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An evaluation method of vibration severity of rotating machines under earthquake loading

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ABSTRACT

An evaluation method based upon Fourier spectrum analysis is developed to estimate the vibration severity of the rotating machine foundations under earthquake loading. A weighting factor expressed by the aspect ratio of the structure is employed to consider the effects of torsional coupling. And, the RMS sum of the maximum velocity response of the translational mode and the maximum velocity response of the torsional mode multiplied by the weighting factor are suggested to compare with the vibration severity recommended by ISO.

1. INTRODUCTION

The ISO(International Standards) requires specific diagnostic treatment to apply to the evaluation of machine vibration with respect to unusual machines that are not normally produced in significant quantities. Therefore, additional care must be taken for the evaluation of vibration severity of rotating machines under external vibration such as earthquake loading. However, there is no evaluation method to estimate vibration severity of rotating machines subjected to external vibration such as earthquake loading. In this paper, an attempt was made to develop an evaluation method based upon Fourier spectrum analysis of the rotating machine foundations considering the effects of torsional coupling to evaluate vibration severity for rotating machines under earthquake loading. The Fourier spectra of acceleration have a velocity dimension that is different from that of normal machine vibration by ISO. Thus, in an effort to relate these two physical values, the relationship of the Fourier spectrum with the velocity response of a Single-Degree-of-Freedom(SDOF) system should be derived. The velocity response of a SDOF system with 0 damping can be represented as follows from Duhamel's integration.

$$\dot{X}(t) = \int_0^t \ddot{y}(\tau)\cos\omega(t-\tau)d\tau$$
$$= \sqrt{\left[\int_0^t \ddot{y}(\tau)\cos\omega\tau\right]^2 + \left[\int_0^t \ddot{y}(\tau)\sin\omega\tau\right]^2}\cos(\omega t+\phi) \quad (1)$$
$$\phi = \tan^{-1}\frac{\int_0^t \ddot{y}(\tau)\sin\omega\tau d\tau}{\int_0^t \ddot{y}(\tau)\cos\omega\tau d\tau}$$

Since the response spectrum represents the maximum value during the duration of earthquake, the maximum value of Eq.(1) can be expressed as follows.

$$S_{v,\xi=0} = \left| \sqrt{\left[\int_0^t \ddot{y}(\tau)\cos\omega\tau d\tau \right]^2 + \left[\int_0^t \ddot{y}(\tau)\sin\omega\tau d\tau \right]^2} \cos(\omega t + \phi) \right|_{max} \tag{2}$$

which is the velocity response spectral value of 0 damped SDOF system.

On the other hand, if the duration of earthquake is from 0 to T, the Fourier transform of the earthquake acceleration can be expressed as follows.

$$F(\omega) = \int_o^T \ddot{y}(\tau)e^{-i\omega\tau}d\tau = \int_0^T \ddot{y}(\tau)\cos\omega d\tau d\tau - i\int_0^T \ddot{y}(\tau)\sin\omega d\tau$$

Accordingly, the Fourier spectrum can be expressed as follows.

$$|F(\omega)| = \sqrt{\left[\int_0^T \ddot{y}(\omega)\cos\omega\tau \right]^2 + \left[\int_0^T \ddot{y}(\omega)\sin\omega\tau \right]^2} \tag{3}$$

Here, regardless of the physical meaning of the period for velocity response and the Fourier spectrum, the comparison of these two quantities given in Eq.(1), and Eq.(3) shows that the velocity response spectrum has the maximum value of the Fourier spectrum.[1] This relationship exists because the velocity response spectrum represents the maximum response at any time during the duration of the earthquake and the Fourier spectrum represents the response after the earthquake has gone. Table 1 [2] provides a tabulation of Newmark's amplification factors as they relate to the damping of a SDOF system. This is considered to be the general characteristics of earthquake loading. Velocity amplification factors decrease from 2.8 down to 1.5 as the damping ratio increases from 2% to 7%. Therefore, it is suggested that the Fourier spectrum can be divided by the corresponding ratio of about 2 to roughly calculate the maximum velocity from the Fourier spectrum in case that the damping ratio of the system is approximately 5%.

2. EFFECT OF SHAPE RATIO OF THE STRUCTURE ON TORSIONAL MODE

From the past record reporting earthquake damage, it is clear that rectangular structures such as schools have been damaged severely by torsional motion rather than by translational motion. Especially, this tendency has been clearly observed in non-symmetric structures. Accordingly, it is expected that non-symmetric structures such as turbine foundations will be more vulnerable to the torsional mode in the presence of earthquake. In this paper, a 3-D Finite Element Method(FEM) analysis of the simplified rectangular structure model shown in Fig.1 was carried out using the SAP90 compute program to investigate the effects on the torsional mode caused by changes in the shape ratio of the structure(the ratio of the length to the width of the rotating machine foundation). For the analysis of this model, the properties of material are assumed to have the values: design concrete strength σ=400kg/cm^2, elastic constant E=300,000 kg/cm^2, and density ρ=2500kg/m^3. As shown in Table 2, it is observed that the larger the aspect ratio is, the more flexible the structure becomes. Also, the structure is influenced by the effects of torsional coupling as the resonance frequency of the torsional and translational modes become closer to the predominant frequencies of the earthquake.

3. CONSIDERATION OF THE EFFECTS OF TORSIONAL COUPLING

Based upon the understanding above, an attempt was made to derive a new method for the evaluating vibration severity of rotating machines subjected to external vibration. ISO 2372 [3] and 3945 [4] are to evaluate normal machine vibration occurring on the surfaces of the machine, on the bearings, or at the mounting points. Vibration severity is used as the basis of quality judgement of machine vibration and the root-mean-square value of vibration velocity has been chosen as the unit of measurement for indicating vibration severity. The ISO evaluation standard, based upon the velocity under normal vibration, suggests that an average value should be adopted when there is more than one fundamental frequency. When the vibration of the rotating machine is beat vibration which consists of two major fundamental frequency components, the RMS of $V_{\max}$ and $V_{\min}$ is defined as in the following equation, and the vibration severity is evaluated according to Table 3.[3,4]

$$V_{rms} = \sqrt{1/2(V_{\max}^2 + V_{\min}^2)} \tag{4}$$

where, V_{rms} = range of vibration severity(mm/s or inch/s)

In the translational mode, relative displacement related to the rupture of the structure may not be of serious concern because the rotating machine and the foundation move in same direction, even though the vibration severity is somewhat large. Conversely, the displacement at both ends of the machine have the opposite direction(180^0 phase difference) in the torsional mode. Moments occur at the supporting points of the shaft even though there is no relative displacement on the shaft. The value of the rotational moment, which is equal to the arm length multiplied by the applied force, depends on the shape of the structure because it determines the arm length. Non-symmetric or rectangular structures rather than symmetric or square structures are more likely to experience the rotational mode. In this respect, the aspect ratio of the structure, L/W can be considered a weighting multiplication factor. Thus, the multiplication of the weighting factor with the maximum response of the Fourier spectrum of the torsional mode could be used to consider the effects of the torsional mode. Then, the Root-Mean-Square(RMS) sum of this multiplied value and the maximum value of the translational response could be used to estimate the vibration severity of rotating machine under external vibration by eq. (5).

$$V'_{rms} = \sqrt{\tfrac{1}{2}\left[V'^2_{\max} + W\left(V'^2_{\min}\right)\right]} \tag{5}$$

where, V'_{rms} = Range of vibration severity of the machine due to external vibration

$V'_{\max}$ = Max. response of translational mode

W = weighting multiplication factor for the aspect ratio of the structure

$V'_{\min}$ = Max. response of torsional mode

4. PROCEDURE FOR THE EVALUATION OF VIBRATION SEVERITY

Therefore, an evaluation of vibration severity of rotating machines under external vibration could be made by the following procedure.

(1) Calculation of Fourier Spectrum from the Relative Acceleration Time-History

Calculate the Fourier spectrum from the relative acceleration time-history that is obtained by deducting input time-histories of acceleration from output time histories of acceleration.

(2) Conversion of Fourier Spectrum to Velocity Response Specrum

Convert the Fourier spectrum to velocity response spectrum considering damping ratio of the system.

(3) Calculation of Vibration Severity

Calculate the RMS sum of the maximum velocity response of the translational mode and the maximum velocity response of the torsional mode multiplied by the weighting factor according to Eq.(5).

(4) Evaluation of Vibration Severity

Compare the vibration severity given in Table 3 recommended by ISO and the vibration severity of the system calculated in Step (3) under both external and normal machine vibration.

5. EVALUATION OF THE PROTOTYPE STRUCTURE FOR VIBRATION SEVERITY

5.1 Determination of Damping Ratios of the System

In this study, based on the artificial mass simulation, a 1/10 scaled test model structure employing plate spring elements that satisfy similitude requirements was simulated for the prototype structure in order to obtain the dynamic characteristics of the system(See Fig. 2). The prototype structure is the 400 MW spring-mounted turbine-generator foundation of Ulsan fossil power plant with dimensions of 41.5m(length) x 14m(width) x 12m(height). A 6 DOF shaking table test program was conducted at the Korea Institute of Machinery and Metals. Natural frequencies, natural modes and damping ratios were obtained from the results of the test. The mode superposition method was used for dynamic analysis and the time-history analysis method was used for modal response. Since the time-history of each mode in the test cannot be separated, the natural frequency and damping ratio of each mode were estimated by comparing the amplitude of response at each frequency range after test results and 3-D analytical solutions were Fourier transformed. Table 4 shows the natural frequencies of the test model structure. The damping ratios of the system(See Table 5) were evaluated by the bandwidth method (half-power method) [5] from the resonance curves obtained by the test. These same damping ratios were applied to the following analytical modeling.

5.2 Analytical Modeling of the Prototype Structure

A 3-D space frame structure was idealized by FEM for analytical modeling of the prototype structure and ADINA computer program was used for the analysis. The model consists of 98 nodes, 93 beam elements and 8 spring elements. The deck and the columns were idealized as beam elements, and spring elements were idealized as internal spring elements.

5.3 Evaluation of Vibration Severity

The vibration severity of the prototype structure shown in Table 6, of which the Fourier spectrum is shown in Fig.3 under the simulated El Centro earthquake with a base acceleration of 0.2g, was obtained by the procedure suggested in Secion 4. The comparison of this result with the machine vibration severity shown in Table 3 leads us to the belief that the prototype will be likely to exhibit unacceptable performance because its vibration severity(V_{rms})exceeds

28mm/s. In addition, since the vibration severity exceeds 82mm/s(half the value of Table 6) under a more typical earthquake such as Operating Basis Earthquake of 0.1g, the prototype structure will be likely to exhibit unacceptable performance.

6. CONCLUSION

In this study, a method to evaluate the vibration severity of rotating machines subjected to external vibration is developed using a comparison of the range of machine vibration severity by ISO and the RMS of the maximum response of the translational mode and torsional mode of the Fourier spectrum. A weighting factor expressed by the aspect ratio of the structure is employed to consider the effects of torsional coupling. And, the RMS sum of the maximum velocity response of the translational mode and the maximum velocity response of the torsional mode multiplied by the weighting factor are suggested to compare with the vibration severity recommended by ISO. A case study with a 3-D space frame structure which was idealized by FEM for analytical modeling of the prototype structure demonstrates an applicability of the evaluation method.

References

1. 大崎順彦 1976, 地震動のスペクトル解析入門, *鹿島出版會*, 216-217.
2. Mario Paz 1985, Structural Dynamics, *Van Nostrand Reinhold, N.Y.*, 29-59, 162-205, 505-509.
3. ISO 1983, International Standard ISO 2372, *ISO*, 1-7.
4. ISO 1985, International Standard ISO 3945, *ISO*, 1-9.
5. Committee on the Turbine Foundation 1987, Design of Large Steam Turbine-Generator Foundations, *ASCE*, 3-40.

Table 1 Relative values of spectrum amplification factors

Percent Damping	Amplification factors		
	Displacement	Velocity	Acceleration
0	2.5	4.0	6.4
0.5	2.2	3.6	5.8
1	2.0	3.2	5.2
2	1.8	2.8	4.3
5	1.4	1.9	2.6
7	1.2	1.5	1.9
10	1.1	1.3	1.5
20	1.0	1.1	1.2

Table 2 Frequency variation due to the aspect ratio of the simplified model

Ratio(L/W)	Width(W)	Length(L)	Y-direction	
			Translational Frequency	Torsional Frequency
1.00	5.00m	5.00m	6.04Hz	7.67Hz
1.33	5.00m	6.67m	5.64Hz	6.90Hz
2.00	5.00m	10.00m	5.03Hz	6.01Hz
4.00	5.00m	20.00m	3.93Hz	4.80Hz

Table 3 Quality judgement of vibration severity by ISO recommendation

<table>
<tr><th colspan="2">Vibration Severity</th><th colspan="2">Support Classification</th></tr>
<tr><th>V_{rms}(mm/sec)</th><th>V_{rms}(in/sec)</th><th>Rigid Supports</th><th>Flexible Supports</th></tr>
<tr><td>0.46</td><td>0.018</td><td rowspan="4">good</td><td rowspan="5">good</td></tr>
<tr><td>0.71</td><td>0.028</td></tr>
<tr><td>1.12</td><td>0.044</td></tr>
<tr><td>1.8</td><td>0.071</td></tr>
<tr><td>2.8</td><td>0.11</td><td rowspan="2">satisfactory</td></tr>
<tr><td>4.6</td><td>0.18</td><td rowspan="2">satisfactory</td></tr>
<tr><td>7.1</td><td>0.28</td><td rowspan="2">unsatisfactory</td></tr>
<tr><td>11.2</td><td>0.44</td><td rowspan="2">unsatisfactory</td></tr>
<tr><td>18.0</td><td>0.71</td><td rowspan="3">unacceptable</td></tr>
<tr><td>28.0</td><td>1.10</td><td rowspan="2">unacceptable</td></tr>
<tr><td>71.0</td><td>2.80</td></tr>
</table>

Table 4 Estimated natural frequencies of the system

X-Direction	Y-Direction		Z-Direction
Translational Mode	Translational Mode	Torsional Mode	Vertical Mode
7.0Hz	8.3Hz	10.4Hz	16.5Hz

Table 5 Estimated damping ratios of the system

X-Direction	Y-Direction		Z-Direction
Translational Mode	Translational Mode	Torsional Mode	Vertical Mode
4.8%	2.6%	2.6%	3.1%

Table 6 Calculated ranges of vibration severity

Peak value(mm/sec)		Weight(L/W)	Calculated Range of Vibration Severity
Translational mode(A}	Torsional mode(B)	(C)	$V_{rms}=\{1/2[A^2+C(B^2)]\}^{1/2}$
160	96	3	163

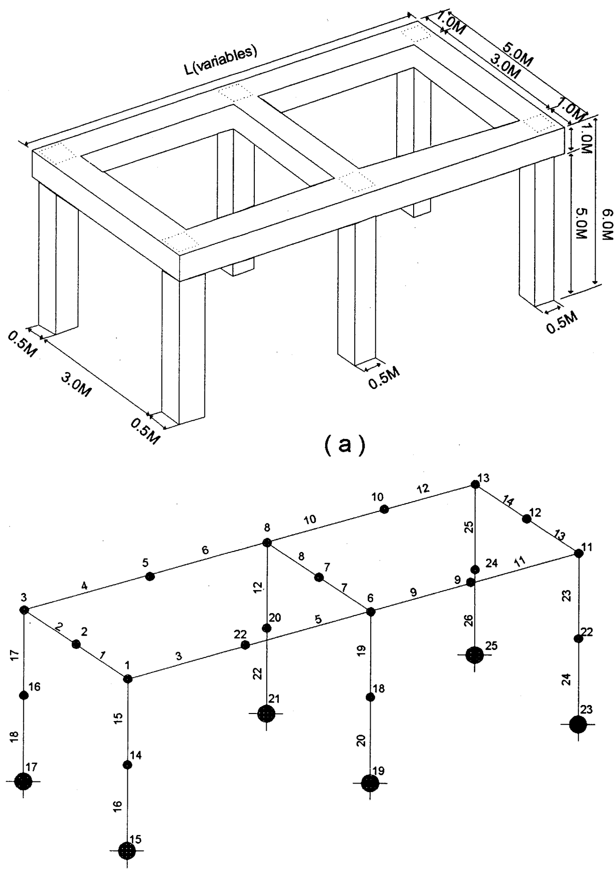

Fig. 1 Simplified turbine-generator foundation(a) and its FEM modeling(b)

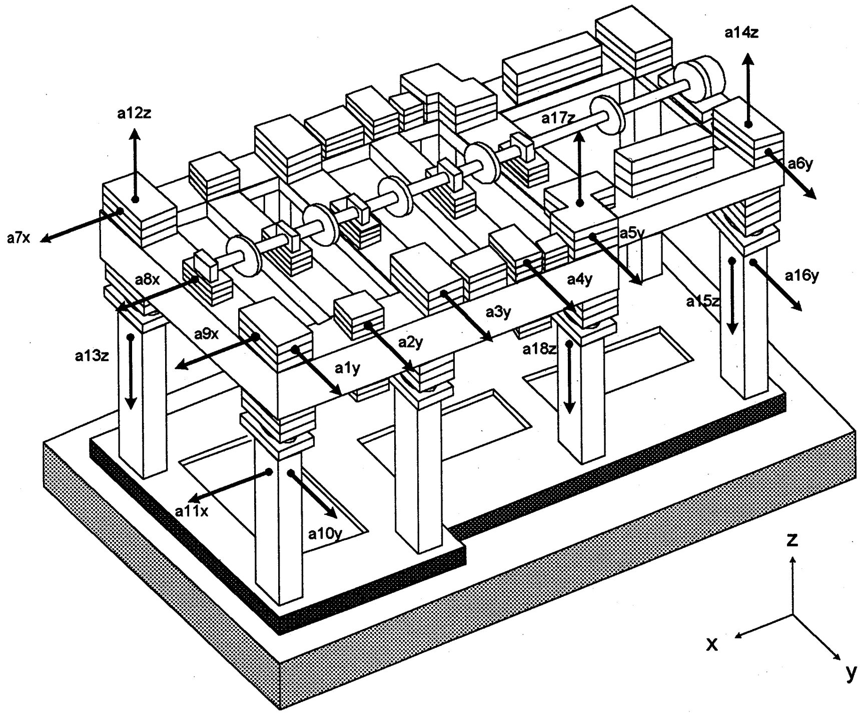

Fig. 2 1/10 scaled test model structure employing plate spring elements

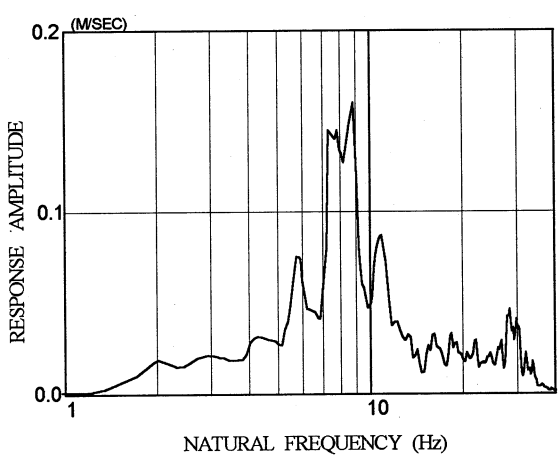

Fig. 3 Fourier Amplitude in the Y-direction