A STUDY ON THE NONLINEAR CHARACTERISTICS OF ELECTRICAL EQUIPMENT CABINETS UNDER STRONG SEISMIC MOTION

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ABSTRACT

The electrical equipment cabinet would show nonlinear behavior under strong seismic motion. It is required to construct a rational analysis model reflecting the nonlinear dynamic characteristics of the cabinets to perform seismic evaluation or qualification more reliably. This study identifies the nonlinear characteristics of the equipment cabinet via experimental program. A simplified analytical model is presented to describe the nonlinear characteristics of dynamic behavior of cabinet. A slender beam theory is applied to model the cabinet. The present model accounts for the stiffness softening behavior of the cabinet by incorporating the Duffing's type of restoring force. The characteristic of nonlinear restoring force for the analytical model is constructed on the basis of the relationship of stress-strain of the element. Experiments are performed on a typical equipment cabinet of nuclear power plant (NPP) to validate the model. Shaking table tests were performed to derive the modal properties and modes of the cabinet under base excitations. The transfer functions calculated from the time history acceleration responses obtained in accordance with excitation amplitudes were analyzed to extract the dynamic properties at each excitation level. The softening or reduction in dynamic stiffness of cabinets with increase in the excitation levels was observed in the experiments. The results obtained from the analysis using the proposed model are found to be in good agreements with the experimental results. The proposed model is expected to be useful for the prediction of seismic behavior of cabinets, particularly during the operation, owing to less computational effort required, accurate prediction of softening and no requirement of tests.

INTRODUCTION

There are many electrical equipment cabinets in the NPP to be seismically qualified to demonstrate their ability to function as required during and after the time it is subjected to the forces resulting from an earthquake. The seismic qualification can be achieved through testing or analysis. Generally, the testing method is preferred for the qualification of electrical equipment with small components or devices, inside it, which are not easy to be mathematically modeled. Accurate dynamic model of electrical equipment can be just identified by performing the experiments such as modal identification tests.

A majority of the cabinet structures in seismically active regions have been reevaluated to verify their seismic capacity against more severe earthquake. More researches are needed to expand knowledge on seismic behavior and design for cabinet structures. The dynamic properties of electrical equipment have been estimated based on the experimental data obtained from either shaking table tests or in-situ modal tests. The calculated dynamic properties of cabinets are then used to evaluate the earthquake input for safety related instruments. Several researchers have studied the cabinet dynamic behavior either analytically [1] or experimentally [2]. Almost all the studies have focused on developing simple methods or evaluating cabinet amplifications.

The seismic qualification for the cabinets can be conducted by experimental testing or numerical analysis prior to the installation of cabinet in its place. In the analysis, it is difficult to construct effective numerical model because of complex structure and assembly of cabinet. Therefore, the experimental testing using the shaking table or the shaker is usually preferred. The cabinets would show very complicated nonlinear dynamic characteristics under earthquake. The dynamic properties of the cabinet could be changed according to excitation level. The nonlinear dynamic characteristics of cabinets should be reflected when seismically evaluating the cabinet and generating the in-cabinet response spectra.

Several simplified methods to construct the analytical model of the electrical equipment cabinet have been proposed [3], [4]. The nonlinear behavior of the cabinet was, however, not discussed in the previous studies. It has been found by Yang et al. [5] and Rustogi and Gupta [6] that the cabinet mounting arrangement can significantly affect the nonlinear behavior and therefore need to properly be accounted for in seismic works. Similar observations were made earlier by Llambias et al. [7] for a suite of two electrical cubicles tied together at top and bottom.

In this study, experiments have been performed to investigate the nonlinear dynamic characteristics of equipment cabinet. A procedure to model the nonlinear dynamic characteristics of cabinets by using the finite element (FE) method is also introduced. Duffing's type of restoring force is adopted, and its corresponding equation of motion is derived. Assuming the nonlinear stiffness matrix to be diagonal around the fundamental frequency, the equation of motion becomes uncoupled. The equipment cabinet is considered to be welded rigidly to the base, as is the practice in Korea [8]. Beam elements are employed to model the cabinet. The model accounts for the softening behavior of the cabinet structure by incorporating the Duffing's type of the restoring force.

Another experiment program is performed on the equipment cabinet to validate the numerical model. In order to obtain the more realistic information of the cabinet, this study adopts the modal identification results obtained from shaking testing with portable shaker at the various excitation levels. Comparing the numerical results with the experimental ones, the FE model is updated. Finally, the seismic responses of the cabinet are obtained. The results obtained from the proposed nonlinear model are compared to those obtained from shaking table tests.

NONLINEAR DYNAMIC CHARACTERISTICS OF ELECTRICAL CABINET

Experiments

This study performed experiments to investigate the nonlinear characteristics of dynamic behavior of cabinets. The test specimen is a typical electrical equipment cabinet of thin steel plate used in the NPP. The cabinet is 1900mm high, 700mm deep, and 635mm in width, as shown in Figure 1(a). The cabinet has two doors in the front and rear sides. Total weight of the cabinet is 311kg including the door's weights of 66.4kg. As shown in Figure 1(b), the cabinet includes complex components and devices inside. The cabinet consists of an internal steel frame of channels and angles and the thin steel plate covers enwrapping the frame. All structural components are interconnected with each other by bolting and spot welding. The cabinet is expected to behave in a complexly nonlinear manner under strong motion of earthquake excitation. The specimen is bolted onto the prefabricated mounting fixtures which are connected with bolts to the test frame structure.

(a)Dimension (b) Internal view

Figure 1. Test cabinet on the mounting fixture

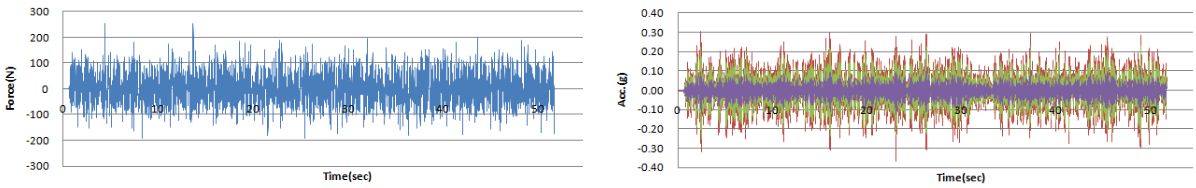
To derive the modal properties of the cabinet, a set of dynamic test were performed. The test cabinet was excited by a portable shaker. Broad band random waves were used as exciting motion. Experiments were performed in one, selected horizontal (x) direction, i.e., side-to-side direction. A series of vibration tests whose amplitudes of the forces varied from small to relatively large have been performed.

As shown in Fig. 2, six PCB PIEZOTRONICS accelerometers model 393B12 were attached to the cabinet for acceleration measurements in side-to-side direction. The portable shaker was APS400 model of APS Dynamics company which can excite a specimen with 0 ~ 200 Hz of the frequency range. APS 145 model of the amplifier was used to amplify the input signal. The shaking point is at the middle of the top of cabinet.

(a)Location of Accelerometers (b)Excitation Point

Figure 2. Test Setup

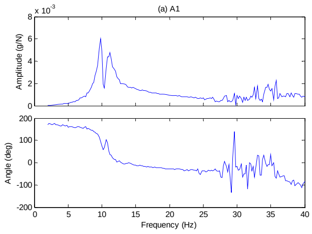
The cabinet with a door was excited for about 60 seconds at each excitation level: 69, 127, 156, 188, 217, 240, 255 and 277N. And the cabinet without a door was also excited for about 60 seconds at each excitation level: 37.1, 53.8, 66.9, 110, 138, 178, 207, 228 and 259N. Low pass filter below 40Hz was applied. The time history forcing function and the acceleration response were recorded from all the sensors. The typical time history signals recorded at 240N of peak excitation are shown in Fig. 3.



(a) Excitation Force (b) Response Accelerations

Figure 3. Typical Recorded Signal

The typical transfer functions calculated for the cabinet with a door and for the cabinet without a door are shown in Figs. 4 and 5 respectively.



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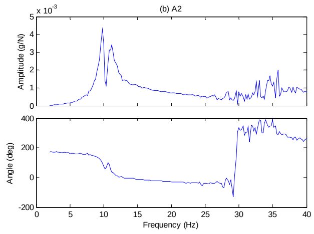

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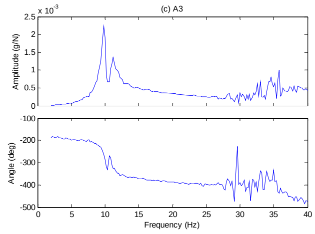

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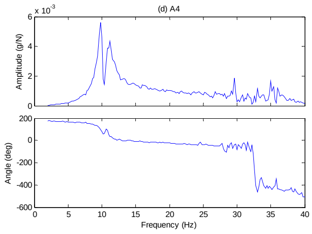

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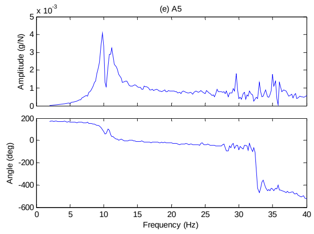

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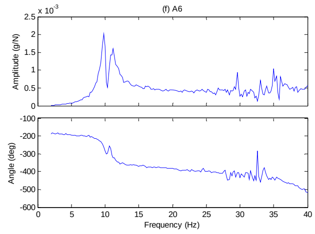

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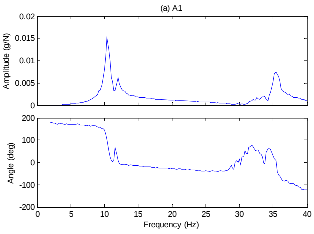
Figure 4. Recorded Signal of the specimen with a door (excitation level of 240N)



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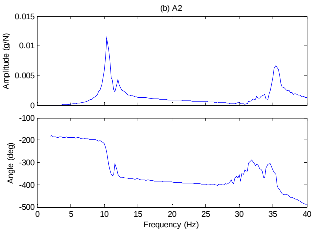

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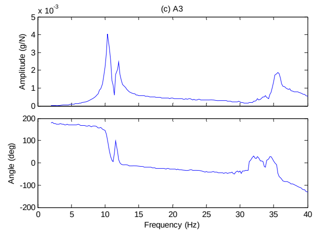

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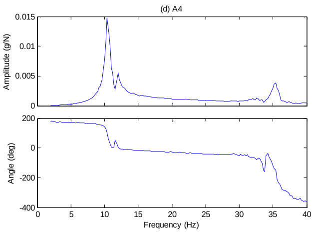

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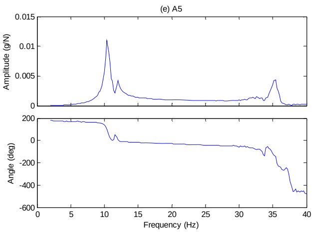

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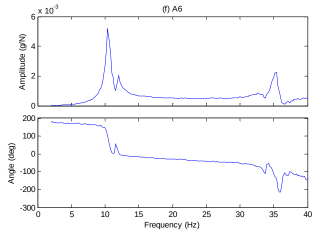

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Figure 5. Recorded Signal of the specimen without a door (excitation level of 228N)

Evaluation of Dynamic Behavior

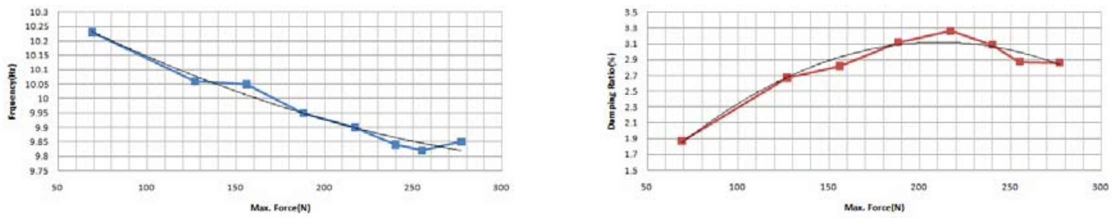
Polynomial curve fitting was applied to identify the modal properties of the cabinet, and the results are shown in Tables 1 and 2.

Table 1. Modal Properties for the cabinet with a door

Peak Excitation Force (N)	69	127	156	188	217	240	255	277
Peak Response Acceleration(g)	0.098	0.190	0.249	0.301	0.338	0.369	0.385	0.406
Frequency (Hz)	10.23	10.06	10.05	9.95	9.90	9.84	9.82	9.85
Damping Ratio (%)	1.87	2.67	2.82	3.12	3.26	3.09	2.87	2.86

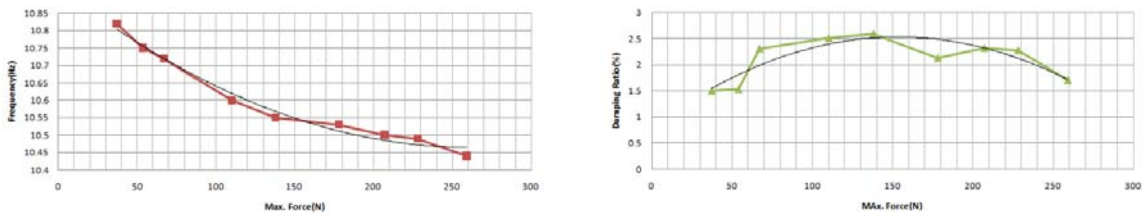
Table 2. Modal Properties for the cabinet withouta door

Peak Excitation Force (N)	37.1	53.8	66.9	110	138	178	207	228	259
Peak Response Acceleration(g)	0.113	0.155	0.188	0.297	0.353	0.448	0.606	0.546	0.609
Frequency (Hz)	10.82	10.75	10.72	10.6	10.55	10.53	10.5	10.49	10.44
Damping Ratio (%)	1.5	1.53	2.31	2.51	2.59	2.13	2.32	2.28	1.71



(a)Frequency (b)Damping Ratio

Figure 6. Variation of Fundamental Modal Properties by Excitation Levels (with door case)



(a)Frequency (b)Damping Ratio

Figure 7. Variation of Fundamental Modal Properties by Excitation Levels (without door case)

NONLINEAR ANALYTICAL 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.

In this study, Duffing's type of restoring force is adopted to model the nonlinear behavior of cabinets with the increase of earthquake amplitude. 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.

When considering the softening stiffness, the equation of motion of a beam element is

$$[m]\{\ddot{u}\} + [k]\{u\} - \beta[k_N]\{u^3\} = \{f\} \tag{1}$$

where $\{u\}$ and $\{f\}$are element displacement and force vectors, respectively; and $[m]$, $[k]$, and $[k_N]$ are element mass, linear stiffness and nonlinear stiffness matrices, respectively. In case of $b \times h$ rectangular type cross-section of the beam, the coefficient is noted as $\beta = \frac{3}{20}\gamma h^2$ where γ is a proportional coefficient of strain.

The equation of motion of a beam system can be obtained by assembling element matrices as follows

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

where $\{U\}$ and $\{F\}$, $[M]$, $[K]$, and $[K_N]$ are system matrices corresponding to element matrices where $\{u\}$,$\{f\}$, $[m]$, $[k]$, and $[k_N]$, respectively.

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

$$\{U\} = [\Phi]\{\xi\} \& [\Phi] = [\phi_{ij}] \,, (i = 1, \cdots, n \;\&\; j = 1, \cdots, m) \tag{3}$$

where$n$ is the number of degrees of freedom, and$m$ is the number of modes. Pre-multiplying both sides of the equation of motion by$[\Phi]^T$, the equation is expanded to the nonlinear modal equation as

$$[\mu]\{\ddot{\xi}\} + [\kappa]\{\xi\} - \beta[\Phi]^T[K_N]\{U^3\} = [\Phi]^T\{F(\Omega, t)\} \tag{3}$$

where $[\mu] = [\Phi]^T[M][\Phi]$ and $[\kappa] = [\Phi]^T[K][\Phi]$, respectively, and Ω is an forcing frequency. Usually, $\{U^3\} = ([\Phi]\{\xi\})^3$ is a coupled nonlinear form of modal displacements.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

$$\{\ddot{\xi}\} + [\omega_i{}^2]\{\xi\} - \beta\left[\frac{k_{N,i}}{\mu_i}\right]\{\xi^3\} = \left[\frac{1}{\mu_i}\right][\Phi]^T\{F(\Omega, t)\}, \; (i = 1, \cdots, m) \tag{4}$$

where, $\omega_i^2 = k_i/\mu_i$.The non-linear equations can then be solved by using the different methods available in the literature. This study applies Runge-Kutta method (Dormand and Prince, 1980) to solve the nonlinear equations developed in the model.

EXPERIMENTS AND NUMERICAL MODEL UPDATING

A series of excitation tests whose maximum amplitudes varied from very small (about 37.1N) to relatively large (about 277N) have been performed for the cabinets with and without a door. Acceleration responses in the time domain obtained in accordance with maximum excitation amplitudes. Acceleration responses in the frequency domain are obtained against maximum excitation amplitudes (69N and 66.9N respectively for the cabinets with and without a door). The corresponding response on the top shows higher spectrum level than the bottom place near the fist natural frequency (10.23Hz and 10.72Hz respectively for the cabinets with and without a door). 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 fundamental frequency is important in seismic analysis. Nonlinear responses of cabinet near the fundamental frequency are analyzed according to the proposed method in Figs. 8-10. The responses show well the softening nonlinear characteristic of cabinet. By applying the proposed FEM formulation of nonlinear equation, the nonlinear response showed shifting in responses. As a result, the proposed method of nonlinear analysis is effective and it is believed that the proposed methodology will contribute to the seismic analysis.

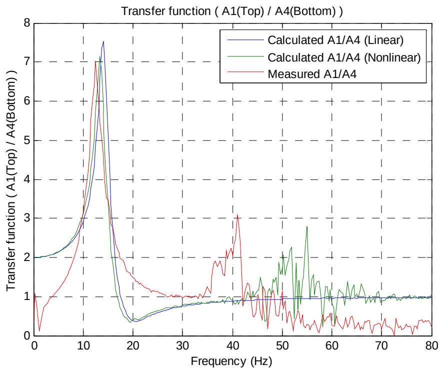


Figure 8. Transfer Function Obtained from Experiments and Numerical Analysis

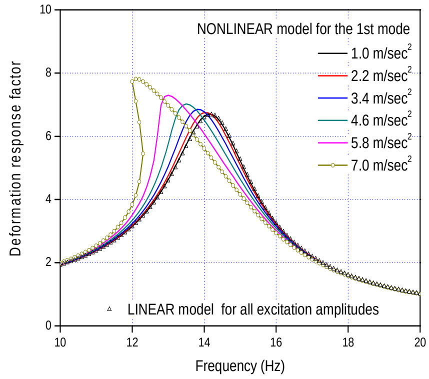


Figure 9. Response Factors Obtained from the Proposed Model for Deformation

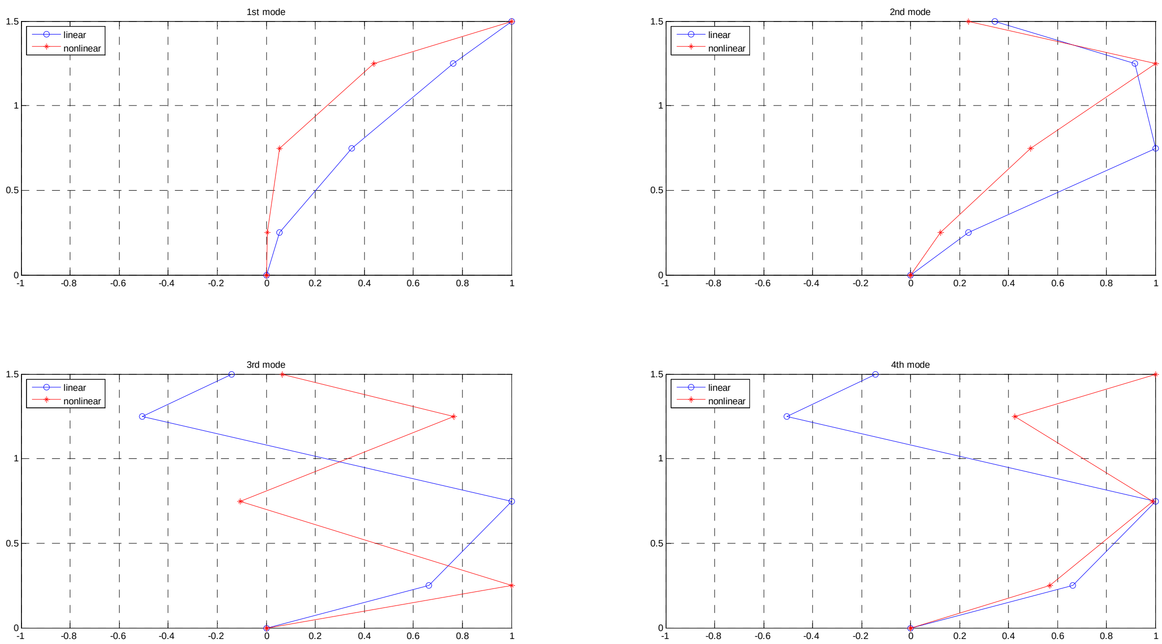


Figure 10. Comparison of Linear and Nonlinear Modes

CONCLUSION

In this study, the vibration analysis method of a nonlinear cabinet system is formulated considering earthquakes 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 the results of the present study can be utilized in the dynamic design of the nonlinear system. For further studies, these dynamic relationships can reflect the nonlinear dynamic characteristics of cabinets when performing the seismic qualification.

ACKNOWLEDGEMENTS

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