An Experimental Study on the Acoustic Instabilities of Tube Arrays in Cross Flows*

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Tube arrays in cross flows sometimes excite the acoustic vibrations of sound fields in ducts. It is very important, for practical applications, to understand the conditions of acoustic resonances in ducts with tube bundles. Exciting forces due to tube arrays of pitch-to-diameter ratio of 1.57 were measured in a water model duct and the characteristics of acoustic resonance in ducts with tube arrays were revealed. The results of the experiments agree with the field test data very well.

Key Words: Oscillatory Flow, Unsteady Flow, Fluctuating Flow, Flow-Induced Vibration, Tube Arrays, Cross Flow, Acoustic Vibration, Resonance, Exciting Force

1. Introduction

Acoustic resonance which is sometimes observed in boilers or heat exchangers, is mainly caused by the exciting forces of tube arrays employed in those machines. A lot of research has been carried out on the forces exciting acoustic vibrations around tube arrays^{(1)~(4)}. Most of this research, however, has been devoted to frequency analysis on the fluctuations of fluid forces around tube arrays placed in steady cross flows, while there has been no research on the detail analysis of the characteristics of fluid forces in fluctuating flows which are caused by acoustic resonance.

In order to study fluid forces in fluctuating flows, we have designed an oscillating flow generator to

simulate acoustic vibrations in a water flow duct, measured fluid forces acting on tube arrays, and analyzed the conditions under which acoustic resonances occur, based on the experimental results.

Performing the experiments on square-lattice tube arrays with a pitch-to-diameter ratio of 1.57 we measured dynamic lifts and dynamic drags in three cases. These include (1) steady flows, (2) addition of oscillating flows in the direction along the width of the duct, and (3) addition of oscillating flows in the direction along the main flows.

We then obtained the results on the conditions under which acoustic resonances occur in the direction along the width of the duct or in the direction along the main flow. These results agree well with those obtained in the field tests.

2. Symbols

C_D : dimensionless amplitude of dynamic drag

C_{D0} : coefficient of steady drag

C_L : dimensionless amplitude of dynamic lift

C_M : dimensionless amplitude of inertia force due to the added mass

D : dynamic drags or its amplitude

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- d : diameter of tubes
 f : frequency
 f_p : frequency of forced oscillation
 f^* : dimensionless frequency
 L : dynamic lift or its amplitude
 l : length of tubes
 Re : Reynolds number
 St : Strohal number
 t : time or tube pitch (in the direction along the width of ducts)
 u_p : fluctuating flow velocity caused by forced oscillation
 V_{ap} : main flow velocity
 V_g : velocity of flow between tubes
 x_0 : amplitude of forced oscillation (in the direction along the main flow)
 y_0 : amplitude of forced oscillation (in the direction along the width of duct)
 ω : angular frequency
 ω_p : angular frequency of forced oscillation
 Φ : the phase difference of the fluid force relative to the acceleration of forced oscillation
 ρ : density of fluid
 ν : kinematic viscosity

3. Experimental Methods

3.1 Experiments in steady cross flows

Applying steady flows to the model tube arrays in the water duct, we measured spanwise and flow-wise components of the fluctuating forces of a tube due to the ambient flows. Here, 'spanwise' and 'flow-wise' are defined as being 'parallel' and 'perpendicular' to the main flows, respectively. Based on the measurements for the two components, we derive the amplitudes of drag forces D and those of lift forces L . D and L were converted into the dimensionless values C_D and C_L by Eqs. (1) :

$$\left. \begin{aligned} C_D &= D / \left(\frac{1}{2} \rho V_g^2 dl \right) \\ C_L &= L / \left(\frac{1}{2} \rho V_g^2 dl \right) \end{aligned} \right\}, \quad (1)$$

where V_g is the maximum average velocity of the flow between tubes (see Fig. 1).

Since C_D and C_L are functions of frequencies, they are derived as the spectra of the dimensionless frequency defined by Eq. (2).

$$f^* = fd / V_g \quad (2)$$

3.2 Experiments in the spanwise forced oscillating flows

The ambient flow around the tube arrays during acoustic vibration is composed of the main flow and oscillating particle velocity due to acoustic vibration. We can thus generate the equivalent flow around the tube arrays during acoustic vibration by adding the

oscillating flow, which is excited by external oscillation with a constant amplitude, to the main flow from the left and right sides of the tube arrays (oscillating flow is perpendicular to the main flow). If the dynamic fluid force acting on tubes is able to damp down the fluctuating flow due to the excitation, the corresponding acoustic vibration will be damped accordingly and acoustic resonance will never take place. On the other hand, if the fluid force is able to amplify the fluctuating flow from the left and right sides of the tube arrays, the amplitude of the acoustic vibration grows, eventually resulting in acoustic resonance.

Let the velocity of the oscillating flow due to the excitation be given by $u_p = -y_0 \omega_p \cos(\omega_p t)$ and $|u_p| \ll V_g$ (see Fig. 2). Then the dynamic lift L can be represented as shown by Eq. (3).

$$\begin{aligned} L &= \frac{\pi}{4} \rho d^2 l C_{M y_0} \omega_p^2 \sin \omega_p t \\ &\quad - \frac{1}{2} \rho V_g d l C_{D_0} (y_0 \omega_p) \cos \omega_p t \\ &\quad + \frac{1}{2} \rho V_g^2 d l C_{L^*} \sin(\omega_p t + \phi) \end{aligned} \quad (3)$$

In Eq. (3), the first term expresses the inertial force due to the added mass, the second the damping due to drag, and the third the dynamic lift due, for example, to Karman vortices where C_{L^*} is the amplitude. C_{D_0} stands for the coefficient of drag on the tube

Fig. 1 Velocity of flow between tubes

Fig. 2 Spanwise fluctuating flow

and ϕ the phase difference of the dynamic lift relative to the acceleration of the forced oscillation.

Equation (3) can be alternatively represented by the following equations:

$$L = L_0 \sin(\omega_p t + \Phi) \quad (4)$$

$$\left. \begin{aligned} L_0 &= [(Ka + K \cos \phi)^2 + (K_0 - K \sin \phi)^2]^{1/2} l \\ \Phi &= \tan^{-1} [(-K_0 + K \sin \phi) / (Ka + K \cos \phi)] \end{aligned} \right\} \quad (5)$$

where

$$\left. \begin{aligned} Ka &= \frac{1}{2} \rho V_\theta^2 y_0 \frac{\pi}{2} C_M (\omega_p d / V_\theta)^2 \\ K_0 &= \frac{1}{2} \rho V_\theta^2 y_0 C_{D0} (\omega_p d / V_\theta) \\ K &= \frac{1}{2} \rho V_\theta^2 d C_L^* \end{aligned} \right\} \quad (6)$$

In Eq. (4), Φ is the phase difference of L relative to the acceleration of the forced oscillation. In the case in which Φ is positive ($0 < \Phi < \pi$), the fluid force is able to amplify the fluctuating flow from the left and right sides of the tube arrays, while for negative Φ ($-\pi < \Phi < 0$) it is able to damp the fluctuating flow.

Let us represent the phase difference by Φ_0 for the case in which $K=0$ in Eqs. (5), that is, dynamic lift due to Karman vortices C_L^* is zero, and Φ_0 can be represented by the following equation.

$$\Phi_0 = \tan^{-1} \left[\left(\frac{-1}{\pi^2} \right) \left(\frac{C_{D0}}{C_M} \right) / (f_p d / V_\theta) \right] \quad (7)$$

In this case, Φ_0 is always negative, and approaches zero with dimensionless frequency. Since Φ in Eqs. (5) is Φ_0 in Eq. (7) with the effects of C_L^* , Φ can be represented by the curved line ② shown in Fig. 3.

3.3 Experiments in the flow-wise forced oscillating flows

As in the case of the spanwise forced oscillating flow, the oscillating flow running parallel to the main flow can be obtained by adding the oscillating flow by the excitation to the main flow from the upper and lower streams of the tube arrays. We can judge the characteristics of the fluid force from the phase difference as well, as we have seen in the spanwise oscillating flow. In this case, x_0 is represented by the amplitude of the excitation and C_D^* by the dimensionless amplitude of dynamic drag due to Karman vor-

Fig. 3 Phase difference curve

tices for example.

$$D = D_0 \sin(\omega_p t + \Phi) \quad (8)$$

$$\left. \begin{aligned} D_0 &= [(Ka + K \cos \phi)^2 + (K_0 - K \sin \phi)^2]^{1/2} l \\ \Phi &= \tan^{-1} [(-K_0 + K \sin \phi) / (Ka + K \cos \phi)] \end{aligned} \right\} \quad (9)$$

$$\left. \begin{aligned} Ka &= \frac{1}{2} \rho V_\theta^2 x_0 \frac{\pi}{2} C_M (\omega_p d / V_\theta)^2 \\ K_0 &= \frac{1}{2} \rho V_\theta^2 x_0 (2 C_{D0}) (\omega_p d / V_\theta) \\ K &= \frac{1}{2} \rho V_\theta^2 d C_D^* \end{aligned} \right\} \quad (10)$$

4. Experimental Equipments

4.1 A water flow duct

Figure 4 shows the entire water flow duct which was used in the experiments. The central part of the duct is the measuring part, where the model tube arrays are installed. Figure 5 shows the measuring part in detail.

4.2 The model tube arrays and model tubes

Figure 6 shows the model tube arrays used in the experiments. The model tube arrays are composed of gas tubes with the inner circular diameter of 16.1 mm and the outer diameter of 21.7 mm, which are fixed on the support plate in the upper part of the duct. The length of the tubes is 198 mm. This corresponds to the length over which the tubes receive the water flow.

Fig. 4 Water flow duct

Fig. 5 Measuring section

The tube marked with ● in Fig. 6 is the measuring tube which can measure lift and drag, as shown in Fig. 7. This tube is judged to be rigid from its flexure of about 0.03 mm at the bottom when it is receiving the water flow of 1.0 m/s, and has a natural frequency of about 40 Hz in water.

4.3 The oscillating flow generator

Figure 8 shows the oscillating flow generator used to excite flow in the direction along the width of the duct. Oscillating flow along the width of the duct can be obtained in the measuring part by cycling the piston with a certain number of strokes. The oscillating flow travelling along the main flow is obtained by

Fig. 6 Model of tube arrays

applying the oscillating flow excited by the piston from the upper stream and lower stream of the tube arrays. The cycle of the piston is continuously changed from 0 to 7 Hz for about 15 min.

5. Experimental Results

5.1 Dynamic forces in the steady cross flows

In Fig. 9, we plot dynamic lift and dynamic drag which were measured on the model tube arrays in the steady cross flow. The fluid forces acting on tubes in the arrays are generally considered to have different values at different Reynolds numbers ($V_0 d / \nu$). In our experiments, we set the Reynolds number to be about 10^4 , similar to the values of used in practice, around which fluid forces are considered not to change rapidly.

As one can see from the results in Fig. 9, dynamic lift in steady flow is considerably small in the tube arrays with the pitch-to-diameter ratio of 1.57. If the Strouhal number is defined as the dimensionless fre-

Fig. 8 Oscillating flow generator

Fig. 7 Measuring tube

Fig. 9 Fluctuating lifts and drags in steady cross flows

quency at the maximum amplitude of dynamic force, we obtain $St=0.25$, in good agreement with the value obtained by Ishigai et al.⁽²⁾.

5.2 Dynamic lifts in spanwise forced oscillating flows

We added the oscillating flows traveling along the width of the duct by using the generator described in subsection 4.3, and then measured the dynamic lifts acting on the tubes. From those values, we calculated the amplitudes of the dynamic lift and the phase differences relative to the acceleration of the forced oscillation; the results are shown in Fig. 10.

In this figure, C_L is the dimensionless quantity of L_0 in Eq. (4), which is defined by the following equation.

$$C_L = L_0 / \frac{1}{2} \rho V_g^2 dl \quad (11)$$

Φ , which corresponds to the phase difference in Eq. (4), remarkably resembles the curve (the curve ② in Fig. 3) predicted at the point at which it starts with negative values, passes peaks and dips, and converges into zero with changing of the dimensionless frequency ($f_p d / V_g$). One can judge from this behavior of Φ that the regions for acoustic resonance exist intermittently in a range of $f d / V_g = 0.13 \sim 0.47$.

Fig. 10 Fluctuating lifts in spanwise forced oscillating flows

5.3 Dynamic drags in the flow-wise forced oscillating flows

With the same procedure in the case of the oscillating flow along the width of the duct, we measured dynamic drags acting on the tubes in the oscillating flows traveling along the main flow; the results are shown in Fig. 11. The measured amplitudes and phase differences of dynamic drags correspond to D_0 and Φ in Eq. (8). C_D is the dimensionless coefficient defined by Eq. (12).

$$C_D = D_0 / \frac{1}{2} \rho V_g^2 dl \quad (12)$$

One can judge from Φ 's behavior in Fig. 11 that acoustic vibration is amplified in a range of $f d / V_g = 0.35 \sim 0.7$, where acoustic resonance occurs. Φ in Fig. 11 also remarkably resembles curve ② in Fig. 3 plotted against the dimensionless frequency ($f_p d / V_g$), as in the case of oscillating flow along the width of the duct.

While acoustic resonances occur in actual practice, the acoustic vibrations are continuously maintained with some magnitude of amplitudes, which means that there is a boundary at which the amplitude has already grown to a certain magnitude and stops its growth. This is due to the damping effects at the boundaries in the acoustic fields, and probably to the

Fig. 11 Fluctuating drags in flow-wise forced oscillating flows

fact that the exciting forces of the tubes are changed by the amplitudes of the acoustic vibration. The values of Φ in Fig. 12 were derived from the dynamic drags measured with various amplitudes of the oscillating flows traveling along the main flow at the fixed value of $f_p d/V_g = 0.45$. Fig. 12 shows that $\Phi = 0$ for $x_0 2\pi f_p/V_g = 0.3$, and that it becomes negative after this point. Hence, when the amplitude x_0 crosses the corresponding point, the system enters into a range for damping. In other words, the amplitudes in the acoustic resonances cannot become greater than those in the above status. The amplitudes of the acoustic pressures in actual plants, which we have measured for other purposes, also agree with the values predicted from the above hypothesis.

In order to confirm the relationships in the phase of each tube in the tube arrays during acoustic vibrations, we measured the dynamic drag forces at five points in the model arrays. The results at $f_p d/V_g = 0.45$ are shown in Fig. 13, which indicates that drags of all the tubes are on the same phase. Hence, the data of one tube can be applied to other tubes.

6. Discussion

6.1 Comparisons with field data

Based on the data described in subsections 5.2 and 5.3, we derived the conditions for acoustic vibrations traveling along the width of the duct or along the main flow. Since those conditions are represented by the range of dimensionless frequency, we first converted the dimensionless range into the range which is used to express the relationships between the fluid velocities and the frequencies for actual plants, and we then plotted data in Fig. 14 together with the field test data.

The field test data in the directions along the width of the duct and along the main flow were obtained from different plants. As one can see from comparisons of those data, the ranges of the dimensionless frequency for acoustic resonances which are

Fig. 12 Amplitudes of forced oscillations and the phase differences of fluctuating drags

predicted from our experiments on the water flow duct agree well with those obtained from actual plants.

6.2 Dynamic fluid forces and acoustic resonances in steady cross flow

As is shown in Fig. 9, dynamic fluid forces in a steady cross flow are very small. However, this does not necessarily mean that acoustic resonance does not take place. It is not desirable to predict whether acoustic resonance takes place or not from the amplitudes of dynamic fluid forces acting on the tubes in steady cross flow.

6.3 Concluding remarks

The acoustic resonance in the direction along the

Fig. 13 Features of fluctuating drags of 5 tubes in flow-wise forced oscillating flows

Fig. 14 Regions for acoustic resonances

width of the duct can be suppressed by making the natural frequency of acoustic vibration higher by inserting compartment plates. For tube arrays with a pitch-to-diameter ratio of 1.57, as were used in our experiments, acoustic resonance can be suppressed by making the lowest order of the characteristic values higher until the dimensionless frequencies always remain larger than 0.5 in the actual plants. On the other hand, compartment plates cannot be used for the direction along the main flow. To suppress the acoustic resonance along the main flow, we require some special designs, for example, employing acoustic absorbers on the upper or lower stream of the tube arrays.

we used the tube arrays with a pitch-to-diameter ratio of 1.57 in a square pitch configuration. However, it is recognized that the exciting force of a tube array depends on the pitch-to-diameter ratio and its configuration. Further studies on various pitch-to-diameter ratios and tube configurations are required.

7. Conclusions

In order to study the forces of tube arrays which excite the acoustic vibrations, we carried out experi-

ments on the water flow duct which is composed of the tube arrays of the pitch-to-diameter ratio of 1.57 in the square pitch configuration, and examined the conditions under which acoustic resonances take place. Using a model we derived the ranges of the dimensionless frequency where acoustic resonances take place in either direction along the width of the duct or along the main flow; these ranges agree well with those derived from the field test data.

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