SEISMIC RESPONSE ANALYSIS OF NUCLEAR POWER PLANT AUXILIARY MECHANICAL EQUIPMENT

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SUMMARY

Seismic response analysis of nuclear power plant auxiliary mechanical equipment is a vital link to the successful design of the plant in assuring its safe operation either during or after a postulated earthquake. Notwithstanding its importance, this subject has not been treated sufficiently in the literature. In an earlier publication, the author presented some modeling techniques for the analysis of tanks, heat exchangers, valves, and pumps. These modeling techniques have since become the main-stay in the qualification of the auxiliary mechanical equipment.

In this paper, the prevailing modeling techniques will be reviewed in accordance with current technological development.

For instance, in the modeling of valves, it has been recognized that the majority of valves are rigidly designed (i.e., having natural frequencies greater than 33 Hz). Substantial simplification has been made by vendors to reduce the models. A one mass representation of the rigid valves is not an uncommon assumption. However, some recent test results have shown that there are valves with flexible parts which can only be qualified using proper assumptions and models with sufficient details. In this paper, some results obtained in these test and analysis programs will be discussed to provide insight to the proper modeling of these components.

Also, considerable amount of equipment data have been accumulated for plants under construction or in operation. A compilation of these data into usable forms, such as through the statistical means, would no doubt be a very useful tool for the safe and economical design and analysis of the nuclear power plant equipment. Some of the data obtained in reaching this goal are illustrated.

1. Introduction

Nuclear power plant auxiliary mechanical equipment consists of pumps, valves, tanks, heat exchangers, filters, demineralizers, flow indicators, strainers and cranes. They, together with the electrical equipment, form the largest percentage of the equipment which needs to be seismically qualified.

To date, discussions on seismic response analysis of nuclear power plant systems and components have been centered on the reactor coolant system and its associated components such as reactor pressure vessel, steam generator, reactor coolant pump, and pressurizer. This has been basically due to the fact that these components are the most important part of the nuclear power plant safe operation, that they generally subjected to multiple design and hypothsized extreme loads (such as Loss of Coolant Accident and seismic events), and that they are massive and with flexible and sometimes nonlinear structure design. As a result, sophisticated analytical techniques are usually adopted for their qualification. These techniques have been adequately discussed in the literature (Refs. 1, 2). As for the electrical equipment, other than control boards and racks which can be analyzed (Ref. 3), there is extensive structural nonlinearity which requires testing. Discussions about testing of electrical equipment to meet the 1975 revision of the IEEE-344 have been many (for instance, Refs. 4 and 7).

Unlike the electrical equipment, auxiliary mechanical equipment is basically linear. Furthermore, most of this equipment is rigid in nature (i.e., the natural frequency is above the frequency content of the input motion, say 33 Hz currently defined for the design earthquakes). Consequently, this equipment can and has been analyzed in a very simple manner. It is only for flexible equipment such as some tanks, heat exchangers, filters and demineralizers that a more elaborate response spectrum analysis is used. A discussion of the mathematical model for such flexible equipment and an effort to establish the characteristics of the rigid pumps and valves is contained in Ref. 5. Further development has led to the simplified analytical technique reported in Ref. 6 for some vertical components. It is the interest of this paper to summarize and discuss the equipment analysis approaches for a current account of the state-of-the-art in this area.

2. Equipment Classification by Dynamic Characteristics

Auxiliary mechanical equipment has been shown in the past to possess distinctive dynamic characteristics such that it can be easily classified and cross referenced with the most appropriate qualification methods. The following is a detailed discussion of each class of equipment according to its dynamic characteristics.

2.1 Rigid Equipment

This class of equipment has a sufficiently high fundamental natural frequency which is not excited by the input motion at its base. For a seismic input at free field, such frequency is generally indicated by the (rigid) frequency at which the response spectral acceleration converges to the maximum ground acceleration. Not only each equipment modal spectral response above this rigid frequency will be the same as the maximum ground motion, the total response of the rigid equipment can also be shown to be the same as the maximum ground acceleration. Therefore, a rigid equipment essentially moves with its base and has no further amplification.

The equipment rigid frequency as determined by the ground response spectra has been 20 Hz in the past when Housner response spectra were used as the nuclear power plant design input. This frequency has been increased to 33 Hz to be consistent with the Regulatory Guide 1.60 response spectra. However, strictly speaking, this rigid frequency is applicable only to ground supported structures, systems and components. For systems and components located and supported by building structures, the rigid frequency needs to be adjusted to be the frequency at which there is no further response (or amplification) from the building motion. This frequency is usually substantially less than the rigid frequency at the free field due to the building filtering effect. Consequently, it should be noted that designing the equipment to be rigid does not necessarily mean to be 33 Hz. Although 33 Hz is a conservative value, relaxation to this requirement can be made depending on the dynamic characteristics of the supporting structure. Equipment that can be included in this class consists of mainly pumps, valves, flow indicators, and strainers.

2.2 Flexible Equipment With One Predominate Mode

Vertical tanks, heat exchangers, and filters which are supported either by a skirt or columns at the bottom part of the vessel can be flexible. Experience has shown that this equipment has only one flexible mode along one horizontal axis. For this equipment, only one mode could be excited by the response spectral peak value. The remaining modes will all be subjected to the constant maximum base acceleration. Although any analysis conducted for this equipment should include the modal amplification for the flexible mode, the equipment need not be subjected to very conservative analysis which includes the multiple mode effect and the effect of closely spaced modes.

2.3 Flexible Equipment With Multiple Modes

This equipment includes flexible vertical components supported at two or more elevations, horizontal components supported by saddles, and refueling equipment such as cranes.

For this equipment, amplification on each of the flexible modes is possible. Also, more than one mode could be subjected to the peak response motion. Therefore, any analysis conducted for this equipment should include the multi-mode response effect and the special method to compute the response from the closely spaced modes (Regulatory Guide 1.92).

3. ANALYSIS METHODS

There are numerous analysis methods that can be used to qualify the auxiliary mechanical equipment. However, for this type of equipment which is basically linear in nature, sophisticated approaches used in the reactor coolant system analysis are not appropriate. More convenient and economical methods described in the following can be tailored to meet the needs of each class of the equipment.

3.1 Rigid Equipment

Being rigid, the equipment has no amplification. All parts of the equipment is vibrating with the same maximum base input acceleration. Therefore, a static analysis approach represents adequately the method of qualification for this equipment.

Some of the auxiliary mechanical equipment has special design requirements which necessitated that it be designed rigid. For instance, for horizontal pumps, the constantly rotating machinery requires that the vibration and noise level be low. Therefore, the running speed of the pump should be sufficiently separated from the predominate natural frequencies of the rotor and shaft. Hence, even for the only possible flexible areas, such as rotor and shaft, the fundamental natural frequencies are usually much higher than the rigid frequency. This is particularly true for smaller pumps.

To justify that the pump is indeed rigid, as suggested by its design characteristics, a simple method to investigate natural frequencies of the shaft and the rotor can be used. These components can be treated conservatively as single or multi-mass models. Or, if a computer program is available, one may want to model the entire pump (including motor and coupling) in a lumped mass model. In this manner, the stresses of every important cross sections and the displacement of the moving parts (such as rotor and shaft), relative to the stationary parts can be determined from the same model used to study the natural frequency. This provides necessary information to justify the equipment can continuously operate during and after the earthquake.

For vertical pumps, such as sump pumps and some residual heat removal pumps, they are mostly flexible. The former is a result of its length and built in flexibility in the shaft. The latter is due to the one level of support usually provided at the bottom of the pump. In the meantime, the pump needs to slide along the axial direction of the piping to accommodate the thermal expansion. This equipment has to be qualified using the response spectrum approach with a multi degree-of-freedom lumped mass model. In this model, the moving parts such as shafts and rotors have to be separated from the casing, in order that the natural frequencies, stresses and displacements of these flexible parts can be properly determined.

Similar to the horizontal pumps, the valves are usually rigid. This is particularly true for small manual valves where the extended structure, extending beyond the pressure retaining valve body, is very short. The valve body has an intrinsic rigid design in order to hold the high fluid pressure. The extended structure which include the yoke and portion of the stem is the only possible area of flexibility. Past analysis has tended to ignore the possible flexibility of stem and yoke, especially the stem. It is not uncommon to assume that the valve is rigid and then results to a simple model to justify the correctness of the assumption. A simple model with one or two masses tends to over simplify the valve design and the possible flexibility of the stem. Some test programs conducted recently have shown that for air operated valves, the spring and the stem together with its upper connecting members may have little horizontal support at the top which could result in a fundamental natural frequency as low as 10 Hz. A more sophisticated model which can determine all pertinent design information for this area will be necessary.

Another area which has shown to create low flexibility is in the flange and bolts. To insure that slippage does not occur in this area, selective in-situ testing or factory testing is required.

Notwithstanding the unforeseen design flexibility of some values, experience has shown that values have the lowest seismic stresses in all the auxiliary mechanical equipment.

3.2 Flexible Equipment With One Predominate Mode

For this equipment, two modes will be sufficient to account for the total equipment response. One mode is to represent the flexible mode and the other is to include the rigid mode. Therefore, a two mass model discussed in Ref. 8 will be sufficient. Or, if one has to use a single mass model, solution from such a model when superimposed with a rigid frequency acceleration should provide a sufficiently conservative result.

3.3 Flexible Equipment With Multiple Modes

For the equipment where multiple flexible modes exist, it is necessary to use a mathematical model which accounts for all the flexible modes. Or, alternatively, a pseudo static approach may be used which uses, as the static acceleration, the maximum response spectral value in the range of the equipment of natural frequencies and multiplied by a commonly accepted coefficient of 1.5. This coefficient, although very conservative and has been justified for piping systems, has not been established for mechanical equipment in general.

Some equipment, such as horizontal tanks and heat exchangers having more than one flexible mode along one axis has only few flexible modes. There are, on the other hand, equipment such as a long sump pump which possesses a large number of flexible modes. It is important that the closely spaced modes described in Regulatory Guide 1.92 be accounted for. Also, sufficient number of modes needs to be considered. For a complex equipment where each individual response of interest may depend on different combinations of modes, it is difficult to require that every item of interest to reach certain percentage of total response.

As a result of the complexity in this problem, a simple and practical guide has been to sum all the modes up to and including the first rigid frequency mode. This is helpful in general, but not necessarily conservative for situations where rigid frequency modes may provide significant response motions.

As a remedy, modal effective mass can be used. Modal effective mass is defined as the following:

$$(M_{eff})_j = \frac{(\Sigma_i M_i \phi_{ij} d_i)^2}{\Sigma_i M_i \phi_{ij}^2} \qquad (1)$$

where M is the mass, ϕ the modeshape, and d is the unit directional vector for the earthquake. Subscripts i and j represents the ith node and jth mode, respectively.

It is quite obvious that Eq. (1) is always positive. It is useful to determine whether all the important modes have been considered as a result of the following relationship:

$$\Sigma_j (M_{eff})_j = \Sigma_i M_i \qquad (2)$$

Eq. (2) shows that the total modal effective mass should equal the total mass used in the model. Therefore, it is possible to keep a running count of the sum of the response calculation. Typically, a modal effective mass of 80 percent would imply that sufficient number of modes have been considered, unless, of course, that the total number of degrees-of-freedom in the model is small.

This method of determining the modal response contribution is more effective than the use of generalized mass which is defined as the demoninator of Eq. (1). Since generalized mass is still a function of ϕ, the result will critically depend on how ϕ is normalized. The relative significance of the generalized mass between modes cannot be known unless all ϕ's are normalized with a common demoninator for all modes.

4. BUILDING AND EQUIPMENT RESPONSE DATA

The number of commercial nuclear power plants currently on order, under construction, or in commercial operation has been increasing during the last decade. This has resulted in a vast amount of seismic information available both for plant and equipment design. This information can be advantageously utilized to upgrade future plant seismic design. For this purpose, a study has been conducted using seismic related data on more than one hundred pressurized water reactor plants. A detailed account of the study and a discussion of the currently adopted qualification approaches for electrical equipment and the reactor coolant system is presented in Ref. 7. To summarize the equipment response data for past plants, Figure 1 shows the first mode spectral acceleration for all equipment excluding rigid pumps and valves. For the equipment included in this figure, first mode accounts for at least 80 percent of the total modal effective mass. Maximum first mode equipment acceleration shown is less than 2.0g for SSE for at least 95 percent of the equipment.

For other equipment damping values, the spectral acceleration at different frequencies cannot be easily determined. However, as a conservative measure, one can determine from the following formulae, the building and equipment amplification values. By multiplying the two amplification values and the maximum design ground acceleration value for either OBE or SSE, one arrives at a realistic estimate for the floor response spectra peak.

For the building amplification, the following formula applies:

$$A_{BLDG} = \left(\frac{1}{2\xi}\right)^{1/2} \tag{3}$$

where ξ is the building damping, and A_{BLDG} is the building amplification. As for the estimation of the floor response spectrum amplification, the following formula is appropriate:

$$A_{EQPT} = \left(\frac{1}{2\xi}\right)^{2/3} \tag{4}$$

where ξ is the equipment damping and A_{EQPT} represents the equipment amplification. For a combination of 7 percent building and 4 percent equipment damping and a maximum ground acceleration of 0.2g for SSE, the response spectrum peak value is estimated by Eqs. (3) and (4) to be 2.87g. This is in line with the data obtained for past plants as presented in Ref. 7. Assuming that the equipment natural frequency will not necessarily coincide with the building natural frequency, root mean square value of the spectral peak estimated (2.87g) may provide the most probable equipment spectral acceleration. This yields 1.69g. This value is somewhat on the high side as compare with the spectral acceleration shown in Figure 1. Consequently, Eqs. (3) and (4) are conservative.

5. SUMMARY

A detailed discussion has been provided concerning the analysis approach that can be used for the seismic qualification of auxiliary mechanical equipment. The equipment is first divided into three classes, namely, rigid, flexible with one mode predominate, and flexible with multiple modes. A most appropriate method and required special consideration are provided for each class of the equipment.

In addition, to facilitate the future design of the equipment, a study result is presented which covers large numbers of existing PWR plants. Methods to estimate equipment spectral responses for various dampings are also recommended.

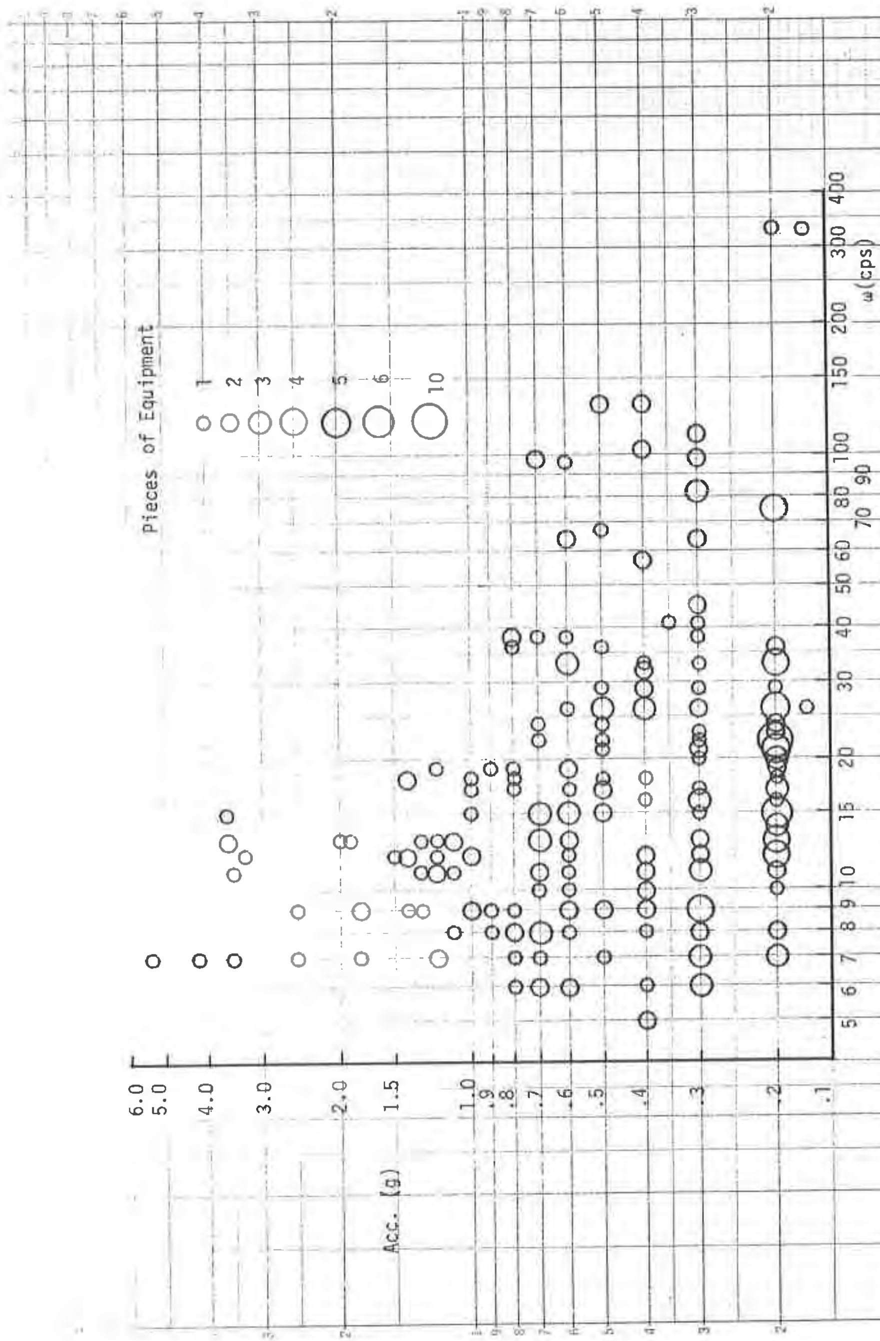

Fig. 1 FIRST MODE EQUIPMENT ACC. FOR SSE vs. EQUIPMENT NATURAL FREQUENCY

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