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VORTEX-INDUCED IN-LINE VIBRATION AND RANDOM VIBRATION OF A CIRCULAR CYLINDER UNDER VARIOUS INFLOW CONDITIONS

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ABSTRACT

The flow-induced vibrations (FIV) of cylindrical structures subjected to a cross flow have been investigated in a number of studies owing to their practical importance. The results of these studies have been compiled into guidelines for the evaluation of FIV of cylindrical structures. However, the applicability of the guidelines to cases with upstream structures and with an oblique inflow has not been fully examined. In this paper, we describe the response characteristics of vortex-induced in-line vibration and random vibration of a circular cylinder under various inflow conditions. Water tunnel tests were conducted on a circular cylinder in a cross flow at three yaw angles of 0, 30 and 45 degrees to clarify the effects of the yaw angle on vortex-induced in-line vibration and random vibration. The vibration amplitudes of a circular cylinder downstream of another circular cylinder of five times larger diameter were also measured to investigate the effects of inflow turbulence generated by an upstream cylinder. The tests were conducted for two different relative locations of the upstream cylinder in the same reduced- velocity range as that of the single-cylinder tests. The response amplitudes and onset flow velocities obtained by the tests were compared with values predicted using the cosine law and the guidelines to verify the evaluation methods in the guidelines.

INTRODUCTION

Nuclear power plants include a number of circular cylindrical structures subjected to a flow. It is essential to evaluate flow-induced vibrations (FIV) when designing such structures to ensure their long-term integrity. Many studies have been conducted on the FIV of a circular cylinder subjected to a cross flow, as reviewed in depth by King [1], Sarpkaya [2, 3], Bearman [4], Naudascher [5] and others. The results of these studies have been compiled into handbooks by several researchers [6 – 9], and the American Society of Mechanical Engineers (ASME) published a nonmandatory code for the evaluation of the FIV of cylindrical structures in 1992 [10]. The Japan Society of Mechanical Engineers (JSME) also published a guideline for FIV evaluation in 1998 [11, 12].

The JSME guideline states that vortex-induced vibrations should be avoided or suppressed and that the stress amplitude caused by random excitation force should be evaluated when designing cylindrical structures in a cross flow. The ASME code also describes methods for evaluating both vortex-induced vibrations and random vibration. In these guidelines, the following criterion for the reduced velocity V_r is given for avoiding vortex-induced vibrations of a circular cylinder in a cross flow in both the transverse and in-line directions:

$$V_r = U/(f_0\, d) < 1, \qquad (1)$$

where U, f_0 and d are the incident flow velocity, the fundamental natural frequency and the cylinder diameter, respectively. This criterion is based on King's experiments on in-line vibration [13], and has been examined by many experiments even in the supercritical Reynolds number range [14 – 18]. For nuclear power plants, this criterion is usually used to avoid high-cycle fatigue caused by vortex-induced in-line vibration when designing cylindrical structures subjected to a flow.

In actual cases of flow such as in the inner structures in a reactor vessel, the incident flow to a cylinder is not usually a pure cross flow. The incident flow is sometimes inclined to the cylinder axis and its velocity varies, and the JSME guideline cannot be applied in the case of a yawed cylinder. For a yawed cylinder, the ASME code states that the vortex-shedding frequency can be predicted using the component of the flow velocity normal to the cylinder axis. This is called the *cosine law* and has been examined by several researchers [19, 20]. In this

context, the reduced velocity should be defined in terms of the normal component of flow velocity. In fact, the water channel tests performed by King [21] demonstrated that the dependence of the onset flow velocity of in-line vibration can be removed by using the normal component of flow velocity when calculating the reduced velocity. However, the Reynolds number at the onset of in-line vibration in the experiments was about 10^4, and few other experiments on the onset of in-line vibration of a yawed circular cylinder have been conducted. The *cosine law* for in-line vibration should be examined in the higher Reynolds number range.

A cylinder is sometimes surrounded by other cylindrical structures of different diameters. For example, many thermometers and core monitors with small-diameter wells will be installed in the gaps of control rod guide tubes of a larger diameter in the upper inner structure of the reactor vessel of Japanese sodium-cooled fast reactor [22]. In this case, the vibration response of a cylinder can be affected by the wake of other cylinders located upstream. However, a method for estimating the excitation force has not been established and the criterion for avoiding vortex-induced vibration has not been fully examined for a cylinder in the wake of different-diameter cylinders.

In this study, free vibration tests were conducted on a circular cylinder in a cross flow at three yaw angles of 0, 30 and 45 degrees to investigate the effects of the yaw angle on vortex-induced in-line vibration as well as random vibration. The vibration amplitudes of a circular cylinder behind another circular cylinder of five times larger diameter were also measured to investigate the effects of the wake of the upstream cylinder. The tests were conducted for two different relative locations of the upstream cylinder in the same flow velocity range as that of the single-cylinder tests. The measured response amplitudes were compared with values predicted using the cosine law and the guidelines of ASME and JSME to verify the evaluation methods in the guidelines.

NOMENCLATURE

a_x	:	Response acceleration at center of test cylinder in x direction
a_y	:	Response acceleration at center of test cylinder in y direction
A_x	:	R.M.S. displacement amplitude in x direction
A_y	:	R.M.S. displacement amplitude in y direction
C_n	:	Reduced damping
d	:	Diameter of test cylinder
D	:	Diameter of dummy tube
f_{na}	:	Natural frequency in air
f_{nw}	:	Natural frequency in water
f_x	:	Response frequency of vibration in x direction
f_y	:	Response frequency of vibration in y direction
l_s	:	Length of cylinder between supports
l_e	:	Length of cylinder subjected to flow
Re	:	Reynolds number
V_r	:	Reduced velocity obtained using normal component of flow velocity
U	:	Incident flow velocity
U_e	:	Local flow velocity
x	:	Axis in the plane parallel to flow direction and normal to cylinder axis
y	:	Axis in transverse direction
z	:	Axis in spanwise direction
ϕ	:	Fundamental mode shape
μ	:	Mass ratio
ν	:	Kinematic viscosity
θ	:	Yaw angle
ρ	:	Density of fluid
ζ_s	:	Structural damping ratio

EXPERIMENTAL ARRANGEMENT

Water Tunnel Facility

A gravitational-draining-type vertical water tunnel [23, 24] is used in the tests. It consists of an underground reservoir with a 1700 m^3 capacity, an upper reservoir with a 40 m^3 capacity, a vertical water tunnel section, a draining valve and a water pump. The test section is rectangular with dimensions of 1000 mm × 500 mm. In the water tunnel, the pressure pulsation at peak frequencies is very small since the water flows only by gravitational force without pumping during the tests.

The descent rate of the water level in the upper reservoir was correlated to the incident flow velocity on the basis of preliminary measurements by an electromagnetic current meter (ACM-200, ALEC Electronics). The incident flow velocity during the free vibration tests was obtained from the descent rate of the water level. The turbulence intensity of the incident flow at the center of the test section is about 1.5 – 3% of the steady flow velocity according to preliminary measurements by laser Doppler velocimetry.

Specifications of Test Cylinders and Experimental Conditions

Three test cylinders were used in the yawed cylinder tests. The cylinders were stainless-steel pipes of 25.4 mm outer diameter, 22.2 mm inner diameter and three spanwise lengths. A cylindrical stainless-steel sleeve of 30 mm outer diameter and 30 mm length was welded to each end of the pipes. Figure 1 shows the schematic arrangement of the free vibration tests on the yawed circular cylinders. Both ends of the pipe were fixed to the sidewalls of the water tunnel, and the yaw angle was set at 0, 30 or 45 degrees. A cylindrical support and four bolts were used at each end to support the test cylinder as shown in Fig. 2. Two uniaxial accelerometers (ARK-1000A, Tokyo-Sokki) were installed at the center of each test cylinder to measure the vibration responses in the x and y directions as shown in Fig. 3. The x axis is normal to the cylinder axis and is in the plane parallel to the incident flow direction. The y axis is in the transverse direction to the flow. The specifications of the three test cylinders are shown in Table 1. The natural

Table 1 Specification of test cylinders

	Cylinder No.1	Cylinder No.2	Cylinder No.3
Cylinder diameter d	25.4 mm	25.4 mm	25.4 mm
Yaw angle θ	0 deg.	30 deg.	45 deg.
Cylinder length subjected to flow l_e	1000 mm	1155 mm	1414 mm
Cylinder length between supports l_s	1198 mm	1368 mm	1599 mm
Natural frequency in air*1 f_{na}	76 Hz	60 Hz	45 Hz
Structural damping ratio*1 ζ_s	0.20%	0.19%	0.17%
Natural frequency in water*1 f_{nw}	67Hz	52Hz	39Hz
Mass ratio μ	3.4	3.4	3.2
Reduced damping C_n	0.09	0.08	0.07

*1: Measured values in hammering tests

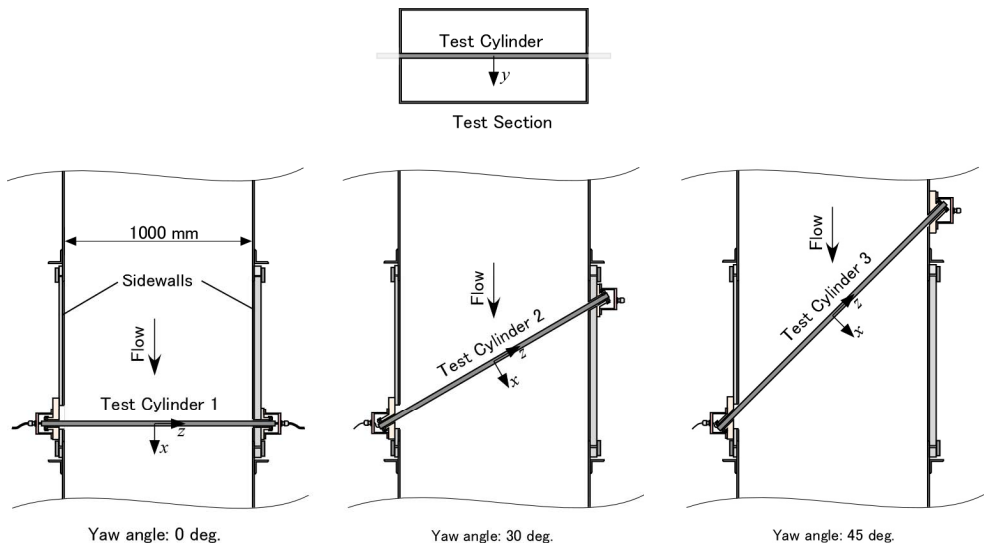

Fig. 1 Schematic arrangement in tests of yawed circular cylinder

frequencies and damping ratios were obtained by hammering tests in air and water. The mass ratio μ and the reduced damping C_n given in the table are defined as

$$\mu = \frac{m_s + m_a}{\rho d^2}\,, \tag{2}$$

$$C_n = 4\pi\zeta_s\mu \cdot \int_0^{l_s} \phi(z)^2\,dz \Big/ \int_{\frac{l_s-l_e}{2}}^{\frac{l_s+l_e}{2}} \phi(z)^2\,dz\,, \tag{3}$$

where $m_a = \rho d^2\pi/4$ is the added mass per unit length. The equivalent structural mass m_s was estimated from the natural frequencies in air and water as follows:

$$\frac{{f_{na}}^2}{{f_{nw}}^2} = \frac{m_s + m_a}{m_s}\,, \tag{4}$$

$$\therefore m_s = m_a \frac{{f_{nw}}^2}{{f_{na}}^2 - {f_{nw}}^2}\,. \tag{5}$$

Both ends of the test cylinder are clamped and the fundamental mode shape is assumed to be

$$\phi(z) = \cosh\left(\frac{\lambda z}{l_s}\right) - \cos\left(\frac{\lambda z}{l_s}\right) - \sigma\left\{\sinh\left(\frac{\lambda z}{l_s}\right) - \sin\left(\frac{\lambda z}{l_s}\right)\right\},$$

$$\lambda = 4.73,\ \sigma = 0.983. \tag{6}$$

The reduced damping of each test cylinder is rather low with a value of approximately 0.1.

In the tests with an upstream cylinder, a rigid cylinder of five times larger diameter was fixed upstream at one of the locations shown in Fig. 4. The center of the test cylinder was located at a distance of 2 D downstream from the center of the upstream rigid cylinder in case 4, whereas the position of the upstream cylinder was offset in the transverse direction in case 5. It is supposed that the separated shear layer from the upstream cylinder impinges to the test cylinder in the case 5.

Table 2 shows a summary of the experimental conditions.

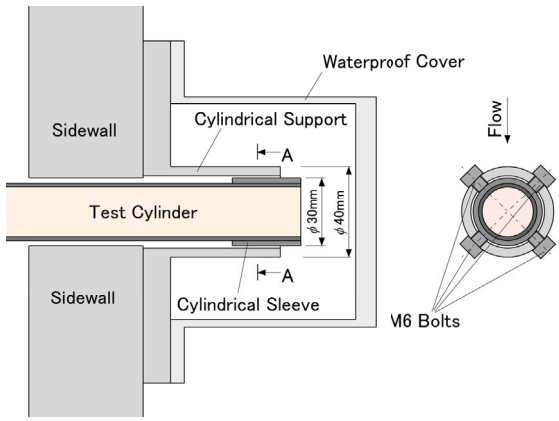

Fig. 2 Support condition of test cylinder

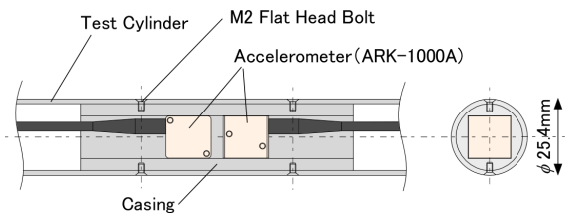

Fig. 3 Accelerometers installed in test cylinder

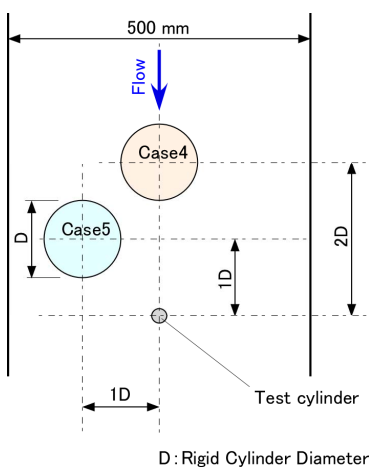

Fig. 4 Layout for cases when a rigid cylinder of larger diameter is placed upstream

Table 2 Experimental conditions

	Test cylinder	Yaw angle	Longitudinal pitch	Transverse pitch
Case 1	No.1	0 deg.	-	-
Case 2	No.2	30 deg.	-	-
Case 3	No.3	45 deg.	-	-
Case 4	No.1	0 deg.	2 D	0
Case 5	No.1	0 deg.	1 D	1 D

D: Rigid cylinder diameter

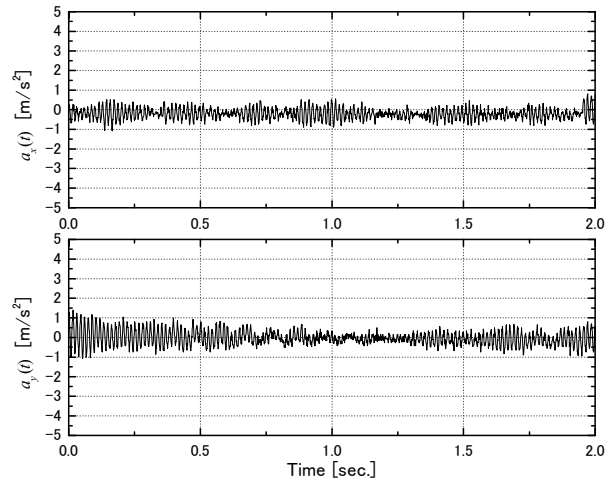

Fig. 5 Time traces of response acceleration (θ = 0 deg., U = 1.4 m/s)

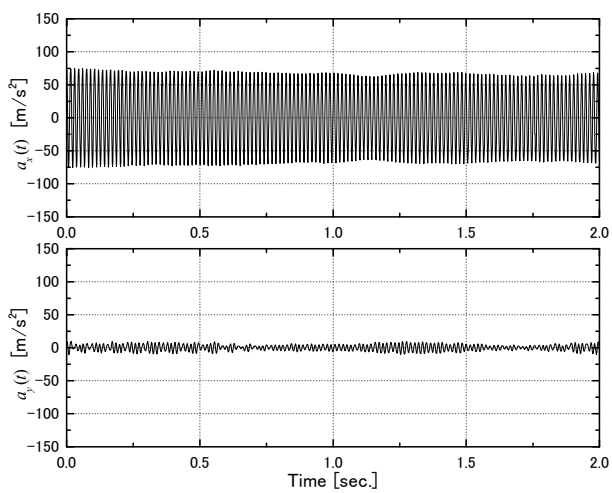

Fig. 6 Time traces of response acceleration (θ = 0 deg., U = 2.1 m/s)

EXPERIMENTAL RESULTS

Yawed Circular Cylinder

Figure 5 shows typical time traces of the response acceleration in the x and y directions for the case of θ = 0 degree and U = 1.4 m/s. The responses of the fundamental natural frequency are dominant in both the x and y directions although their amplitude is low with random perturbations. Figure 6 shows time traces for the case of θ = 0 degree and U = 2.1 m/s. Harmonic oscillation with a large and constant amplitude is observed, which is considered to be caused by the vortex-induced in-line vibration.

Figure 7 shows the R.M.S. amplitude and frequency of the response acceleration measured for the case of pure cross flow (θ = 0 degree) versus the incident flow velocity U and Reynolds number $Re = Ud/\nu$. Each frequency plotted in the

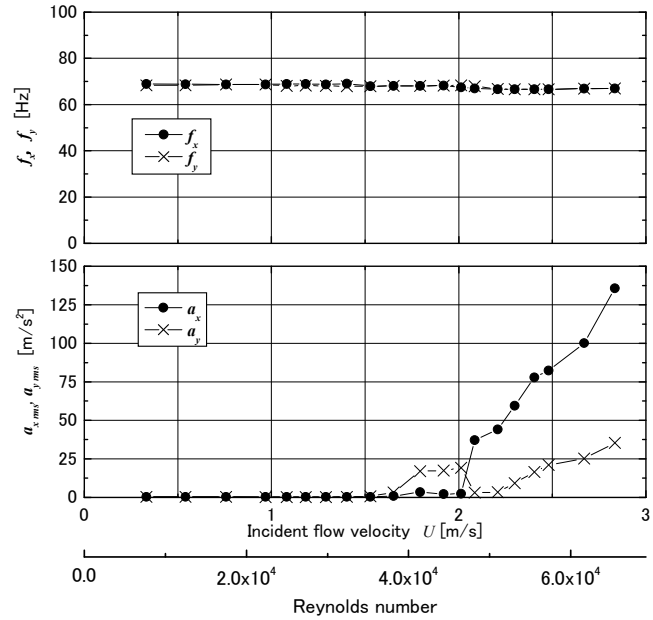

Fig. 7 R.M.S. amplitude and dominant frequency of response acceleration (θ = 0 deg.)

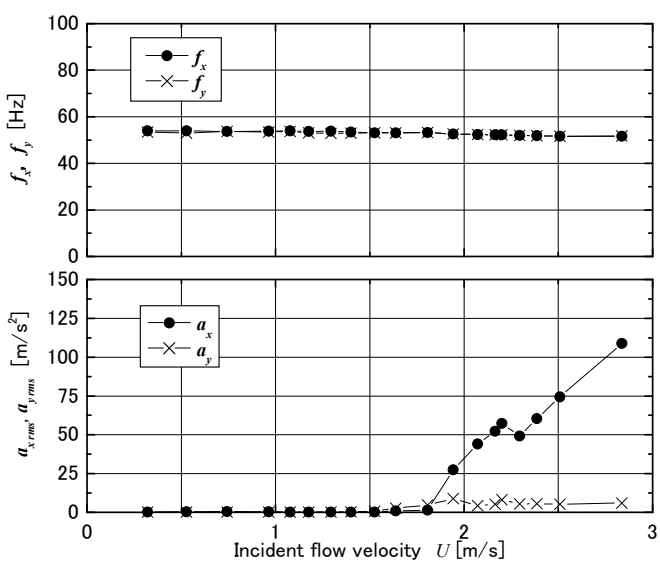

Fig. 8 R.M.S. amplitude and dominant frequency of response acceleration (θ = 30 deg.)

figure is the most dominant peak frequency in the power spectral density derived from a set of 2^{13} values of the measured response acceleration. The R.M.S. amplitudes are derived by integrating the power spectral density around the peak component. The dominant frequency does not vary and agrees with the fundamental natural frequency. A small-amplitude oscillation in the transverse direction appears when U = 1.8 – 2.0 m/s, and the response amplitude in the in-line direction is dominant for U > 2.1 m/s. Similar responses can be

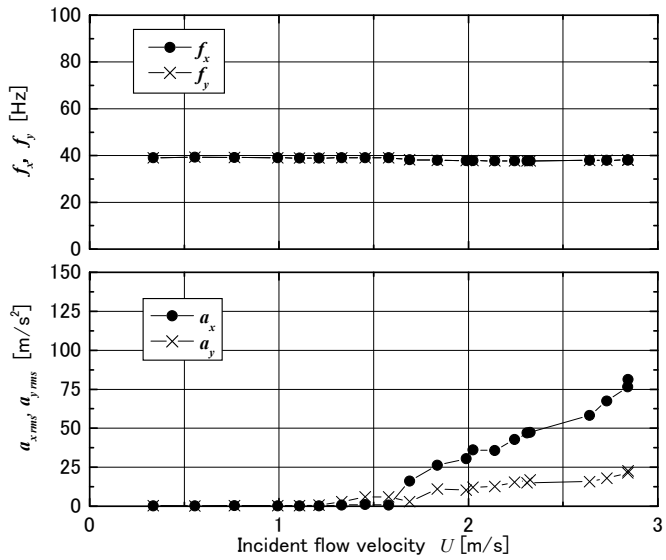

Fig. 9 R.M.S. amplitude and dominant frequency of response acceleration (θ = 45 deg.)

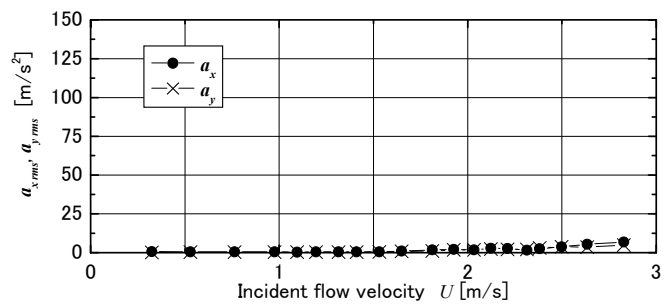

Fig. 10 R.M.S. amplitude of response acceleration (case 4)

seen in Figs. 8 and 9 for cases 2 and 3, respectively, although the onset flow velocity is lower and the amplitude of response acceleration is smaller than those in the pure cross-flow case. The reason for this is that the cylinder is longer and the fundamental natural frequency is lower in cases 2 and 3 although the normal component of the flow velocity is also lower.

Cylinder behind Larger Diameter Cylinder

Figure 10 shows the R.M.S. amplitude of response acceleration versus the incident flow velocity for case 4, in which another cylinder is placed upstream. The response amplitude is small and vortex-induced vibration does not occur. On the other hand, responses similar to those in case 1 (without an upstream cylinder) can be seen in Fig. 11 for case 5, and the onset flow velocity is clearly lower and the amplitude is larger than those in case 1. These results clearly show that the vibration response of the test cylinder is affected by the cylinder set upstream and greatly depends on the relative location of the upstream cylinder.

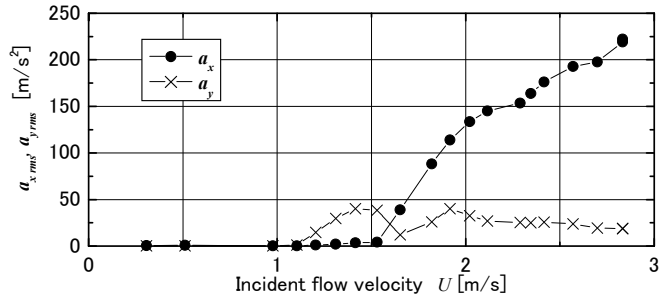

Fig.11 R.M.S. amplitude of response acceleration (case 5)

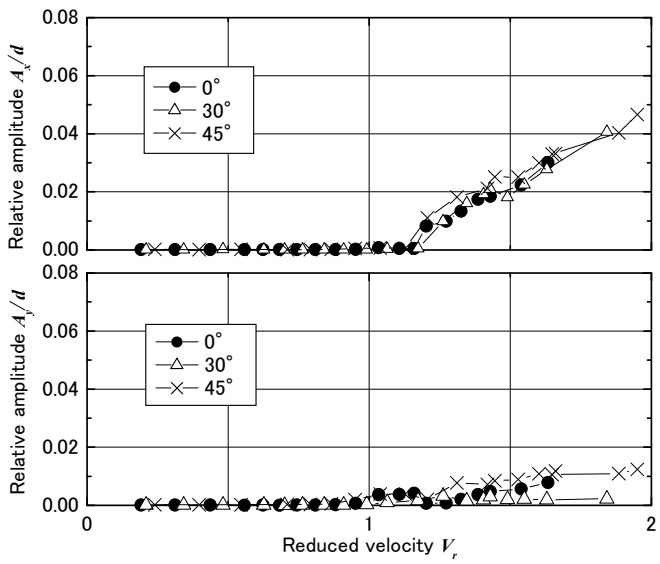

Fig.12 Relative R.M.S. amplitude versus reduced velocity obtained using the normal component of flow velocity

DISCUSSION

Onset of Vortex-Induced In-Line Vibration

In the tests on a single yawed cylinder, the onset flow velocity and R.M.S amplitude of response acceleration are lower in cases 2 and 3 than in case 1. These are reasonable results since the fundamental natural frequency of the test cylinder is lower in cases 2 and 3 than in case 1. The response amplitudes in cases 1 – 3 are re-plotted in Fig. 12, where the abscissa shows the reduced velocity V_r obtained from the normal component of the incident flow velocity as follows:

$$V_r = U\cos\theta / (f_{nw} d). \tag{7}$$

The ordinate shows the relative R.M.S. amplitudes A_x/d in the x direction and A_y/d in the y direction. The R.M.S amplitudes A_x and A_y of the response displacement were calculated as follows:

$$A_x = a_{xrms} / (2\pi f_x)^2, \tag{8}$$

$$A_y = a_{yrms} / (2\pi f_y)^2, \tag{9}$$

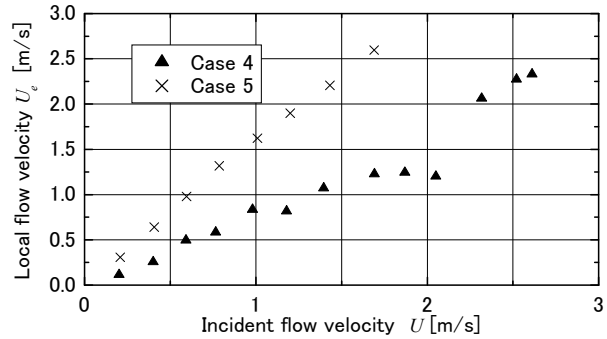

Fig. 13 Effective incident flow velocity in cases 4 and 5

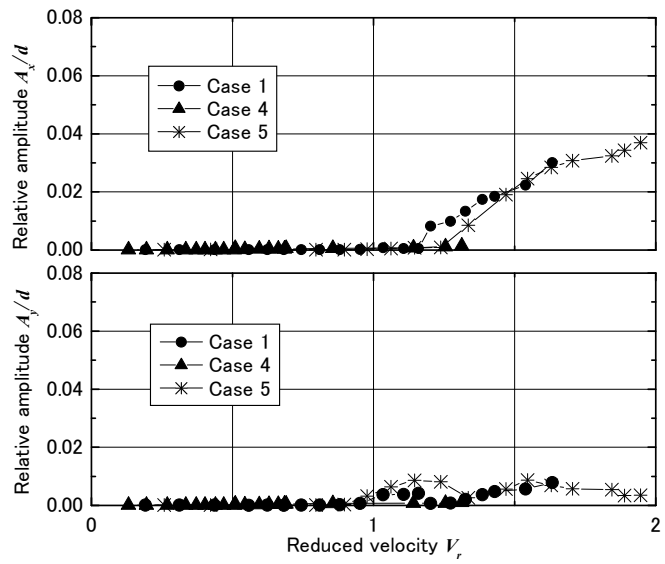

Fig. 14 Relative R.M.S. amplitude versus reduced velocity using the effective flow velocity at the test cylinder

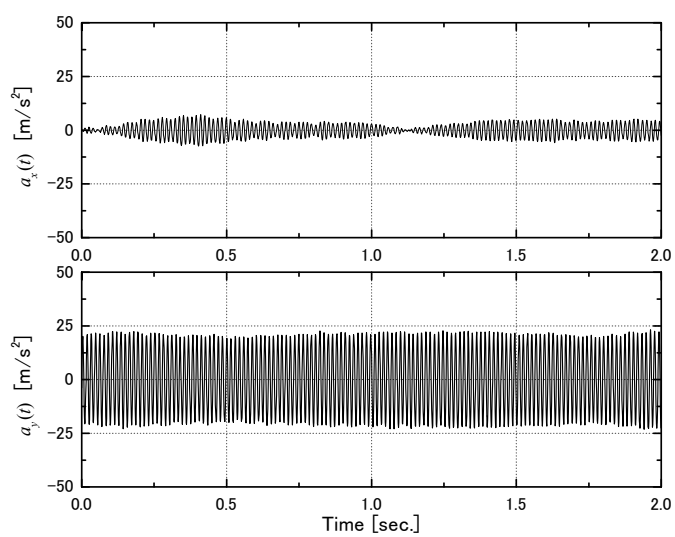

Fig. 15 Time traces of response acceleration (case 1, U = 1.9 m/s)

where a_{xrms} and a_{yrms} express the R.M.S. amplitudes of response accelerations in the x and y directions, respectively. The response curves for the three different yaw angles collapse onto a single curve, which demonstrates the validity of the cosine law for in-line vibration at least at intermediate subcritical values of the Reynolds number. This result clearly shows that vortex-induced in-line vibration can be avoided in the case of a yawed cylinder by using the criterion $V_r < 1$ where V_r is calculated using the normal flow velocity.

In cases 4 and 5 with an upstream fixed cylinder, the vibration response clearly differs from that in the single-cylinder case. This mainly results from the difference in the effective incident flow velocity to the test cylinder. Figure 13 shows the local flow velocity U_e at the location where the test cylinder is installed in the free vibration tests. It was measured by an electromagnetic current meter in case that the upstream cylinder was fixed in the test section without a test cylinder. The local flow velocity U_e is markedly lower in case 4 and is higher in case 5 than the incident flow velocity. The R.M.S. amplitudes of the response displacement in cases 4 and 5 as well that as in case 1 are shown in Fig. 14. The abscissa shows the reduced velocity $V_r = U_e/(f_{nw}\ d)$. The response curves in cases 1 and 5 reasonably agree well with each other. This graph also indicates the reason why vortex-induced vibration does not occur in case 4, that is, the reduced velocity does not reach the onset value. This result shows that the criterion $V_r < 1$ can be applied to the case of a circular cylinder with an upstream larger-diameter one if the effective incident flow velocity at the cylinder can be accurately estimated.

In cases 1 and 3, considerable transverse responses can be seen at approximately $V_r = 1$ to 1.2, which is a lower value of V_r than that causing the onset of in-line vibration. Figure 15 shows time traces in case 1 at $U = 1.9$ m/s. The response is not random but exhibits harmonic oscillation with a constant amplitude. Similar responses in the transverse direction were observed around $V_r = 1$ in previous free vibration tests on a cantilevered cylinder [18, 19]. These transverse responses are considered as being due to 1/5 lock-in vibration since the Karman vortex shedding frequency is nearly 1/5 of the natural frequency of the cylinder at $V_r = 1$. Transverse responses also occur around $V_r = 1$ to 1.2 in case 5 with an upstream cylinder, with an amplitude larger than in cases 1 and 3. The responses are also considered as being due to 1/5 lock-in vibration. However, the relative R.M.S. amplitude of the transverse response is small and the vibration can also be avoided for $V_r < 1$.

Random Vibration Response

According to the JSME guideline [11], the R.M.S. amplitude of random vibration can be estimated as follows:

$$y_R(z) = \sqrt{\frac{\beta_0{}^2 \cdot G(f_0)}{64 \cdot \pi^3 \cdot M^2 \cdot f^3 \cdot (\zeta_s + \zeta_f)}} \cdot \phi_0(z)\,, \qquad (10)$$

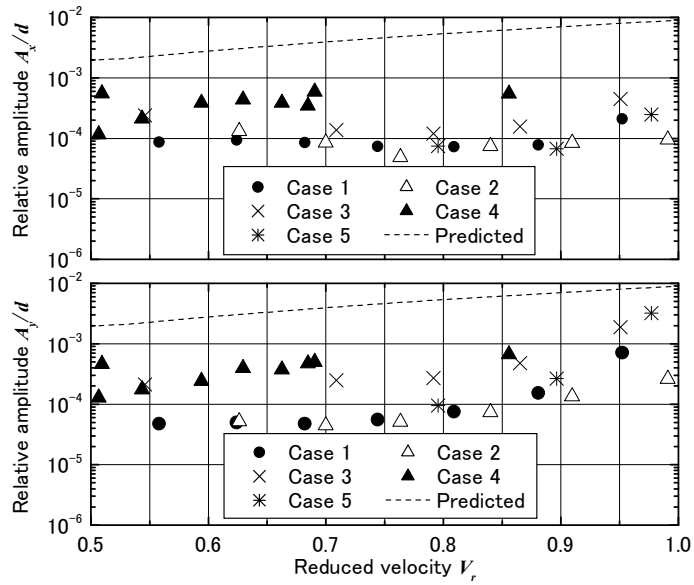

Fig. 16 Comparison between the measured relative R.M.S. amplitude of random vibration and the predicted amplitude based on the JSME guideline

where M, f, $G(f_0)$, ζ_s and $\phi_0(z)$ are the modal mass per unit length including the added mass, the fundamental natural frequency, the single-sided power spectral density of the random excitation force, the structural damping ratio and the fundamental mode shape of vibration, respectively. β_0 is the participation factor, defined as

$$\beta_0 = \int_{l_e} \phi_0(z)dz \Big/ \int_{l_s} \phi_0{}^2(z)dz\,. \qquad (11)$$

Figure 16 shows the relative R.M.S. amplitude of random vibration responses measured in the reduced velocity range of 0.5 to 1.0. The broken line shows the predicted R.M.S. amplitude at the center of the span based on the JSME guideline. In the calculation, the random vibration responses caused only by s cross flow are estimated since those caused by the parallel-flow component are usually small. The fluid damping ζ_f is set at 0, and ϕ_0 is assumed to be given by eq. (6) since both ends of the test cylinder are clamped.

The power spectral density of the random excitation force is estimated using Mulcahy's normalized power spectral density in the drag direction [25]. The response in case 4 is larger than those in the other cases, which results from the turbulence caused by the upstream cylinder. However, all the measured response amplitudes are lower than the predicted values since the random vibration response is conservatively estimated assuming a perfect spanwise correlation for a random excitation force in the JSME guideline. This result indicates that the method of estimating the random response in the JSME guideline includes an adequate safety margin even for the case with an upstream larger-diameter cylinder.

CONCLUDING REMARKS

Water tunnel tests on an elastic circular cylinder supported at both ends were conducted at three yaw angles of 0, 30 and 45 degrees. The vibration amplitudes of a circular cylinder in the wake of an upstream circular cylinder of five times larger diameter were also measured for two different relative locations of the upstream cylinder. The following results were obtained.

(1) The vortex-induced in-line vibration of a single circular cylinder for the case with a nonzero yaw angle can be avoided at least at intermediate subcritical values of the Reynolds number, by using the criterion $V_r < 1$, where the reduced velocity V_r is calculated using the normal component of the incident flow velocity.

(2) The in-line vibration for the case with an upstream larger-diameter cylinder can also be avoided by using the criterion $V_r < 1$, where V_r is calculated using the effective incident flow velocity.

(3) A transverse vibration due to 1/5 lock-in vibration can occur at a reduced velocity lower than that causing the onset of in-line vibration. However, the vibration can also be avoided for $V_r < 1$.

(4) The random vibration responses for $V_r < 1$ can be conservatively estimated on the basis of the JSME guideline even for cases with a nonzero yaw angle and those with an upstream larger-diameter cylinder.

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