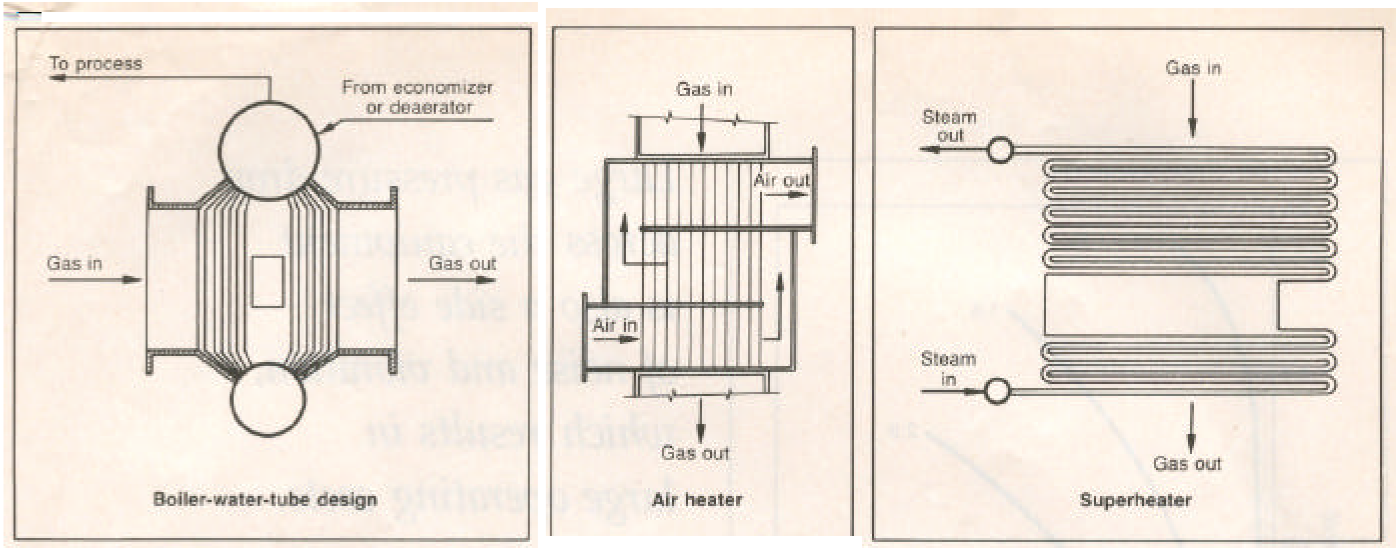

Fig. 1—Cross flow of gas in heat transfer equipment.

Avoid heat transfer equipment

Plant heat transfer modifications and additions can change equipment dynamics. Use these design checks to predict and reduce vibration and noise problems

V Ganapathy, ABCO Industries, Inc., Abilene, Texas

TUBE BUNDLES in heat exchangers, boilers, superheaters and heaters are often subject to vibration and noise problems. Vibration can lead to tube thinning and wear, resulting in tube failures. Excessive noise can be a problem to plant operating personnel. Large gas pressure drop across the equipment is also a side effect, which results in large operating costs. [1,2] With the design checks presented here, one can predict during design if problems associated with noise and vibration are likely to occur.

Vibration causes. Vibration and noise problems are caused when air or flue gases flow over tube bundles, which may be arranged inline or staggered (Fig. 1). Vortices are formed and shed beyond the wake of the tubes, resulting in harmonically varying forces on the tubes perpendicular to the flow direction. It is a self-excited vibration. If the frequency of vibration of the Von-Karman vortices, as they are called, coincide with the natural frequency of vibration of the tube bank, resonance occurs which leads to tube vibration.

Another phenomenon that occurs with vortex shedding is acoustic vibration, leading to noise and high gas pressure drop. The duct or the bundle enclosure vibrates when the acoustic oscillation frequency coincides with the vortex shedding frequency.' The acoustic oscillation is normal to both the direction of gas flow and tube length.

Design methods to check vibration and noise. The first step in the analysis for possible vibration or noise is the estimation of the vortex shedding frequency, fe Vortex shedding is prevalent in the Reynolds number range of 300 to 100,000, which is the operating range of many boilers, heaters and exchangers. The vortex shedding frequency may be estimated once the Strouhal number, S, is known which is given by the expression:

$$S = fe\, d/(12V) \qquad (1)$$

Here d is the tube outer diameter, V is the average gas velocity and S is a function of tube geometry. Figs. 2 to 5 give typical values of S.

The natural frequency of vibration of the tubes is then determined. For a uniform beam supported at each end, fn is given by the expression [8]:

$$fn = C(EI/M_eL^4)^{0.5}/2\pi \qquad (2)$$

C is a constant depending on end conditions and is given in Table 1. The tube length in feet is l and Me is the total weight of the tube, which includes the contribution of the fluid weight inside and outside the tubes. For carbon steel tubes the above equation may be simplified and written as [8]:

$$fn = 90C[(d^4 - d_i^4)/M_e]^{0.5}/l^2 \qquad (3)$$

The next step is estimation of acoustic frequency, f_a

$$fa = Vs/l \qquad (4)$$

Vs is the sonic velocity of the gas and λ is the wave length. $\lambda = 2w/n$ where w is the width of the duct in feet and n is the mode of vibration. For air or flue gases, Vs is approximately $49\sqrt{T}$ where T is the gas temperature in degrees R. For a cylindrical duct

$$f_a = NV s/D \qquad (5)$$

N is a constant - 0.5681 for mode 1, 0.9722 for mode 2 and 1.337 for mode 3.

Checks and analysis for vibration and noise. To analyze for possible vibration or noise in the tube bundles caused by flow of gases across tube banks, the following calculations are performed:

1. Calculate fn for different modes and load conditions. Compute fe. If fn and fe are within 20% of each other, vibration is almost certain to occur.
2. Estimate fa at different loads. Compare fa with fe. If

Large gas pressure drop across the equipment is also a side effect

o f noise and vibration, which

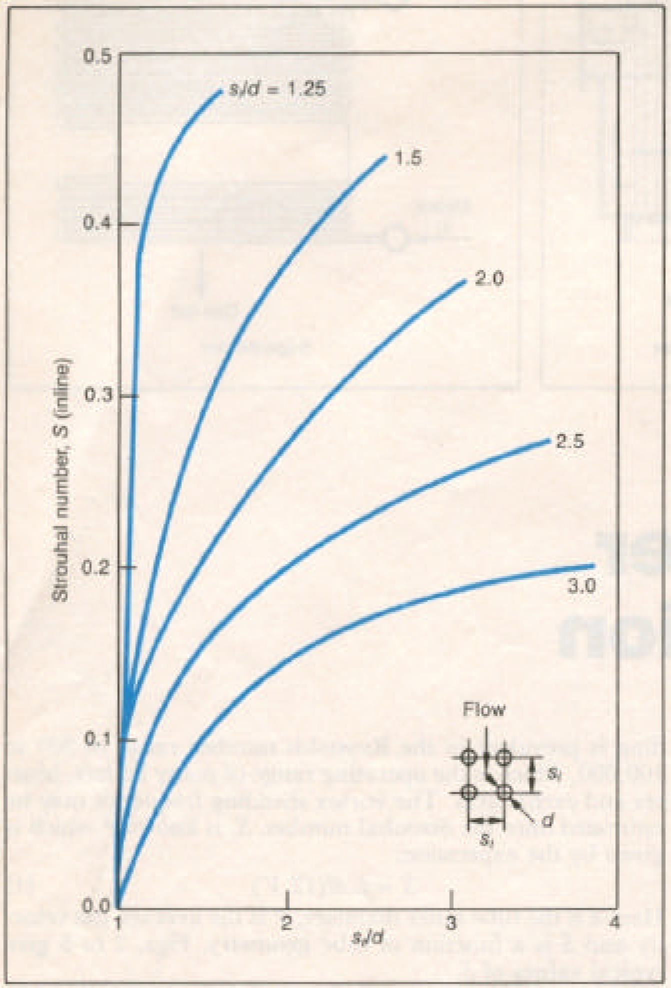

Fig. 2—Strouhal number for inline bank of tubes (Chen).[1]

TABLE 1-Values of C

End support conditions	Mode of 1	2	3
Both ends clamped	22.37	61.67	120.9
One clamped, one hinged	15.42	49.97	104.2
Both hinged	9.87	39.48	88.8

TABLE 2-Summary of results

n	1	2	3
fn	33.1	91	179
fe	54	54	54
fa (no baffles)	56.1	112.2	168.3
fa 1 baffle)	112.2	224	336.6
fa (2 baffles)	168.3	336	504

they are within 20% of each other, excessive noise is likely.[4,5,6] The first mode of vibration is the most critical one as the amplitude of vibrations is large.

Eliminating noise and vibration problems. By changing the tube span, tube pitch, or end conditions, the natural frequency may be altered keeping„ and fe apart to avoid vibration problems. Gas velocity can also be changed so that fe is altered. This may be done by changing the tube length and number of tubes wide.

Primary correction devices for noise are baffles. [5,7] Baffles

divide the gas column into smaller channels or ducts and thereby increase the acoustic frequency, moving it away from the vortex shedding frequency. If the gas temperature is high, the materials for baffles must be chosen with care. Acoustic vibrations usually lie in the range of 40 to 100 Hz.

Example problem. A tubular air heater 11.7 ft wide, 17.5 ft deep and 10 ft long is used in a plant. Carbon steel tubes of 2 in. OD and 0.08 in. thick are arranged inline with a transverse and longitudinal pitch of 3.5 in. The bundle is 40 tubes wide and 60 tubes deep.

Air flows over the tubes, while flue gas flows inside. Air flow is 300,000 lb/h at an average temperature of 260°F. The tubes are fixed at each end in tube sheets.

Analyze the bundle for possible noise and vibration problems.

Solution: Estimate fe. For st/d = s_1/d = 3.5/2 = 1.75, from Fig. 3, S = 0.3. From Fig. 5, we see that S = 0.31. Calculate the air velocity, V. Air density = 0.081(492)/(460 + 260) = 0.055 lb/ft.' V = 300,000(12)/[366(0.055)40 (3.5 - 2)10] = 30 ft/s. Hence fe = 12SV/d = 12(30)0.30/ 2 = 54 Hz.

Estimate fn using Eq. 3. l = 10, d = 2, di = 1.84, M = 1.67 lb/ft = M_e (neglecting weight of air/gas). For the first three modes, C_1 = 22.37, C_2 = 61.67 and C_3 = 120.9, from Table 1.

Then, fn1 = 33.1,fn2 = 91 and fn3 = 179 Hz, using Eq. 3. Let us compute the acoustic frequencies, fa. Sonic velocity, Vs = 49(460 + 260)°-' = 1,315 ft/s. Width, w = 11.7 ft and X = 2(11.7)/n, fa1 = Vs/λ = 56.1, fa2 = 112.2, f a3 = 168 Hz. The summary of results is shown in Table 2, which also shows the fa data with one and two baffles (w being 11.7/ 2 = 5.85 ft and 11.7/3 = 3.9 ft).

Note that fa and fe are very close to each other in the very first mode. Hence, acoustic vibration leading to noise is likely. If one baffle is used, fa and fe are kept well apart in all the modes. Also, fa and fn are well apart in all modes, and tube vibrations are unlikely.

Conclusion. The above calculations show how one can check a tube bundle design for possible vibration or noise problem. A simple approach was discussed. For elaborate analysis, one would use the methods discussed in literature.-`'

However, noise and vibration problems are better predicted based on field operating experience of similar sized units. Performing the above calculations and modifying a design to keep the forcing frequencies well apart may not avoid noise/vibrations in all cases, as vibration and noise phenomenon are inexplicable at times.

Damping effect of finned tubes, presence of' ash in flue

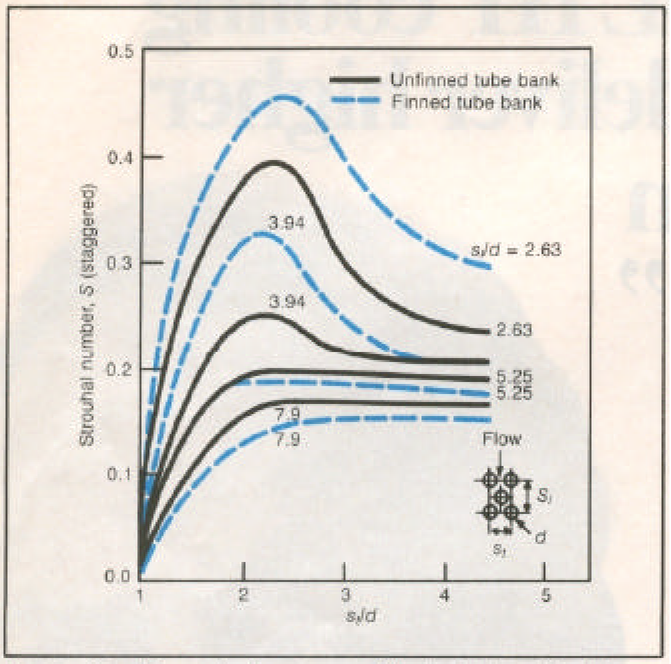

Fig. 3—Strouhal number for staggered bank of tubes (Chen).[1]

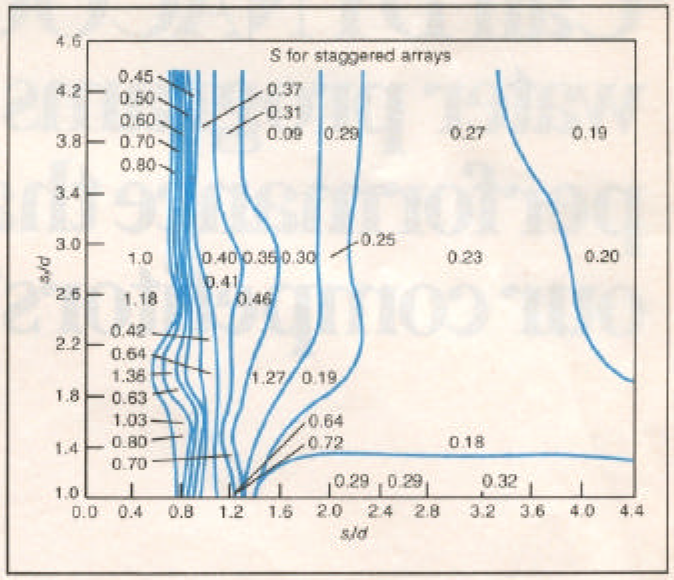

Fig. 4—Strouhal number for staggered bank of tubes (Fitzhugh).[3]

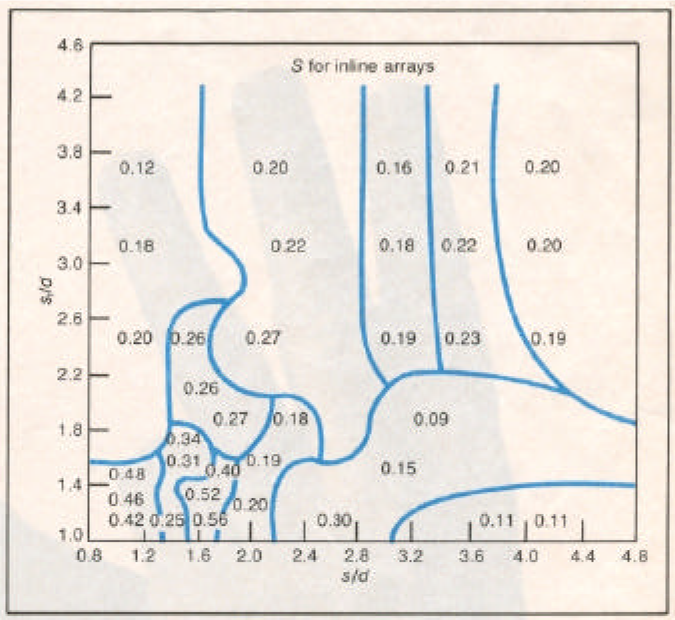

Fig. 5—Strouhal number for inline bank of tubes (Fitzhugh).[3]

gases, manufacturing **tolerances** *used and effect of end con-*nections *are variables that cannot be quantified.* Hence, field experience coupled *with analysis* would *be the ideal way to deal with the problem* of *noise and vibration.*

NOMENCLATURE

- C Constant used in Eq. 3
- d Tube outer diameter, in.
- d_i Tube inner diameter, in.
- E Youngs modulus of elasticity, psi
- f_a Acoustic frequency, hertz
- f_a Vortex shedding frequency, hertz
- f_n Natural frequency of vibration of tubes, hertz
- I Moment of inertia of tube
- L Tube length, ft
- M Total weight of tube per foot, lb
- n Mode of vibration
- S Strouhal number
- s_l Longitudinal pitch, in.
- s_t Transverse pitch, in.
- T Gas temperature, R
- V Gas velocity, ft/s
- V Sonic velocity, ft/s
- w Width of duct, ft
- λ Wave length, ft

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The author

V. Ganapathy is a heat transfer specialist with ABCO Industries Inc., Abilene, Texas. He is engaged in the engineering of heat recovery boilers *for process, incineration and cogeneration applications. He also develops software for engineering of heat recovery systems and components. He holds a B Tech degree in mechanical engineering from Indian* [...] *nology from Madras University. Mr. Ganapathy is the author of over 125 articles on boilers, heat transfer and steam plant systems and has written four books: Applied Heat Transfer, Steam Plant Calculations Manual, Nomograms for Steam Generation and Utilization and Basic Programs for Steam Plant* **Engineers** *(book and diskette), copies of which are available from him. He also has contributed several chapters to the* **Encyclopedia** *of Chemical Processing and Design, Vol. 25, Marcel Dekker New York.*