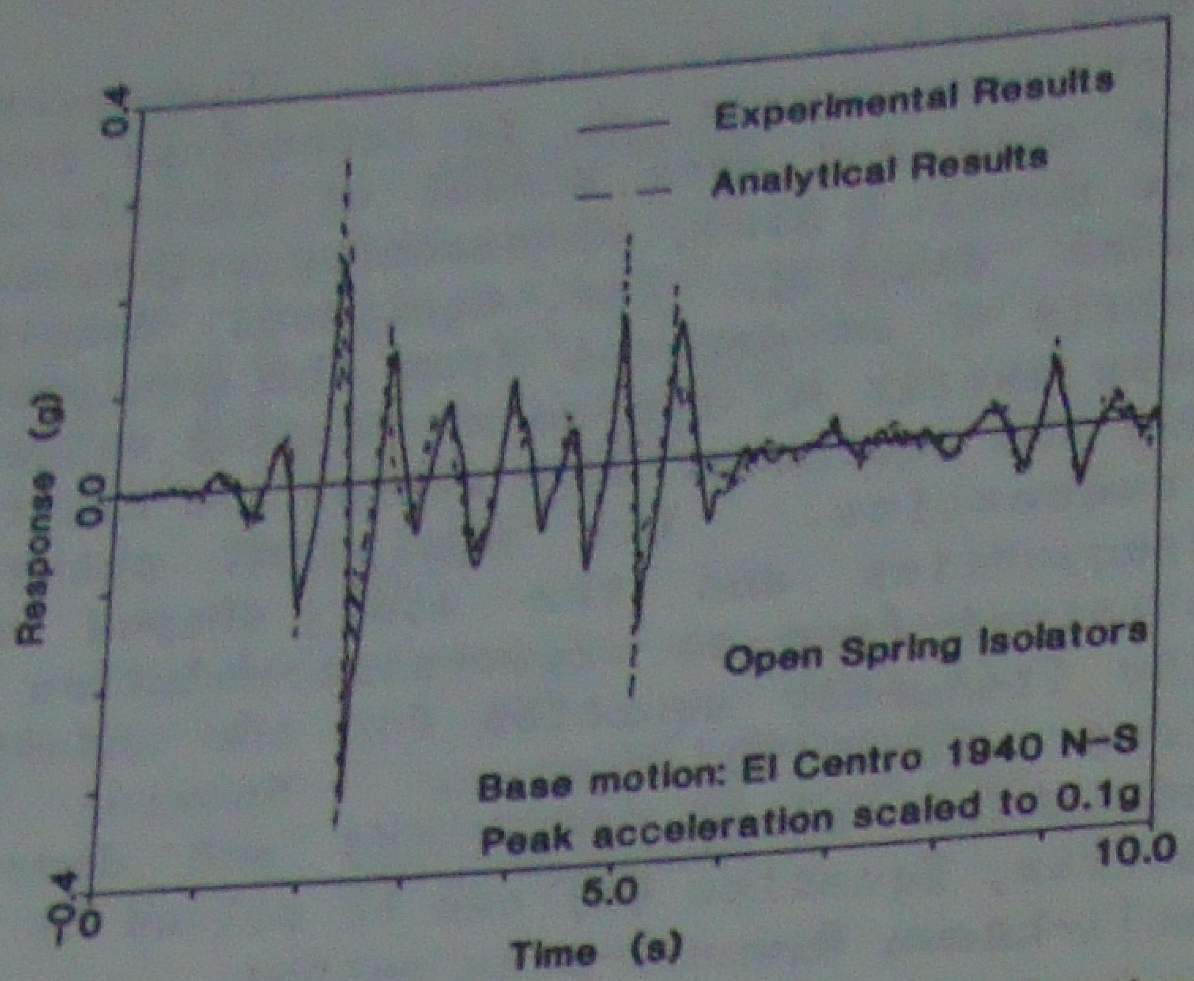

Fig. 13 Predicted and measured horizontal acceleration response time-history

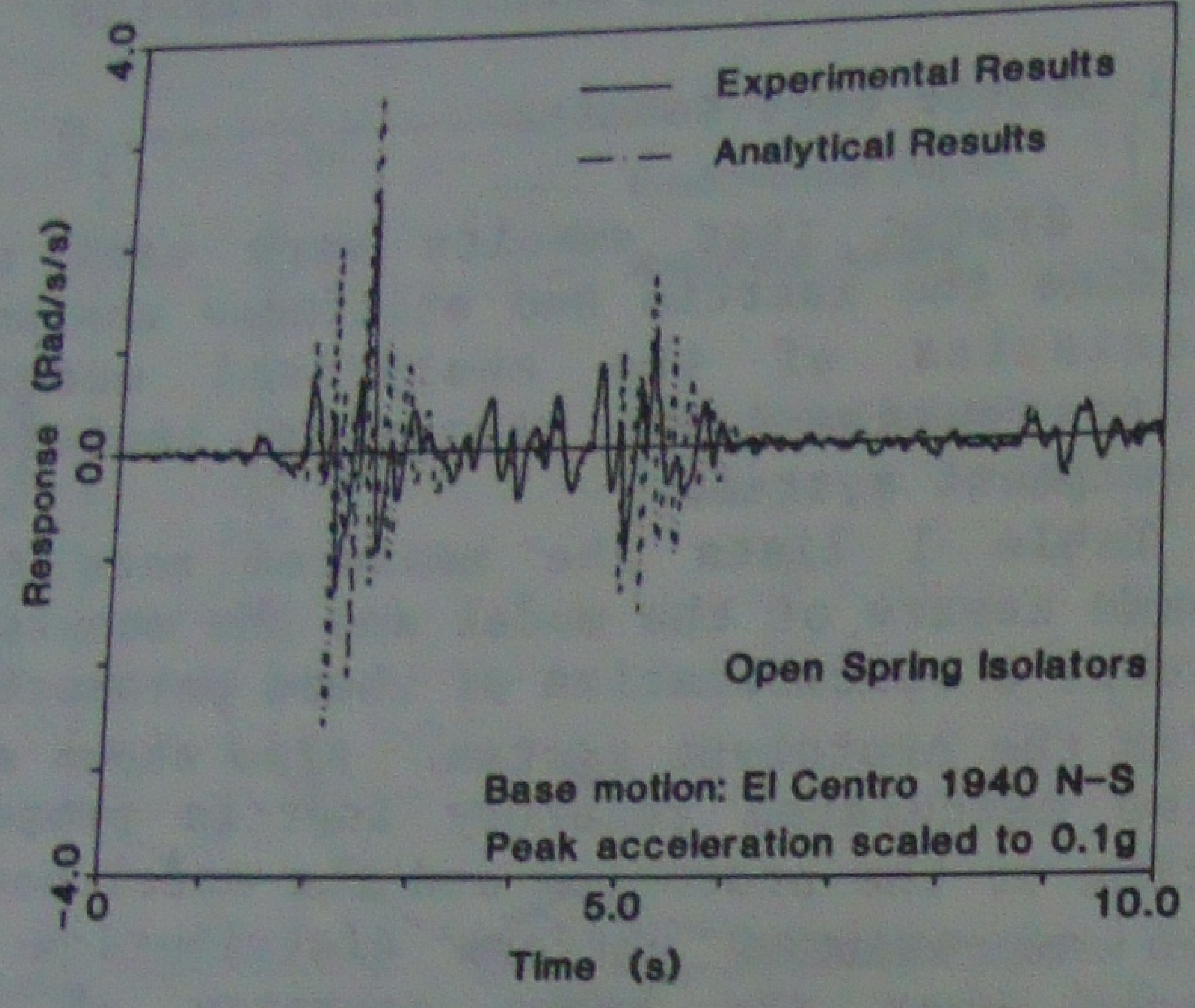

Fig. 14 Predicted and measured rotational acceleration response time-history

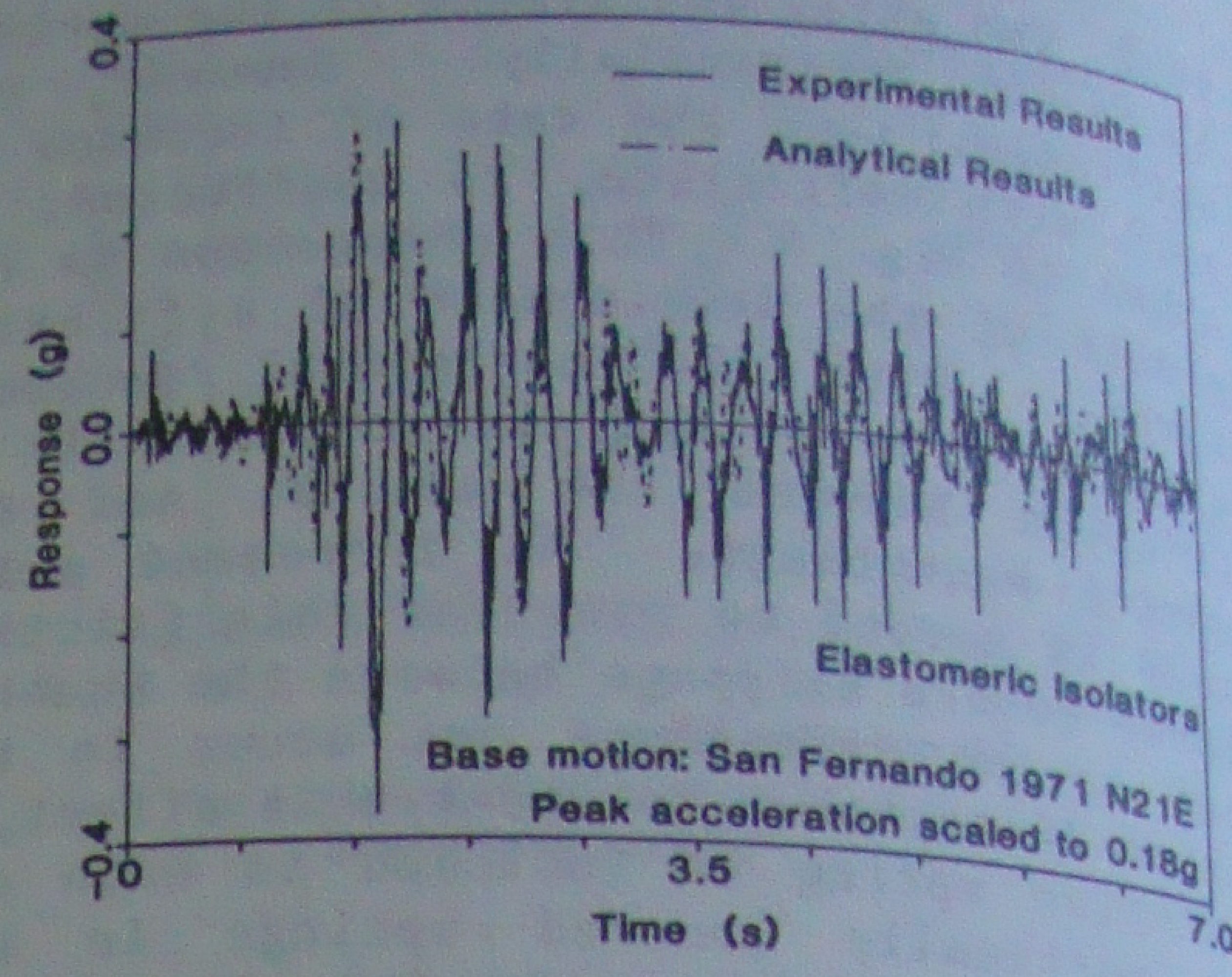

Fig. 15 Predicted and measured horizontal acceleration response time-history

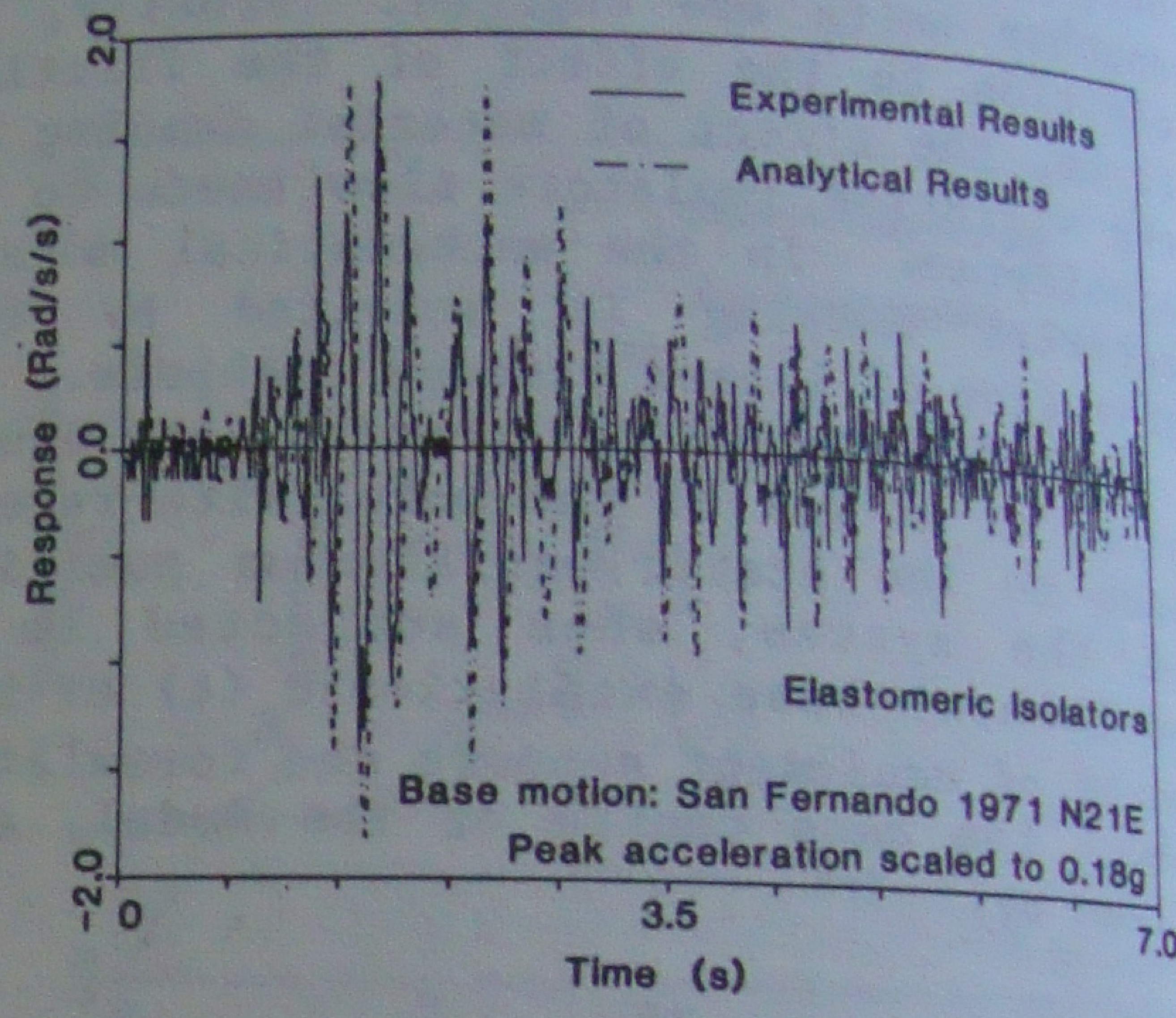

Fig. 16 Predicted and measured rotational acceleration response time-history

Table 1. Model and prototype inertia properties

System	Mass (kg)	Mass moment inertia w.r.t. 2-2 axes through c.of.g. (kg.m.m)*	Mass centre location w.r.t. top of isolators (m)
Model	2136.6	1510.8	.965
Prototype	1852.9	1772.4	.950

*Assumes the equipment mass is uniformly distributed over its shape.

Table 2. Vertical and lateral stiffness of open spring and elastomeric isolators.

Stiffness	Open Spring Isolators		Elastomeric Isolators	
	Axial (kN/m)	Lateral (kN/m)	Axial (kN/m)	Lateral (kN/m)
Initial Stiffness	394	98	1349	1750
Final Stiffness (after gap closure)	3500	839	--	--

In the case of the EI, the measured load-displacement relationships are extremely complicated; they cannot be represented by simple mathematical expressions. However, from the natural frequency system identification tests (under sinusoidal motions) for the prototype model an equivalent linear stiffness could be extracted which gave reasonable acceleration response predictions for different earthquake excitations. An equivalent linear axial stiffness of 1349 kN/m (7700 lb/in) and an equivalent linear lateral stiffness of 1750 kN/m (9900 lb/in) were used in the analytical model.

4.2 Dynamic model identification test results

The dynamic model identification tests yielded decaying sinusoidal curves and frequency response curves for the two vibration isolated equipment systems.

Based on the logarithmic decrement method, the modal damping ratios were found to be 0.103 and 0.105 for the first and third modes of the open spring system. The second (vertical) mode is not excited sufficiently by the essentially uni-directional horizontal motion of the table to secure a damping estimate in this mode. 10% damping was used in all modes in the analytical model.

Figs. 9-10 show the horizontal and rotational acceleration frequency response functions of this system. These responses were measured at the center of mass of the system. The graphs provide a comparison of the results obtained from sinusoidal tests with those determined analytically. In general, the analytical and experimental results agree reasonably well. Based on these results, the natural frequencies of this system were identified as 1.8 Hz and 4.9 Hz.

The modal damping ratios of the elastomeric isolator system as calculated by the logarithmic decrement method were 0.17 and 0.18 for the first and third modes respectively; these modal values were used in the analysis.

Figs. 11-12 show the horizontal and rotational acceleration frequency response function of the elastomeric isolator. These responses also were monitored at the centre of mass of the system. Good agreement between the measured and predicted natural frequencies and response amplitudes was obtained. The two measured natural frequencies of the system were 4.6 Hz and 15 Hz.

The measured and predicted seismic response time-histories of the two vibration isolated equipment systems are presented in Figs. 13-16. Figs. 13-14 give the results for horizontal and rotational acceleration response at the centre of mass of the open spring system when excited by the El Centro N-S (1940) earthquake record. Figs. 15-16 show the corresponding graphs for the elastomeric isolator equipment system when excited by the San Fernando N21E (1971) earthquake record. Good agreement is observed between the measured and predicted response for both systems. The mathematical models used in the analyses leads to response predictions which are slightly more severe than the corresponding values measured experimentally.

5. SUMMARY AND CONCLUSIONS

1. An experimental study of the seismic response characteristics of two different equipment isolation systems was conducted. A model of a prototype air handling unit was used to represent the equipment to be mounted on vibration isolators. The vibration isolators investigated in this study were of the elastomeric type and the open spring type with uni-directional restraint. Static and dynamic tests were carried out to provide information on the nonlinear axial and lateral stiffness of these isolators and on their damping characteristics. The elastomeric isolator exhibited nonlinear stiffness characteristics and developed higher damping values than the open spring isolators. No simple representation could be obtained for these stiffness results. It was found that the stiffness of the open spring isolator can be represented by bi-linear springs.

2. Based on system identification test results an equivalent linear stiffness for the elastomeric isolators can be obtained. Analytical models of the two vibration isolated equipment systems were formulated. A computer program, utilizing time-step analysis and incorporating the system nonlinearity, was developed to solve the equations of motion of the systems. Analytical results, such as frequency response curves and seismic response, were compared with the experimental results to verify the analytical model.

The predicted responses using the

analytical models agreed reasonably well with the response values obtained experimentally. It can be concluded that the analytical models developed were satisfactory for the seismic analysis of the vibration isolated equipment systems considered in this study.

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