SEISMIC QUALIFICATION OF PRESSURE RELIEF VALVES FOR A NEGATIVE CONTAINMENT SYSTEM

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SYNOPSIS

A few nuclear power stations designed and built in Canada (e.g. Pickering Generating Station) utilize a multi-containment arrangement with a common vacuum building to provide a negative pressure containment envelope should a postulated accident occur in one of the containments. In the event that the pressure rises in one of the containments to a certain level, the pressure relief valves which are located in a vacuum duct joining the different containments to the vacuum building will open to relieve the pressure to the vacuum building where a spray system is actuated to condense the incoming steam. These safety related valves should remain intact and operational, and cause no loss of containment following a Design Basis Earthquake (DBE). In this paper the basis of the seismic qualification of these valves by a nonlinear transient dynamic analysis is presented. The nonlinear analyses conducted take into consideration the true nature of the behaviour of the piston during opening and accounts for piston rocking and sway effects, diaphragms folding, eccentricity of the center of mass and center of rigidity as well as the nonlinearities generated by gaps and friction in the system among others. Response quantities such as accelerations, displacements, rotations, diaphragm forces as well as opening time during a design basis earthquake are obtained. The results of the different analyses, as related to the functional operability of the valves, are evaluated and discussed.

RESUME

Un nombre restreint de centrales nucléaires construites au Canada utilisent un vaisseau à vide qui englobe plusieurs récipients. Les clapets, pour réduire la pression dans ces boîtes, doivent pouvoir fonctionner dans l'éventualité d'un séisme majoré (DBE). On présente une analyse dynamique non-linéaire qui tient compte du basculement et du déplacement latéral, en plus de l'excentricité possible de la masse du piston, des écarts possibles dans le système, des effets de frottement et du pliage des diaphragmes.

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INTRODUCTION

A few nuclear power stations designed and built in Canada (e.g. Pickering Generating Station) utilize a multi-containment arrangement with a common vacuum building to provide a negative pressure containment envelope should a postulated accident occur in one of the containments. In the event that the pressure rises in one of the containments to a certain level, the pressure relief valves (PRV) which are located in a vacuum duct joining the different containments to the vacuum building will open to relieve the pressure to the vacuum building where a spray system is actuated to condense the incoming steam.

The negative pressure containment system consists of the Reactor Building, the pressure relief system and the Vacuum Building. The arrangement of the structures for the Pickering Generating Station is shown in Figure 1.

The pressure relief system consists of the pressure relief louvers, the ductwork which interconnects the reactor buildings, and the vacuum building, and 12 pressure relief valves which isolate the atmosphere of the reactor building from that of the vacuum building during normal operating conditions.

During normal operation of the station two pressure zones are maintained within the containment envelope. The reactor buildings and the pressure relief ducts are kept at atmospheric or slightly negative pressure. The vacuum building and the vacuum ducts are maintained at a pressure lower than 14KPa absolute (2.0psia). In the event of any accident in a reactor which causes the pressure to rise to a positive gauge pressure of 4.4KPa (0.6 psi), the pressure relief valves in the pressure relief system will open to relieve the pressure through the ducts to the vacuum building.

The spray system provided in the vacuum building is actuated at a predetermined pressure to condense the incoming steam and cool the air. Fig. (2) shows a typical reactor building pressure transient.

The twelve pressure relief valves are installed in parallel in steel receptacles between the steel vacuum ducts and the concrete relief ducts. These safety related valves have a seismic Category 'B' according to Canadian Codes and Standards (1). Thus, they should remain intact and operational, and cause no loss of containment following a Design Basis Earthquake (DBE). These valves are approximately 6ft in diameter and consist of a housing in which a piston moves up and down. Two rolling neoprene diaphragms serve to prevent leakage and act as guides to reduce friction around the piston during vertical movements. Fig. (3) shows the different components of the Pickering valves.

OBJECTIVE OF INVESTIGATION

The general objective of the investigation is to demonstrate that the pressure relief valves can remain intact and operational and cause no loss of containment during and after a Design Basis Earthquake (DBE). In fact, since the Canadian approach to seismic design is to qualify all the piping and components of the primary and secondary heat transport system inside containment, the valves are not required to function during a DBE. The valves are required only to be operational after a seismic event. The large size of the valves somewhat precludes any possibility for a full scale shaking test. Thus, the basis for the seismic qualification as presented herein is a dynamic analysis followed by an investigation of any possible failure modes due to a seismic event. It is worth mentioning that the valves themselves are tested for leak tightness and operability at the nuclear installation and AECL laboratories. In addition the valves opening and closing performance is recorded experimentally. Thus, the current investigation is limited to the performance of the valves during or after a seismic event as compared to the same performance in the absence of such a seismic event. In other words the objective of the investigation is to quantify the possible changes in performance and operability of the valves during and after a seismic event knowing beforehand that the valves have operated satisfactorily in the absence of a seismic event.

Different analysis techniques have been utilized in the investigation to quantify the performance of the valves. These techniques included linear elastic analysis, nonlinear time-history analysis and complex frequency-domain analysis. Whenever any uncertainty existed in the dynamic properties to be used, a conservative choice of the parameters was made or the analysis was conducted for a range of these parameters. It is concluded from all these analyses as discussed later that the valves are capable of performing their intended function during and after a DBE event and thus are considered to be seismically qualified as a Category 'B' component. The rest of the paper covers the different aspects of the investigation and the analysis techniques used.

SEISMIC DESIGN INPUT MOTIONS

The pressure relief valves are located in a concrete relief duct. The duct is a bridge type structure, 25' high by 20' wide supported on columns about 78' above grade level. The ground motion which may occur at the site will be amplified by this structure. A dynamic analysis for the duct structure has been conducted first. The model used in this dynamic analysis is three-dimensional in nature (Figure 4) and was subjected to three components of the design basis ground motion to be expected at the site. The resulting three components of motion (two horizontal components and a vertical component) were developed at the floor level where the valves are located. These motion components constitute the seismic input for which the valves should be qualified. The acceleration time-histories for the horizontal and vertical design basis ground motions are shown in Figure 5 and 7 respectively. The horizontal design basis ground motion is assumed to occur in the East-West or the North-South direction. The resulting time-histories of longitudinal (East-West), Transverse (North-South), and Vertical floor acceleration are shown in Figure 9, 11 and 13 respectively. To obtain some insights into the characteristics of these motion; Fourier amplitude spectra were obtained for the design basis input ground motions and the resulting design basis floor motions. These spectra are shown in Figures 6, 8, 10, 12, 14. It can be clearly observed from these figures that the design basis input ground motion is broad-band in nature, while the floor motion is somewhat narrow-band in nature.

ELASTIC LINEAR ANALYSIS

A detailed linear dynamic analysis model was first formulated for the whole valve assembly which include the top and the bottom housing, vent pipe, receptacle and its four supporting rods, and the piston as supported laterally by the diaphragms.

The linear elastic analysis was conducted utilizing the SAP IV program (2). The main conclusions of the elastic analysis are:

1) All the components of the valve assembly are rigid. The frequencies calculated are close to 33Hz. Thus, very little amplification of motion is expected. The valve components will experience basically the floor acceleration in a DBE event.

2) The only component which shows some flexibility is the piston itself as supported by the diaphragms. The level of acceleration experienced by the piston depends on the diaphragm properties and the amount of gaps between the piston and the casing.

3) Because of the rigidity of the valve assembly, the input motion to the inertial reference frame in which the piston moves is basically the floor acceleration. Thus an evaluation for the impact forces and the response of the piston is required. The details of this evaluation are discussed later.

EIGENVALUE ANALYSIS OF PRV PISTON

Having established that all the components of the valve assembly are rigid; the dynamic behaviour of the piston as supported laterally by the diaphragms is studied here.

During pressure excursion, the lateral behaviour of the piston changes with the piston location. This is due to the fact that the location of the center of gravity of the piston changes with respect to its center of rigiidity. Thus, even if the rolling diaphragms can be simulated by linear elastic springs, the stiffness matrix is space-dependent as well as time-dependent.

Assuming the center of gravity is located at a height X from the lower spring (Fig. 15) the following stiffness relationships can be derived:

$$\begin{Bmatrix} F_1 \\ F_3 \end{Bmatrix} = \begin{bmatrix} K_1+K_2 & K_1X-K_2(b-X) \\ K_1X-K_2(b-X) & K_1X^2+K_2(b-X)^2 \end{bmatrix} \begin{Bmatrix} u_1 \\ u_3 \end{Bmatrix} \tag{1}$$

or:

$$\{F\} = \left[\mathbb{K}\ (X,b)\right] \{u\} \tag{2}$$

Thus the stiffness matrix of the piston with respect to its center of gravity is dependent on K_1, K_2, b, and X.

For the special case of $K_1 = K_2 = K$ and b is constant (because of the nature of the rolling diaphragms kinematics, the stiffness matrix with respect to the center of gravity becomes:

$$\left[\mathbb{K}\right] = \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix} \tag{3}$$

where:

$$K_{11} = 2K \tag{4}$$

$$K_{12} = K_{21} = Kb\left[\left(\frac{X}{b}\right) - (1-\frac{X}{b})\right] \tag{5}$$

$$K_{22} = Kb^2\left[\left(\frac{X}{b}\right)^2 + (1-\frac{X}{b})^2\right] \tag{6}$$

The inertia matrix of the piston is given by

$$M = \begin{bmatrix} m & o \\ o & I \end{bmatrix} \tag{7}$$

where m is the mass of the piston and I is its mass moment of inertia around the center of gravity.

The frequencies of vibration are given by the solution of the following eigenvalue problem:

$$\text{Det.} = \left| \left[K\right] - \omega^2 \left[M\right] \right| = o \qquad (8)$$

Because $K_{12} = K_{21} \neq 0$, stiffness coupling exists. The effect of this coupling is studied in what follows:

Defining:

$$\omega_H = \text{Horizontal frequency in the absence of coupling} \qquad (9)$$

$$= \sqrt{\frac{K_{11}}{m}}$$

$$\omega_R = \text{Rocking frequency in the absence of coupling}$$

$$= \sqrt{\frac{K_{22}}{I}} \qquad (10)$$

χ = nondimensional coupling parameter which is a quantitative measure of the off-diagonal terms $K_{12} = K_{21}$ of the stiffness matrix

$$\chi = \frac{K_{12}K_{21}}{K_{11}K_{22}} = \frac{(K_{12})^2}{K_{11}K_{22}} \qquad (11)$$

$$\chi = \frac{\left[\left(\frac{x}{b}\right) - \left(1 - \frac{x}{b}\right)\right]^2}{2\left[\left(\frac{x}{b}\right)^2 + \left(1 - \frac{x}{b}\right)\right]^2} \qquad (12)$$

The solution of equation (8) gives the eigenvalue (square of the frequencies) of the coupled system in terms of the conventional uncoupled eigenvalues (square of the frequencies) and the coupling term 'χ'. The solution is given by:

$$\omega_{\substack{1\\2}}^2 = \frac{\omega_H^2 + \omega_R^2}{2} \mp \sqrt{\left(\frac{\omega_H^2 + \omega_R^2}{2}\right)^2 - (1-\chi)\,\omega_H^2\,\omega_R^2} \qquad (13)$$

where:

$$\omega_H^2 = \frac{2K}{m} \qquad (14)$$

$$\omega_R^2 = \frac{kb^2}{I}\left[\left(\frac{X}{b}\right)^2 + \left(1 - \frac{X}{b}\right)^2\right] \qquad (15)$$

$$\omega_R^2 = \xi \frac{kb^2}{I} \tag{16}$$

$$\xi = \left(\frac{X}{b}\right)^2 + \left(1 - \frac{X}{b}\right)^2 \tag{17}$$

$$\frac{\omega_R^2}{\omega_H^2} = \frac{\xi}{2} \left(\frac{b}{r}\right)^2 \tag{18}$$

where r is the radius of gyration of the piston (i.e. $r = \sqrt{\frac{I}{m}}$)

Equation (13) can also be put in the following form:

$$\omega_{\frac{1}{2}}^2 = \frac{\omega_H^2}{2} \left[1 + \frac{\xi}{2} \left(\frac{b}{r}\right)^2\right] \mp \frac{\omega_H^2}{2} \sqrt{\left[1 - \frac{\xi}{2} \left(\frac{b}{r}\right)^2\right]^2 + 2\chi\xi\left(\frac{b}{r}\right)^2} \tag{19}$$

It can be concluded from the above formulation that the basic parameters which govern the extent of coupling between the horizontal and the rocking vibrations are $\frac{X}{b}$ and $\frac{b}{r}$. The effect of ω_H^2 is to modify the coupled frequencies by the same amount as shown in equation (19). This last effect may be important considering that the piston is located in a structure which exhibits well defined frequencies of its own.

In a typical design situation the mass of the piston 'm' is somewhat determined by the requirements for the opening pressure. The stiffness parameter 'K' is somewhat determined by the type and thickness of the diaphragms used and the radius of folding. This may suggest that the values of ω_H is somewhat limited in its typical range of variability. Unfortunately this typical range of variability corresponds to the range of variability of frequencies of civil structures like the one in hand. This simply means that resonance phenomenom between the piston as supported by the diaphragms and the supporting structure may typically occur.

In order to gain some insight into the effect of the different parameters, nondimensional frequency calculations were made. Fig.(16) shows the variation of the conventional horizontal frequency 'ω_H' and the conventional rocking frequency 'ω_R' with the distance $\frac{X}{b}$ and three ratios of $\frac{b}{r}$. It can be observed from the figure that while ω_H is constant with $\frac{X}{b}$, ω_R is variable. For small ratios of $\frac{b}{r}$, the variation of ω_R is small with the ratio of $\frac{X}{b}$. For larger ratios of $\frac{b}{r}$, the variation of ω_R is large with the ratio $\frac{X}{b}$. The figure also shows from the figure the ranges where $\omega_R > \omega_H$ and the ranges where the opposite is true.

When coupling is taken into account, the frequencies of vibration become ω_1 and ω_2 as given by equations (13) or (19). These coupled frequencies were calculated for different ratios of $\frac{b}{r}$ and $\frac{X}{b}$. Typical results are presented in Figure (17) for a ratio of $\frac{b}{r}$ or 1.0. From this figure it can be seen that the lowest natural coupled frequency may be of the translational type or the rocking type. It can also be seen that the effect of coupling is small for $\frac{X}{b}$ ratios around 0.5 and is large for $\frac{X}{b}$ ratios away from 0.5. The frequency behaviour is symmetrical with respect to this characteristic ratio of 0.5.

MODE SHAPES AND PARTICIPATION FACTORS

The coupled frequencies studied before are important as an indication of the possibility of a resonance phenomenon. However, the frequencies alone are not adequate to quantify the potential response of the valve piston. A parametric study is conducted here to quantify the mode shapes and the participation factors.

The piston as supported by the diaphragms exhibits two coupled mode shapes defined as follows:

$$\phi_1 = \text{first mode shape} = \begin{Bmatrix} \phi_{11} \\ \phi_{21} \end{Bmatrix} \tag{20}$$

$$\phi_2 = \text{second mode shape} = \begin{Bmatrix} \phi_{12} \\ \phi_{22} \end{Bmatrix} \tag{21}$$

The mode shapes can be obtained from the solution of the following equation:

$$\begin{bmatrix} (K_{11} - \omega_i^2 m) & K_{12} \\ K_{21} & (K_{22} - \omega_i^2 I) \end{bmatrix} \begin{Bmatrix} \phi_{1i} \\ \phi_{2i} \end{Bmatrix} = 0 \tag{22}$$

Since the horizontal motion is the one of interest, we will choose as customarily is done $\phi_{11} = \phi_{12}$ = unity.

From equation (22) it can be shown that

$$\phi_{2i} = - \frac{(K_{11} - \omega_i^2 m)}{K_{12}} \cdot \phi_{1i} \tag{23}$$

or

$$\phi_{2i} = \frac{-K_{21}}{(K_{22} - \omega_i^2 I)} \cdot \phi_{1i} \tag{24}$$

Since:

$$K_{11} = \omega_H^{\,2} \cdot m \tag{25}$$

$$K_{12} = \sqrt{2\,\xi\,\chi}\;\; kb \tag{26}$$

$$\text{Then } \phi_{2i} = \frac{-(\omega_H^{\,2} - \omega_i^{\,2}) \cdot m}{\sqrt{2\,\xi\,\chi}\;\; kb} \tag{27}$$

$$\text{i.e. } \phi_{2i} = \frac{-(\omega_H^{\,2} - \omega_i^{\,2})}{\omega_H^{\,2}} \cdot \frac{2}{b\sqrt{2\,\xi\,\chi}} \tag{28}$$

and the mode shapes are as follows:

$$\phi_{11} = 1.0 \tag{29}$$

$$\phi_{21} = \frac{-(\omega_H^{\,2} - \omega_1^{\,2})}{\omega_H^{\,2}} \cdot \frac{2}{b\sqrt{2\,\xi\,\chi}} \tag{30}$$

$$\phi_{12} = 1.0 \tag{31}$$

$$\phi_{22} = \frac{-(\omega_H^{\,2} - \omega_2^{\,2})}{\omega_H^{\,2}} \cdot \frac{2}{b\sqrt{2\,\xi\,\chi}} \tag{32}$$

The participation factor for the i^{th} mode is given by:

$$\Gamma_i = \frac{\{\phi_i\}^T \left[M\right] \{J\}}{\{\phi_i\}^T \left[M\right] \{\phi_i\}} \tag{33}$$

where

$$J = \begin{Bmatrix} 1 \\ 0 \end{Bmatrix} \text{(since no rocking motion is applied)} \tag{34}$$

Carrying out the substitution,

$$\Gamma_1 = \frac{\phi_{11} m}{\phi_{11}^{\,2} m + \phi_{21}^{\,2}\, I} \tag{35}$$

$$\Gamma_1 = \frac{1}{\phi_{11}} \cdot \frac{1}{1 + \left(\dfrac{\phi_{21}}{\phi_{11}}\right)^2 \cdot \dfrac{I}{m}} \tag{36}$$

$$\Gamma_1 = \frac{1}{1 + \frac{2\left(\frac{r}{b}\right)^2}{\xi\chi}\left[1 - \left(\frac{\omega_1}{\omega_H}\right)^2\right]^2} \tag{37}$$

Similarly

$$\Gamma_2 = \frac{\phi_{21} m}{\phi_{21}^{\,2}\, m + \phi_{22}^{\,2}\, I} \tag{38}$$

$$\Gamma_2 = \frac{1}{\phi_{21}} \; \frac{1}{1 + \left(\frac{\phi_{22}}{\phi_{21}}\right)^2 \frac{I}{m}} \tag{39}$$

$$\Gamma_2 = \frac{1}{1 + \frac{2\left(\frac{r}{b}\right)^2}{\xi\,\chi}\left[1 - \left(\frac{\omega_2}{\omega_H}\right)^2\right]^2} \tag{40}$$

It is clear from the above derivation that the two basic parameters which govern the participation of the modes are $\frac{b}{r}$ and $\frac{X}{b}$. To demonstrate the variability in Γ_1 and Γ_2, Fig. (18) was calculated for the case of $\frac{b}{r} = 1.0$.

Since $\phi_{11} = \phi_{12} = 1.0$, it is worth pointing out that both Γ_1 and Γ_2 are positive and that $\Gamma_1 + \Gamma_2$ is equal to unity. Thus the linear response of the piston does not exceed the maximum ordinate of the Floor Response Spectrum if the system behaves as a linear elastic system.

To demonstrate the previous concepts, the results of analysis of the piston in its closed position are presented here:

$m = 8.0$ lb.in^{-1}. sec^2

$I = 5832.0$ lb.in.sec^2

$h_1 = 16.0$ in.

$h_2 = 43.0$ in.

$h_{c.g} = 24.0$ in.

$K = 3000.$ lb/in.

$b = 27.0$ in.

$X = 8.0$ in.

$\frac{X}{b} = 0.2963$

$r = 27.0$

$\frac{b}{r} = 1.00$

The frequencies obtained are:

$f_1 = 2.12314$ Hz (rocking)

$f_2 = 4.4843$ Hz (swaying)

The mode shapes are given by:

$$\phi_1 = \begin{Bmatrix} 1.0 \\ .1387 \end{Bmatrix} \qquad \phi_2 = \begin{Bmatrix} 1.0 \\ -.0099 \end{Bmatrix}$$

$\Gamma_2 = 0.9333$

$\Gamma_1 = 0.0667$

$\Gamma_1 + \Gamma_2 = 1.0$

Thus it can be concluded that although the rocking frequency is lower than the sway frequency, the response generated is still primarily a sway response. This behaviour is due to the fact that the input motion is predominantly of the sway or horizontal type which does not excite a substantial rocking response if the rocking and the sway frequencies are widely separated.

COMPLEX FREQUENCY DOMAIN ANALYSIS

To obtain the response of the piston as supported by the diaphragms in the linear elastic model, described before, the complex frequency-domain solution was utilized as an alternative to the conventional time-domain solutions. The transfer functions and the results obtained for the piston response when subjected to the longitudinal duct motion are shown in Figures (19) to (22). The maximum response acceleration obtained in this analysis is 54.57 in/sec^2 for a floor acceleration of 44.24 in/sec^2 (amplification factor of 1.23). This horizontal acceleration will generate a total force in the upper and the lower diaphragms of 437 lb. This horizontal force when applied to the top housing (approximately weighs 6450 lb) does not affect the integrity of the valve assembly and in the actual installation may be resisted by the natural friction available (the required coefficient of friction to resist this force is less than 7%).

NONLINEAR DYNAMIC ANALYSIS OF PRV

To achieve an understanding of the behaviour of the PRV piston under the simultaneous action of a pressure excursion and a DBE event, a nonlinear dynamic analysis model which account for impact and friction was formulated. It has been established before that all the components of the PRV are rigid in nature, and thus experience the duct floor accelerations. The PRV piston located inside the housing will vibrate horizontally impacting the housing through the diaphragms in the horizontal direction. The rolling diaphragms in the Pickering PRV are intended to have a 1/8" gap all around between the piston and the housing. In addition due to the fact that the piston is laterally supported somewhat at random in the circumferential direction by the rolling diaphragms the effective gap size may be greater than the nominal intended gap of 1/8". Because of this uncertainty in the gap size as well as the uncertainty in the actual lateral stiffness of the rolling diaphragms, the nonlinear analysis is conducted for two extreme cases. The first case, Case (I) has no gaps and the second Case (II) has relatively large gaps of 0.5 in. on each side.

The actual behaviour of the piston should fall somewhere between these two extreme cases. Due to the fact that impact forces are generated in a DBE event, the opening of the valve may be delayed because of the corresponding friction forces which may be developed. The mathematical model used, the degrees of freedom, and the notations are shown in Figure (23). The analysis is based on the following assumptions:

a) The piston can be treated as a rigid body.
b) Pressure excursion is known beforehand and does not get affected either by the piston or its motion.
c) The piston housing is rigid (established before by the linear elastic analysis).

The equations of motion for the piston vibrating in an inertial reference frame can be written as:

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + \{F\} = -[M]\begin{Bmatrix}\ddot{u}_H\\ \ddot{u}_V\\ \ddot{u}_{HV}\end{Bmatrix} + \begin{Bmatrix}o\\ p(t)\\ o\end{Bmatrix}.A \qquad (41)$$

where:

M = Mass matrix = $\begin{bmatrix} m & o & o \\ o & m & o \\ o & o & I \end{bmatrix}$

$\{u\},\{\dot{u}\},\{\ddot{u}\}$ = Relative displacement, velocity and acceleration vectors respectively

$\{F\}$ = Force vector which includes spring forces as well as frictional forces

$\ddot{u}_H$ $\ddot{u}_V$ and $\ddot{u}_{HV}$ are the duct floor horizontal, vertical, and rocking accelerations respectively

A = effective area which is subjected to the accident pressure excursion

p(t) = effective applied pressure time-history (the resultant upward pressure after subtracting gravity effects)

$[C]$ = damping matrix (Reference 4)

$$= [M] [\emptyset] [\beta] [\emptyset]^T [M] \tag{42}$$

Where:

$[\emptyset]$ = A matrix whose columns are the eigenvectors of the initial elastic system (normalized with respect to M)

β_i = Percentage of critical damping in the i^{th} mode

ω_i = Circular frequency of the i^{th} mode

$[\beta]$ = Diagonal matrix with elements $2\beta_i\omega_i$.

The previous set of coupled equations of motion are marched in the time domain and the time history of any response quantity is obtained. The marching scheme utilizes the explicit impulse acceleration method (3,4) (known alternatively as the central difference method). The nonlinear analysis takes into consideration the sway and the rocking effects of the piston, the eccentricity of the center of mass and the center of rigidity, gaps, friction among other effects. Table (1) gives the different response quantities of the PRV piston for the two extreme cases discussed before (with no gaps and with gaps of 0.5 in.). It is clear from the table that due to the impact phenomenon the response accelerations, displacements, and forces are very high compared to a linear analysis or a system with no gaps. Most important the accelerations experienced by the piston vibrating in an environment with a 0.5 in. gap are almost four times those of a system with no gaps. In addition the forces which the rolling diaphragms experience are almost 8 times more because of impact. The maximum force generated on either the upper or lower diaphragms was found to be 1437 lb. This force although very high compared to what would be predicted by a linear elastic analysis should not constitute any engineering concern as far as the structural integrity of the valve assembly. Figure 24, 25, 26 and 27 show the nature of the diaphragm reactions for a system with no gap versus a system with gaps.

To obtain a quantitive assessment of the effects of these impact forces on generating frictional forces which may delay the opening of the valve, the model was subjected to the longitudinal motion up till attainment of peak level (approx. 7.0 sec). A pressure which would cause a net upward constant force of 2000.0 lb was then applied and was maintained constant. This constant force would be equivalent to approximately 0.5psi. Since in the design concept a 0.65psi is needed to overcome the self weight of the piston, the total pressure differential is approximately 1.15psi. Obviously a higher differential would lead to a lower opening time. Thus the above pressure was considered to be the minimum pressure which can be realistically available to open the valve under a DBE event. The coefficient of friction was assumed to be 0.30 and a coefficient of restitution was assumed as 1.0. Both coefficients are believed to be very conservative. The results of the analysis indicated that the piston motion in the vertical direction affected the lateral response very little. The opening time for a travel distance of 42.0 in. under 1.15psi as determined by the analysis under a DBE event is 0.61 sec. The time the piston would normally take to travel 42.0 in. under a net upward force of 2000 lb. is 0.57 sec. Thus the effect of the DBE event is a delay in opening of 0.04 sec. This delay in opening is very small and does not constitute any harmful consequences. This delay in opening time is equivalent to approximately 7% increase in the regular opening time. Since the opening time is relatively small, such a small increase is not important from the engineering point of view.

VALVE PERFORMANCE IN A DBE EVENT AND CONCLUSIONS

Based on the previous studies, the following can be concluded as far as the valve performance in a DBE event:

a) The DBE accelerations generated in all the valve components apart from the piston are equal to the duct floor accelerations (approximately 0.12g horizontal and 0.12g vertical). The level of accelerations does not cause any disruption of seals. A tilt-up of the valve housing will not occur and a loss of containment anywhere will not happen. The stresses generated under these accelerations considering the rugged nature of the different components are very small and of academic nature.

b) The DBE event when combined with a pressure excursion causes some delay in opening. Using very conservative assumptions, (e.g. 1% damping, coefficient of restitution of unity, gap size of 0.5 in. on each side, coefficient of friction of 0.30 and a net opening pressure of 0.5 psi only). The delay in opening amounted to less than 10% of the regular opening time. Such a very small delay neither affects the concept of the negative containment system nor the results of the LOCA analysis already conducted.

c) When considering the most unfavourable dynamic conditions for possible 'ratchet' effects (e.g. free gaps of 0.5 in. on each side and a spring constant available of 3000 lb/in. only for each diaphragm after impact. It was found that the piston does rock and does impact the housing. This rocking was found not to cause a pronounced delay in opening. This is primarily because the vertical travel velocity did not change substantially by the frictional forces. The maximum impact forces at the level of a diaphragm considering what is believed to be the most unfavourable conditions was found to be 1437.0 lb.The maximum instantaneous total impact force on the housing was found to be 1606 lb. If no gap was assumed rather than the 0.5 in. gap on each side, this force drops to 437 lb. These instantaneous impact forces which change signs with a somewhat high frequency content, when applied to a rigid housing weighing approximately 6540 lb, do not endanger the structural integrity of the valve assembly. In addition these forces do not cause any damage to the rolling diaphragms which are believed to have a tremendous capacity to absorb impact forces.

The final overall conclusion is that the functional operability of the valve is not affected by a DBE event and the valves can be certified to be qualified as seismic Category 'B' components.

Table 1

Effect of Gaps on Response

	Quantity of Interest	System with No Gaps	System with Gaps (0.5" Each Side)	Ratio
1.	Max. Input Horizontal Acc. (in/sec^2)	44.24	44.24	1.0
2.	Max. Response Horizontal Acc. (in/sec^2)	54.47	200.49	3.67
3.	Max. Response Rotational Acc. (Rad/sec^2)	0.83	4.67	5.63
4.	Max. Response Horizontal Disp. (in)	0.10	0.81	8.10
5.	Max. Response Rotation (Rad)	0.0057	0.056	0.82
6.	Max. Horizontal Force on Piston (lb)	437.	1606.	3.67
7.	Max. Moment on Piston (in.lb)	4824.7	27296.	5.66
8.	Max. Force in Lower Spring (lb)	417.3	1075.5	2.58
9.	Max. Force in Upper Spring (lb)	185.6	1436.6	7.74

*Input Motion is the Duct Longitudinal Horizontal Motion with a duration of 30.0 sec.

Mass of Piston 'm'	= 8.0 lb.in^{-1} sec^2
Moment of inertia 'I'	= 5832.0 lb.in. sec^2
h_1	= 16.0 in. (See Figure 15)
h_2	= 43.0 in.
K	= 3000. lb/in
H_{cg}	= 24 in
β	= 1%
Damping type	= Modal Damping (equation 42)

REFERENCES

1. 'Seismic Qualification of CANDU Nuclear Power Plants', Canadian Standard Association (CSA) Seismic Design Standard, CSA-N289 (in preparation).

2. Bathe, K.J., et al 'SAP IV: A Structural Analysis Program for Static and Dynamic Response of Linear Systems' Report No. EERC-73-11 June 1973, Berkeley, California.

3. Biggs, J.N., 'Introduction to Structural Dynamics' McGraw-Hill Book Company.

4. Aziz, T.S., 'Inelastic Dynamic Analysis of Building Frames' Publication No. R76-37, M.I.T. Cambridge, Massachusetts.

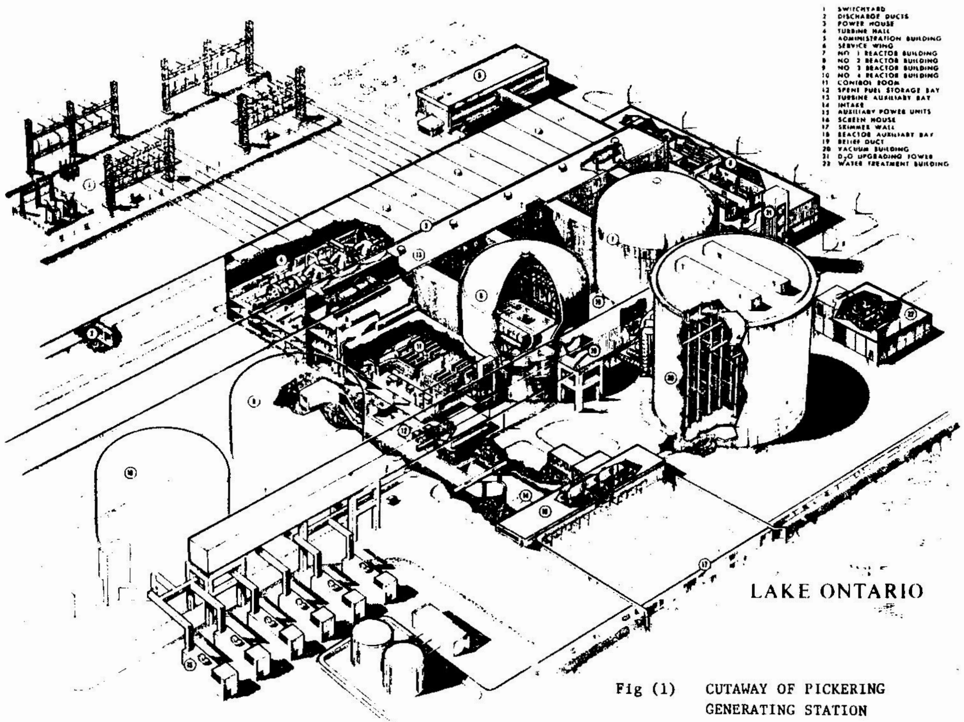

Fig (1) CUTAWAY OF PICKERING GENERATING STATION

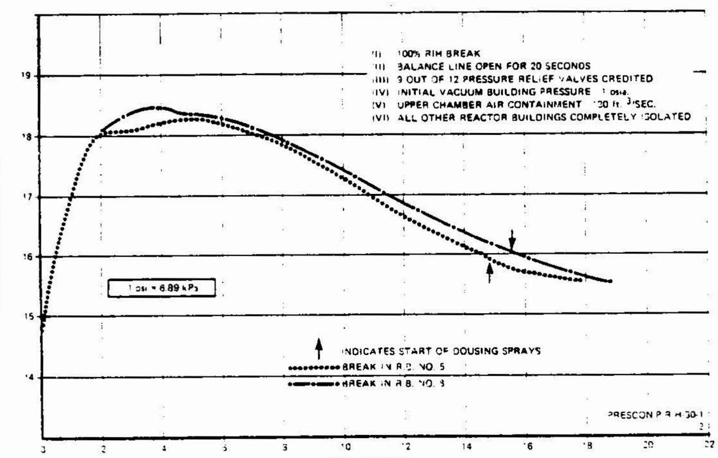

Fig (2) REACTOR BUILDING PEAK PRESSURE TRANSIENT

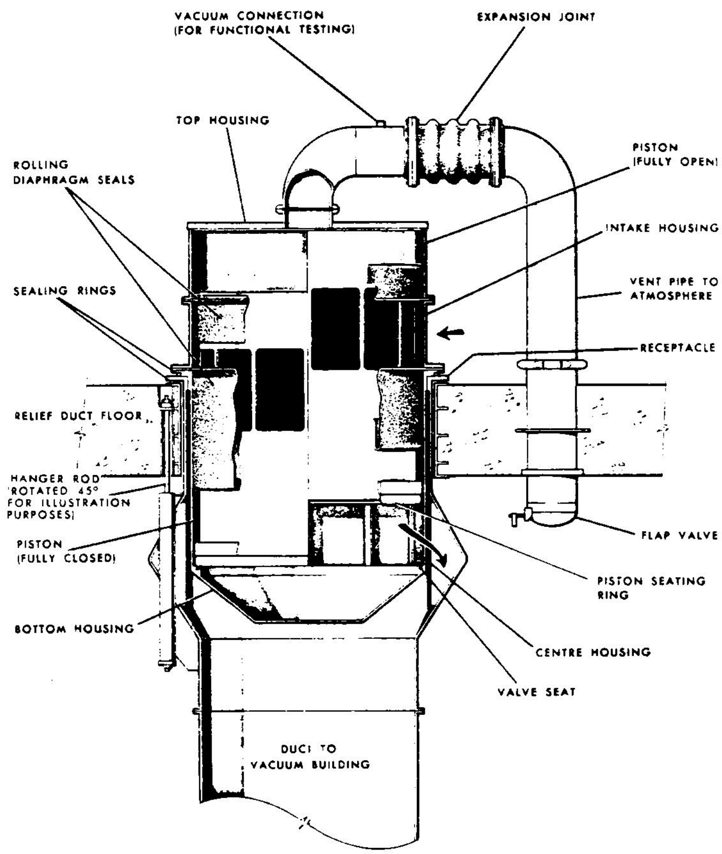

Fig (3) PRESSURE RELIEF VALVES TO VACUUM BUILDING

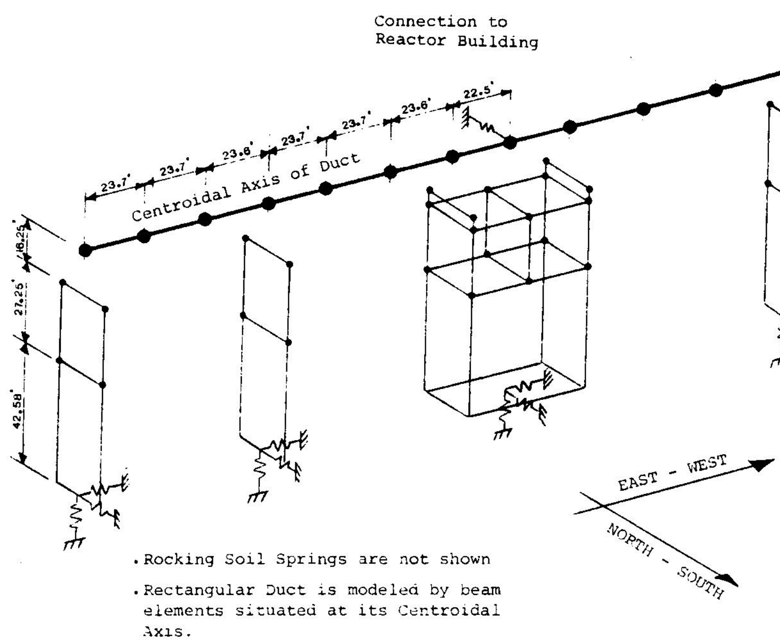

Fig (4) PRESSURE RELIEF DUCT DYNAMIC MODEL

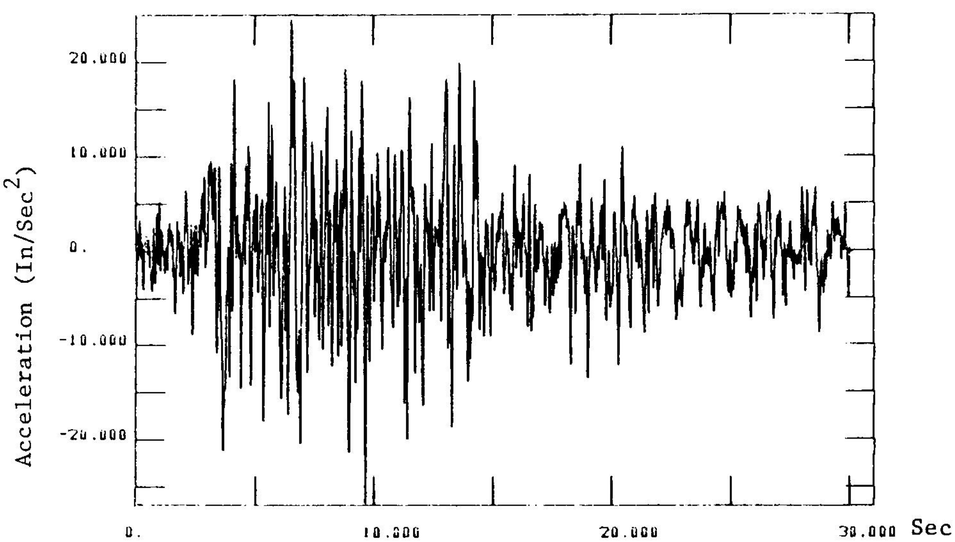

Fig (5) DBE GROUND MOTION - HORIZONTAL

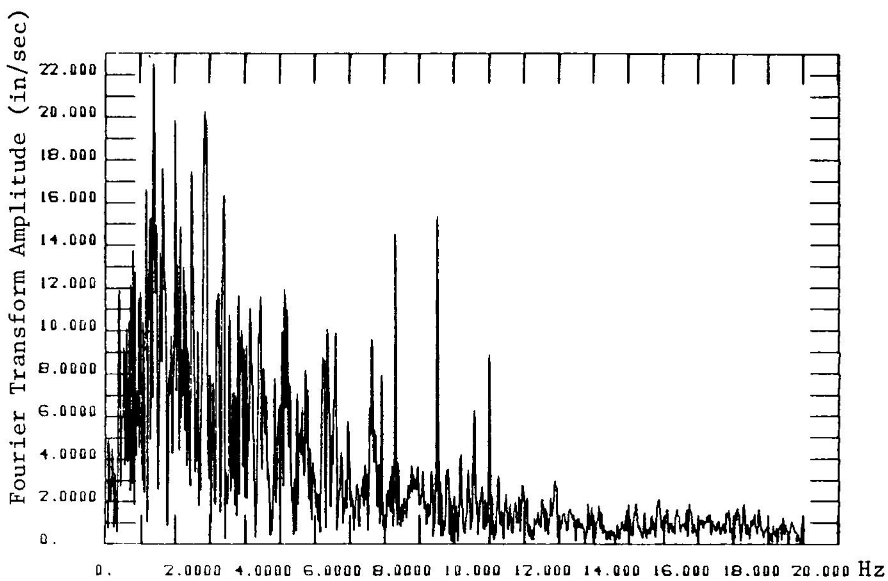

Fig (6) DBE - GROUND MOTION - HORIZONTAL

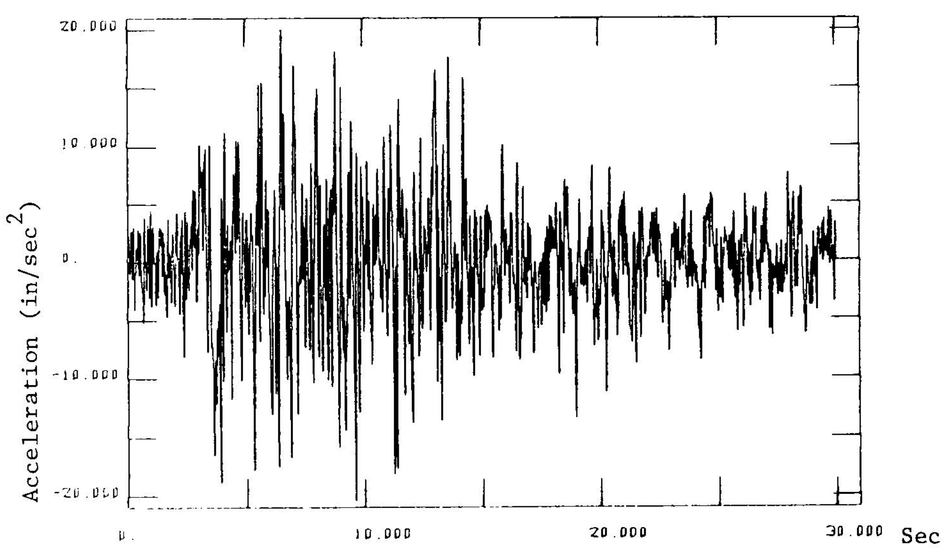

Fig (7) DBE GROUND MOTION - VERTICAL

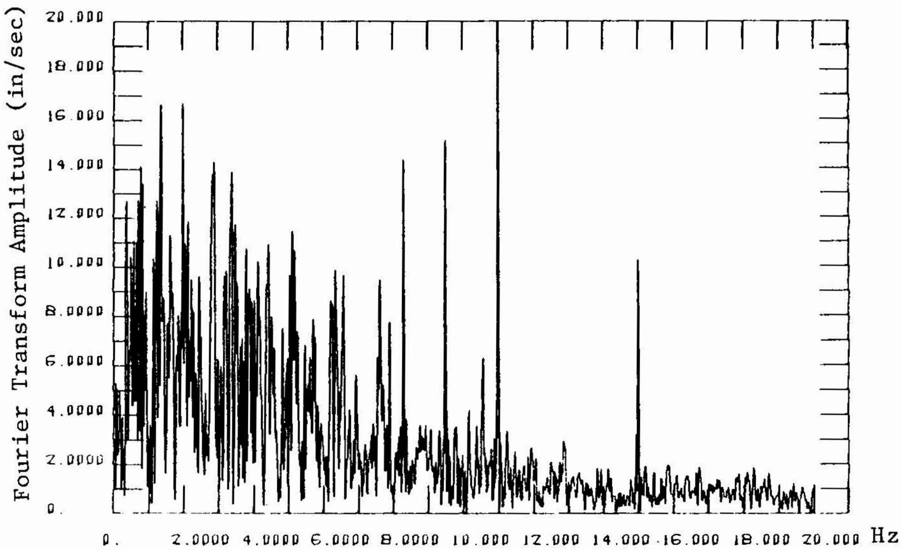

Fig (8) DBE GROUND MOTION - VERTICAL

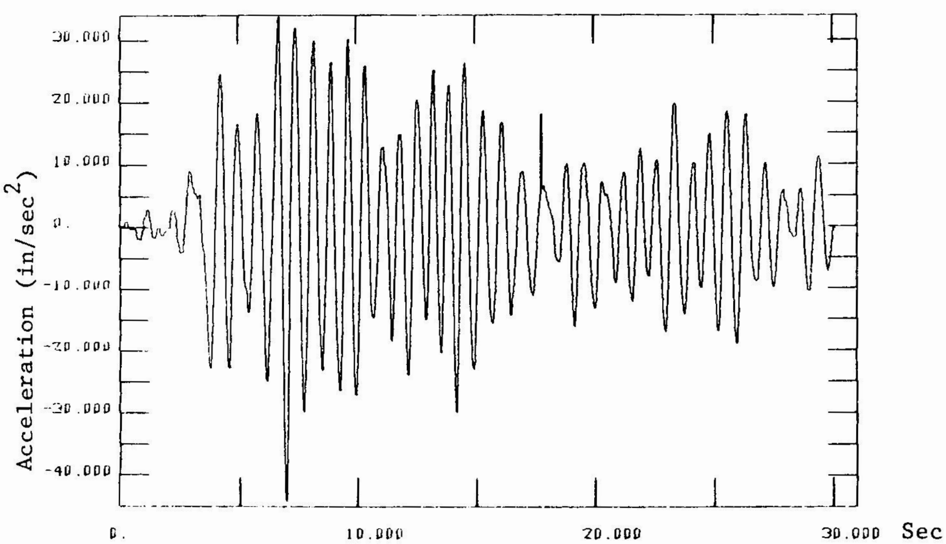

Fig (9) DUCT MOTION - LONGITUDINAL

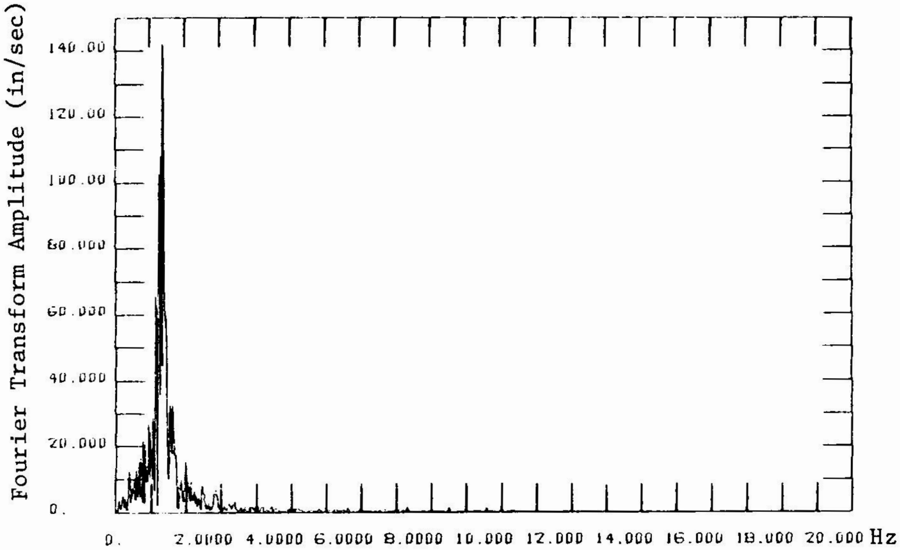

Fig (10) DUCT MOTION - LONGITUDINAL

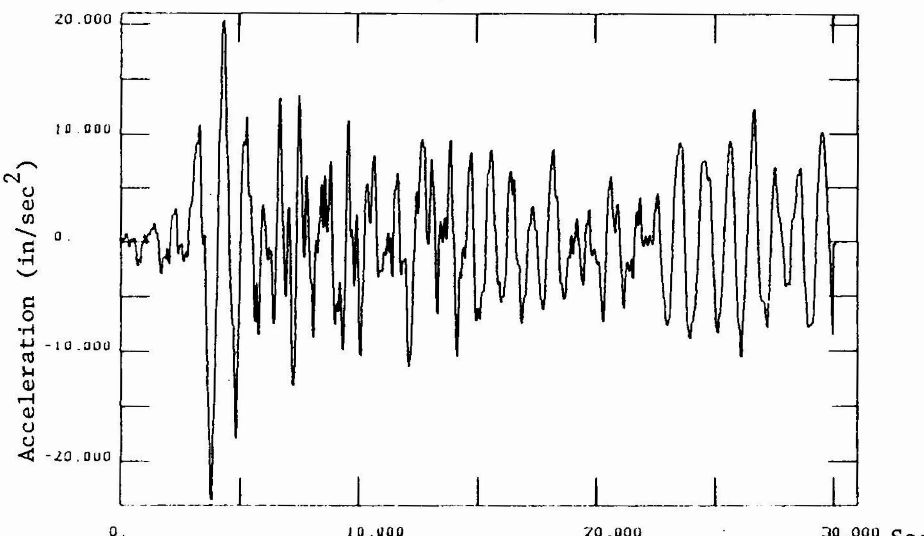

Fig (11) DUCT MOTION - TRANSVERSE

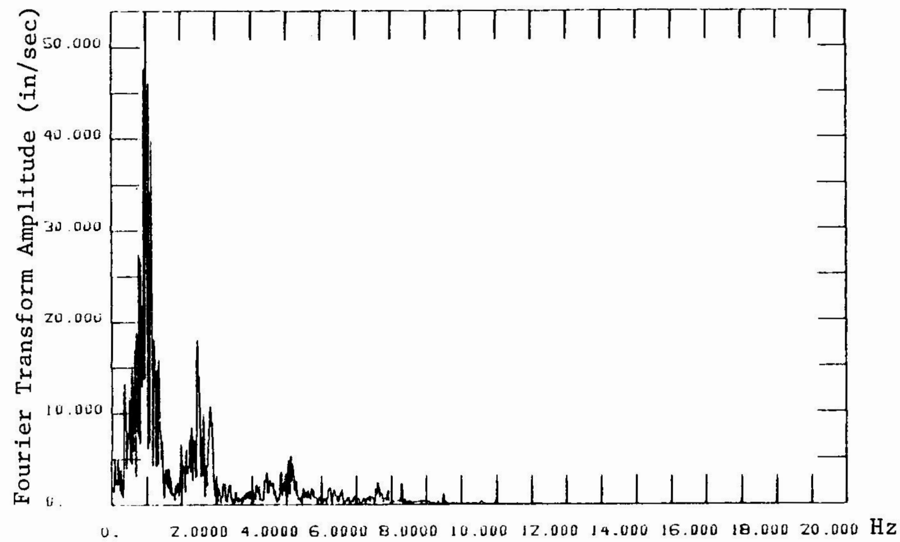

Fig (12) DUCT MOTION - TRANSVERSE

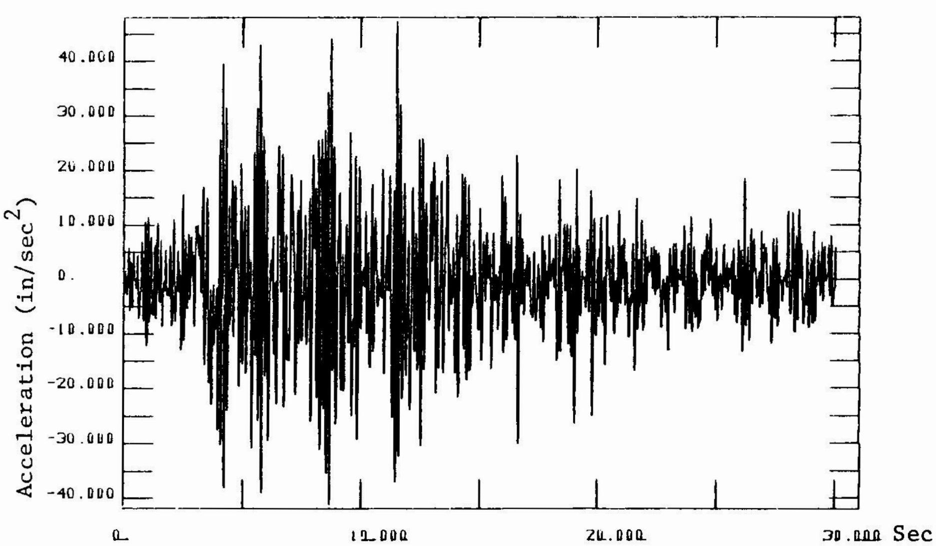

Fig (13) DUCT MOTION - VERTICAL

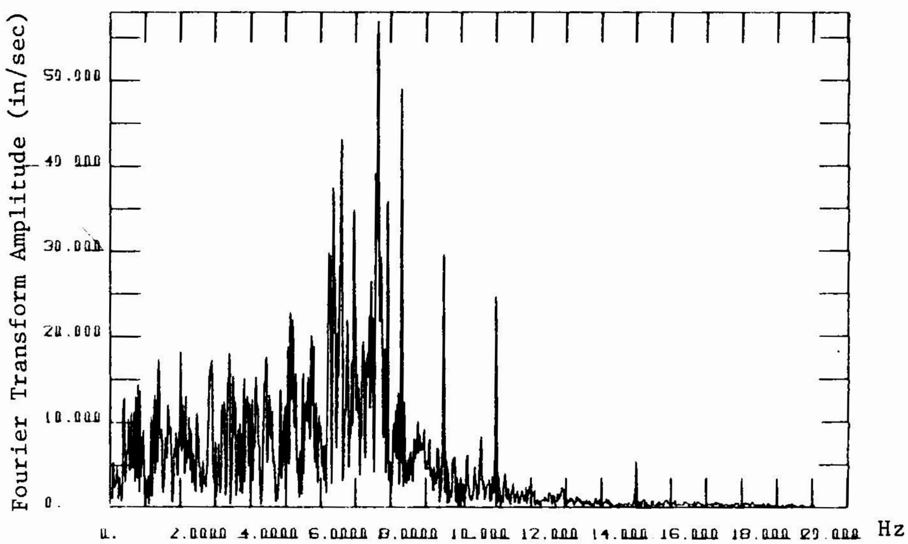

Fig (14) DUCT MOTION - VERTICAL

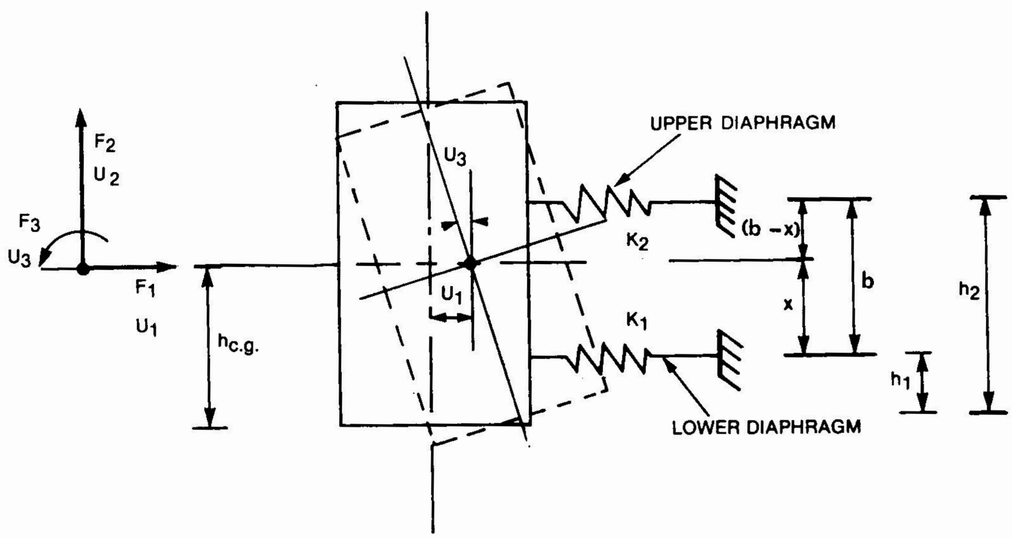

FIGURE (15) SIGN CONVENTIONS UTILIZED IN THE ANALYSIS

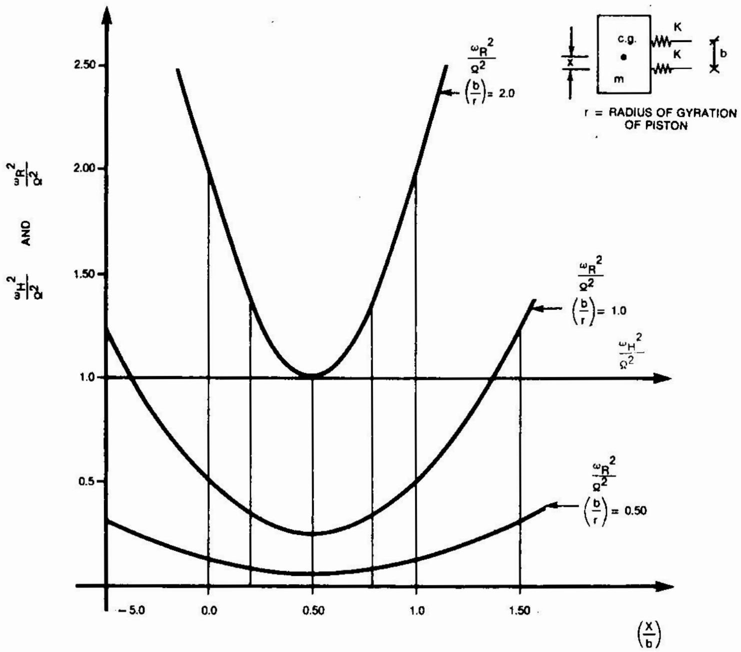

FIGURE 16 VARIATION OF $\frac{\omega_H^2}{\Omega^2}$ AND $\frac{\omega_R^2}{\Omega^2}$ WITH $\frac{x}{b}$

$\Omega^2 = \frac{2K}{m}$

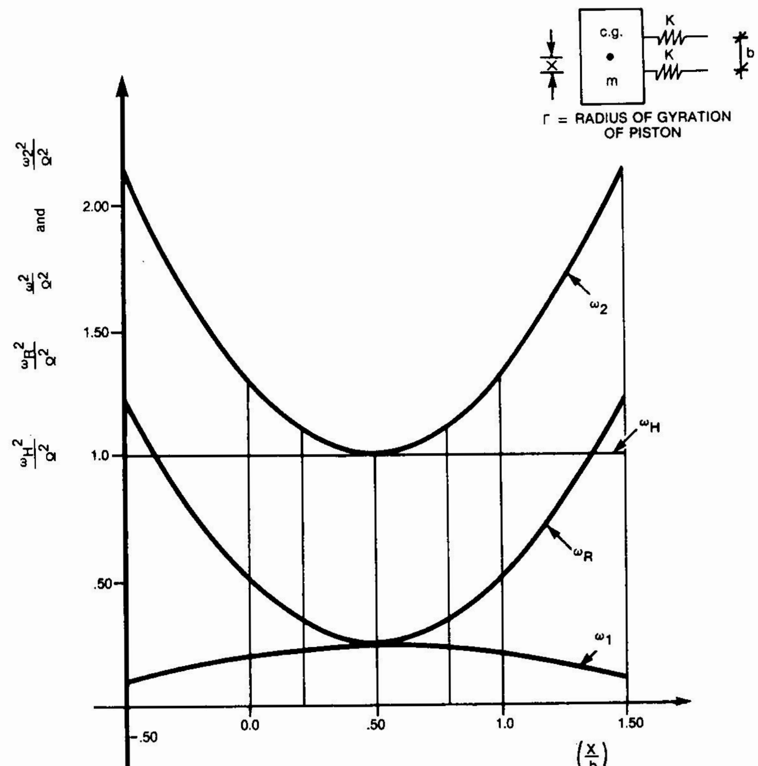

FIGURE 17 FREQUENCY BEHAVIOR ($\frac{b}{r}$ = 1.0)

$\Omega^2 = \frac{2K}{m}$

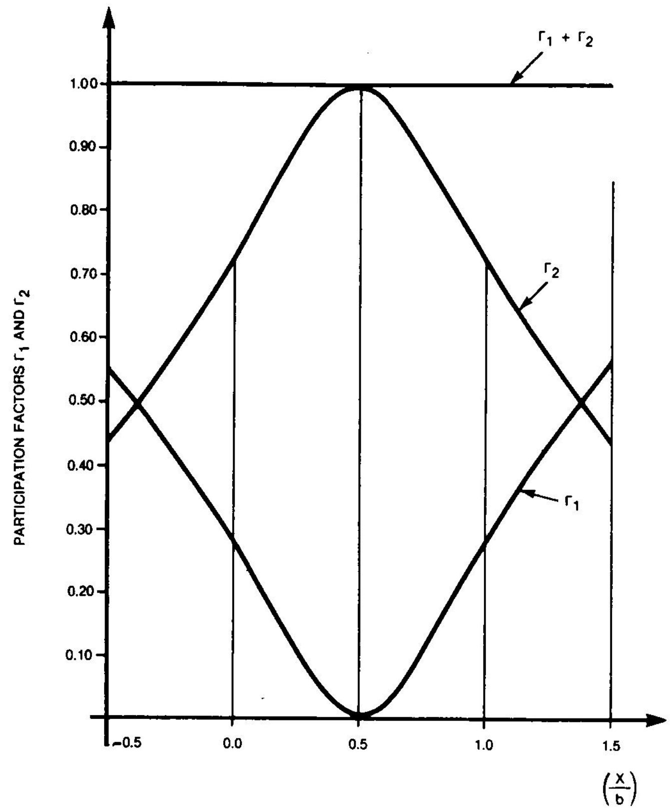

FIGURE 18 BEHAVIOUR OF PARTICIPATION FACTORS Γ_1 AND Γ_2

$\left(\frac{b}{r}\right) = 1.0$

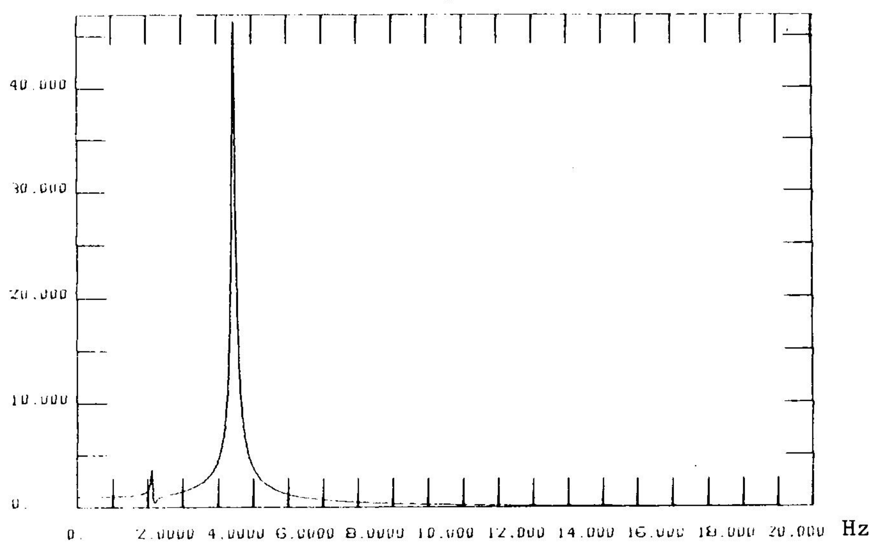

Fig (19) ACCELERATION TRANSFER FUNCTION - AMPLITUDE

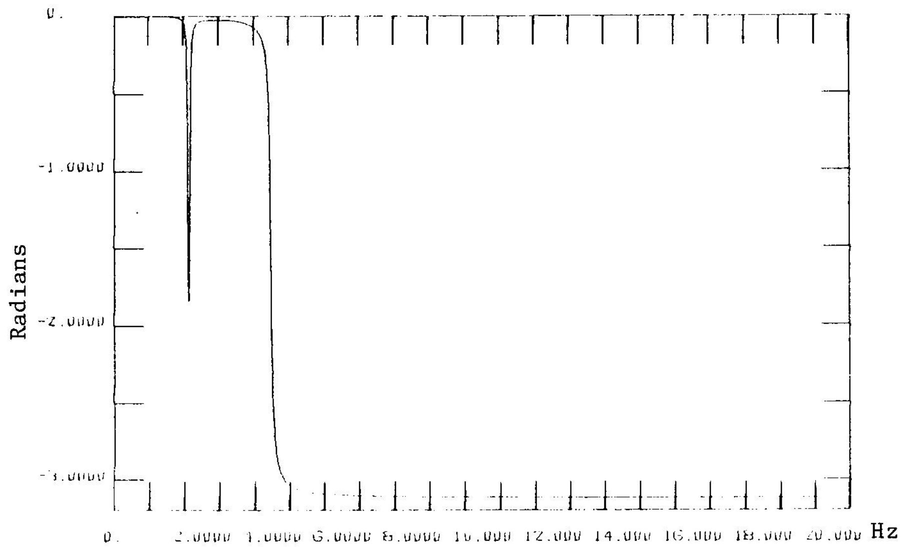

Fig (20) ACCELERATION TRANSFER FUNCTION - PHASE ANGLE

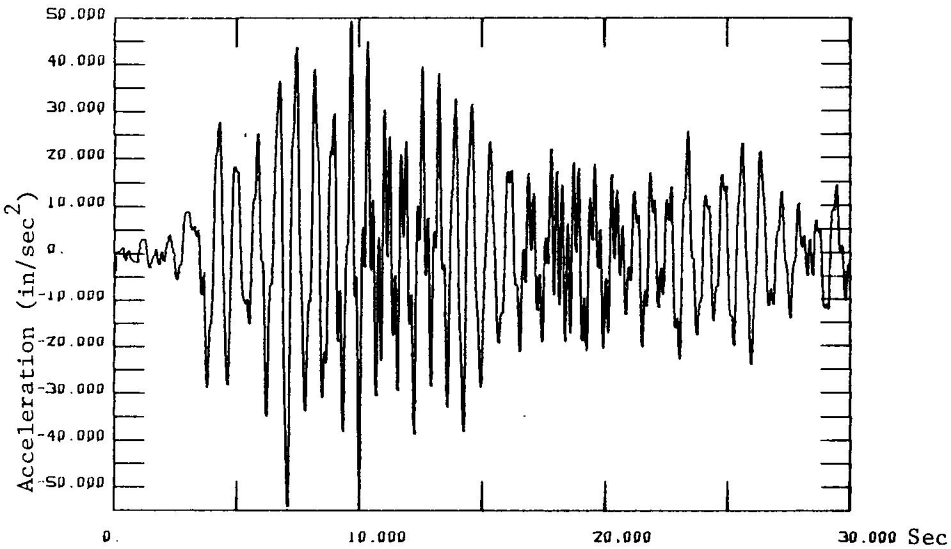

Fig (21) PISTON RESPONSE - LONGITUDINAL

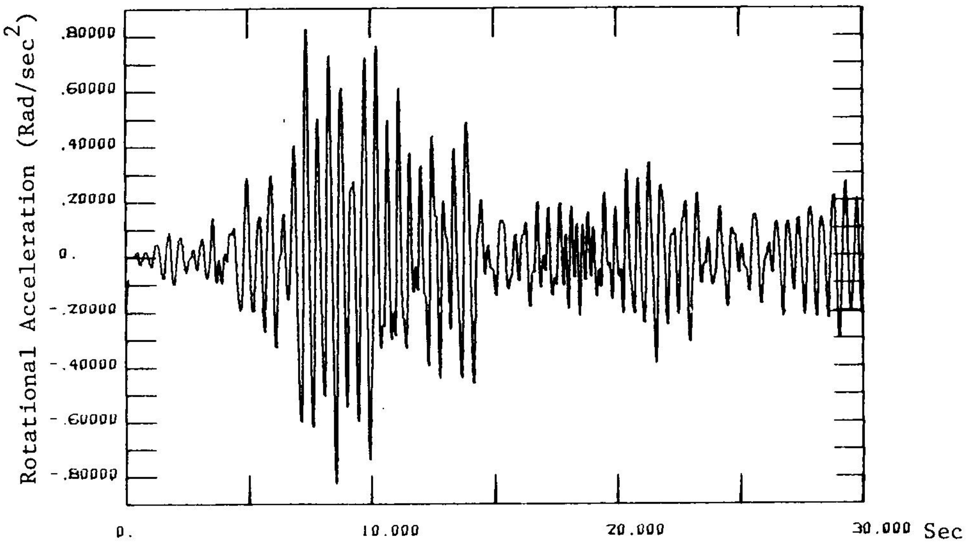

Fig (22) PISTON RESPONSE - LONGITUDINAL

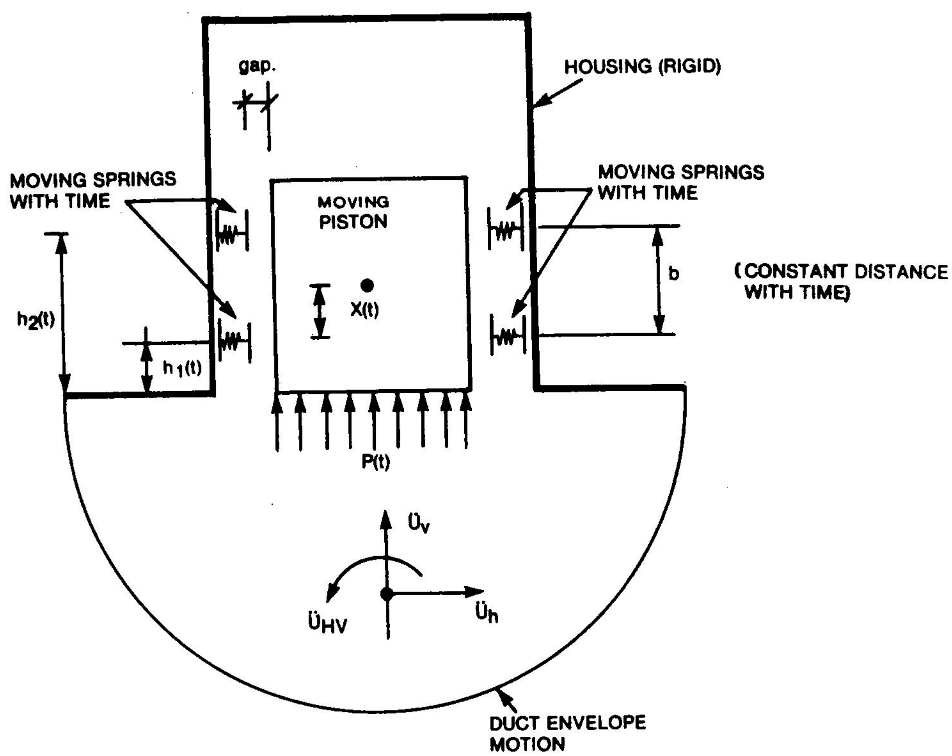

FIGURE (23) NONLINEAR DYNAMIC MODEL

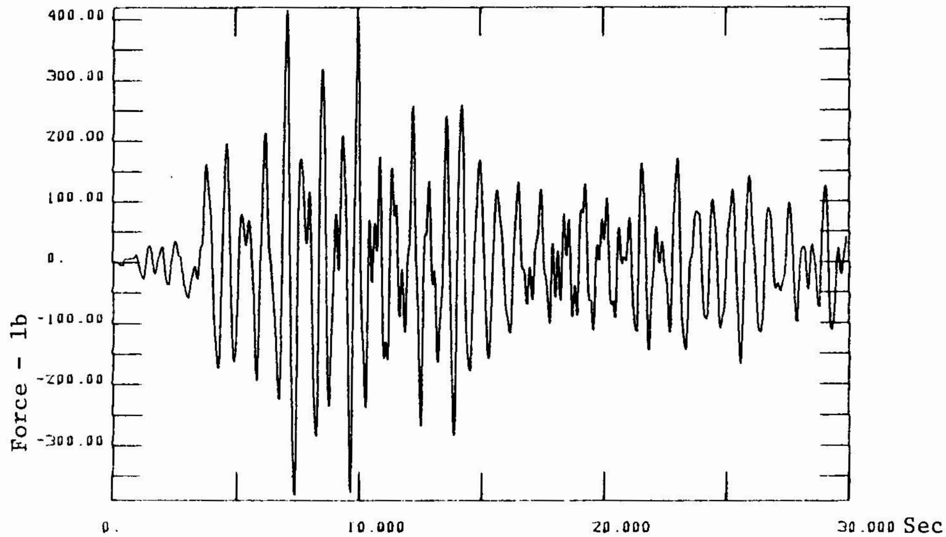

Fig (24) LOWER SPRING FORCE TIME-HISTORY (No gap, β=1%)

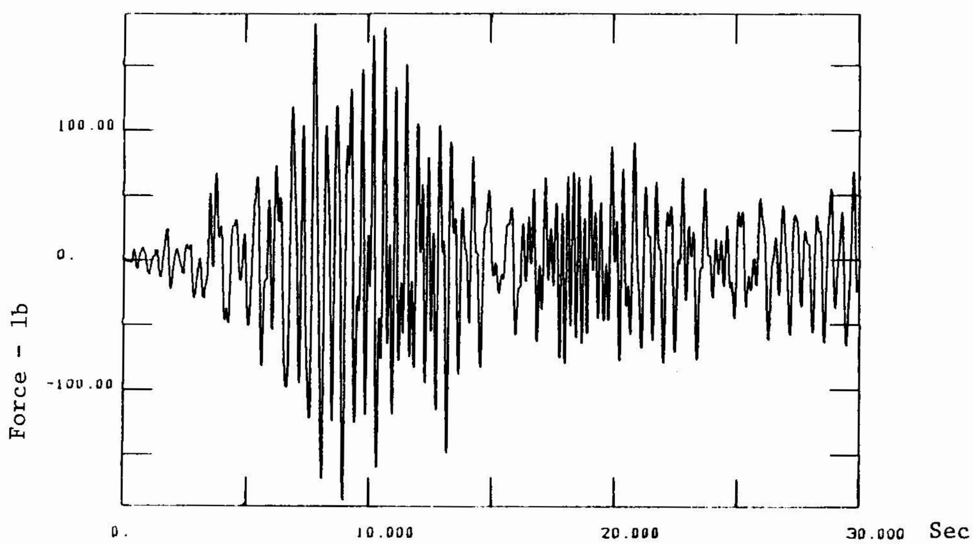

Fig (25) UPPER SPRING FORCE TIME-HISTORY (No gap, β=1%)

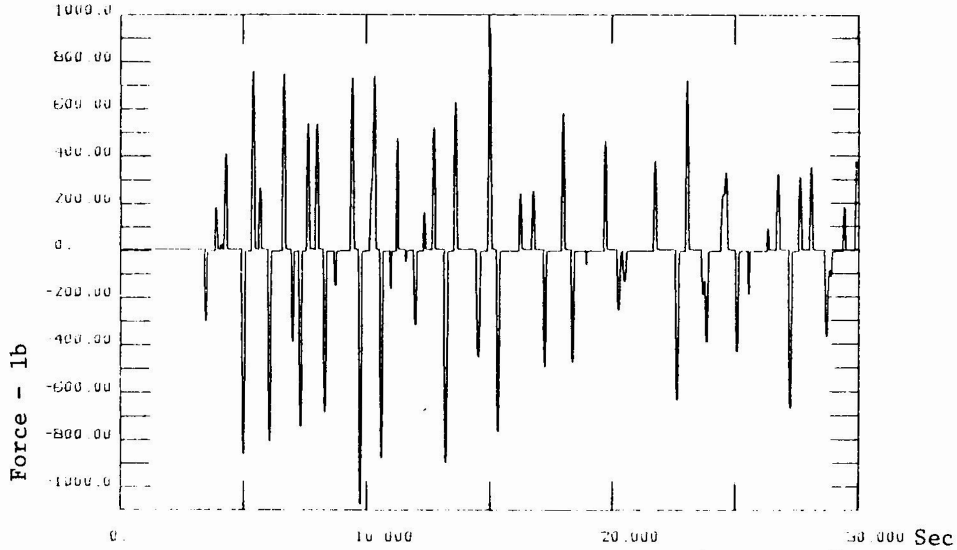

Fig (26) LOWER SPRING FORCE TIME-HISTORY (Gap = 0.5", β=1%)

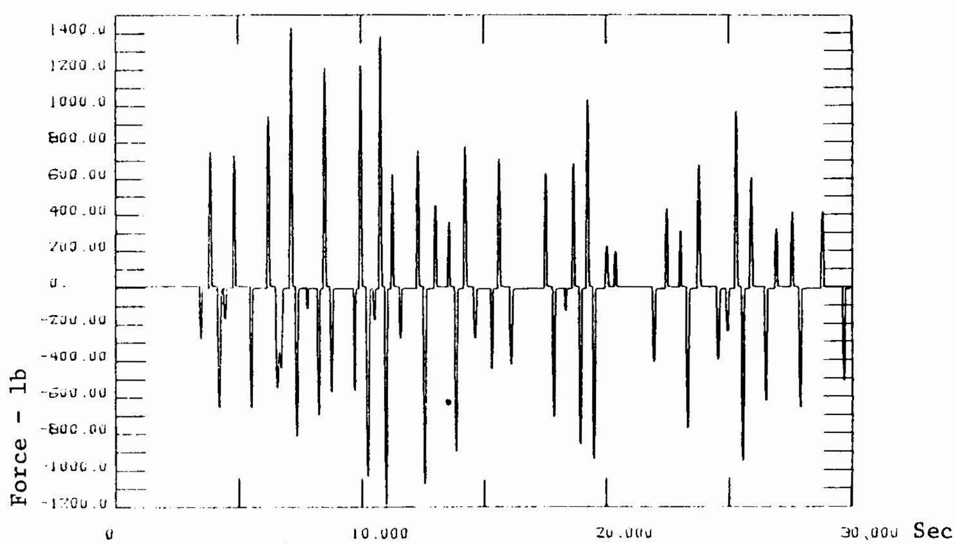

Fig (27) UPPER SPRING FORCE TIME-HISTORY (Gap = 0.5", β=1%)