



Study on 3-Dimensional Base Isolation System Applying to New Type Power Plant Reactor (Hydraulic 3-Dimensional Base Isolation System: No.1)

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ABSTRACT

In Japan, a number of three-dimensional base isolation systems have been studied for application to new nuclear plant concepts such as the fast breeder reactor (FBR), but these efforts have not so far yielded practically applicable results. The impeding factor has been the difficulty of obtaining an adequate capacity on the vertical isolator for supporting the mass of an actual structure and for suppressing rocking motion.

In this paper, we propose a new three-dimensional base isolation system that should solve the foregoing problem. The system is constituted of a set of hydraulic load-carrying cylinders connected to accumulator units containing a compressed gas, another set of rocking-suppression cylinders connected in series, and a laminated rubber bearing laid under each load-carrying cylinder. The present paper covers a basic examination for applying the proposed system to a commercialized FBR now under development in Japan, together with static and dynamic loading tests performed on a scale model to verify expected system performance. Response analysis reflecting the test results has indicated the proposed system to be well applicable to the envisaged commercialized FBR.

The study was undertaken as part of an R&D project sponsored by the government for realizing a three-dimensional seismic isolation system applicable to future FBR's.

KEY WORDS: 3-Dimensional, base isolation, rocking suppression, hydraulic system, air spring, accumulator, FBR, nuclear power plant, laminated rubber bearing

INTRODUCTION

Seismic base isolation calls for abating seismic force not only in horizontal but also in vertical direction. To this end, R&D efforts in Japan have envisaged application to new nuclear plant concepts such as FBR and the fusion reactor, with the aim of simultaneously ensuring high earthquake resistance and cost economy. An effective device of such description should also fill the needs of non-nuclear structures[1].

In the present state of development, three-dimensional seismic isolation has been realized for the floor level of buildings, but it still remains to develop a system that would abate response of the building as a whole. The difficulty lay in the lack of an effective means of supporting the mass of an actual structure, having a vertical natural period exceeding say 1 second. Another impediment was the difficulty of suppressing rocking motion.

The solution to the foregoing problem, as proposed by the present authors, is the application of a hydraulic mechanism[2], which would permit realization of a compact and large-capacity three-dimensional seismic isolation system.

The present paper covers a basic examination envisaging application of the proposed system to a commercialized FBR now under development in Japan, together with static and dynamic loading tests performed on a scale model to verify expected system performance.

The study was undertaken as part of an R&D project sponsored by the government for the purpose of realizing a three-dimensional seismic isolation system applicable to future FBR's.

SYSTEM OUTLINE

System Structure

The system proposed is constituted of the components depicted in Fig. 1, arranged as shown in Fig. 2. The three key components are the isolators (Fig. 1(a)) inserted at intervals between the ground and the structure to be supported, the accumulator unit (Fig. 1(b)) providing the vertical isolation function, and the rocking suppression cylinders (Fig. 1(c)). Isolator function is ensured by the combination of the load-carrying cylinder and the laminated rubber bearing providing isolation function against horizontal excitation. The space above the piston in an isolator is connected by piping—either directly or via rocking suppression cylinder—to one of four first-stage tanks of the accumulator unit.

Vertical isolation

In the event of seismic excitation, the fluctuating vertical seismic force is converted by the load-carrying cylinders (such as shown in Fig. 1(a)) into pressure fluctuations of hydraulic fluid, each cylinder being connected through piping to the first-stage tank of the accumulator unit (Fig. 1(b)). In this tank, the hydraulic fluid space is bounded by a flexible bladder containing nitrogen gas, to which the fluctuating pressure is transmitted. The vertical restoring force to be applied to structure is generated by the bulk modulus of the gas contained in this first-stage tank and in the second-stage tank of constant volume. An orifice installed in the pipe connecting the first- to the second-stage tanks generates requisite vertical damping force.

In particular, as indicated in Fig. 2, the load-carrying cylinders situated along the structure periphery are connected together in groups of four, the piping from each group then being led to a rocking-suppression cylinder.

Rocking suppression

The rocking-suppression cylinders (Fig. 1(c)), arranged along the structure periphery (Fig. 2), are connected in series through each side of the structure by means of swivel joints. As depicted in the inserted diagram of Fig. 2, each cylinder has one chamber connected to four load-carrying cylinders, and the other chamber to one of the four accumulator units. The swivel joints serve to equalize vertical displacement of the load-carrying cylinders connected to the rocking-suppression cylinders, to firmly ensure suppression of the rocking motion.

System Specification

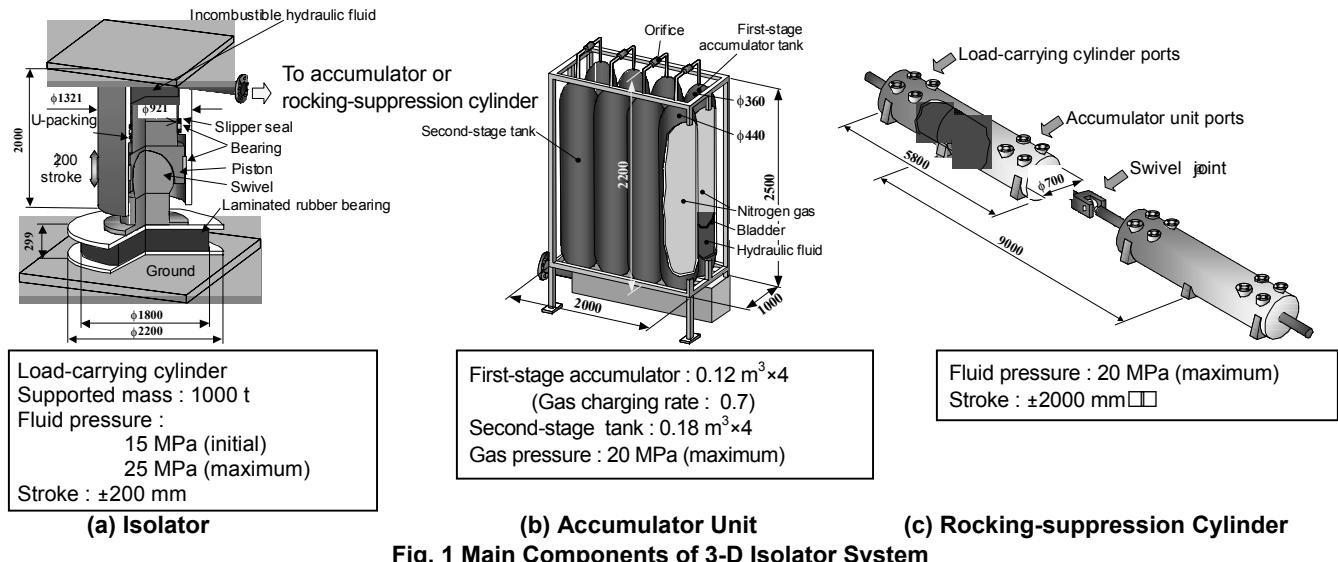


Fig. 1 Main Components of 3-D Isolator System

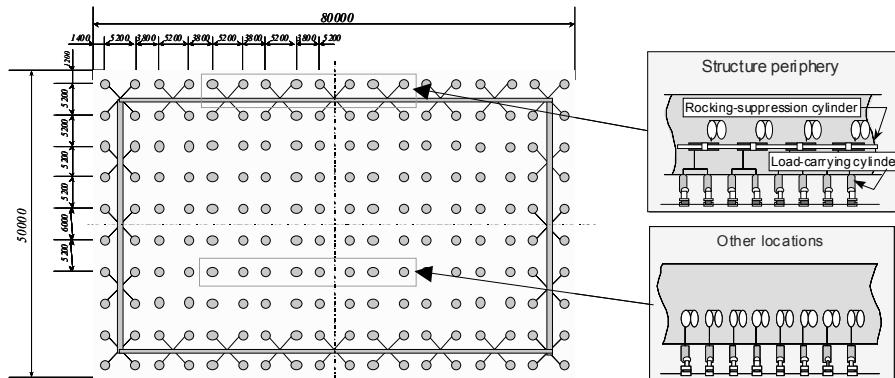


Fig. 2 Arrangement of Components

The basic specifications of the system adapted to the commercialized FBR mentioned earlier are as given in Table 1. For the commercialized FBR[3], a natural frequency below 1.0 Hz and a damping ratio above 20% in vertical direction has been specified, in keeping which, 0.5 Hz natural frequency and 20% damping ratio have been adopted as target of the proposed system.

The 180 isolators, each of 1000-ton loading capacity, are adapted to support the FBR structure of about 180,000 tons. Taking account of the sealing performance of the hydraulic fluid, each load-carrying cylinder of the isolator is dimensioned to let the fluid come to a pressure of 15 MPa when supporting a mass of 1,000 tons (not to exceed 25 MPa under added load due to seismic excitation).

The natural frequency and damping ratio in the horizontal direction have been derived without change from the dynamic characteristics of an existing 1,000-ton class high damping laminated rubber bearing, adopted for reference.

In the vertical direction, on the other hand, the natural frequency and damping ratio have been determined from the accumulator unit specification. Envisaging the natural frequency of 0.5 Hz and damping ratio of 20% mentioned earlier, the calculation described in the following section has indicated that the first-stage accumulator with bladder should comprise four tanks of 120 liters each, filled with nitrogen gas at 15 MPa to 70% of the tank volume. From similar calculation, the

Table 1 Main Isolation Specification for Commercialized FBR

Isolator	Supported mass	1000t per Isolator
	Number	180
Horizontal	Natural frequency	0.54 Hz (equivalent stiffness at 200 mm deformation)
	Damping	16% (equivalent damping ratio at 200 mm deformation)
Vertical	Natural frequency	0.5 Hz
	Damping ratio	20%
	Stroke	±200 mm
Supplementary device		Rocking-suppression unit

second-stage constant-volume accumulator has been made to comprise four tanks of 180-liter capacity each, individually connected to one of the first-stage tanks.

VERIFICATION TEST

Purpose of Test

With the intention of verifying the feasibility of the proposed system, a test was conducted covering the following items:

- a. Natural frequency and damping ratio

To ascertain the feasibility of realizing the envisaged conditions of 0.5 Hz natural frequency and 20% damping ratio.

- b. Friction between piston and cylinder of load-carrying cylinder

To verify the efficacy of the bearings adopted for diminishing friction, which could affect system performance.

Method of Test

The scale factor to hold between the model adopted for the test and the system to be adopted in actual practice is indicated in Table 2. A ratio of 7.5:1000 has been adopted for the supported mass, for the piston area and for the volume of gas space in the first- and second-stage accumulator tanks. A scale factor of unity has been set for the hydraulic fluid pressure and for the pressure on the bearing subjected to horizontal seismic force (Fig. 1(a)), which respectively affect sealing performance and friction characteristics. The above scheme proved to equalize the calculated natural frequency between the systems of model and actual practice.

The testing arrangement is depicted in Fig. 3(a), and the equipment is illustrated in Fig. 3(b). An actuator is set to impart load to the upper part of the model for forcing its dynamic displacement. The flow-resisting orifice takes the form of a needle valve, whose opening is varied by turns of the valve spindle, to examine the effect on system functioning that is brought by changes in orifice opening.

For piston friction testing, the model load-carrying cylinder is statically loaded horizontally to an intensity corresponding to seismic force.

Table 2 Scale Factors between Test Model and Actual System

Item	Actual system	Test model	Scale factor
Supported mass (t)	1000	7.5	7.5/1000
Piston area (m^2)	0.667	0.005	7.5/1000
Gas volume of first-stage accumulator tank (m^3)	0.333	2.5×10^{-3}	7.5/1000
Gas volume of second-stage tank (m^3)	0.733	5.5×10^{-3}	7.5/1000
Natural frequency (Hz)	0.5	0.5	1/1
Fluid pressure (MPa)	14.7	14.7	1/1

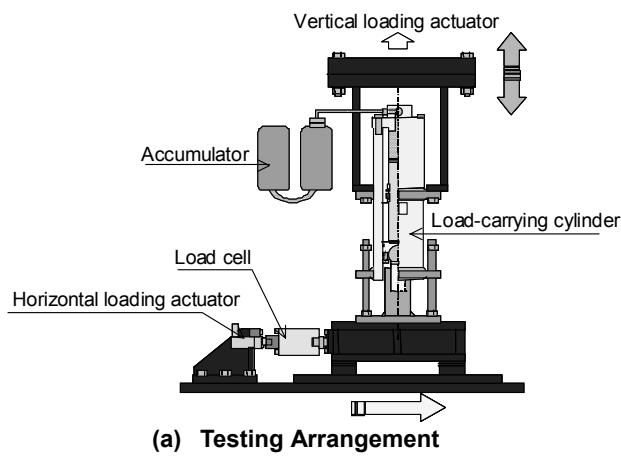


Fig. 3 Test Model

TEST RESULTS

Theory of orifice damping

The natural frequency of the system in the vertical direction is determined by the formula

$$f = \frac{1}{2\pi} \sqrt{\frac{\gamma g A}{V_1 + V_2}} \quad (1)$$

where γ : Polytropic index

A : Piston surface area

V_1 : Volume of gas space in first-stage tank

V_2 : Volume of gas space in second-stage tank

The above relation is conditioned on the provision of an orifice with a sufficiently large opening located between the first- and second-stage accumulator tanks. As explained further on, the natural frequency is affected by the orifice opening. The natural frequency and damping ratio are evaluated applying Kunieda's "theory of orifice damping effect by air spring"[4]. This theory permits deriving through linearization the energy dissipation brought by the pressure difference between the two sides of an orifice, assuming the energy dissipation to be proportional to the square of gas velocity at the orifice.

Figure 4 presents resonance curves obtained using the above theory, indicating—for different orifice openings—the relation between amplification factor and excitation frequency.

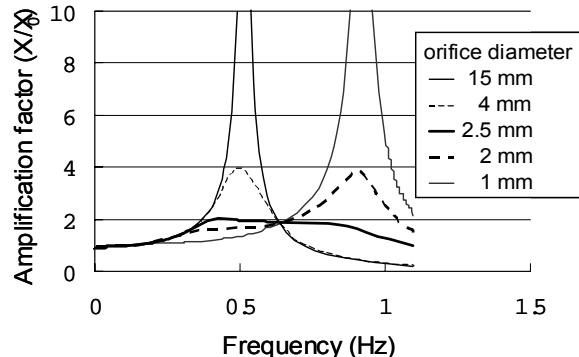


Fig. 4 Frequency Response Function based on Orifice Damping Theory

In Fig. 4 X_0 and X are respectively excited ground displacement and absolute response displacement.

It can be seen that, with a very large orifice opening, when little orifice resistance is presented, the two accumulator tanks would function as a single vessel, to produce theoretical infinite resonance at a low frequency (natural frequency calculated from Eq. (1)). In the converse case of very small orifice opening, only the first-stage tank would effectively function, to bring about infinite resonance at a high frequency.

With an intermediate orifice opening, the orifice resistance will lower the resonance to a finite level at an intermediate frequency, with curves for different orifice openings all passing through a fixed point. The orifice opening that brings the resonance frequency to this point is that which would minimize the amplification factor—i.e. which would maximize damping—and which would thus present the optimum orifice opening (2.5 mm orifice diameter in Fig. 4).

It must be noted, however, that different excitation amplitudes would produce different resonance frequencies and amplification factors, i.e. that damping ratio and resonance frequency present nonlinear characteristics with change in orifice opening and excitation amplitude.

In the instance of the model used in the verification test, different orifice openings gave the performances presented in Table 3. The damping ratio ζ under the optimum orifice condition is approximately expressed by the formula

$$\zeta = \frac{1}{2\left(1 + 2\frac{V_1}{V_2}\right)} \quad (2)$$

indicating that increasing the volume of the second-stage tank relative to that of the first stage will enhance damping.

Making effective use of the above nonlinear relationship should permit devising measures for obtaining high resonance frequency with high damping effect, and for avoiding excessive piston stroke in the event of inordinately large seismic excitation.

Table 3 Resonance Frequency and Damping Ratio based on Orifice Damping Theory

Orifice opening	Resonance frequency (Hz)	Damping ratio (%)
Fully opened	0.52	0
Shut	0.94	0
Optimum opening	0.65	25

Natural frequency, damping coefficient

As an example of test result, Fig. 5 presents the hysteresis curves relating force to displacement for a typical case of 90 mm forced displacement at 0.06 Hz. The force is equal to [pressure \times piston area] of the load-carrying cylinder. From the inclinations and surface areas of the three hysteresis curves, the stiffness and damping capacity have been derived as given in Table 4. In respect of stiffness, it is seen to increase with diminishing orifice opening, which behavior agrees with the theoretical data of Table 3.

Table 4 Test Results of Resonance Frequency and Damping coefficient

Orifice opening	Spring Constant (N/m)	Resonance frequency (Hz)	Equivalent damping coefficient (N·s/m)
0 turn (Fully opened)	8.09×10^4	0.52	7.25×10^3
6.25 turns	8.35×10^4	0.53	6.55×10^4
6.5 turns (Shut)	2.69×10^5	0.95	2.45×10^4

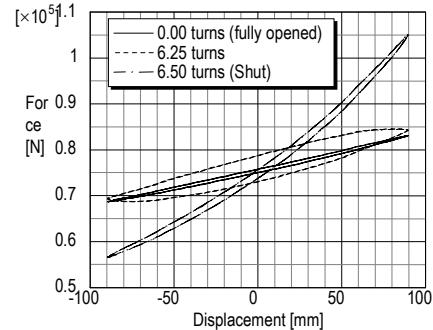


Fig. 5 Test Results of Hysteresis Curves

In respect of damping, comparison between theoretical and empirical data calls for evaluating the relation between flow velocity and the pressure difference generated by the orifice and for quantitatively expressing the differences in loading condition. As regards the first above requirement, the evaluation was made using the C_v value of needle valve, which is expressed by

$$C_v = \sqrt{\frac{\rho}{\Delta P} V} \quad (3)$$

where ρ : Specific gravity of gas

ΔP : Pressure difference generated by orifice

V : Flow velocity

The above equation indicates that the pressure difference ΔP is proportional to the square of flow velocity. The relation between valve opening and C_v in the case of the orifice used in test was as shown in Fig. 6.

As regards the second requirement concerning differences in loading condition, this was evaluated by inversely deriving the flow velocity at orifice from the displacement forced on load-carrying cylinder.

The resulting comparison between theoretical and empirical data for equivalent damping coefficient is presented in Fig. 7. Good agreement between theory and test is seen for the orifice openings at 6.25 and 5.5 turns of spindle. With the orifice fully opened, the test data exceed the theoretical value, which can be ascribed to the resistance generated by the connecting piping two tanks, not considered in the theoretical calculation.

It can be concluded from the foregoing findings that, all in all, the system ensures good agreement with theory on values

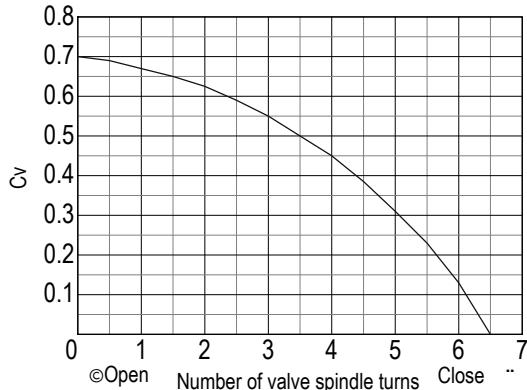


Fig. 6 Characteristic C_v Value of the Valve used in Experiment

of natural frequency and damping coefficient.

Friction characteristics

Test data obtained on friction characteristics are as presented in Fig. 8, which indicates the values of force registered on load cell mounted on actuator and those derived from the load-carrying cylinder fluid pressure. Friction resistance is the difference in force values between that from actuator load cell—which includes friction force—and that from cylinder fluid pressure—which it does not include.

Even in the absence of applied horizontal force, friction would be generated from such factors as sealing action. Actually, however, the friction resistance proved to be only 4.4 kN at 29.4 kN horizontal force, corresponding to a friction coefficient of 0.15, which can be considered quite small.

The above performance can be attributed to the swivel mechanism provided on the load-carrying cylinder (see Fig. 1(a)), that avoids causing reaction on the upper bearing with the moment that would be generated when the load-carrying cylinder is subjected to horizontal seismic force.

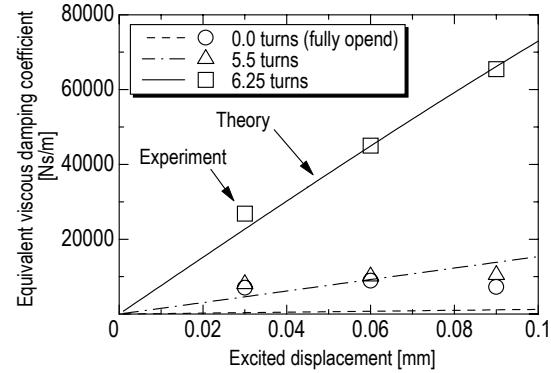


Fig. 7 Comparison Between Theoretical and Experimental Equivalent Damping Coefficients

SIMULATION OF ISOLATION PERFORMANCE

Modeling of Isolation System

For modeling the function of isolation system in horizontal, vertical and rotational directions, the characteristics of the system are represented by a spring and a damping element concentrated at the center of the actual isolation system. The horizontal spring element is represented by a modified bilinear model derived from a 1000-ton class high-damping laminated rubber bearing.

Envisaging an accumulator unit such as shown in Fig. 1(b), with the orifice generating a damping ratio of 20%, modeling of the vertical spring and damping element is obtained by combining a spring element with a damping element proportional to the square of flow velocity.

Modeling of the rotating element is obtained by means of the spring function provided by the hydraulic fluid column contained in the connecting piping between the load-carrying cylinder and the rocking-suppression cylinder. The bulk modulus of hydraulic fluid is assumed to be somewhat smaller than that of the hydraulic fluid in natural condition, in consideration of air immixture into the system, and elasticity of sealing material. This has led to a value of 4.3 Hz for the rocking mode natural frequency.

Consideration is also given to the friction of the piston in the load-carrying cylinder, and the effect is examined analytically.

As input seismic waves for analysis, the S2 waves from a case study were used, which were estimated for application to the envisaged commercialized FBR.

Results of Horizontal and Rocking Response Analyses

Figure 9 presents the analyzed data of response to excited horizontal ground acceleration (diagram (a)), resulting in the horizontal response acceleration shown in diagram (b), the horizontal relative displacement of diagram (c), and the vertical displacement due to rocking motion of structure of diagram (d).

It is seen that, against a maximum ground acceleration of 831 gal, the response acceleration is 0.42 times. Relative displacement is 400 mm maximum, corresponding to about 250% in laminated rubber deformation, which is amply small.

Rocking of the structure (evaluated on the vertical displacement of structural edge) is suppressed to about 2 mm maximum, indicating markedly good rocking suppression performance.

Results of Vertical Response Analyses

Figure 10 shows frequency response curves for relative displacement, evaluated from the vertical response of an actual system, estimated by applying the theory of gas orifice damping[4].

Comparing in Fig. 10 the results of analysis with linear model assuming 20% viscous damping, it is revealed that, for small amplitude, orifice damping yields a larger relative displacement response compared with viscous damping, but that increasing the input amplitude diminishes the response of ori-

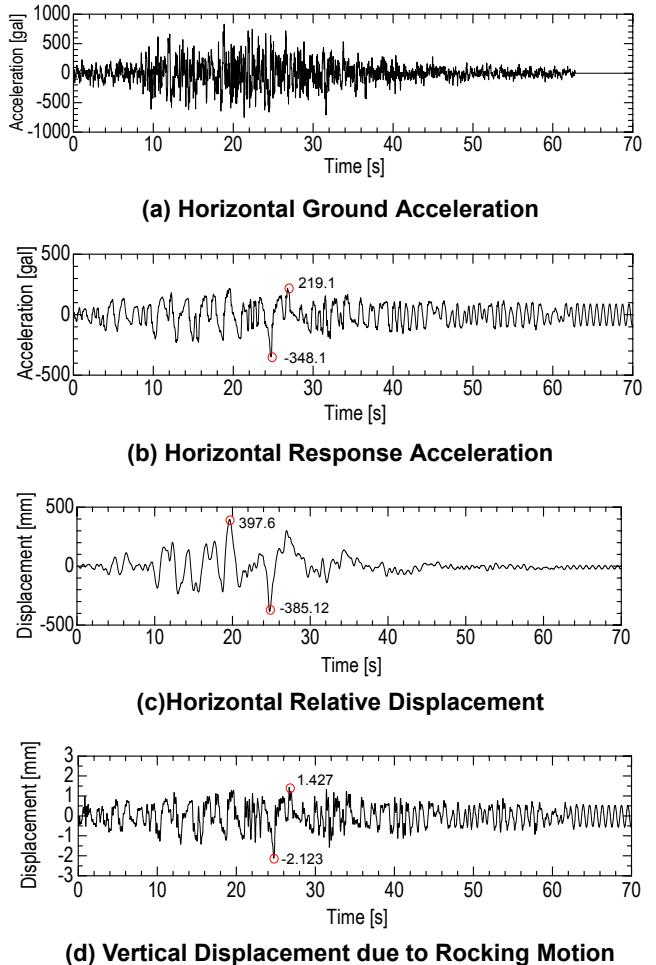


Fig. 9 Analyzed Data of Horizontal and Rocking Responses

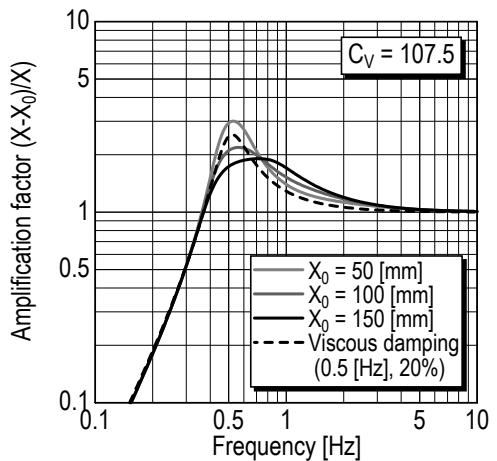


Fig. 10 Vertical Frequency Response Curves

fice damping below that of viscous damping, and raises the resonance frequency.

Using the same model, the response of an actual system to seismic acceleration was analyzed, the results being as shown in Fig. 11. This figure presents the analyzed data of response to excited vertical ground acceleration (diagram (a)), resulting in the vertical response acceleration shown in diagram (b), and the vertical relative displacement of diagram (c).

As it was in Fig. 10, the above results again indicate response characteristics similar to those obtained from analysis with the 20% viscous damping model. It can thus be concluded that orifice damping is roughly equivalent to 20% viscous damping.

Shown in Fig. 12 are the same response waves as in Fig. 11, but with account taken of the friction characteristics of the load-carrying cylinder obtained from test. In the analysis, the relation between horizontal force and friction force was adapted for applying the test results to the actual system, and the resulting relation was applied to the horizontal force derived from horizontal response analysis. It can be concluded that a friction force of level such as applied in test should, if anything, rather contribute to diminishing structural displacement.

CONCLUSIONS

Model tests have been performed to verify the expected performance of a three-dimensional seismic base isolation system, as proposed by the authors, incorporating hydraulic mechanism to ensure adequate vertical isolation. The tests have proved good agreement of performance between theoretical calculation and test data in respect of natural frequency and damping ratio, thus confirming the feasibility of realizing the envisaged conditions of 0.5 Hz natural frequency and 20% damping ratio.

From the response analysis reflecting the test results, it was indicated that, upon practical application on the envisaged commercialized FBR, the system should reduce vertical acceleration by around 2/3. Marked suppression of structural rocking motion was also indicated, that would limit vertical displacement due to rocking motion to about 2 mm under seismic excitation. The proposed system is concluded to be well applicable to the envisaged commercialized FBR.

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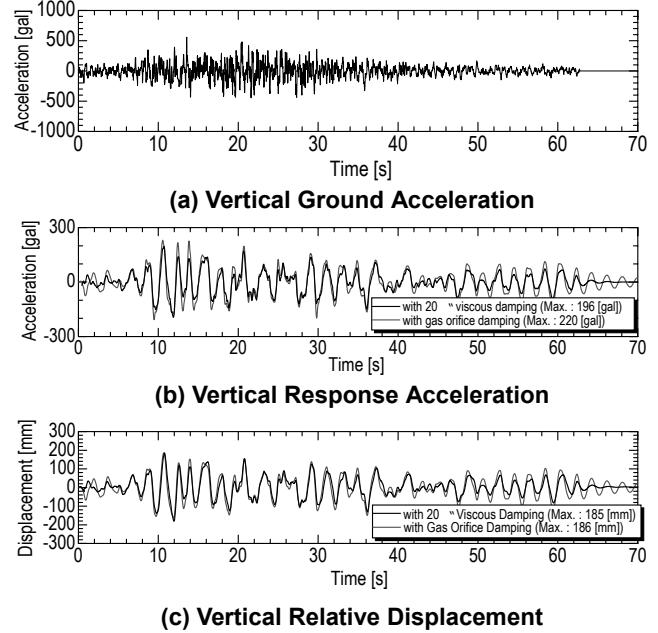


Fig. 11 Analyzed Data of Vertical Response

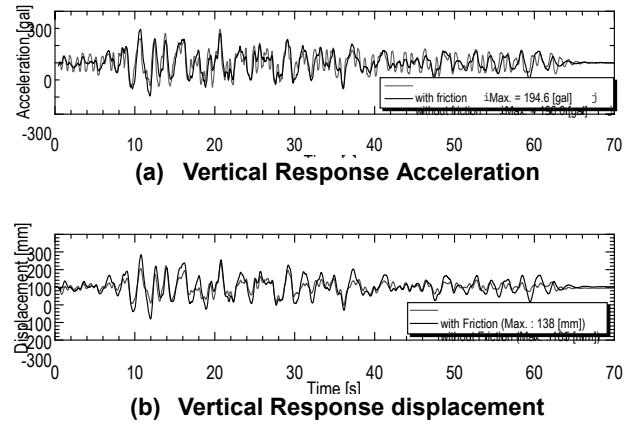


Fig. 12 Analyzed Data of Vertical Response, Considering the Friction Characteristics of Load-carrying Cylinder