



Flow induced acoustic resonance in tubular heat exchangers

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FLOW INDUCED ACOUSTIC RESONANCE IN TUBULAR HEAT EXCHANGERS

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FLOW INDUCED ACOUSTIC RESONANCE IN TUBULAR HEAT EXCHANGERS

1. NOTATION AND UNITS

		Units	
		SI	British
c	velocity of sound (Equation (4.5))	m/s	ft/s
D	inside diameter of shell of circular duct	m	ft
d	outside diameter of tube	m	ft
f	frequency	Hz	Hz
g_c	constant of proportionality in Newton's second law of motion	– (1.0)	lb ft/s ² lbf (32.18)
L	rectangular duct dimension	m	ft
M	Mach number, ratio of flow velocity to local speed of sound in fluid medium	–	–
MW	relative molecular mass (molecular weight)	kg/kmol	lb/lbmol
n, m, p	rectangular duct mode numbers for transverse, tube axis and gas flow directions respectively (Equation (4.1))	–	–
P	pressure	N/m ² (Pa)	lbf/ft ²
p	tube pitch (Sketch 4.3)	m	ft
p_l	longitudinal tube pitch (Sketch 5.1)	m	ft
p_t	transverse tube pitch (Sketch 5.1)	m	ft
R	universal gas constant	J/kg mol K (8314.3)	ft lbf/lb mol°R (1545)
Re	Reynolds number = $\rho Vd/\eta$	–	–
S	Strouhal number = $f d/V$	–	–
T	absolute temperature	K	°R
V	flow velocity based on straight line gap or transverse flow area (Sketch 5.1)	m/s	ft/s
X_l	longitudinal tube pitch ratio, p_l/d (Sketch 5.1)	–	–

X_t	transverse tube pitch ratio, p_t/d (Sketch 5.1)	—	—
x_g	vapour mass fraction (mass flow rate of vapour/total mass flow rate)	—	—
Z	gas compression factor	—	—
γ	ratio of gas specific heat capacities	—	—
Δ^*	Fitzpatrick's acoustic damping parameter, defined in Equation (6.3)	—	—
η	dynamic viscosity of gas	kg/m s	lb/ft s
λ	mode constant (Equation (4.3))	—	—
ρ	density	kg/m ³	lb/ft ³
ρ_g, ρ_l	gas and liquid phase densities respectively in a two-phase vapour-liquid flow	kg/m ³	lb/ft ³
σ	bundle solidity (Sketch (4.3))	—	—
ψ	Chen's acoustic damping parameter, defined in Equation (6.2)	—	—

Extra subscripts

a	acoustic
e	excitation
eff	effective
g	gas or vapour phase
tp	two-phase
x, y, z	rectangular duct dimensions in transverse, tube axis and gas flow directions respectively
α	number of diametral pressure nodes in circular duct
β	number of concentric circular pressure nodes in circular duct

2. PURPOSE AND SCOPE

Noise may occur due to the flow of a gas over a heat exchanger tube bundle. Under resonant conditions noise levels in excess of 160 dB have been reported^{18, 20}. In some circumstances this acoustic resonance may also result in structural damage to various components such as the tubes, baffles, bundle containment, pipework and ducting, supports and foundations. Additionally, the gas pressure drop is increased.

High noise levels may affect the health and safety of plant personnel and generate complaints if the plant is situated near residential areas.

ESDU 87019²¹ was primarily concerned with tube vibration as a result of fluid/structural interaction. This Item is concerned with the fluid/acoustic interaction. Its purpose is fivefold:

- (i) to predict the necessary conditions for the onset of acoustic resonance,
- (ii) to predict whether the conditions present are sufficient to sustain acoustic resonance,
- (iii) to indicate steps that may be taken at the design stage to eliminate potential acoustic resonance problems,
- (iv) to indicate measurements necessary to identify and clarify the nature of a problem, should one arise,
- (v) to suggest solutions to problems with installed exchangers.

3. THE CHARACTERISTICS OF ACOUSTIC RESONANCE IN HEAT EXCHANGER TUBE BUNDLES

The flow field between the tubes of a heat exchanger is made up from steady (mean) components and fluctuating components. The fluctuating components comprise:

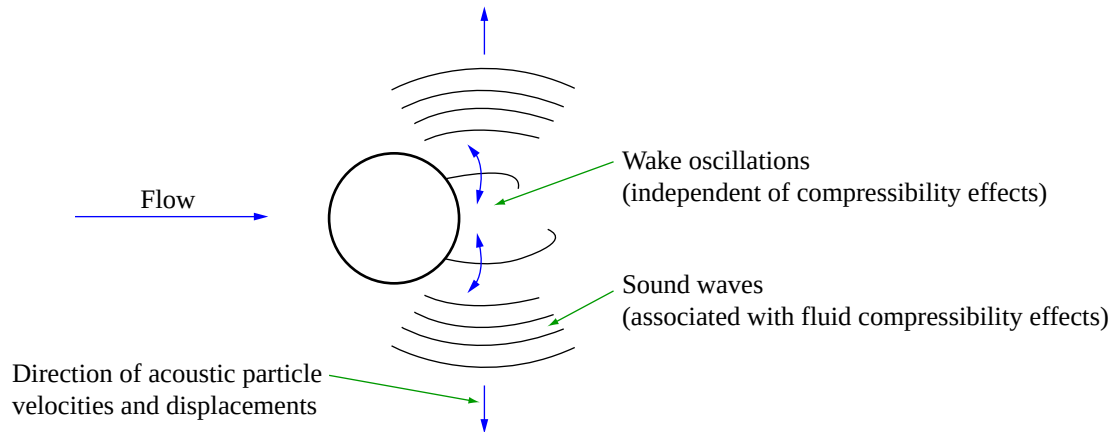
- (i) random turbulence which is convected by the mean flow,
- (ii) periodic oscillations due to the motion of the wakes behind the tubes,
- (iii) driven oscillations due to the motion of the tubes.

The interactions between these fluctuating components of the flow and the tubes have been described in ESDU 87019²¹.

The fluctuating components of the flow can also excite acoustic oscillations in the fluid. These acoustic oscillations are associated with the compressibility of the fluid and constitute sound waves. All fluids are compressible, to some extent. However, gases are much more compressible than liquids and, as a consequence, are more readily set into acoustic oscillation by fluctuations in the flow field. These acoustic effects are the subject of this Data Item. The sound waves are not significantly attenuated by the flow or the presence of the tubes. Under resonant conditions a standing wave is formed in which relatively small fluctuations in the flow may cause significant noise.

A standing wave results from an outgoing wave being reflected back on itself by the walls of the containment and returning to the location of the excitation with a phase relationship which augments the pressure field. This reflection occurs directly in the case of parallel walls, but by multiple reflections in all other cases.

Oscillations in the wake are the usual cause of acoustic resonance in heat exchangers. A simplified representation of the interaction between the wake and the sound field is shown in Sketch 3.1.



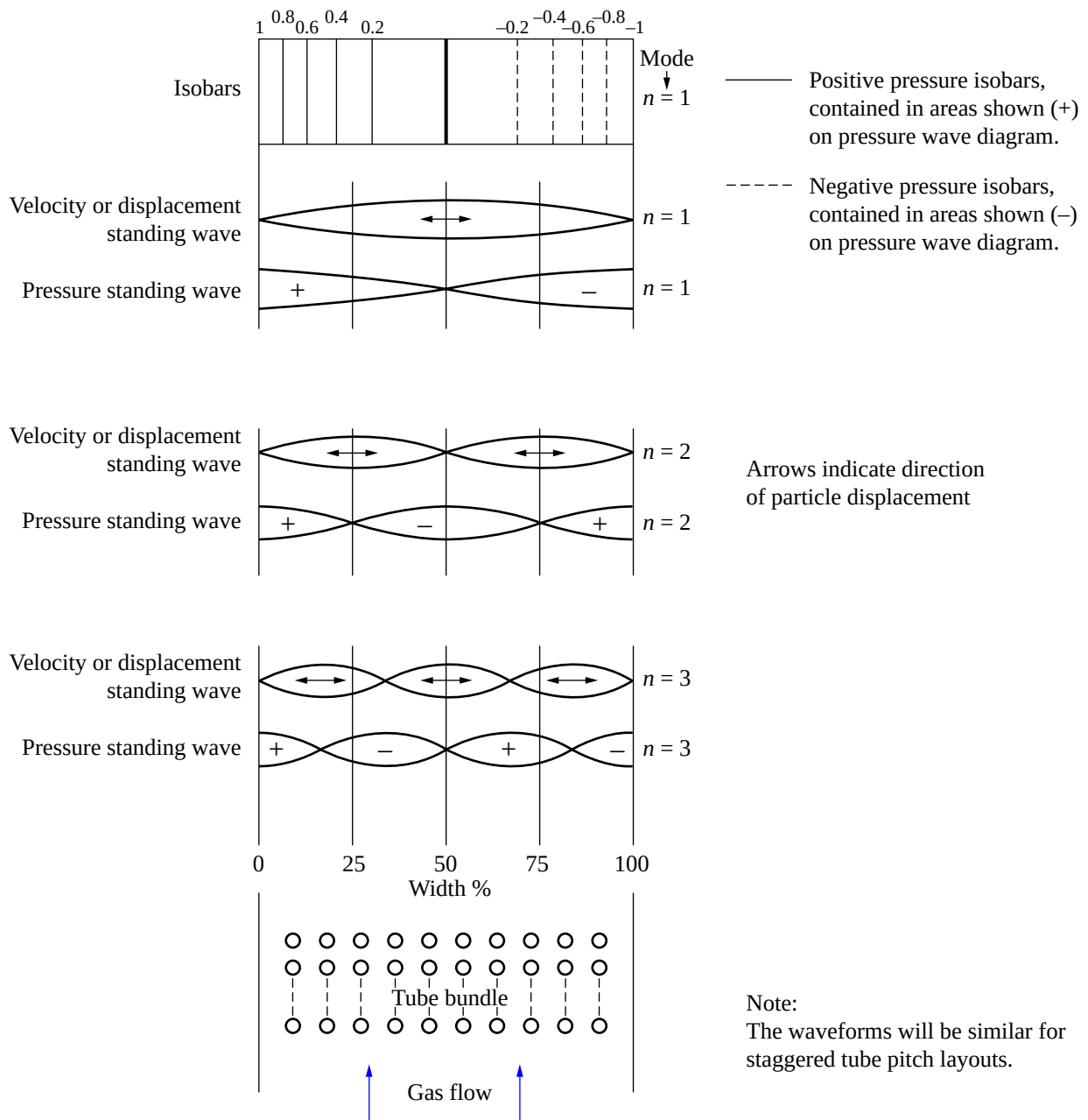
Sketch 3.1 Diagrammatic representation of relationship between flow and generated sound waves

The fluid-acoustic interaction has the following characteristics.

- (i) The tube axis, flow direction and direction of propagation of the standing wave are mutually perpendicular; this will be called the perpendicular relationship.
- (ii) The mechanism does not require tube motion; however, if tube motion does occur, the interaction between the sound field and the wake may be enhanced, resulting in higher sound pressure levels and amplified tube motion.
- (iii) The sound field may, under certain circumstances, control the wake excitations (see Section 5).

The acoustic standing wave may be represented in terms of particle motion or the associated fluid pressure. Pressure nodes are located at displacement antinodes and vice-versa. Typical waveforms for flow over a bundle in a rectangular duct are shown in Sketch 3.2. Here, pressure isobars and particle displacements are illustrated for the fundamental, second and third modes*. The isobars have been normalised to unity at the maximum pressure location.

* In the literature the terms 'mode', 'harmonic' and 'overtone' will be found. In this Data Item 'mode' is used throughout. Harmonics and overtones strictly apply only when their frequencies are integer multiples of the fundamental frequency. In the case of rectangular ducts the modes are harmonically related and the first, second and third modes correspond respectively to the fundamental frequency and second and third harmonics whilst the first and second overtones correspond to the second and third modes respectively. In ducts of circular cross-section this is not the case and 'modes' are not synonymous with 'harmonics' and 'overtones'.



Sketch 3.2 Typical standing waveforms for flow in rectangular ducts

In shell-and-tube heat exchangers, the cylindrical shell gives rise to more complicated mode shapes. These mode shapes are discussed in Sections 4.1 and 9.1.2 and shown in Sketch 4.2.

4. PREDICTION OF ACOUSTIC FREQUENCY

4.1 Calculation for a Vessel with no Internals

The acoustic natural frequency of a rectangular cavity is given¹ by the expression

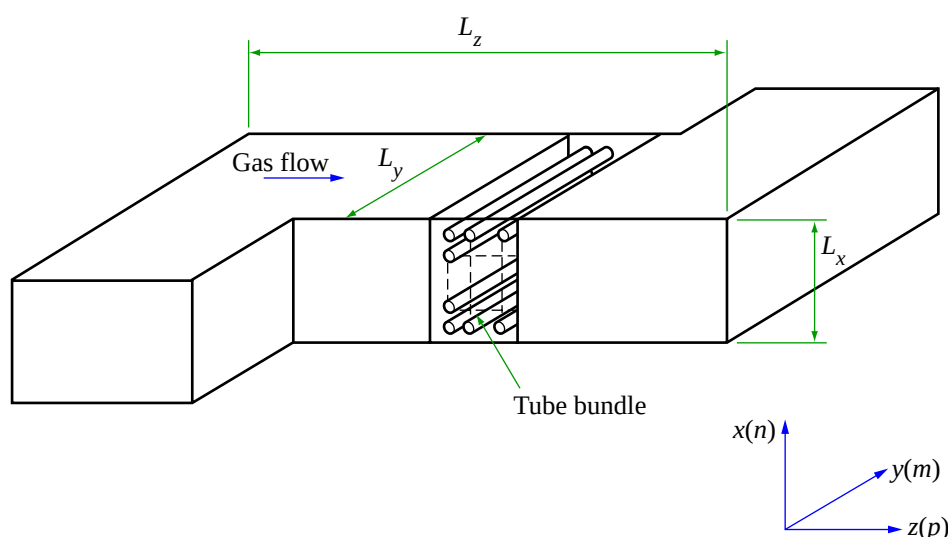
$$f_{nmp} = \frac{c}{2} \left[\left(\frac{n}{L_x} \right)^2 + \left(\frac{m}{L_y} \right)^2 + \left(\frac{p}{L_z} \right)^2 \right]^{1/2}, \quad (4.1)$$

where c = velocity of sound in the gas,

n, m, p = mode numbers for the transverse, tube axis and gas flow, or x -, y - and z -directions,

L_x, L_y, L_z = duct dimensions in the transverse, tube axis and gas flow directions respectively.

The geometry is illustrated in Sketch 4.1.



Sketch 4.1 Notation for a rectangular duct

The mode numbers, n , m and p , take integer values. Normally, standing waves do not occur with an axial component and p can be set to zero. In the great majority of cases the situation is further simplified in that it is unusual for the m modes to be excited. Equation (4.1) therefore reduces to

$$f_n = \frac{nc}{2L_x}. \quad (4.2)$$

The expression corresponding to Equation (4.2) for a cylindrical duct, and applicable to shell-and-tube heat exchangers, where the axial component has been ignored, is¹

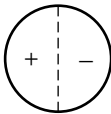
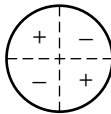
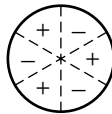
$$f_{\alpha\beta} = \frac{\lambda_{\alpha\beta} c}{\pi D}, \quad (4.3)$$

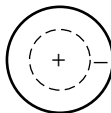
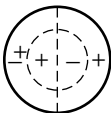
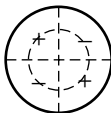
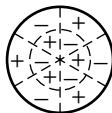
where D = internal diameter of the duct ,
 $\lambda_{\alpha\beta}$ = mode constant ,

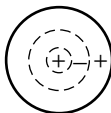
subscript α = number of diametral pressure nodes (lines of zero pressure),
subscript β = number of concentric circular pressure nodes.

Values of the mode constant, $\lambda_{\alpha\beta}$, along with the corresponding nodal lines, are presented in Table 4.1.

TABLE 4.1 Mode constants and nodal line notation for acoustic modes in circular ducts

			
Mode (α, β)	1, 0	2, 0	3, 0
Mode order	1st	2nd	4th
$\lambda_{\alpha\beta}$	1.84	3.05	4.20

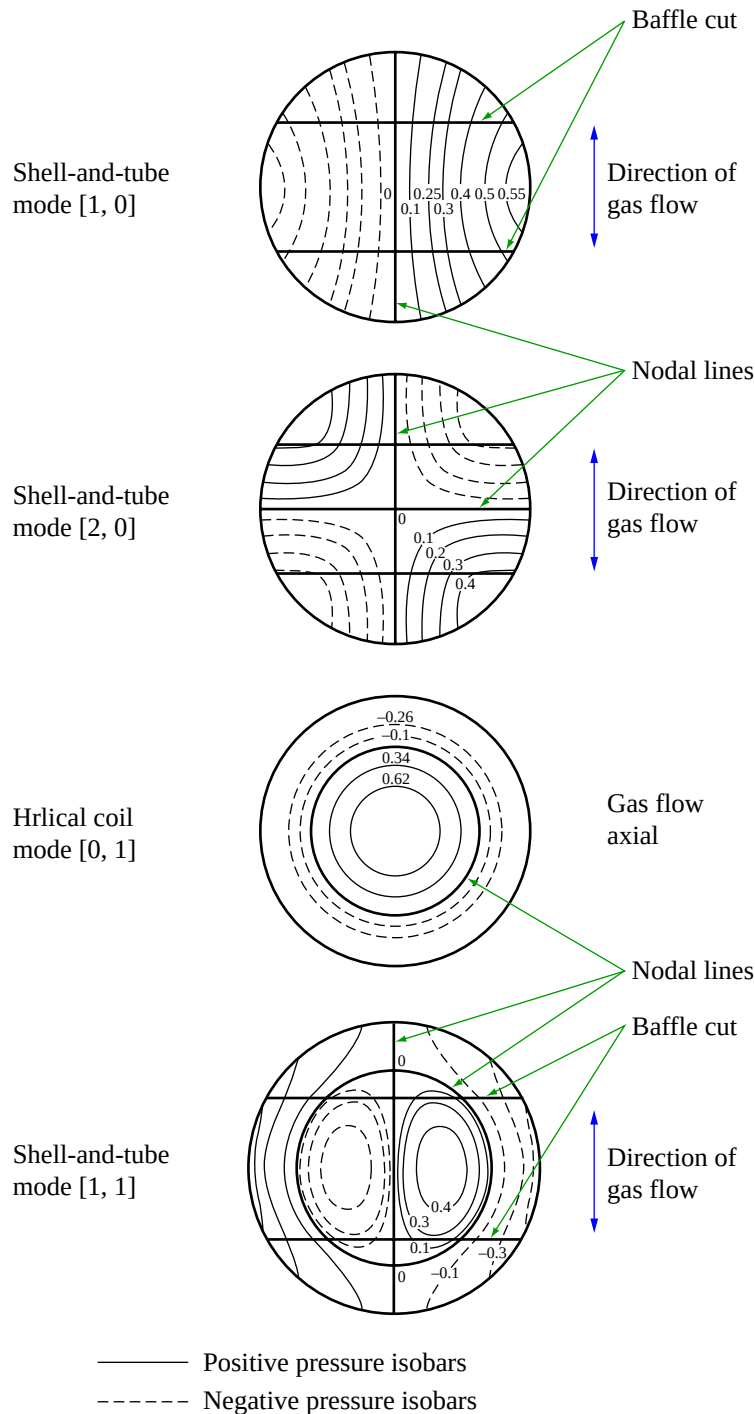
				
Mode (α, β)	0, 1	1, 1	2, 1	3, 1
Mode order	3rd	5th	6th	8th
$\lambda_{\alpha\beta}$	3.83	5.33	6.71	8.02

				
Mode (α, β)	0, 2			
Mode order	7th			
$\lambda_{\alpha\beta}$	7.02			
α = number of diametral pressure nodes. β = number of concentric circular pressure nodes. Dashed lines --- are the nodal lines.				

Note: (i) for $\alpha = 0$ the modes are purely radial, for $\beta = 0$ they are purely diametral
(ii) mode order is the **order** in which modes occur in terms of natural frequency.

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The modes of importance in shell-and-tube exchangers are $[1, 0]^*$, $[2, 0]$ and, possibly, $[1, 1]$, where the flow at right angles to both the tube axes and the direction of propagation of the standing wave occurs over a large percentage of the tubes. Mode $[0, 1]$ is of major importance with helical coil arrangements. Sketch 4.2 shows the pressure distribution as isobars together with the pressure nodal lines for these modes. As in rectangular ducts, the velocity antinodes are at the location of the pressure nodes. Acoustic velocity vectors are always normal to the pressure isobars.



Sketch 4.2 Instantaneous isobars on plane normal to shell axis

* The fundamental, or $[1, 0]$, mode involves particle motion from one side of the diametral node to the other, resembling liquid 'sloshing' in a barrel. For this reason it is sometimes called the sloshing mode. Similarly, mode $[0, 1]$ is sometimes called the 'breathing' mode.

In the case of baffled shell-and-tube heat exchangers, three-dimensional acoustic standing waves, involving significant reflection from the baffles, are unlikely because (i) a substantial proportion of the wave will propagate through the baffle window and (ii), the perpendicular relationship will not occur over a large portion of the tube bank.

4.2 Calculation of Acoustic Velocity in a Vessel with no Internals

The velocity of sound in a gas, c , is given by

$$c = \sqrt{\frac{Z \gamma R g_c T}{MW}} = \sqrt{\frac{\gamma g_c P}{\rho}}. \quad (4.4)$$

Values of γ may be obtained from References 15 and 17 and values of Z from the Nelson-Obert generalised compressibility chart presented in References 10 and 17 and elsewhere. Equation (4.4) is not applicable to liquids which are relatively incompressible. The speed of sound in a liquid is very high and consequently acoustic resonances are rarely excited by periodic wake shedding.

Due to the variation in temperature within a vessel the velocity of sound will vary from location to location. Consequently the acoustic natural frequency will be different in different regions. In addition, the range of operating conditions, with associated temperature profiles, may result in a range of acoustic natural frequencies.

From Equation (4.4) it is apparent that the speed of sound in a particular gas at low and moderate pressures is dependent solely on its temperature. Only when the gas compression factor, Z , departs appreciably from unity at high pressures is this not so.

4.3 Correction for the Presence of Tubes

The speed of sound in a vessel is lowered when internals, such as tubes, are present. Parker¹³, from theoretical considerations, has proposed the expression

$$\frac{c_{eff}}{c} = \frac{1}{(1 + \sigma)^{1/2}}, \quad (4.5)$$

where c_{eff} = effective speed of sound with tubes present,

σ = bundle solidity or the fraction of the bundle volume occupied by the tubes.

Parker also found Equation (4.5) was in agreement with his data for 60° triangular tube pitch layouts having p/d ratios of 1.34 ($\sigma = 0.51$) and 1.95 ($\sigma = 0.24$).

Expressions for σ in terms of tube pitch and diameter for common tube pitch layouts are given in Sketch 4.3.