Evaluation of Turbulence-Induced Vibration of a Circular Cylinder in Supercritical Reynolds Number Flow*

Koji IWATA**, Masaki MORISHITA**,
Takaaki SAKAI**, Akira YAMAGUCHI**
and Kenji OGURA***

Experiments have been conducted on turbulence-induced vibration of a circular cylinder in water flow with supercritical Reynolds number ranging from 3×10^5 to 3×10^6 . Based on the power spectral density of cylinder vibrations measured at several Reynolds numbers, fluctuating force coefficients, Strouhal number and correlation length were evaluated. As a result, it was clarified that the prediction method based on the random vibration theory introducing the correlation length has sufficient margin for actual turbulence-induced response.

Key Words: Flow-Induced Vibration, Supercritical Reynolds Number, Turbulence-Induced Vibration, Circular Cylinder, Power Spectral Density, Correlation Length

1. Introduction

In industrial plants such as power and process plants, cylindrical wells are frequently used to house and protect instruments monitoring the state of the fluid in the piping. Thermometer wells, where thermocouples are inserted to measure fluid temperature in the piping, are typical examples. They are installed in the piping and are subjected to fluid flow. Such structures must be designed taking the flow-induced vibrations caused by the fluid into consideration.

To date, practical engineering knowledge on flow-induced vibrations has been well systematized in a number of monographs⁽¹⁾⁻⁽⁴⁾ and review papers^{(5),(6)}. Not a few papers proposing relevant design guides appeared in the 1980s⁽⁷⁾. However, it is in 1992 that the American Society of Mechanical Engineers (ASME) issued nonmandatory code N-1300 in Appendix N of

Section III of the ASME Boiler and Pressure Vessel Code (hereafter ASME N-1300)⁽⁸⁾. This code provides designers with basic design guidelines as well as useful information on flow-induced vibrations, but does not specify the rules in detail. One of the reasons for this may be the fact that the development of design rules is still underway in related areas.

The sodium leakage incident, which occurred at the prototype fast breeder reactor Monju in 1995, was due to flow-induced vibration of a thermometer well in the secondary piping system⁽⁹⁾. Since then, in Japan, much effort has been made toward the standardization of design methods for preventing flow-induced vibrations. The Japan Nuclear Cycle Development Institute (JNC), the governmental organization responsible for the development of Monju, prepared a design guide (hereafter the JNC design guide) for thermometer wells against flow-induced vibrations so as to examine the safety of existing thermometer wells of Monju and to modify their design when necessary⁽¹⁰⁾⁻⁽¹²⁾. Along these lines, in 1998, Japan Society of Mechanical Engineers (JSME) published guidelines for evaluation of flow-induced vibration of a cylindrical structure in a pipe (hereafter the JSME

* Received 2nd April, 2001

** Oarai Engineering Center, Japan Nuclear Cycle Development Institute, 4002 Narita, Oarai, Ibaraki 311-1393, Japan. E-mail: iwata@oec.jnc.go.jp

*** Toshiba Corporation, 8 Shin-sugita, Isogo-ku, Yokohama 235-8523, Japan

guideline)⁽¹³⁾. In these guidelines, two potential vibration sources, that is, vortex-induced vibration in the flow and the cross flow directions and turbulence-induced vibration are considered. It is required in design that turbulence-induced vibration be assessed after the avoidance or suppression of vortex-induced vibration is achieved.

Available experimental data on flow-induced vibration of cylinders in cross flow are obtained mostly in the subcritical Reynolds number region, i.e. $Re < 2 \times 10^5$; little data is obtained in the region beyond the critical Reynolds number because of experimental difficulty. The situation is the same for turbulence-induced vibration, which is the subject of the present study. In particular, with regard to the turbulent fluid force spectral density data, which are essential in assessing turbulence-induced vibration, only a few data from wind tunnel tests⁽¹⁴⁾ and water tests in a pipe⁽¹⁵⁾ have been reported to date. More experimental data in practical conditions such as in high Reynolds number water flow in a pipe are necessary to establish the evaluation methods for turbulence induced vibration.

With the above points as background, the authors conducted a series of experiments for a single cylinder subjected to water cross flow in a pipe to obtain fundamental characteristics of both vortex-induced and turbulence-induced vibrations in the critical through the supercritical regions, $3 \times 10^5 < Re < 3 \times 10^6$, where experimental data are lacking in spite of their practical importance. Some of the results have already been reported⁽¹⁶⁾⁽¹⁷⁾. This paper presents the evaluation of the results on turbulence-induced vibration, by following up on the previous report⁽¹⁷⁾; the results on vortex-induced vibration are reported elsewhere⁽¹⁸⁾.

The objectives of this paper are: first to present the experimental results for single cylinders with and without a taper, discussing power spectral density of fluid forces, correlation length, fluctuating force coefficients, the effects of Reynolds numbers and turbulence intensity, and so forth, and second to assess the response prediction method adopted in the JNC design guide by comparison with the experimental data.

2. Nomenclature

- C'_D : fluctuating drag coefficient
 C_{LR} : random component of fluctuating lift coefficient
 C_R : random fluid force coefficient
 C_0 : peak factor
 d_m : mean outer diameter of tapered cylinder
 $= (d_0 + d_{\max})/2$, m

- $d_{\max}$: maximum (root) outer diameter of tapered cylinder, m
 d_0 : outer diameter at tip of tapered cylinder, m
 f_i : natural frequency of i -th vibration mode, Hz
 f_s : frequency of periodic vortex shedding, Hz
 $G_F(f)$: single-sided power spectral density of fluid force
 $J_i^2(f)$: joint acceptance for i -th vibration mode
 L : cylinder length, m
 L_c : axial correlation length, m
 L_e : cylinder length subject to fluid flow, m
 $m(x)$: total mass per unit length of cylinder including cylinder mass, contained materials mass and cylinder displaced fluid mass, kg/m
 M_i : effective modal mass per unit length of i -th vibration mode, kg/m
 Re : Reynolds number $= Vd_m/\nu$
 S : reduced frequency $= fd_m/V$
 St : Strouhal number $= f_s d_m/V$
 T : degree of taper $= (d_{\max} - d_0)/L$
 V : mean velocity, m/s
 V_r : reduced velocity $= V/(fd_m)$
 $y_R(x)$: turbulence-induced displacement amplitude of cylinder, m
 ζ_i : fluid damping factor for i -th vibration mode
 ν : kinematic viscosity, m^2/s
 ξ_i : structural damping factor in the air for i -th vibration mode
 ρ : fluid mass density, kg/m^3
 $\phi(f)$: normalized power spectral density function of fluid force
 $\varphi_i(x)$: i -th vibration mode shape of cylinder

3. Test Method and Specimens

3.1 Test apparatus

The experiments were performed using the hydraulic test loop shown in Fig. 1. The test section, a rigid block with a 318 mm-diameter cylindrical hole, is placed in the loop, and a test specimen is mounted in it horizontally (see Fig. 2). A 6 m-long running

Fig. 1 Test loop

pipe with the same inner diameter (318 mm) is connected to the test section to reduce the upstream flow effect. A rectification lattice is placed across the entrance of the running pipe. The turbulence intensity level in the test section is changed via a lattice. Three lattices with different densities were used in this study. A laser Doppler velocimeter is used to measure turbulence intensity. The system pressure can be increased to 0.8 MPa by a pressurizer, and the water temperature can be controlled by the heat exchanger line. More details on the test apparatus are given by Sakai et al.⁽¹⁶⁾.

3.2 Test specimens

Four test specimens were prepared: two straight (D, E) and two tapered (B, C) cylinders. The dimensions of the specimens are given in Figs. 3(a), (b). Specimens D and C are used for the measurement of spectra of fluctuating forces and tip displacements using strain gages mounted inside, at the root of the specimens; specimens E and B are used for the measurement of spanwise pressure distributions using five pressure gages placed in the spanwise direction as shown in Fig. 3. The degree of taper of a tapered cylinder is defined by $T = (d_{\max} - d_0)/L$, where $d_{\max}$ and d_0 are the root and the tip diameters, respectively, and L is the length. The value of T was 0.06.

A pressure gage with a 6 mm-diameter sensing area was placed at the angle 150° from the forward

stagnation point on the surface of a cylinder, to avoid its influence on turbulence separation. Natural frequencies in the air were measured to be about 780 Hz and 970 Hz for specimens D (straight) and C (tapered), respectively.

3.3 Test method

Measurement of turbulence-induced vibration behaviors was carried out under the condition satisfying both the reduced frequency range $V_r < 1$, where no vortex-induced vibration takes place, and the Reynolds number range $3 \times 10^5 < Re < 3 \times 10^6$. The test conditions were determined as follows:

(1) Average flow velocity in a pipe: 5 - 15 m/sec.

The average flow velocity in a pipe was varied between 5 and 15 m/sec by controlling the pump revolution rate.

(2) Water temperature: 50°C and 190°C

Water under pressure was heated up to 50°C or 190°C to raise its kinematic viscosity. This makes higher Reynolds number flow possible in the above flow velocity range. The kinematic viscosities of water used in this study were $\nu_{50^\circ\text{C}} = 5.5 \times 10^{-7} \text{ m}^2/\text{s}$ and $\nu_{190^\circ\text{C}} = 1.6 \times 10^{-7} \text{ m}^2/\text{s}$, respectively. Temperature was controlled within $\pm 0.1^\circ\text{C}$. Measurements were made over 5 min at a steady state constant flow velocity, thereby yielding spectra of the fluctuating forces, tip displacement amplitudes, spanwise pressure distributions, and so forth. In the experiments, three levels of turbulence intensities were generated using lattices with different densities, according to the method described in a paper by Sakai et al.⁽¹⁶⁾, resulting in turbulence intensities of 2%, 3% and 5% in the central region of the pipe cross section.

4. Test Results and Discussions

4.1 Power spectral density of fluid force

The measured results of normalized power spectral density of lift force ϕ for specimen D in the case of turbulence intensity of 2% are shown in Figs. 4(a),

Fig. 2 Test section

(a) Straight cylinder (D: w/o pressure sensor, E: with pressure sensor)

(b) Tapered cylinder (B: w/o pressure sensor, C: with pressure sensor)

Fig. 3 Schematics of test specimens (Unit: mm)

Fig. 4 Normalized power spectral density of lift force (Cylinder: D, Turbulence intensity: 2%)

(b). The range of the Reynolds number was $2.9 \times 10^5 < Re < 2.7 \times 10^6$ in this case. The power spectra globally tend to decrease as the reduced frequency $S (=fd/V)$ increases, but in the cases $Re > 7.6 \times 10^5$, a distinct peak gradually appears around $S=0.2$ as Re is increased; however, a peak is unnoticeable in the range of $2.9 \times 10^5 < Re < 4.3 \times 10^5$. Another peak appears around $S=0.05$ for $Re > 1.9 \times 10^6$, as is observed in Fig. 4(b). For $Re > 2.2 \times 10^6$, the peak around $S=0.2$ becomes relatively smaller and smaller as the peak around $S=0.05$ increases.

Figures 5(a), (b) show normalized power spectral density of lift force ϕ obtained for the flow with a turbulence intensity of 6%. Similar trends are observed in terms of peaks around $S=0.2$ and $S=0.05$. In view of these trends, it can be said that the effects of turbulence intensity are relatively small.

Figures 6(a), (b) show normalized power spectral density of lift force ϕ for tapered specimen C in the flow with a turbulence intensity of 6%. The range of the Reynolds number, based on the mean outer diameter, was $4.4 \times 10^5 < Re < 2.8 \times 10^6$. The outer diameter at the tip of the specimen, d_0 , was used to evaluate reduced frequencies S . In this case, the peak in the spectrum around $S=0.2$ became gradually distinctive under the condition $Re > 7 \times 10^5$; another peak around $S=0.05$ appeared under the condition $Re > 1.5 \times 10^6$. The general trends of tapered specimens

Fig. 5 Normalized power spectral density of lift force (Cylinder: D, Turbulence intensity: 6%)

Fig. 6 Normalized power spectral density of lift force (Tapered cylinder: C, Turbulence intensity: 6%)

are almost the same as those of cylindrical specimens. One difference to be pointed out is that the relative magnitude of two peaks, one around $S=0.2$ and the

Fig. 7 Normalized power spectral density of fluctuating pressure (Cylinder: E)

Fig. 8 Strouhal number

other around $S=0.05$, is different between cylindrical and tapered cylinders; for instance, the peak values around $S=0.05$ relative to those around $S=0.2$ in cylindrical specimen D are obviously large compared with those in tapered specimen C.

The cause of the peak appearing around $S=0.05$ may be due to an edge vortex which occurs at the tip of a cylinder, which is inferred from the Strouhal number distribution data along a cantilevered cylinder by Fox and Apelt⁽¹⁹⁾. This can also be deduced from Fig. 7, where the power spectra of fluctuating pressure measured at five locations along the cylinder are compared. The spectral data by a pressure sensor nearest to the tip (28 mm from the tip) show that the peak around $S=0.05$ is prominent compared with that around $S=0.2$. In contrast, the spectral data at all the other locations display a peak value only around $S=0.2$. In view of this characteristic, it is highly probable that the peak around $S=0.05$ is due to an edge vortex caused at the tip.

The Strouhal numbers estimated from the power spectra of lift force are presented in Fig. 8. In the range $3 \times 10^5 < Re < 7 \times 10^5$, it was not possible to deter-

mine the Strouhal numbers, because there is no identifiable peak in the spectra. In the region $Re > 7 \times 10^5$, the Strouhal numbers could be identified without ambiguity, remaining almost constant around 0.2 as shown in Fig. 8. This implies that the periodicity in vortex shedding recovered with the increase of the Reynolds number in the supercritical region. The Strouhal numbers for tapered cylinders appear to be a little smaller than those for straight cylinders, which is ascribed to the fact that the tip diameter was used in the calculation of the Strouhal numbers. Hence, the data suggest that an actual characteristic diameter for a tapered cylinder, which is closely associated with vortex shedding, lies between the tip and the root diameters.

The power spectral density functions ϕ cited for comparison in Figs. 4 thru 6 are the ones which Mulcahy⁽¹⁵⁾ proposed for turbulence-induced fluctuating lift and drag forces, based on the pipe water flow tests in the subcritical region. It should be noted that the effects of vortex-induced fluctuating forces are excluded in Mulcahy's functions, while the present data include them despite measurements in the region $V_r < 1$. This yields big discrepancies of power spectral densities around $S=0.05$ and $S=0.2$ between the present data and Mulcahy's functions. Apart from this point, it can be considered that the general shape of measured power spectral density in the critical through the supercritical regions is highly consistent with Mulcahy's functions for the subcritical region.

The prediction method of turbulence-induced vibration amplitude, described in section 4.2, employs Mulcahy's fluctuating drag spectrum function, which gives conservative results in $S > 0.3$, a practically important region. It seems suitable to consider that this function can represent measured spectra adequately in $S > 0.3$, although the measured values slightly exceeded the function's values in a very limited region of $0.3 < S < 0.4$.

4.2 Evaluation for turbulence-induced vibration response

4.2.1 Prediction method for turbulence-induced vibration amplitudes

Responses of structures excited by flow turbulence can be treated by the random vibration theory. In the JNC design guide⁽¹⁰⁾, the prediction method of turbulence-induced vibration amplitudes for thermometer wells in piping is given. The methods generally agree with those recommended by ASME N-1300, but differ in detail. The outline of the methods of the JNC design guide⁽¹¹⁾ is presented here.

From the practical viewpoint, it is assumed that turbulence in a uniform cross flow is homogeneous, isotropic, and stationary, and that structural damping

is small. The following equation using the acceptance integral approach is used to estimate the mean square response of a cylindrical structure (beam) excited by turbulence, as in ASME N-1300 :

$$\overline{y_R^2(x)} = \sum_i \left\{ \frac{G_F(f_i) L^2 J_i^2(f_i)}{64 \pi^3 f_i^3 M_i^2 (\xi_i + \zeta_i)} \varphi_i^2(x) \right\} \quad (1)$$

where $\overline{y_R^2(x)}$ is the total mean square displacement amplitude or the sum of the mean square displacement amplitudes of the individual vibration modes, $\varphi_i(x)$ is the i -th vibration mode of the cylinder, $G_F(f_i)$ is the single-sided power spectral density (PSD) of turbulent fluid force, $J_i^2(f_i)$ is called the axial joint acceptance and is defined as⁽²⁰⁾

$$J_i^2(f_i) = \frac{1}{L^2} \int_0^L \int_0^L \varphi_i(x_1) \varphi_i(x_2) \times r(x_1, x_2, f_i) dx_1 dx_2 \quad (2)$$

with $r(x_1, x_2, f_i)$ the correlation function of the pressure (fluid force) distribution in the axial (x) direction of the cylinder, and M_i is the generalized mass of the cylinder for i -th vibration mode given by $M_i = \int_0^L m_i(x) \varphi_i^2(x) dx$, with $m_i(x)$ the total cylinder mass (structural mass, contained materials mass, and structure displaced fluid mass) per unit length. In Eq. (1), structure damping in the air ξ_i and fluid damping ζ_i for the i -th vibration mode are considered.

The turbulent fluid force PSD is expressed as

$$G_F(f) = \left(\frac{1}{2} C_R \rho V^2 D \right)^2 \left(\frac{D}{V} \right) \phi(f) \quad (3)$$

where C_R is the random fluid force coefficient, and $\phi(f)$ is the normalized PSD function. The random fluid force coefficient $C_R = 0.2$, which was determined based on the existing literature data, is used in the JNC design guide both for the lift and the drag components. The normalized PSD function in the JNC design guide has the form

$$\phi(f) = \frac{4}{1 + \left(2\pi \frac{fD}{V} \right)^2} \quad (4)$$

which is the same function that Mulcahy⁽¹⁵⁾ adopted for the fluctuating drag force spectra, that is, the normalized theoretical velocity fluctuation spectrum for one-dimensional, homogeneous, isotropic turbulent flow. Equation (4) is also used commonly for both the lift and the drag force spectra.

The joint acceptance $J_i^2(f_i)$ can be simplified, by assuming that the correlation length for random fluid forces is small in comparison to the cylinder length, L , and is uniform over the cylinder, the vibration mode is unchanged within the correlation length, and the cylinder length subjected to fluid flow is L_e ,

$$J_i^2(f_i) = \frac{L_c}{L_e^2} \int_{L_e} \varphi_i^2(x) dx \quad (5)$$

where L_c is the axial correlation length. The JNC design guide adopts $L_c = 3d_m$, using the mean diameter

of a cylinder, which is often cited in the literature^{(8),(15)}. Thus, substituting Eqs. (3) and (5) into Eq. (1) and introducing the peak factor C_0 yields the following expression for the peak displacement amplitude :

$$y_R(x) = C_0 \sqrt{\sum_i \left\{ \frac{L_c G_F(f_i) \int_{L_e} \varphi_i^2(x) dx}{64 \pi^3 f_i^3 M_i^2 (\xi_i + \zeta_i)} \right\} \varphi_i^2(x)} \quad (6)$$

where C_0 is the peak factor, which estimates the peak amplitude from the rms amplitude, and is taken equal to 3.0. This value implies that the probability that a randomly chosen peak will not exceed the amplitude estimated with $C_0 = 3$ is about 98.9%.

Equation (6) is the final expression for turbulence-induced displacement amplitude in the JNC design guide. It is found from Eq. (6) that fluctuating fluid forces effectively acting on the cylinder are proportional to $C_R \sqrt{L_c}$. The validity of the setting, $L_c = 3d_m$ and $C_R = 0.2$, in the JNC design guide is assessed in the following sections based on the present test data in the range of high Reynolds numbers.

The JSME guideline provides a simplified displacement amplitude prediction method employing not the correlation length explicitly but the random fluid force coefficient including the axial correlation effects implicitly⁽¹³⁾.

4.2.2 Correlation length The correlation length was evaluated by the present experiments, using correlation factors for pressure variation measured at five locations along the cylinder axis (see Fig. 3).

The correlation length is defined by

$$L_e = \int_0^{L_e} \int_0^{L_e} r(x_1, x_2) dx_1 dx_2 \quad (7)$$

where $r(x_1, x_2)$ is the correlation factor between two arbitrary pressure sensors at the locations x_1 and x_2 . The discretized form of Eq. (7), in the case of five pressure gages, becomes

$$L_e = \sum_{i=2}^5 \left[\frac{1}{2} \left\{ \sum_{j=2}^5 \frac{(r_{i,j} + r_{i,j-1})}{2} \Delta L_{i-1,j} + \sum_{j=2}^5 \frac{(r_{i-1,j} + r_{i-1,j-1})}{2} \Delta L_{j-1,i} \right\} \right] \frac{\Delta L_{i-1,i}}{\Delta L_{1,5}} \quad (8)$$

where $r_{i,j} \equiv r(x_i - x_j)$ and $\Delta L_{i,j} \equiv |x_i - x_j|$ are the correlation factor and the distance, respectively, between two arbitrary pressure sensors, i and j ($=1 \sim 5$), selected out of five pressure sensors shown in Fig. 3.

As shown in Fig. 9, the correlation lengths obtained for the straight and the tapered cylinders were about $1d_m$ in the region $Re < 10^6$, which suggests that the fluid force's correlation in the axial direction is quite small. Beyond $Re < 10^6$, however, the correlation length tends to increase gradually, indicating a sign of enhancement of periodic vortex shedding. From this figure, it is sufficiently conservative to use

Fig. 9 Correlation length

the correlation length $L_c = 3d_m$ in Eq. (6).

4.2.3 Turbulence-induced fluid force coefficients

The turbulence-induced random fluid force coefficient C_R was evaluated by separating turbulence-induced fluid forces from the total fluid forces. With regard to the drag forces, however, the total fluid forces were treated to be equal to the turbulence-induced forces, because there is no peak in the power spectra of the drag force. On the other hand, the turbulence-induced lift forces were determined by subtracting fluid forces around $S=0.2$ from the total fluid forces calculated by the integration of the power spectra over the whole reduced frequency range. The random fluid force coefficients corresponding to C_R in Eq. (3), that is the fully correlated one by its definition, were evaluated from the above calculated fluid force coefficients, which are inclusive of the axial correlation effects in nature, multiplied by the factor $\sqrt{(L_e/L_c)}$: actually, the random lift coefficient C_{LR} and the random drag coefficient C_{DR} were evaluated from the lift and the drag forces measured by the experiments. The correlation length $L_c = d_m$ was used for the calculation of C_{LR} and C_{DR} .

The results for C_{LR} and C_{DR} which were obtained by the method described above are shown in Fig. 10, where they are compared with Mulcahy's experimental data which were obtained under the conditions close to full correlation⁽¹⁵⁾. The random fluid force coefficient C_R in Eq. (3) is set to be constant ($C_R = 0.2$) in the JNC design guide independently of the Reynolds number, based on the assumption of isotropic turbulence. Figure 10 demonstrates that this setting is effective throughout the whole Reynolds number region including the supercritical region.

4.2.4 Comparison of displacement amplitudes

Peak tip displacement amplitudes for the cylinders tested in this study were estimated using Eq. (6). Fluid damping was neglected in the calculation, because its effect is considered to be quite small in

Fig. 10 Random fluctuating force coefficients

Fig. 11 Comparison of the measured and the estimated tip displacement amplitudes

turbulence-induced vibration. One vibration mode, or the fundamental vibration mode, sufficed for this calculation. The estimated peak displacement amplitudes normalized by the measured ones are shown in Fig. 11. It is noted that the measured amplitudes here are evaluated by the vector composition of the two maximum displacement amplitudes in the flow and the cross flow directions. The estimated displacement amplitudes are shown to be roughly five to ten times the measured values. It can be considered that such conservatism mainly arises from the conservatism included in $\phi(f)$ and L_c . Through the comparison, it was confirmed that Eq. (6) yields more than sufficient margins when applied to the design.

5. Conclusions

Experiments on turbulence-induced vibration of a cylindrical structure subjected to cross water flow in piping was performed in the supercritical Reynolds number range of $3 \times 10^5 < Re < 3 \times 10^6$ and the response prediction method of the JNC design guide was assessed based on the measurement results. The following conclusions were drawn.

1. The shape of the power spectrum for fluid force, particularly the peak behavior corresponding to the

vortex shedding frequency, varied with the Reynolds number, while influences of the turbulence intensity were small within the turbulence intensity levels of 2–6% in the present tests.

2. The shapes of power spectra of fluid force were roughly the same between straight and tapered cylinders tested, except that the peaks associated with the tip edge vortex appeared in straight cylinders more clearly than in tapered cylinders.

3. The Strouhal numbers estimated from the power spectra of fluid force were approximately 0.2 in the range $Re > 7 \times 10^5$; however, it was not possible to determine Strouhal numbers in the range $3 \times 10^5 < Re < 7 \times 10^5$ because of the absence of identifiable peak in the spectra.

4. The correlation length of cylinders was about $1d_m$ in the range $3 \times 10^5 < Re < 10^6$, but appears to increase gradually in the range $Re > 10^6$ with the recovery of the periodicity in the vortex in the wake.

5. The prediction method for turbulence-induced vibration response in the JNC design guide has sufficient margin for actual displacement amplitudes. The random fluid force coefficient $C_R = 0.2$ and the correlation length $L_c = 3d_m$, which are prescribed there, are adequate in a wide range of Reynolds number up to 3×10^6 .

References

- (1) Blevins, R.D., *Flow-Induced Vibration*, 2nd ed., (1994), Krieger Publishing Company.
- (2) Chen, S.S., *Flow-Induced Vibration of Circular Cylindrical Structures*, (1987), Hemisphere Pub. Co.
- (3) Naudasher, E. and Rockwell, D., *Flow-Induced Vibrations Engineering Guide*, (1994), Int. Assoc. Hydraul. Res., A.A. Balkema.
- (4) Sumer, B.M. and Fredsoe, J., *Hydrodynamics around Cylindrical Structures*, (1997), World Scientific.
- (5) Paidoussis, M.P., *A Review of Flow-Induced Vibrations in Reactors and Reactor Components*, *Nuclear Engineering Design*, Vol. 74 (1982), pp. 31–60.
- (6) Weaver, D.S., Ziada, S., Au-Yang, M.K., Chen, S. S., Paidoussis, M.P. and Pettigrew, M.J., *Flow-Induced Vibration in Power and Process Plant Components—Progress and Prospects*, *Trans. ASME, J. Pressure Vessel Technology*, Vol. 122 (2000), pp. 329–348.
- (7) Au-Yang, M.K., *Design Guides, Codes, and Standards in Flow-Induced Vibration and Fluid-Structure Interaction*, *Technology for the '90s*, Au-Yang, M.K. ed., (1993), pp. 958–972, ASME.
- (8) ASME Boiler and Pressure Vessel Code, Section III, Division 1, Appendix N-1300, (1995), ASME.
- (9) Itoh, K., Yamaguchi, A., Wada, Y., Iwata, K., Morishita, M., Miyake, O., Aoto, K., Okabayashi, K., Shiba, K., Anoda, Y. and Matsuoka, S., *Technical Report on Monju's Sodium Leak Incident*, *Journal of Atomic Energy Soc. of Japan*, (in Japanese), Vol. 39, No. 9 (1997), pp. 704–725.
- (10) *Proposed Flow-Induced Vibration Design Guide for Thermometer Wells*, (in Japanese), PNC TN9410 97-042, (1997), Japan Nuclear Cycle Development Institute (formerly PNC).
- (11) Iwata, K., Morishita, M., Yamaguchi, A. and Wada, Y., *Flow-Induced Vibration Design Guide for Thermometer Wells—Background Document* —, (in Japanese), JNC TN9400 2000-088, (2000), Japan Nuclear Cycle Development Institute.
- (12) Dozaki, K., Morishita, M. and Iwata, K., *Modification and Design Guide of Thermowell for FBR*, PVP-Vol. 363, *Flow-Induced Vibration and Transient Thermal-Hydraulics*, Au-Yang, M.K. et al. eds., (1998), pp. 125–133.
- (13) JSME Standard S012, *Guidelines for Evaluation of Flow-Induced Vibration of a Cylindrical Structure in a Pipe*, (in Japanese), (1998), JSME.
- (14) Fung, Y.C., *Fluctuating Lift and Drag Acting on a Cylinder in a Flow at Supercritical Reynolds Numbers*, *J. Aeronaut. Sci.*, Vol. 27, No. 11 (1966), pp. 801–814.
- (15) Mulcahy, T.M., *Design Guide for Single Circular Cylinder in Turbulent Cross Flow*, ANL-CT-82-7, (1982).
- (16) Sakai, T., Iwata, K., Yamaguchi, A., Morishita, M. and Ogura, K., *Vortex-Induced Vibration of a Circular Cylinder in High Reynolds Number Flow*, *Trans. Jpn. Soc. Mech. Eng.*, (in Japanese), Vol. 65, No. 635, B (1997), pp. 44–50.
- (17) Iwata, K., Yamaguchi, A., Morishita, M., Sakai, K. and Ogura, K., *Turbulence-Induced Vibration of a Circular Cylinder in High Reynolds Number Flow*, *Trans. Jpn. Soc. Mech. Eng.*, (in Japanese), Vol. 65, No. 635, B (1997), pp. 51–56.
- (18) Sakai, T., Iwata, K., Morishita, M. and Kitamura, S., *Vortex-Induced Vibration of a Circular Cylinder in Super-Critical Reynolds Number Flow and Its Suppression by Structure Damping*, *JSME Int. J. Ser. B*, Vol. 44, No. 4 (2001), pp. 712–720.
- (19) Fox, T.A. and Apelt, C.J., *Fluid-Induced Loading of Cantilevered Circular Cylinders in a Low Turbulent Uniform Flow*, Part 3, *J. fluids Struct.*, Vol. 7 (1993), pp. 375–386.
- (20) Powell, A., *On the Fatigue Failure of Structures due to Vibrations Excited by Random Pressure Fields*, *Journal of the Acoustical Society of America*, Vol. 30, No. 12 (1958), pp. 1130–1135.