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Dynamic Model of Electrical Equipment Cabinet Representing the Stiffness Softening under Strong Seismic Motion

Sung Gook Cho [a*], Dookie Kim [b]
[a]*JACEKOREA Co. R&D CENTER, #506 Sondo Bridge Hotel, 10-2 Songdo_dong, Yeonsu_gu, Incheon 406-840*
[b]*Department of Civil & Environmental Eng., Kunsan Nat'l Univ., Miryong_dong, Kunsan 573-701*
[*]*Corresponding author: sgcho@jacekorea.com*

1. Introduction

This paper presents a procedure to model the nonlinear dynamic characteristics of cabinets by using the finite element (FE) method. Duffing's type restoring force is adopted, and its corresponding equation of motion is derived. Assuming the nonlinear stiffness matrix to be diagonal around the first natural frequency, the equation of motion becomes uncoupled. Comparing the numerical results with the experimental ones, the FE model is updated. Finally, the seismic responses of the cabinet are obtained.

2. Nonlinear Model

This study represents the cabinet as a lumped-mass beam stick model. The stress-strain relation of the beam element and the dynamic equation of motion are described in this section. The uncoupled nonlinear equation of motion is developed.

2.1 Force-displacement Relation

In this study, Duffing's type restoring force is adopted to model the nonlinear behavior of cabinets with the increase of earthquake amplitude as shown in Figure 1. If the stress-strain relation of the material shows the softening spring type, which is equivalently regarded as Duffing's type force-displacement relation, then the bending stiffness of a beam decreases with the large displacement of vibration. Therefore, the relation of stress (σ_x) and strain (ε_x) of the beam element can be expressed as

$$\sigma_x = E(\varepsilon_x - \gamma\varepsilon_x^3) \tag{1}$$

where E and γ are a modulus of elasticity and a proportional coefficient of strain respectively.

2.2 Equation of Motion

The equation of motion of a beam element is

$$[M^{(e)}]\{\ddot{U}^{(e)}\} + [K^{(e)}]\{U^{(e)}\} - \beta[K_N^{(e)}]\{U^{(e)3}\} = \{F^{(e)}\} \tag{2}$$

where $\{U^{(e)}\}$ and $\{F^{(e)}\}$ are element displacement and force vectors respectively; and $[M^{(e)}]$, $[K^{(e)}]$, and $[K_N^{(e)}]$ are element mass, linear stiffness and nonlinear stiffness matrices, respectively.

Therefore, the equation of motion of a beam system can be obtained by assembling element matrices as follows

$$[M]\{\ddot{U}\} + [K]\{U\} - \beta[K_N]\{U^3\} = \{F\} \tag{3}$$

where $\{U\}$, $\{F\}$, $[M]$, $[K]$, and $[K_N]$ are system matrices corresponding to element matrices $\{U^{(e)}\}$, $\{F^{(e)}\}$, $[M^{(e)}]$, $[K^{(e)}]$, and $[K_N^{(e)}]$ respectively.

Therefore, the equation of motion of a beam system can be obtained by assembling element matrices as follows

$$[M]\{\ddot{U}\} + [K]\{U\} - \beta[K_N]\{U^3\} = \{F\} \tag{4}$$

where $\{U\}$, $\{F\}$, $[M]$, $[K]$, and $[K_N]$ are system matrices corresponding to element matrices $\{U^{(e)}\}$, $\{F^{(e)}\}$, $[M^{(e)}]$, $[K^{(e)}]$, and $[K_N^{(e)}]$ respectively

2.3 Uncoupled Nonlinear Equation of Motion

The modal coordinate system can be obtained by using the modal matrix $[\Phi]$ of the linear system. The displacement $\{U\}$ in the physical coordinate system can be transformed into the corresponding displacement $\{\xi\}$ in the modal coordinate system as follows

$$\{U\} = [\Phi]\{\xi\} \;\&\; [\Phi] = [\phi_{ij}],\; (i = 1,\cdots,n;\; j = 1,\cdots,m) \tag{5}$$

where n is the number of degrees of freedom. And m is the number of modes.

The following assumptions are adopted to uncouple the nonlinear equation of motion:

1) The nonlinear dynamic responses of the system are strongly governed by the fundamental natural mode.

2) The modal frequencies that are coupled each other are approximately assumed by considering their ratios as the ratios of the first mode amplitudes.

With the above assumptions, the uncoupled nonlinear equation of motion can be obtained in simple diagonal matrices resulting in efficient analysis as

$$\{\ddot{\xi}\} + [\backslash \omega_i^2 \backslash]\{\xi\} - \beta[\backslash \frac{\kappa_{N,i}}{\mu_i} \backslash]\{\xi^3\} = [\backslash \frac{1}{\mu_i} \backslash][\Phi]^T\{F(\omega_e, t)\},$$

$$(i = 1,\cdots,m) \tag{6}$$

where $\omega_i^2 = \kappa_i / \mu_i$.

The above non-linear equations can then be solved by using the different methods available in the literature.

3. Experiments and Numerical Model Updating

The test specimen is a seismic monitoring system central processing unit cabinet for a nuclear power plant.

The test cabinet model shown in Fig. 1 was mounted on the shaking table, the dimension of the test cabinet is 190x75x63cm and its weight is about 310kg. To identify the modal properties and mode shapes of the cabinet, the sine sweep test was performed. When the harmonic excitations were reaching natural frequencies the vibrations of the frame displayed respective natural modes, as shown Fig. 2.

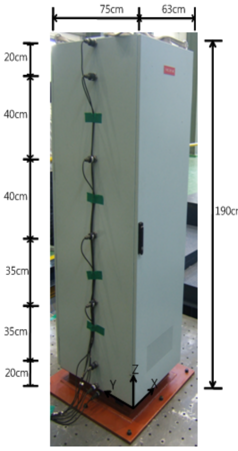

Fig. 1 Cabinet model

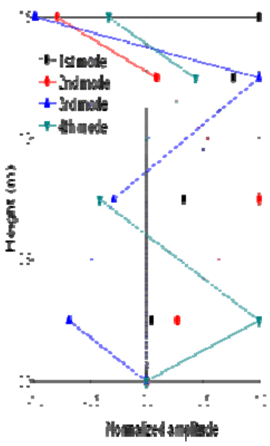

Fig. 2 Mode shapes

A series of sine sweep tests whose amplitudes of the harmonic accelerations varied from very small (about 0.015g) to relatively large (about 0.3g) have been performed. Acceleration responses in the time domain obtained in accordance with excitation amplitudes in RMS (1.0, 2.2, 3.4, 4.6, 5.8, 7.0 m/sec^2, respectively). Acceleration responses in the frequency domain are obtained against excitation amplitude (RMS 1.0 m/sec^2). The response on the top shows higher spectrum level than the bottom place near the fist natural frequency (14Hz). Transfer Function of nonlinear cabinet responses are analyzed and compared with the experimental ones. The calculated nonlinear response according to the proposed method in this paper shows similar tendency of real cabinet test. However, there are differences in frequency near 40 Hz between those methods where the first natural frequency is important in seismic analysis. Nonlinear responses of cabinet near the first natural frequency are analysed according to the proposed method, as shown in Fig. 3. The responses show well the softening nonlinear charateristic of cabinet. By applying the proposed FEM fomulation of nonlinear equation, the nonlinear response near first natural frequency showed shifting in responses.

As a result, the proposed method of nonlinear analysis is effective and it is belived that the proposed methodology will contribute to the stochastic seismic analysis.

4. Conclusions

In this study, a simplified and computationally efficient model has been presented for the earthquake analysis. In which, earthquakes are regarded as a stationary process. It is shown that nonlinear seismic responses can be efficiently calculated according to the selected number of vibration modes. Responses that are of interest in nonlinear vibration applications are reviewed. The results herein will provide a better understanding of the nonlinear vibration against random excitation. Moreover, it is believed that those properties of the results can be utilized in the dynamic design of the nonlinear system. For further studies, these dynamic relationships can reflected the nonlinear dynamic characteristics of cabinets when performing the seismic qualification.

As a future study, the model may be extended for prediction of seismic behavior of devices mounted in the cabinets and used for the modal updating of the cabinets subjected to earthquake loads.

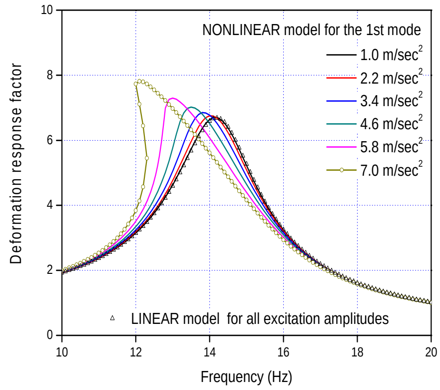

Fig.3 Nonlinear frequency responses of displacement

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