Dynamic Buckling Experiments on Liquid Containing Cantilever Cylindrical Shells under Horizontal Excitation

Mayumi FUKUYAMA**, Masaki NAKAGAWA***, Takeshi YASHIRO***, Yukihiro TOYODA**** and Hiroshi AKIYAMA*****

Dynamic buckling experiments on thin cylindrical shells placed inside a rigid liquid container were carried out by using a shaking table. The shells represent the thermal baffles and the container represents the reactor vessel of a fast breeder reactor. The fluid pressure caused by the horizontal excitation distributes nonuniformly around the cylinders and causes external pressure buckling deformation on them. The buckling pressure on various types of test cylinders and seismic waves was measured, and it was confirmed to be higher than that predicted by static buckling analysis. It was also found that sub harmonic vibration occurs under a certain sinusoidal-wave excitation, and the response displacement increases suddenly at a lower pressure than the buckling pressure measured by seismic-wave-excitation tests. The test results indicated that in seismic design to prevent buckling of the thermal baffles, the static buckling analysis can be used as long as sub-harmonic vibration does not occur.

Key Words: Buckling, Seismic Motion, Vibration Coupled with Fluid Motion, Sub Harmonic Vibration, Fast Breeder Reactor, Seismic Design, Cylindrical Shell

1. Introduction

Fast breeder reactors (FBRs) use hot liquid sodium at 500°C as a primary coolant. The main vessel and other important components must therefore be composed of thin shell structures in order to cope with severe thermal conditions. When designing such thin shell structures, it is important to confirm the stability of them against earthquakes, and one of failure modes that must be prevented is buckling.

Accordingly, a seismic buckling design guideline for preventing shear-bending-type buckling of the main vessel, whose radius-to-thickness ratio is between 100 and 200, was established\(^{1(2)}\).

The scope of this guideline has been extended to the seismic design method for primary coolant systems. Thermal baffles are one of the primary coolant components and prevent the main vessel from contacting hot sodium. Figure 1 shows a design of one

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Fig. 1 FBR structure
demonstration-class FBR and its thermal baffles. The thermal baffles are clamped-free thin cylinders supported by the main vessel, and the main vessel and the thermal baffles comprise co-axial cylinders with narrow fluid gaps between the baffles. Therefore, when they are subjected to seismic excitation, fluid pressure is produced by fluid-structure-coupling vibration, and the difference between the fluid pressures on the inside and the outside surfaces of the thermal baffles acts on them as external pressure. Since the thermal baffles, whose radius-to-thickness ratio is considered to be between 200 and 400, are thin, the most critical mode of failure is external pressure buckling.

Even though structural integrity can be investigated by precise dynamic analysis treating large-deformation when a design seismic wave is given, this method is time-consuming. A quasi-static method is therefore usually adopted. In this method, the maximum fluid pressure calculated by linear analysis is compared with the allowable pressure, obtained by dividing static buckling pressure by a safety factor.

Taking account of external pressure buckling of thin cylindrical shells, formulae to predict the buckling pressure have been provided\(^{33,40}\). These formulae are, however, related to cylinders whose ends are simply supported or fixed and whose pressure distributions are assumed to be hydrostatic or uniformly distributed. The seismic design guidelines for liquid storage tanks, such as LNG tanks, also provide formulae for preventing buckling under hydrodynamic pressure. The main failure mode is, however, buckling under axial compression\(^{33,40}\). These formulae cannot predict the buckling pressure for thermal baffles precisely because this buckling is different from that of other structures.

In order to establish the seismic design method for the thermal baffles, we carried out the dynamic buckling experiments of thin cylindrical shells immersed in fluid by using a shaking table. These dynamic buckling experiments under horizontal-wave excitation revealed buckling modes and buckling pressure. We call the buckling pressure obtained by experiments *dynamic buckling pressure*, and we compared it with the calculated static buckling pressure.

2. Dynamic Buckling Experiments

2.1 Description of test cylinders

Table 1 lists the sizes of the test cylinders, and Fig. 2 shows a schematic diagram of two cylinders. Test cylinder LTP1, with radius, thickness, and height of 250, 1, and 350 mm, is called a *reference model* and the other cylinders have different geometry. LTP3 is a co-axial cylinder with a narrow gap of 30 mm. LTP4 is a single-cylinder model stiffened at its top edge. And LTP5 is a thicker cylinder with the same radius and height as LTP1. And TTP1 is a broader cylinder with the same height and thickness as LTP1. These test cylinders are about one twentieth of the size of prototype thermal baffles. A test cylinder was made by rolling a piece of polyester plate. To make LTP4, a polyester plate was cut into a 30-mm-wide strip and this strip was adhered to the top of the cylinder. The broader model, TTP1, was made from two pieces of polyester plate. All cylinders have open tops and are fixed to a rigid flange at their bottoms. Test cylinders were mounted on the shaking table so that the seems are normal to the excitation direction.

Initial shape imperfections of the cylinders were measured beforehand, and those of TTP1 were no more than 6 mm, and those of the other cylinders were no more than 2.5 mm. The initial imperfections were, therefore, no more than 1.5% of the cylinder diameters. Material properties were also measured: Young’s modulus was 1.59 GPa and the density was 12,700 kg/m\(^3\).

2.2 Test apparatus and sensor installation

The test cylinders were placed inside a rigid liquid container filled with water and were set on a shaking table as shown in Fig. 3. The gap between test cylinder TTP1 and the rigid container is 50 mm and that of the other cylinders is 30 mm. The water level is 300 mm or 400 mm above the bottom of the test cylinders.

In order to measure deformation, pressure, and

<table>
<thead>
<tr>
<th>No.</th>
<th>Model</th>
<th>Radius R (mm)</th>
<th>Height H (mm)</th>
<th>Thickness t (mm)</th>
</tr>
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<tr>
<td>LTP1</td>
<td>Single</td>
<td>250</td>
<td>350</td>
<td>1</td>
</tr>
<tr>
<td>LTP3</td>
<td>Inner</td>
<td>Double</td>
<td>220</td>
<td>350</td>
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<tr>
<td></td>
<td>Outer</td>
<td></td>
<td>250</td>
<td>350</td>
</tr>
<tr>
<td>LTP4</td>
<td>Stiffened</td>
<td></td>
<td>250</td>
<td>350</td>
</tr>
<tr>
<td>LTP5</td>
<td>Thickened</td>
<td></td>
<td>250</td>
<td>350</td>
</tr>
<tr>
<td>TTP1</td>
<td>Single</td>
<td>400</td>
<td>350</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 1 Test cylinder models

Units (mm) Figure 2 Test cylinder features
acceleration, sensors were installed as shown in Fig. 4. Twenty displacement sensors were attached inside the rigid liquid container to measure radial displacement at the top of the cylinders. Positive displacements correspond to outward displacements and negative ones correspond to inward displacements. Pressure gauges were attached on the inside surface of the cylinders at their bottom and inside the rigid liquid container along an axial line (Fig. 4). These gauges are positioned at angles of 0 and 180 degrees around the circumference of 175 mm. The direction of excitation was parallel to the line from these position angles. Several accelerometers were attached to the shaking table and the rigid liquid container. A higher input acceleration was required to induce buckling on the thicker cylinder, LTP5. Therefore, instead of using a shaking table, another type of shaking apparatus was used. The test apparatus for LTP5 used a horizontal actuator to produce a higher acceleration. The actuator was connected to a rigid cylinder immersed in fluid inside LTP5 so that it produced a high pressure inside the cylinder.

2.3 Test conditions

It is difficult to satisfy the similarity law of fluid and structure simultaneously in a fluid-structure coupling system. Since the buckling mode to be clarified was caused by external pressure, the similarity between the fluid pressure and cylinder elasticity was mainly maintained here. An equation of motion is written as

$$M\ddot{x} + Kx = 0$$  (1)

where $M$, $K$, and $x$ are mass, stiffness, and displacement of a fluid-structure coupling system. A dot indicates differential operation with respect to time. The first term represents the fluid force by assuming that most of $M$ is determined by the fluid mass. The second term represents shell elasticity. The ratio of these two terms should therefore be maintained such that

$$\frac{K_m x}{M_m \dot{x}} = \frac{K_p x}{M_p \dot{x}}$$  (2)

where the subscripts $p$ and $m$ indicate prototype and test models.

Furthermore, fluid density $\rho_f$, Young's modulus $E$, reference length $L$, and frequency $\omega$ are substituted into Eq. (2), which becomes

$$\frac{E_m L_m \omega_m^2}{\rho_m L_m^2} \cdot \frac{L_p}{\rho_p \frac{L_p}{\omega_p}} = \frac{E_p L_p \omega_p^2}{\rho_p L_p^2}$$  (3)

Although a prototype will be made of stainless steel, test cylinders were made of polyester so that the buckling will take place within the capacity of the shaking table. The ratio of Young's modulus $E_m/E_p$ is, therefore, 0.01, the ratio of length $L_m/L_p$ is 1/20, and the ratio of the fluid density $\rho_m/\rho_p$ is 1/0.83, where $\rho_m$ is the density of water and $\rho_p$ is that of liquid sodium. Substituting these values into Eq. (3) gives the ratio of frequency $\omega_m/\omega_p$ as 1.8.

As shown in Fig. 1, the thermal baffles are supported by main vessel. The seismic response of the main vessel is, therefore, considered to be an input excitation on them. Seismic waves and sinusoidal waves were used as the input waves for the shaking table. Since the dominant frequency of the seismic response of the main vessel is 6.5 Hz and the similarity in frequency domain is 1.8, the excitation frequency is 13 Hz. In several tests, excitation frequencies other than 13 Hz were also chosen to clarify the dependence of the buckling behavior on the excitation frequencies.

Test conditions, water depth and input wave types are listed in Table 2. The acceleration response spectrum of the input seismic waves (at a damping ratio of 0.01) are shown in Fig. 5. Wave-A is a

<table>
<thead>
<tr>
<th>Cylinder/wave</th>
<th>Test cylinder</th>
<th>Wave</th>
<th>Water depth (mm)</th>
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<tr>
<td>LTP1-WA</td>
<td>LTP1</td>
<td>Wave-A</td>
<td>400</td>
</tr>
<tr>
<td>LTP1-WB</td>
<td>LTP1</td>
<td>Wave-B</td>
<td>400</td>
</tr>
<tr>
<td>LTP3-WA</td>
<td>LTP3</td>
<td>Wave-A</td>
<td>400</td>
</tr>
<tr>
<td>LTP4-WA</td>
<td>LTP4</td>
<td>Wave-A</td>
<td>400</td>
</tr>
<tr>
<td>TTP1-WA</td>
<td>TTP1</td>
<td>Wave-A</td>
<td>300</td>
</tr>
</tbody>
</table>

Table 2 Test conditions (seismic wave)
3. Sinusoidal Wave Excitation Experiments

3.1 Experimental conditions

During the dynamic buckling experiments, input acceleration of the sinusoidal wave was gradually increased. The condition for buckling were determined by varying the input acceleration and then measuring the corresponding displacement at the top of the cylinders (at D1 (0 degree), D9 (180 degree), D15 (270 degree)) and the pressure at the bottom of the cylinder (P22).

3.1.1 Tapping test results The eigen frequencies and eigen modes corresponding to high-order-circumferential vibration were obtained by tapping tests. Figure 6 shows the spectrum of displacement measured by D9. The lowest eigen frequency for LTP1, denoted by $f_{\text{min}}$, is 6.5 or 9 Hz when water depth $W$ is 400 mm or 300 mm, respectively. The number of lobes of these modes is six.

3.1.2 Excitation frequency Test cylinders, LTP1, LTP3, and LTP4 were subjected to 13-Hz sinusoidal wave excitations. The sub harmonic vibration may occur when the liquid-containing cylinders are subjected to a sinusoidal wave whose frequency is twice the eigen frequency of high-order circumferential modes. Several excitation frequencies, denoted by $f_s$, other than 13 Hz were therefore additionally chosen as follows.

The excitation frequency $f_s$ on LTP5 was 9 Hz; $f_s$ was equal to the $f_{\text{min}}$.

The excitation frequency $f_s$ on LTP1 with $W$ of 400 mm was 11 Hz; $f_s$ was lower than $2 \times f_{\text{min}}$.

The excitation frequency $f_s$ on LTP1 with $W$ of 400 mm was 13 Hz; $f_s$ was equal to $2 \times f_{\text{min}}$.

The excitation frequency $f_s$ on LTP1 with $W$ of 300 mm was 22.6 Hz; $f_s$ was equal to the natural frequency of first-order-circumferential mode.

The frequency of 22.6 Hz was chosen because it is the dominant frequency of general seismic responses. It is noted, however, that the seismic responses, including pressure, are greatly amplified when test cylinders are subjected to the resonant frequency of first-order-circumferential modes. It was, therefore, anticipated that the buckling might be induced at a low acceleration level. Therefore, water depth $W$ was set to 300 mm to prevent the open edge of the cylinders from being loaded by fluid pressure.

3.1.3 Time history and deformation modes

The displacement time histories from the LTP1 test (sinusoidal-wave excitation) are shown in Fig. 7. These displacements were measured by D1 at excitation frequencies of 13 and 11 Hz. Input accelerations are 5.29 and 8.23 m/s² under 13-Hz excitation and 6.47 and 16.56 m/s² under 11-Hz excitation. Under 11-Hz excitation, the frequency component did not change regardless of the input acceleration. Buckling deformation with five or six lobes was observed when the response displacement was large enough. Under 13-Hz excitation with an input acceleration of 8.23 m/s², sub harmonic vibration was observed and the
response displacement increased suddenly when the response displacement exceeded 0.2 mm. When this sub harmonic vibration occurred, high-order-circumferential modes, as shown in Fig. 8, were observed. The number of lobes was different depending on the excitation frequencies; that is, five and six for 13-Hz excitation (Fig. 8(a)) and seven and eight for 22.6-Hz excitation (Fig. 8(b)). It should be noted that, in this study, the occurrence of the sub harmonic vibration is treated as dynamic buckling behavior because high-order-circumferential mode deformation is induced.

Figure 9 shows the pressure time histories of P22. Figures 9(a) and (b) were measured during the same excitation shown in the lower figures of Figs. 7(a) and (b), respectively, when the response displacement exceeded 1 mm. Comparing Fig.9(a) with the lower figure in Fig. 7(a) shows the pressure at the bottom of the test cylinders is not affected by the low-frequency components, which was observed in the displacement time history at the top of the cylinders, and the high-order-circumferential mode deformation appeared locally.

The motion of the water surface was not significant when the response displacement was below 1 mm and the pressure measured near the water surface had the same frequency component as the excitation frequency. When the response displacement reached 2 or 3 mm under sub harmonic vibration, the motion of the surface and pressure near the surface were, however, affected by the radial deformation of the cylinders.

3.1.4 Relation between input acceleration and response displacement In order to clarify dynamic response characteristics of LTP1, the relation between input acceleration (A3X) and response displacement (D1) was investigated (Fig. 10). Figure 10(b) shows a typical eight-character shape representing sub harmonic vibration when the displacement exceeds 0.5 mm. The relation shown in Fig. 10(b) does not drastically change like that in Fig. 10(a); however, the inward deformation was larger than the outward deformation. The maximum outward deformation was 0.7 mm, whereas the inward deformation increased beyond 0.7 mm even at constant acceleration. Therefore, as long as sub harmonic vibration did not occur, displacements, D1 or D9, of 0.7 mm were determined as the displacement at the buckling threshold because the nonlinearity of the system exists at this displacement.

3.1.5 Relationship between pressure and displacement It is also meaningful to investigate the relation between pressure (P22) and displacement (D9 and D15) as shown in Fig. 11. And Fig. 11(a) shows an example under 13-Hz excitation on LTP1. The displacement suddenly increases when the sub harmonic vibration occurs. And the maximum pressure clearly appears at the pressure (P22) of 1.8 kPa, which is determined to be the buckling threshold.
Figure 11(b) shows that the displacement gradually increases with increasing pressure; therefore, the buckling threshold is determined when D1 reaches 0.7 mm, which is the displacement at the buckling threshold as mentioned in Section 3.1.4. The pressure (P22) at the buckling threshold is 2.3 kPa. Furthermore, the displacement (D15) increased when the pressure exceeds 2.3 kPa, and the radial deformation begins to appear at this pressure. And the sudden increase in response displacement under 11-Hz excitation when the pressure exceeds 4 kPa is caused by sub harmonic vibration; however, this vibration occurred as post-buckling behavior.

3.1.6 Characteristics of pressure distribution
As shown in Fig. 9, the pressure measured at the bottom of test cylinders is not affected by the buckling deformation. Figure 12 shows the pressure distribution around a circumference measured by the pressure gauges attached inside the rigid liquid container. These pressures were measured under the same excitations, as shown in the lower part of Fig. 7(a), at the time of 5.05 sec. At this time, D1 is 1 mm and high-order-circumferential modes deformation was observed. The pressure, however, distributed around the circumference in a form proportional to a cosine function with respect to position angle \( \theta \). This proportionality means that the pressure distribution around a circumference is not affected by buckling deformation and that the pressure can be calculated by linear-fluid-structure-coupling analyses.

3.2 Dynamic buckling pressure
The dynamic buckling pressure is determined by subtracting the fluid pressure on the inside surface of the cylinders from that on the outside. However, the pressure gauges could not be sufficiently located on the test cylinders. The response displacement increases linearly with the fluid pressure before the buckling and the pressure is not affected by local deformation at the top of the test cylinders. It is therefore reasonable to assume that only the first-order circumferential mode is produced under horizontal excitation before buckling takes place. Linear vibration analysis of fluid-structure coupling can therefore be adopted and the pressure distribution at the buckling threshold can be calculated.

An example of a numerical model is shown in Fig. 13. Test cylinders and water are modelled by shell elements and fluid elements, respectively. Material properties are shown in Fig. 13; where Poisson's ratio (\( \nu \)) is 0.39 and damping ratio is 0.01. First, the linear analysis of each cylinder under each excitation wave was done. The pressure distributions on the inner and the outer surface of the test cylinders were calculated and the difference of the pressure was obtained. The pressure thus obtained is called a dynamic buckling pressure here. When following the above procedure, it is important to confirm that the correlation between input acceleration and pressure calculated from the linear analyses is the same as that from the experiments.

Dynamic buckling pressure obtained in this way are shown in Fig. 14 at 0 degree (see Fig. 12). The
Table 3  Buckling pressure under sinusoidal wave excitation

<table>
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<tr>
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<th>f (Hz)</th>
<th>Tests</th>
<th>Analysis</th>
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<tr>
<td></td>
<td></td>
<td>without imperfection</td>
<td>with imperfection</td>
</tr>
<tr>
<td>LTP*/W400</td>
<td>13</td>
<td>1.18</td>
<td>1.18</td>
</tr>
<tr>
<td>LTP*/W400</td>
<td>11</td>
<td>1.37</td>
<td>1.37</td>
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<tr>
<td>LTP*/W300</td>
<td>22.6</td>
<td>0.59*</td>
<td>2.06</td>
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<td>LTP*/W400</td>
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<td>1.67</td>
<td>1.67</td>
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<td>LTP*/W400</td>
<td>22.6</td>
<td>3.61</td>
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<tr>
<td>LTP*/W300</td>
<td>13</td>
<td>0.98*</td>
<td>1.18</td>
</tr>
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</table>

(*:Sub-harmonic vibration occurred)

Fig. 15  Buckling mode

Fig. 16  Relation between pressure and displacement

The pressure distribution around the circumference is proportional to a cosine function when adopting linear analyses. The dynamic buckling pressure corresponding to 22.6 Hz excitation is denoted by open squares and is lower than the other pressures. This is because the sub harmonic vibration occurs at lower pressure under this excitation. The dynamic buckling pressure are also listed in Table 3 as the maximum values along an axial line.

We compared the dynamic buckling pressure with the maximum pressure, which is calculated by static buckling analysis, and is called a static buckling pressure. Test cylinders were modelled by three-dimensional thin-shell finite elements and fluid pressure was replaced by quasi-static distributing force. Since the buckling mode observed during the experiments is obviously an external-pressure buckling mode, the inertia force of the structures is neglected. The static buckling analysis can therefore be applied. The static buckling pressure were calculated not only by buckling -eigen-value analyses using cylinders without initial shape imperfections but also by large-deformation step-by-step analyses using cylinders with initial shape imperfections. The shape of initial imperfections was the same as the measured ones.

Calculated buckling eigen mode is shown in Fig. 15. The buckling mode has five-lobe or six-lobe high-order-circumferential modes, which resemble those observed in the experiments. The number of lobes shown in Fig. 8(b) is more than that shown in Fig. 15. This is because number of lobes is different depending on the excitation frequency under sub harmonic vibration.

Step-by-step analysis of cylinders with initial shape imperfections reveals the relation between pressure and displacements as shown in Fig. 16. The maximum pressure is treated as the static buckling pressure. The calculated pressures are also summarised in Table 3. These pressures are the maximum values along the axial line and are different from the measured values, at P22, shown in Fig. 11. Table 3 can be summarised as follows.

- Static buckling pressure on the cylinders with initial shape imperfection is 20% lower than the buckling eigen values. The static buckling pressure depends on water depth, radius, and thickness.

- Dynamic buckling pressures obtained when sub harmonic vibration did not occur are between 1.2 and 1.5 times the static buckling pressure on the cylinders without initial shape imperfections (buckling eigenvalue).

- Dynamic buckling pressures obtained when the sub harmonic vibration occurred are lower than the static buckling pressures (buckling eigenvalue). Except for the dynamic buckling pressure on LTP1 under 22.6-Hz excitation, the dynamic buckling pressures when the sub harmonic vibration occurs are higher than the static buckling pressure (with imperfection).

3.3 Sub harmonic vibration

Although the sub harmonic vibration observed in these dynamic buckling experiments was not studied here, brief summary of our previous report(30) is given as follows.

- By investigating the frequency components of displacements, two peak frequencies were found near half the excitation frequency. They correspond to high-order-circumferential modes and their numbers of lobes were successive pair (Fig. 17).
The sub harmonic vibration depends on both excitation frequency and input acceleration and the number of lobes of these high-order-circumferential vibrations depends on excitation frequency.

This dependence can be considered as the parametric vibration of liquid-containing cylinders studied by Chiba et al.\textsuperscript{15} and Takayanagi\textsuperscript{16}. According to their studies, this dependence can be explained by elastic stability theory.

4. Seismic Wave Excitation Experiments

4.1 Test condition and results

According to the sinusoidal-wave excitation tests, the input acceleration was gradually increased until the buckling deformation was observed. Pressure (P22), input acceleration (A3X) and the displacements (D1 and D15) were measured.

4.1.1 Time histories Figure 18 shows the time histories of response displacements (D1) of LTP1 and TTP1 when subjected to Wave-A. The response displacement nonlinearly increased with increasing acceleration. Furthermore, regarding TTP1, the frequency component changed depending on the input acceleration. Translational motion in the first-circumferential mode occurs when input acceleration is low, whereas buckling (which had high-order-circumferential modes) occurs when input acceleration is high. The buckling could therefore be predicted when nonlinearity appears in the relation between response displacement and input acceleration. The characteristics of the time histories of pressure (P22) do not change (regardless of input acceleration) and the relation between pressure and acceleration is linear. The occurrence of the buckling can therefore be determined when the nonlinearity appears in the relation between response displacement and pressure.

4.1.2 Relation between pressure and displacement To obtain buckling pressure, the relations between pressure (P22) and displacements (D1, D9, and D15) are shown in Fig. 19, which represents LTP1 and TTP1 subjected to excitation Wave-A. The displacements increase gradually with increasing pressure. The nonlinearity, however, appears when the pressure (P22) exceeds 3 kPa when the displacement is 0.7 mm. This displacement value is the same as that at the buckling threshold (as mention in Section 3.2). This pressure is therefore determined as the buckling threshold for LTP1 subjected to Wave-A. As shown in Fig. 19(b), when the pressure (P22) exceeds 1 kPa, the displacement suddenly increases. This pressure is therefore determined as the buckling threshold for TTP1 subjected to Wave-A.

4.2 Dynamic buckling pressure

In this section, the dynamic buckling pressures are
Table 4  Buckling pressure under seismic-wave excitation

<table>
<thead>
<tr>
<th>Cylinder/wave</th>
<th>Wave</th>
<th>Tests without imperfection (kPa)</th>
<th>Analysis with imperfection</th>
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<tr>
<td>LTP1-WA</td>
<td>Wave-A</td>
<td>1.86</td>
<td>1.18</td>
</tr>
<tr>
<td>LTP1-WB</td>
<td>Wave-B</td>
<td>1.37</td>
<td>1.18</td>
</tr>
<tr>
<td>LTP3-WA</td>
<td>Wave-A</td>
<td>2.16</td>
<td>2.16</td>
</tr>
<tr>
<td>LTP4-WA</td>
<td>Wave-A</td>
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<td>1.86</td>
</tr>
<tr>
<td>TTP1-WA</td>
<td>Wave-A</td>
<td>1.59</td>
<td>1.18</td>
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</table>

determined in the same way described in Section 3.2. That is, the linear vibration analysis of fluid-structure coupling is adopted to interpolate the measured pressure and to obtain the difference between inside and outside pressure on the test cylinders at the buckling threshold. Furthermore, the static buckling pressures for cylinders with and without initial shape imperfections were also calculated (as mentioned in Section 3.2). Thus, the dynamic buckling pressures and static buckling pressures were obtained and are listed in Table 4. These buckling pressures have maximum values along the axial line. The static buckling pressures for cylinders with initial shape imperfections are lower than those without initial shape imperfections. Table 4 can be summarised as follows.

- The dynamic buckling pressures are between 1.2 and 1.8 times larger than the static buckling pressures on cylinders without imperfection. The static buckling analyses can therefore be adopted in order to predict the dynamic buckling pressure.

- The static buckling pressure for LTP4 is 60% higher than that for LTP1 because of the model stiffening. However, the dynamic buckling pressures are not significantly different. This may be because the experimental data are not enough to determine the occurrence of buckling.

4.3 Buckling behavior under seismic-wave excitation and sinusoidal-wave excitation

In this section, the response characteristics under sub harmonic vibration caused by sinusoidal-wave excitation are compared with those caused by seismic-wave excitation. Characteristics under 13-Hz sinusoidal-wave excitation on LTP1 and TTP1 and under a narrow-band random-wave (Wave-A) were investigated.

The response characteristics of sub harmonic vibration under sinusoidal wave excitation is that the response displacement suddenly increases. Therefore, the sub harmonic vibration may induce instability and may cause critical damage. Furthermore, this instability may occur at a lower pressure than the static buckling pressure, as shown in Table 3.

On the contrary, sub harmonic vibration rarely occurs under seismic wave excitation, as shown in Fig. 18(a). As shown in Fig. 18(b), a low-frequency component, however, is observed. It is observed when the displacement exceeds 2 mm and can be considered as a post-buckling behavior. Furthermore, considering the fact that the seismic excitation is non-stationary, it can be concluded that the instability that induces the sudden increase in displacement rarely occurs under seismic wave excitation. It was also shown that the dynamic buckling pressure under seismic wave excitation was 1.5 times larger than that under sinusoidal-wave excitation and it was higher than the static buckling pressure. Therefore, it is reasonable to say that the effect of the sub harmonic vibration is not very serious under seismic wave excitation and the allowable pressure can be determined by static buckling analysis. It should, however, be emphasised that sub harmonic vibration on test cylinders of different geometry and under a random sinusoidal-like wave, must be investigated.

5. Summary

Dynamic buckling behaviors of clamped-free thin cylindrical shells immersed in fluid were investigated. In addition, the static buckling pressure is calculated and compared with the experimental buckling pressure. The main results of this investigation are summarized below.

- From the dynamic buckling experiments, buckling mode and buckling pressure with various cylinder models and various excitation conditions were clarified.

- Buckling pressures predicted by the static buckling analysis agree with the dynamic buckling pressures as long as the sub harmonic vibration does not occur.

- Sub harmonic vibration was observed under a sinusoidal wave excitation of a particular frequency, and response displacement increased suddenly at a lower pressure than the static buckling pressure.

- Sub harmonic vibration rarely occurred under seismic-wave excitation and the conditions used in this study.

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References


