# Flow-Induced Vibration of Power and Process Plant Components

## A Practical Workbook

M. K. Au-Yang

Flow-Induced Vibration of Power

With this, we get,

$V = 76.3$ in/s or 1.9 m/s

We would avoid vortex-induced acoustic resonance in this heat exchanger if we keep the cross-flow velocity 30% below the above values (see Chapter 6), or 53 in/s (1.35 m/s). Note that in common with vortex-shedding analysis, the velocity is the free stream velocity, not the pitch velocity $V_p$ commonly used in tube bundle fluid-elastic stability analysis. In terms of the pitch velocity,

$$V_p = \frac{P}{P-d} = 184 \text{ in/s} \quad (15.3 \text{ ft/s or } 4.7 \text{ m/s})$$

Considerable research work has been published in the literature on heat exchanger acoustics. The following section gives a short review of this special topic.

### 12.5 Heat Exchanger Acoustics

The above examples show that once the acoustic modes are "cut on," that is, once the minimum frequency for the formation of standing waves is exceeded, acoustic modes are "dense" in the frequency axis. This, together with the uncertainties in predicting the vortex-shedding frequency, especially in tube bundles, and the ability of vortices to shift their shedding frequencies to "lock-into" the structural modes (see Chapter 6), makes it very difficult to avoid vortex-induced acoustic resonance in heat exchangers once the shedding frequency exceeds the acoustic "cut-on" frequency. However, experience with operating heat exchangers shows that even when the theoretical vortex-shedding frequency coincides with one of the acoustic modal frequencies, very often there is no noise problem. There are two reasons for this: First, probably due to its extremely chaotic nature, no organized vortex-shedding has ever been observed in two-phase flows. As a result, there probably will be no vortex-induced acoustic vibration in heat exchangers with two-phase shell-side fluids. Second, in order to maintain acoustic resonance, there must be a steady supply of energy to drive the acoustic standing waves. Inside a heat exchanger, the flow is highly inhomogeneous. It is not likely that all, or even most, of the tubes would shed vortices at the same frequency. Most likely only a small percentage of the tubes may shed vortices at a certain acoustic modal frequency. If structural, squeeze film and viscous damping of the entire tube bundle dissipate more energy than what a few vortices can generate, steady acoustic modal resonance cannot be established.

At least some of the tubes in some high-performance nuclear steam generators often have to operate above the acoustic cut-on frequency. Thus, more realistic guidelines to avoid heat exchanger acoustic resonance are needed. Before 1970, heat exchanger tube vibration research was much influenced by Chen (1968), who believed heat exchanger tube bundle dynamics was governed by vortex-shedding. After Connors published his

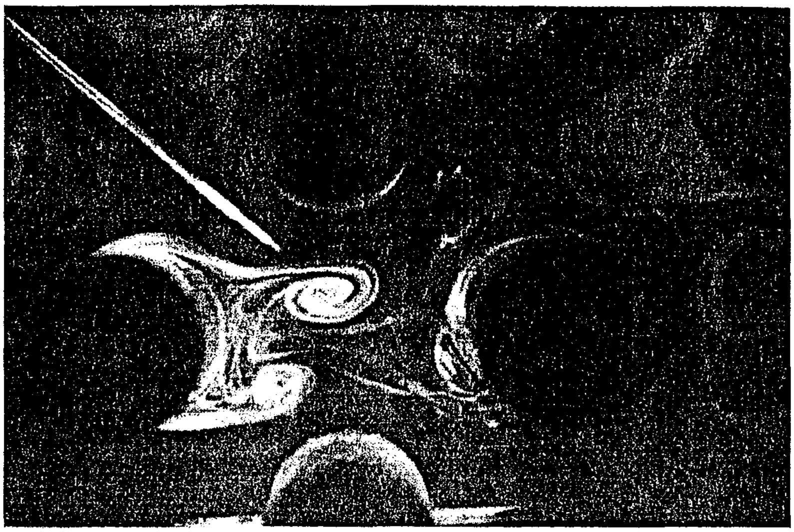


Figure 12.11 Vortex-shedding in a tube bundle (Weaver, 1993)

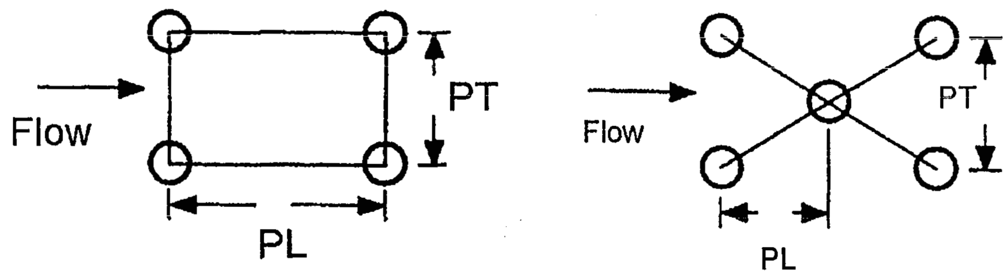
landmark paper on tube bundle fluid-elastic instability (see Chapter 7), the question of vortex-induced vibration and acoustic resonance was largely ignored. From 1970-1990, most of the research activities on tube bundle dynamics focused on fluid-elastic instability. Indeed, the very existence of vortex-shedding in a tube bundle was questioned.

However, noise problems do occur in heat exchangers, though only in those with shell-side gases. Very often the resulting low-frequency noises, which were undoubtedly the result of acoustic resonance in the heat exchanger internal compartments, could be heard even a mile away and were expensive to correct. Some forces are exciting these acoustic modes.

In the late 1980s, there was renewed interest in heat exchanger acoustics by researchers such as Blevins and Bressler (1987), Weaver (1993) and Ziada et al (1989), to mention a few. The readers are referred to an excellent review article by Weaver (1993) for a summary of this effort. This paper also contains an extensive bibliography on related publications. Figure 12.11, reproduced from the above-mentioned paper, conclusively shows the existence of vortex-shedding in a tube bundle. Guidelines for designing heat exchangers to avoid acoustic resonance with vortex-shedding were published.

Resonance Maps (Ziada et al, 1989)

Ziada et al (1989) proposed to classify the susceptibility of a tube bundle to acoustic resonance based on a "resonance parameter" $G$, defined as (refer to Figure 12.12 and Chapter 7 for array geometry definitions),




In-line Staggered

Figure 12.12 Definition of $P_T$ and $P_L$ (Note $P_L$=$L$/2 in Ziada's definition)

$$G_i = \frac{\sqrt{R_c}}{R_a} X_T \quad \text{for in-line arrays} \tag{12.46}$$

$$G_s = \frac{\sqrt{R_c}}{R_a} \frac{[2X_L(X_T - 1)]^{1/2}}{(2X_L - 1)} \quad \text{for staggered arrays} \tag{12.47}$$

where

$$R_c = \frac{V_p d}{\nu} \tag{12.48}$$

is the Reynolds number (see Chapter 6) at the onset of acoustic resonance and $V_p$ is the pitch velocity (see Chapter 7),

$$R_a = \frac{cd}{\nu} \tag{12.49}$$

with $c$ = velocity of sound in the tube bundle (Equation (12.9)), is the "acoustic" Reynolds number,

$$X_L = P_L / d \tag{12.50}$$
$$X_T = P_T / d \tag{12.51}$$

Based on experimental data, the authors gave acoustic resonance maps as functions of tube bundle geometry and the resonance parameter $G$. These are shown in Figures 12.13 and 12.14.

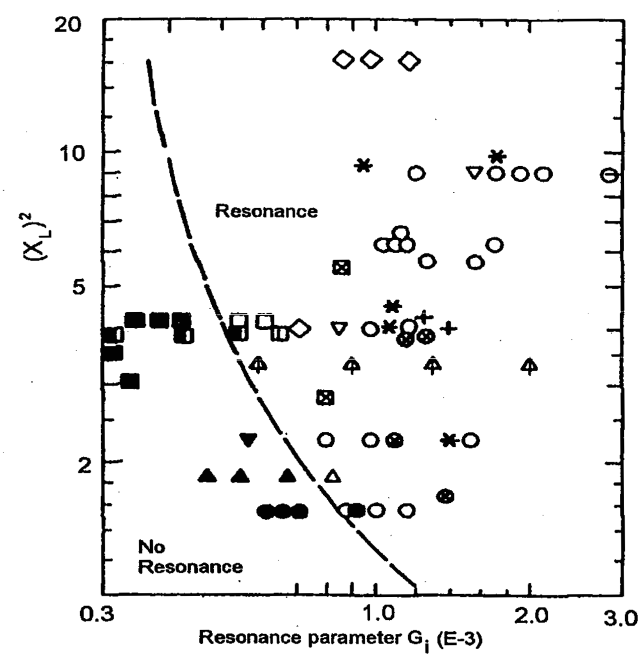



Figure 12.13 Resonance map for in-line array (Ziada et al, 1989)

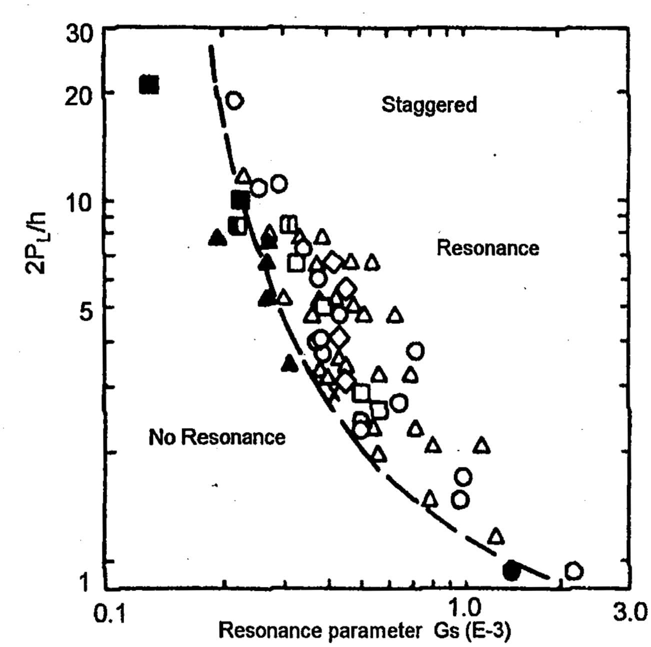



Figure 13.14 Resonance map for staggered array (Ziada et al, 1989)
(*h=(PT-d)/2* or *g* (minimum gap between tubes), whichever is smaller)

Sound Pressure Level

If resonance cannot be avoided, Blevins (1990) suggested the following equation to estimate the resulting acoustic pressure:

$$p_{rms} = \frac{12V_p}{c}\Delta p \tag{12.52}$$

**Example 12.5**

In the heat exchanger in Example 12.4, the maximum pitch velocity is $V_p$=40 ft/s (12.2 m/s). Would there be any acoustic resonance according to Ziada's resonance maps? If the pressure drop across the tube bundle is 3 psi (20,700 Pa), what would be the sound pressure level if acoustic resonance does occur?

**Solution**

From Figures 12.10 and 12.12, we get,

$P_T=P=1.4d$
$P_L=P\sin 60^\circ=1.212d$
$X_T=P_T/d=1.4$
$X_L=P_l/d=1.212$

The gap between adjacent tubes is $P- d=0.4d$, while $(P_T- d)/2=0.2d$. Thus,

$h=0.2d$ and

$2P_l/h=12.1$

To find the resonance parameter $G_s$ at the given pitch velocity, we must calculate the acoustic Reynolds number from Equation (12.49) and the Reynolds number $R_c$ from Equation (12.48). As seen from Example 1.1 in Chapter 1, this can be tricky in the US unit system. At the ambient conditions given in Table 12.12, one can find the dynamic viscosity From the ASME Steam Table (1979). This is given in both the US and SI units in Table 12.13, together with the densities. However, the density given in the ASME Steam Table is in lb/ft$^3$. To form a consistent unit set with the other units used in the US unit system, we must convert the mass density into slug/ft$^3$, then use the equation,

$\nu = \mu / \rho$ =4.144E-7/6.4E-2=6.475E-6 ft$^2$/s=9.32E-4 in$^2$/s

This value is given in Table 12.13. The corresponding calculation in the SI unit system is much more straightforward. Knowing the kinematic viscosity and the velocity of sound in the tube bundle from Example 12.4, we now proceed to calculate the Reynolds number from Equations (12.48) and (12.49),

$R_a = cd/\nu$ =1.15E+7
$R_c$ = 3.2E+5

As expected, the Reynolds numbers are the same in both unit systems. We can now substitute $R_a$, $R_c$, $X_L$=1.212, $X_T$=1.4 into Equation (12.47) to compute,

$G_s$=3.4E-5

From Figure 13.14, we see that the point $2P_L/h$=12.1, $G_s$=3.4E-5 is well in the "no resonance" zone. Thus, there should be no acoustic resonance problem at this pitch velocity.

If acoustic resonance cannot be avoided, from Equation (12.52),

$$p_{rms} = \frac{12V_g}{c}\Delta p = \frac{12*480*3}{17035} = 1.01 \text{ psi} \quad (6900 \text{ Pa})$$

The sound intensity generated is given by Equation (2.10),

$I = c\, p_{rms}$=1.01*17035=17280 (lb-in/s)/in$^2$=302 w/cm$^2$

Using the reference sound intensity, $I_0$=1E-16 w/cm$^2$ at 0 dB (see Chapter 2), we come to the conclusion that this heat exchanger will generate acoustic noise in excess of,

$SPL$=10log$_{10}$(301/1.E-16)=185 dB

Nuclear steam generators with comparable dimensions have been operating under similar conditions for years without emitting any noticeable acoustic noises. This example clearly shows that the mechanism of acoustic resonance in heat exchangers is much more complicated than just the coincidence of vortex-shedding and acoustic modal frequency, and that the conventional "separation" rule of preventing vortex-induced vibration will lead to unreasonably overconservative results when applied to a heat exchanger. However, this example also shows that acoustic resonance in heat exchangers is to be avoided at all costs. An *SPL* of 185 dB is not acceptable by any standard. Furthermore, once the resonance condition is established, any means of reducing the excitation energy or attenuating the acoustic energy will not by itself significantly change the sound intensity much. In the above example, even if we reduce the pressure differential by a factor of 2.0, the reduction in *SPL* will only be 3 dB—a barely noticeable result.

## 12.8 Thermoacoustics

The addition to or removal of heat from a column of gas can cause pressure oscillations in the gas. This leads to the very important subject of thermoacoustics. This section briefly describes the physics of the phenomenon and simple ways to avoid thermoacoustic instability, which may lead to detrimental results. The readers are referred to a review article by Eisinger (1999) and references cited there for further studies.

Figure 12.16 (A) shows a column of gas with one end opened (pressure-released) and the other end closed (pressure reaches maximum). If heat is added either internally or externally to the closed end, or if heat is removed either internally or externally from the open end, spontaneous acoustic pressure oscillation in the column of gas may be initiated. The condition for spontaneous pressure oscillation depends on the temperature gradient between the hot end and the cold end. If there is insufficient temperature gradient between these two ends, spontaneous pressure oscillation cannot be established. This thermoacoustic system is called a Sondhauss tube after the person who first studied it.

Figure 12.16 (B) shows a column of gas with both ends open (pressure-released). If the gas column is positioned vertically and heat is added either internally or externally to the lower part of the column, again spontaneous acoustic pressure oscillation may be initiated. Again, in order to sustain this spontaneous pressure oscillation, there must be sufficient temperature gradient between the cold end and the hot end of the gas column. This thermoacoustic system is called a Rijke tube after the person who first investigated it. Note that in either system, fluid flow is not a requirement for thermoacoustic oscillation, although it may contribute to its initiation or its severity.

Many components in the power and process industries have the Sondhauss or Rijke tube arrangement. Gas furnaces, for example, are Rijke tubes while some nuclear steam generators are Sondhauss tubes. Sometimes thermoacoustic pressure oscillations is designed into the system to help the combustion process, very often unwanted thermoacoustic oscillation may be harmful to the environment or the equipment.

The temperature ratio $\alpha_c = T_h / T_c$, with $T_h$ and $T_c$ in degrees K, between the hot and the cold ends at which spontaneous pressure oscillation is initiated is governed by the ratio of the lengths of the "hot" column to the "cold" column (see Figure 12.16),

$$\xi = (L - \ell)/\ell$$

where $\ell$ is the length of the "cold" column, usually separated from the "hot" column by a furnace grid, a swirler or some other physical object (see Figures 12.16 and 12.17). Or $\ell$ can be the length of the burner while $L$-$\ell$ is the length of the furnace. Thus, $\ell$ is usually well defined. The relationship between the critical temperature ratio $\alpha$ and the geometry parameter $\xi$ was obtained based on experimental data,

$$(\log_{10} \xi)^2 = 1.52(\log_{10} \alpha - \log_{10} \alpha_{\min}) \tag{12.53}$$

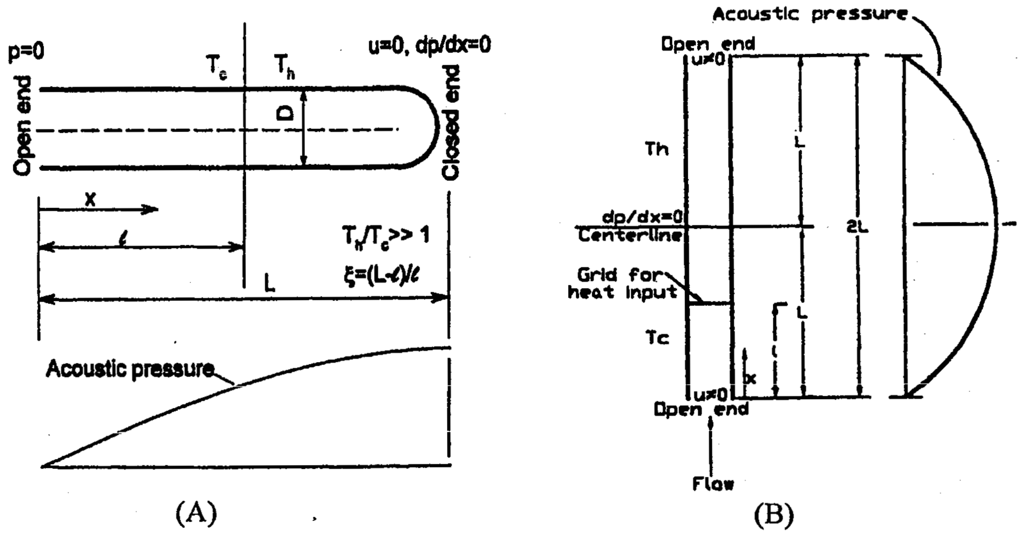



Figure 12.16 Thermoacoustic systems: (A) Sondhauss tube; (B) Rijke tube (Eisinger, 1999)

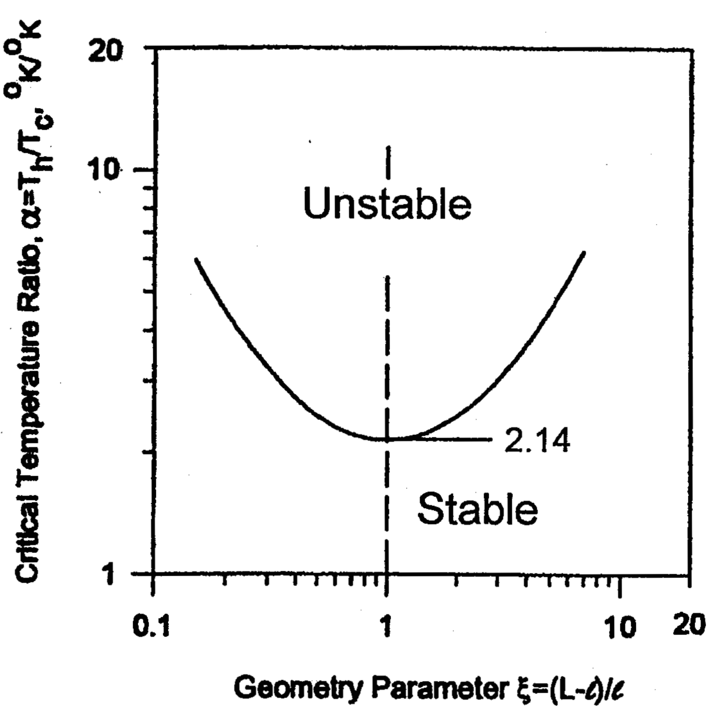



Figure 12.17 Stability diagram for thermoacoustic systems. (Eisinger, 1999; note temperatures must be in deg. Kelvin)

Figure 12.17 is a plot of Equation (12.53). From the diagram, it can be seen that:

- Decreasing the temperature ratio suppresses thermoacoustic vibration.
- Decreasing the length of the "hot" column (by, e.g., moving the furnace grid or swirler) or increasing the length of the cold column suppresses thermoacoustic vibration.

The stability diagram also shows that the worst situation occurs when the swirler or grid is placed at the mid-point of a Sondhauss-type (closed-open) system, or at the quarter point of a Rijke-type (open-open) system. When the condition of instability is established, the gas column will vibrate spontaneously with acoustic frequencies given by Equation (12.12) for the Rijke tube, and Equation (12.16) for the Sondhauss tube.

## 12.9 Suppression of Acoustic Noise

Noise and acoustic forces can excite structural components, causing sonic fatigue failure. Very often, the noise level itself requires corrective action, even if it does not compromise the structural integrity of any components. According to the US Occupational Safety and Health Adminstration (OSHA) Guide, if the ambient noise level exceeds 105 dB (see Chapter 2 for definition of dB), ear protection is required for long-term (more than one hour) exposure to the noise. Even if the noise is below this level, complains from the neighbors or just the feeling that something is not right often requires corrective action. As discussed in Chapter 2, due to the logarithmic nature of the human ear, a tenfold decrease in acoustic energy is often perceived as "half the loudness." Thus, once the condition of excessive noise is established, often it is very difficult and expensive to correct. Post-construction correction to noise problems centers around the following methods:

- Suppression by filtering. This may involve something as simple as enclosing the noise source with sound-absorbing material or protecting the ears with ear plugs or ear cups. This method has limited success in suppressing high-frequency noise such as that generated by cavitation or turbulent shear flow. Low-frequency noises from acoustic resonance are much more difficult to attenuate. Enclosing a heat exchanger suffering from acoustic resonance usually cannot solve the problem. Because human beings "hear" low-frequency noise through their body (face, chest, abdomen and back) rather than through their ears, even ear protection may not help.
- Suppression by energy dissipation. A prime example of this is the automobile muffler. The Helmholtz resonator, because of its energy dissipation ability, is often employed to muffle acoustic noise. This has limited success if the noise level is not excessive and if the component that generates the noise is relatively small. Due to the logarithmic nature of the human ears, at least 90% of the acoustic energy has to