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SEISMIC RESPONSE CONTROL IN NUCLEAR PIPING SYSTEM BY DYNAMIC ABSORBER

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SUMMARY

This paper deals with the application of dual dynamic absorbers to nuclear piping systems to accomplish high damping value and reduce seismic response at resonance frequencies. The transfer matrix method is used for designing the dual dynamic absorbers as well as for determining the optimum mounting location. The effectiveness of the dynamic absorbers is demonstrated by suppressing the first three resonance peaks in the 3-dimensional model piping system.

INTRODUCTION

Is is important to suppress seismic response at resonance frequencies of piping systems to ensure safety in nuclear plants during large earthquakes. Thus, a new trend has recently appeared to use damping devices in designing nuclear piping systems. Kunieda (Ref.1) reported an idea on application of a dynamic damper to piping systems to reduce seismic vibration response. The authors (Ref.2) also reported the basic concept about the application of dual dynamic absorbers to seismic design of nuclear piping systems.

This paper shows how to suppress many resonance peaks of the piping system, since it is affected by complex vibrations with many resonance frequencies throughout the low range of frequency, by using the method of vibration control of multi-degree-of-freedom system by dynamic absorbers which has been established by one of the authors (Ref.3,4).

The essence of this method is in the systematic combining of the modal analysis technique with the design method of dynamic absorbers. By means of modal analysis, the mounting location of the dynamic absorbers to a 3-dimensional piping system as a vibration controlling object is determined, and the equivalent mass at the location, which is the major value on designing the dynamic absorber, is estimated. Established design is then applied to dynamic absorbers by using the equivalent mass obtained for each vibration mode. Effectiveness of this method is demonstrated by suppressing the first three resonance peaks completely through not only numerical calculation but also experimental evaluation.

DESIGN CONCEPT OF DYNAMIC ABSORBERS OF CONTROLLING THE VIBRATION IN PIPING SYSTEM

In order to establish vibration control in a piping system by using dynamic absorbers, it is essential to combine the modal analysis technique with the established design method of the dynamic absorber for controlling one-degree-of-freedom vibration systems. According to modal analysis, nth-degree-of-freedom systems such as the piping system with many resonance frequencies are decoupled into nth one-degree-of-freedom system in the modal domain. This creates better control of the vibration at each one-degree-of-freedom system using the dynamic absorbers designed by the established method.

The modal analysis technique is used to determine a suitable location for installing the dynamic absorber on each vibration mode shape, and to estimate the equivalent mass at the location corresponding to each one-degree-of-freedom system.

Applying the concept for controlling the vibration of the piping system, the design procedure of the dynamic absorber is as follows:

(1) Analyze the vibration mode shapes using modal analysis.

(2) Determine the mounting location of the dynamic absorber at the maximum amplitude point on each vibration mode shape. In principle, the mounting location should be at the vibration node of other mode in order to obtain the decoupled condition of each mode.

(3) Estimate the equivalent mass at each mounting location of the dynamic absorber.

(4) Design the dynamic absorber using the equivalent mass as a basic design parameter.

VIBRATION ANALYSIS IN THE PIPING SYSTEM

The Configuration in the Piping System The configuration of the piping system selected in this paper as a vibration controlling object is shown in Fig. 1. This is made of copper tubing 25 mm in outside diameter and 15 mm in inside diameter, and fixed at both ends of the tube.

In order to analyze the vibration characteristics and examine the vibration mode shapes of the 3-dimensional piping system , the transfer matrix method is used. The piping system is divided in two fundamental elements, which consists of 21 pieces of a piping element and 4 pieces of a coordinate transformation element. In applying the transfer matrix method, it is necessary to prepare a 3-dimensional transfer matrix representation of these two fundamental elements.

For representing 3-dimensional transfer matrix of piping elements, a 13*13 matrix, which includes partial matrixes for the axial and torsional vibration of an elastic shaft, and for the flexural vibration on x direction and y direction of a beam in addition to internal force term, was constructed (Ref.5). For coordinate transformation, also three 13*13 matrixes of rotation about x, y, and z axes owing to right-hand-screw were prepared (Ref.5).

The piping system in Figure 2 consists of a pipe divided into 21 elements with four coordinate transformations expressed by transfer matrixes. Hence B_i is the transfer matrix of ith piping element, T_j is of jth coordinate transformation, and Z_k is the state vector at kth point. State vectors Z_1 and Z_{26} at the ends are given from the boundary conditions as follows:

$$Z_1=\{0\ N\ 0\ T\ 0\ 0\ M_y\ F_y\ 0\ 0\ M_x\ F_y\ 1\}_1^T \quad (1)$$
$$Z_{26}=\{0\ N\ 0\ T\ 0\ 0\ M_y\ F_y\ 0\ 0\ M_x\ F_y\ 1\}_{26}^T \quad (2)$$

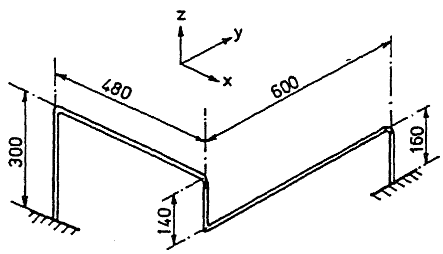


Fig. 1 Construction of a piping system as a vibration control object

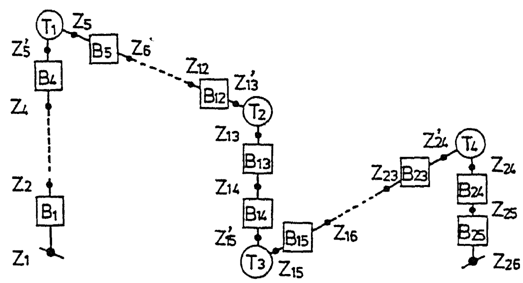


Fig. 2 Transfer matrix representation of of the piping system

Frequency Analysis A transfer matrix operation based on Fig.2 yields the following state vectors at each point:

$$\left.\begin{array}{l} Z_{25}=H_{25}\cdot Z_{26} \\ Z_{24}=H_{24}\cdot Z_{26} \\ Z_{23}=H_{23}\cdot Z_{26} \\ \vdots \quad \vdots \\ Z_2=H_2\cdot Z_{26} \end{array}\right\} \qquad (3)$$

$$Z_1=H_1\cdot Z_{26} \qquad (4)$$

where

$$H_{25}=B_{21},\ H_{24}=B_{20}\cdot H_{25},\ H_{23}=T_4\cdot H_{24},\cdots,$$
$$H_2=B_2\cdot H_3,\ H_1=B_1\cdot H_2$$

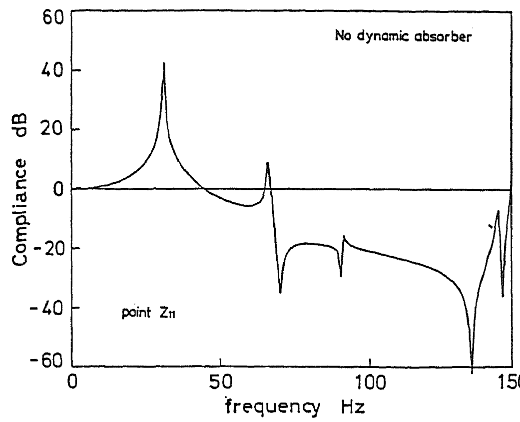


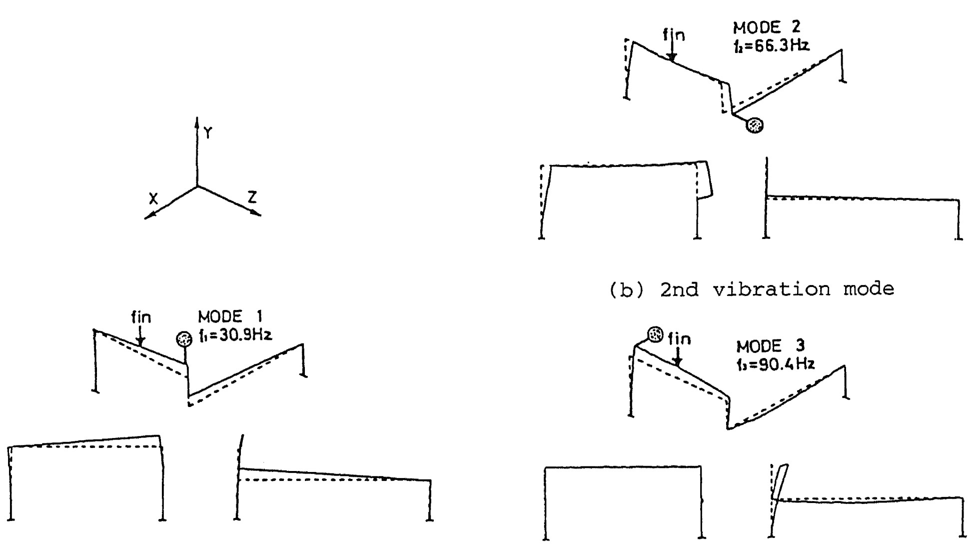
Fig. 3 Calculated compliance

State vector Zn+1 is known if unknown variables are determined using Eqs. (1), (2), and (4), and substituting the result in Eq. (3) enable state vector Zi to be determined at each contact point.

Fig. 3 shows compliance obtained at point 11 when the point 7 is excited in an up and down direction. There are three resonance peaks below 100 Hz: fn1=30.1 Hz, fn2=66.3 Hz, and fn3=90.4 Hz, to be selected for the controlling object.

Vibration Mode Shapes To control the first three resonance peaks, the corresponding three vibration mode shapes must be examined. Figure 4 shows the natural frequencies and the corresponding mode shapes with front and side views calculated by the transfer matrix method. From this figure, maximum amplitudes are located

at the node point 12 toward y axis for the 1st mode;
at the node point 16 toward z axis for the 2nd mode;
and at the node point 6 toward x axis for the 3rd mode.



(a) 1st vibration mode

(b) 2nd vibration mode

(c) 3rd vibration mode

Fig. 4 The first three resonance frequency and corresponding mode shapes

Mounting location for the dynamic absorber and determination of equivalent mass at its location The location for dynamic absorbers to be mounted on the piping system must be at a point where the maximum amplitude appears for each vibration mode shape. Since the modal mass is the smallest at the point, it is an advantage in reducing the mass of the dynamic absorber to be used. In Fig. 4 dynamic absorbers are thus mounted at points marked by a solid dot.

The equivalent mass Mij at jth point on ith vibration mode is calculated by premultiplying and postmultiplying the eigen vector, normalized to the largest element at jth point to be 1, with mass matrix.(Ref.6) The equivalent mass must be calculated at the mounting location on the corresponding vibration mode. Calculation of the equivalent masses Mi (i=1,2,and 3) at the absorber mounting location in the respective vibration modes is shown as follows:

M1=1.42 kg, M2=1.62 kg, M3=1.31 kg

Design of dual dynamic absorbers A dual dynamic absorber attached for controlling ith vibration mode is shown in Fig.5, where Mi and Ki are ith-mode equivalent mass and stiffness, and mri, kri and cri are absorber mass, stiffness, and damping coefficient (r=1,2). When compared with a conventional single dynamic absorber, the dual dynamic absorber has the advantage of broad-band suppressing capacity for resonance peaks. Giving the mass ratio as the basic parameter, the dimensions of the dual dynamic absorber are determined by the following equations established by one of the authors.(Ref.7)

$$\begin{aligned} &m_{1i}=m_{2i}=\mu_i M_i \qquad k_{1i}=m_{1i}\left(\frac{K_i}{M_i}\right)[0.403\cdot(\mu_i+0.13)^{-0.434}]^2 \qquad c_{1i}=2\sqrt{m_{1i}k_{1i}}[(\mu_i/17.6)^{0.285}-0.065] \\ &\qquad\qquad\qquad\qquad k_{2i}=m_{2i}\left(\frac{K_i}{M_i}\right)[1.04-0.72\mu_i]^2 \qquad c_{2i}=2\sqrt{m_{2i}k_{2i}}[(\mu_i/3.06)^{0.377}-0.062] \end{aligned} \tag{5}$$

Therefore, mass ratios are selected for each mode as follows:

$\mu_1=0.05$, $\mu_2=0.05$, $\mu_3=0.04$

Substituting these values in Eqs.(5), dimensions of the optimally designed dual dynamic absorbers are listed in Table 1.

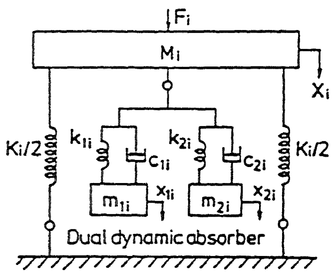


Fig. 5 Dual dynamic absorber

	absorber 1			absorber 2		
Mode	m(kg)	k(N/m)	c(Ns/m)	m(kg)	k(Ns/m)	c(Ns/m)
1st	0.071	1920	2.87	0.071	2690	4.14
2nd	0.081	10100	7.04	0.081	14100	10.16
3rd	0.052	12700	5.77	0.052	17200	8.00

Table 1 Dimensions of dual dynamic absorbers

RESPONSE CALCULATION OF THE PIPING SYSTEM WITH OPTIMALLY DESIGNED DUAL DYNAMIC ABSORBERS

How much the three dual dynamic absorbers suppress vibration is examined below through response calculation, requiring dynamic absorber representation by the 3-dimensional complex transfer matrix. The transfer matrix of the dual dynamic absorber located at the point between state vectors Za and Zb is also expressed as a 13*13 matrix (Ref.5).

In order to confirm the effectiveness of dual dynamic absorbers for controlling the vibration in piping system, numerical evaluation was performed. Figure 6, and 7 give the results of the numerical calculation for compliance

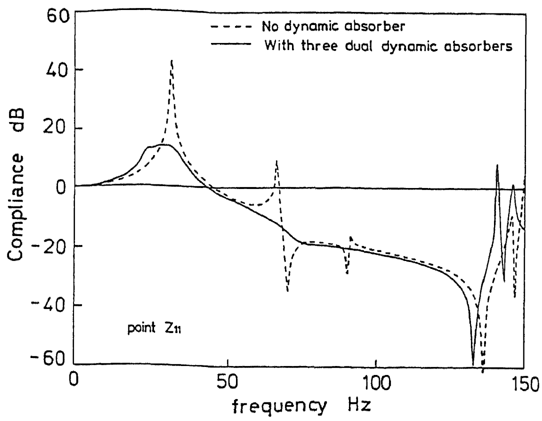


Fig. 6 Compliance measured at point 11 on y axis

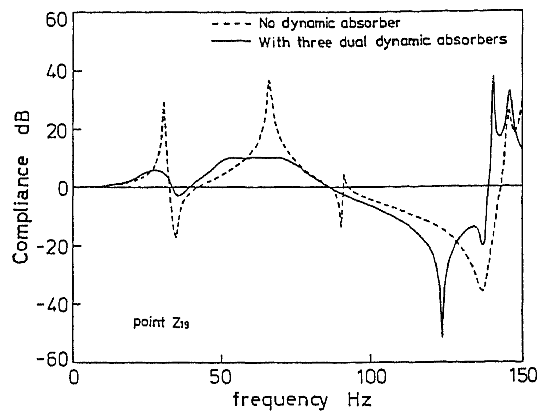


Fig.7 Compliance measured at point 19 on z axis

at point 11 on y axis, and at the point 19 on z axis, respectively, where the excitation point is indicated by the symbol (↓) in Fig. 4. The solid line is the response of the piping on which three dual dynamic absorbers were mounted in conformity to the values given in Table 1, and the broken line for the piping without absorbers. The first three resonance peaks at 30.9 Hz, 66.3 Hz, and 90.4 Hz are suppressed markedly by the three dual dynamic absorbers.

EXPERIMENT

Construction of the Dual Dynamic Absorber The setup of dual dynamic absorber constructed in this study is shown in Fig. 8. The dual dynamic absorber is composed of parallel plate-springs, absorber masses that can slide on the spring, and magnetic dampers that use eddy current loss. In the magnetic damper, the poles of a pair of permanent magnets attached to both sides of a U-shaped magnet holder make a uniform magnetic field in the air gap. The absorber mass made of copper traverses the air gap without mechanical contact and generates a magnetic damping force.

The advantages of the dynamic absorber are that optimum damping given in advance is kept very stable during use and that spring constant and damping are adjustable to obtain optimum tuning and damping.

Experimental Result An experiment was done to confirm the theoretical results. The 3-dimensional piping system with three dual dynamic absorbers were excited by an impulse hammer and the accelerance of the piping was obtained. Fig. 9 shows the experimental results for the accelerance measured at point 11 on y axis, indicating that the first three resonance peaks were well suppressed by the three dual dynamic absorbers. The experimental result agrees well with the result of the theoretical analysis.

Figure 10 compares impulse response with and without three dynamic absorbers. The damping effect of the optimally designed dual dynamic absorbers become more evident. Actual measurement in terms of impulse response also shows the excellent adaptability of the dual dynamic absorbers for reducing seismic response in the piping system.

In order to confirm the above mentioned effect, a random exciting measurement shown in Fig. 11 has been done on a shaking table. It is found that the vibration amplitude is reduced below one fourth when three dual dynamic absorbers are attached. If more reduced results are required, it will be attempted by suppressing resonance peaks above the fourth resonance ones.

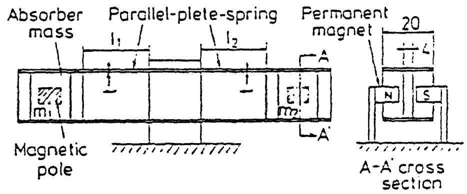


Fig.8 Setup of Dual dynamic absorber

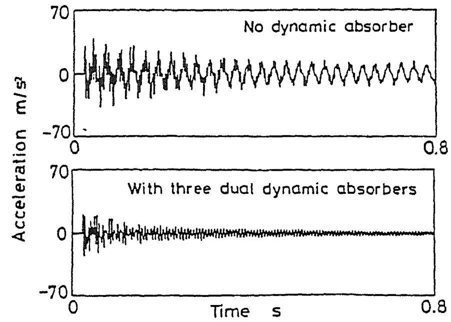


Fig. 10 Impulse response

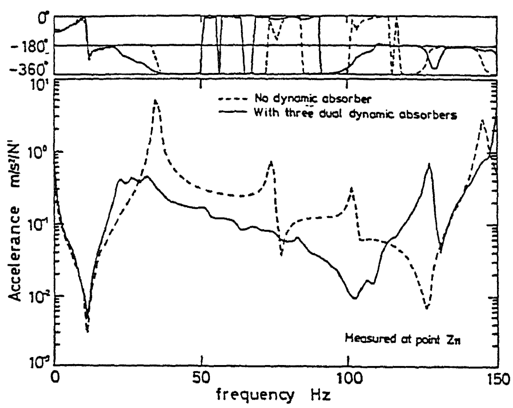


Fig. 9 Accelerance measured at point 11 on y axis

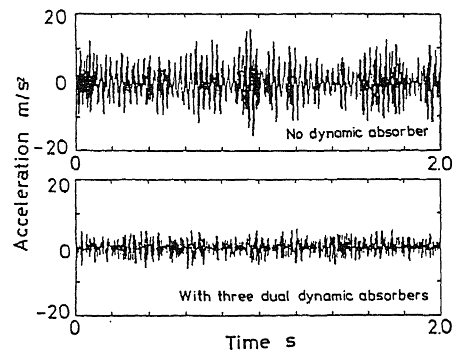


Fig. 11 Experimental result under the random excitation

CONCLUSION

Three dual dynamic absorbers adjusted based on the design procedure proposed here suppress sufficiently the first three resonance peaks of the 3-dimensional piping system. This idea can be used for realizing non-resonance piping systems. Use of dual dynamic absorbers for seismic design of nuclear piping systems may give us freedom from confrontation between high damping and soft piping system.

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