Journal of Fluids and Structures 18 (2003) 469–483

JOURNAL OF FLUIDS AND STRUCTURES

www.elsevier.nl/locate/jnlabr/yjfls

Vibration analysis of shell-and-tube heat exchangers: an overview—Part 1: flow, damping, fluidelastic instability

M.J. Pettigrew[a,*], C.E. Taylor[b]

[a] *BWC/AECL/NSERC Chair of Fluid–Structure Interaction, Department of Mechanical Engineering, École Polytechnique Montréal, Que., Canada H3T 1J4*

[b] *Inspection, Monitoring and Dynamics Branch, Atomic Energy of Canada Ltd., Chalk River Laboratories Chalk River, Ont., Canada K0J 1J0*

Received 12 March 2003; accepted 13 August 2003

Abstract

Design guidelines were developed to prevent tube failures due to excessive flow-induced vibration in shell-and-tube heat exchangers. An overview of vibration analysis procedures and recommended design guidelines is presented in this paper. This paper pertains to liquid, gas and two-phase heat exchangers such as nuclear steam generators, reboilers, coolers, service water heat exchangers, condensers, and moisture-separator-reheaters.

Generally, a heat exchanger vibration analysis consists of the following steps: (i) flow distribution calculations, (ii) dynamic parameter evaluation (i.e. damping, effective tube mass, and dynamic stiffness), (iii) formulation of vibration excitation mechanisms, (iv) vibration response prediction, and (v) resulting damage assessment (i.e., comparison against allowables). The requirements applicable to each step are outlined in this paper. Part 1 of this paper covers flow calculations, dynamic parameters and fluidelastic instability.

1. Introduction

Tube failures due to excessive vibration must be avoided in heat exchangers and nuclear steam generators, preferably at the design stage. Thus, a comprehensive flow-induced vibration analysis is required before fabrication of shell-and-tube heat exchangers. It must be shown that tube vibration levels are below allowable levels and that unacceptable resonances and fluidelastic instabilities are avoided.

The purpose of this overview paper is to summarize our design guidelines for flow-induced vibration of heat exchangers. The overview can be used by the designer as a guideline for vibration analysis, by the project engineer to get an overall appreciation of flow-induced vibration concerns, or by the plant operator to understand tube failures. This paper pertains to shell-and-tube heat exchangers such as nuclear steam generators, heat exchangers, coolers, condensers and moisture-separator-reheaters (MSR).

1.1. Flow-induced vibration overview

The vibration behaviour of heat exchangers is governed by vibration excitation mechanisms and by damping mechanisms. Generally, in heat exchangers there are several significant damping mechanisms: (i) friction damping

*Corresponding author.
E-mail address: michel.pettigrew@polymtl.ca (M.J. Pettigrew).

0889-9746/$ - see front matter
doi:10.1016/j.jfluidstructs.2003.08.007

between tube and tube-support, (ii) squeeze-film damping at the support, and (iii) viscous damping between tube and shell-side fluid. In nuclear steam generators, damping due to two-phase flow is also important.

Generally, the flow in heat exchanger tube bundles can be parallel (axial flow) or transverse (cross flow) to the tube. In nuclear steam generators, the flow is axial for a large portion of the tube bundle. Vibration excitation forces induced by axial flow are relatively small in heat exchangers. Thus, vibration excitation mechanisms in axial flow may generally be neglected. The vibration behaviour is clearly governed by cross flow vibration excitation mechanisms.

Several vibration excitation mechanisms are normally considered in heat exchanger tube bundles in cross flow, namely: (i) fluidelastic instability, (ii) periodic wake shedding, (iii) random excitation, and (iv) acoustic resonance. Fluidelastic instability is by far the most important mechanism and must be avoided in all cases. Periodic wake shedding resonance may be of concern in liquid cross flow where the flow is relatively uniform. It is not normally a problem at the entrance region of steam generators because the flow is very nonuniform and quite turbulent. Turbulence inhibits periodic wake shedding. Periodic wake shedding is generally not a problem in two-phase flow except at very low void fractions (i.e., $\varepsilon_g < 15\%$). Then, the behaviour is similar to that of liquid flow. Random excitation is important in both liquid and two-phase cross flow. Periodic wake shedding resonance and random excitation are not usually of concern in gas flow since the fluid density is generally low thereby resulting in relatively small excitation forces. However, both mechanisms should be considered in some gas heat exchangers such as MSRs where relatively high fluid densities exist. Acoustic resonance must be avoided in gas heat exchangers. However, it is generally not a problem in liquid and two-phase heat exchangers.

1.2. Scope of a vibration analysis

A heat exchanger vibration analysis consists of the following steps: (i) flow distribution calculations, (ii) dynamic parameter evaluation (i.e., damping, effective tube mass, and dynamic stiffness), (iii) formulation of vibration excitation mechanisms, (iv) vibration response prediction, and (v) resulting damage assessment (i.e., comparison against allowables). The requirements applicable to each step are outlined in this paper. It is divided in two parts: Part 1 covers flow calculations, dynamic parameters and fluidelastic instability, and Part 2 (Pettigrew and Taylor, 2003) covers forced vibration excitation mechanisms, vibration response prediction, fretting-wear damage assessment, and acceptance criteria.

2. Flow calculations

Flow-induced vibration problems usually occur on a small number of critical tubes in specific areas of a heat exchanger (e.g., entrance regions, tube free lanes, and U-tubes). Thus, a flow analysis is required to obtain the local flow conditions throughout the heat exchanger.

2.1. Flow parameter definition

The end results of the flow analysis should be the shell-side cross flow velocity, U_p, and fluid density, ρ, distributions along the critical tubes. For flow-induced vibration analyses, the flow velocity is defined in terms of the pitch velocity:

$$U_p = U_\infty P/(P - D), \quad (1)$$

where U_∞ is the free stream velocity (i.e., the velocity that would prevail if the tubes were removed). P is the pitch between the tubes and D is the tube diameter. For finned tubes, the equivalent hydraulic diameter, D_h, is used. The pitch velocity is sometimes called the reference gap velocity. The pitch velocity is a convenient definition since it applies to all bundle configurations.

The situation is somewhat more complex in two-phase flow. Another parameter, steam quality or void fraction, is required to define the flow conditions. Two-phase mixtures are rarely homogeneous or uniform across a flow path. However, it is convenient and simple to use homogeneous two-phase mixture properties as they are well defined. This is done consistently here for both specifying vibration guidelines and formulating vibration mechanisms. The homogeneous two-phase flow model assumes that both liquid and gas phases flow with equal velocity. The homogeneous void fraction, ε_g, is defined in terms of the volume flow rates of gas, $\dot{V}_g$, and liquid, $\dot{V}_\ell$:

$$\varepsilon_g = \frac{\dot{V}_g}{\dot{V}_g + \dot{V}_\ell}. \quad (2)$$

The homogeneous density, ρ, the free stream velocity, U_∞, and the free stream mass flux, $\dot{m}_\infty$, are defined using the homogeneous void fraction:

$$\rho = \rho_\ell(1 - \varepsilon_g) + \rho_g\varepsilon_g, \tag{3}$$

$$U_\infty = \frac{\rho_\ell \dot{V}_\ell + \rho_g \dot{V}_g}{\rho A}, \tag{4}$$

$$\dot{m}_\infty = \rho U_\infty, \tag{5}$$

where A is the free-stream flow path area.

For both liquid and two-phase cross flow, the pitch velocity, U_p, is defined by Eq. (1) and, thus, the pitch mass flux, $\dot{m}_p$, is similarly defined:

$$\dot{m}_p = \dot{m}_\infty \frac{P}{(P - D)} = \rho U_p. \tag{6}$$

2.2. Simple flow path approach

For relatively simple components, where the flow paths are reasonably well defined, a flow path approach may be adequate to calculate flow velocities. This is illustrated in Fig. 1. In the flow path analysis approach, characteristic flow paths (i.e., through the tube bundle, between the tube bundle and the shell, etc.) between regions of common pressures are identified. Flow impedances (i.e., pressure drop coefficients) are estimated. The flows within each path are then calculated. The resulting flow velocity distributions are then used to estimate vibration excitation mechanisms and predict vibration response.

All typical operating conditions must be considered including the following: (i) as-designed operating conditions, from 0% to 100% flow, (ii) operating conditions with fouling of the tubes or crudding of the tube-supports, and (iii) other possible operating conditions (e.g., after chemical cleaning, system testing, etc.).

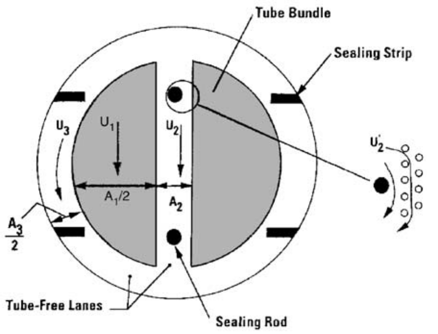
2.3. Comprehensive 3-D approach

For complex components such as nuclear steam generators and power condensers, a comprehensive three-dimensional thermalhydraulic analysis is required. In such analyses, the component is divided into a large number of control volumes. The equations of energy, momentum and continuity are solved for each control volume. This is done with numerical methods using a computer code such as the THIRST code for steam generators (Pietralik, 1995). The numerical grid outlining the control volumes for the analysis of a typical steam generator is shown in Fig. 2. The grid must be sufficiently fine to accurately predict the flow distribution along the tube. Some typical thermalhydraulic analysis results are shown in Fig. 3 for the U-bend region of a steam generator. A hypothetical power condenser tube and detailed flow velocity calculations are displayed in Fig. 4. For flow-induced vibration analyses, the results must be in the form of pitch flow velocity and fluid density distributions along a given tube. These distributions constitute the input to the flow-induced vibration analysis of this particular tube.

2.4. Two-phase flow regime

Some knowledge of flow regime is necessary to assess flow-induced vibration in two-phase flow. Flow regimes are governed by a number of parameters such as surface tension, density of each phase, viscosity of each phase, geometry of flow path, mass flux, void fraction and gravity forces. Flow regime conditions are usually presented in terms of dimensionless parameters in the form of a flow regime map. As yet, very little information is available on flow regimes in tube bundles subjected to two-phase cross flow. Grant (1975), Grant and Chisholm (1977) and, more recently, Ulbrick and Mewes (1994) have contributed most of the available data. The information of Grant and Chisholm (1977) was used to develop the flow regime map shown in Fig. 5. The axes on the Grant flow regime map are defined in terms of a Martinelli parameter, X, and a dimensionless gas velocity, U_g. The Martinelli parameter is formulated as follows:

$$X = \left(\frac{1 - \varepsilon_g}{\varepsilon_g}\right)^{0.9} \left(\frac{\rho_\ell}{\rho_g}\right)^{0.4} \left(\frac{\mu_\ell}{\mu_g}\right)^{0.1}, \tag{7}$$



$$Q = \sum AU = A_1U_1 + A_2U_2 + A_3U_3$$

$$\Delta P = \Delta P_1 = \Delta P_2 = \Delta P_3$$

$$\Delta P = \sum K \rho U^2/2$$

$$\Delta P = \underbrace{K_2 \frac{\rho U_2^2}{2}}_{\text{Tube-Free Lane}} + \underbrace{\frac{K_2' \rho (U_2')^2}{2}}_{\text{Around Sealing Rods}} + \underbrace{\frac{K \rho U_2^2}{2}}_{\text{Turn Around Losses, etc.}}$$

Fig. 1. Flow path approach.

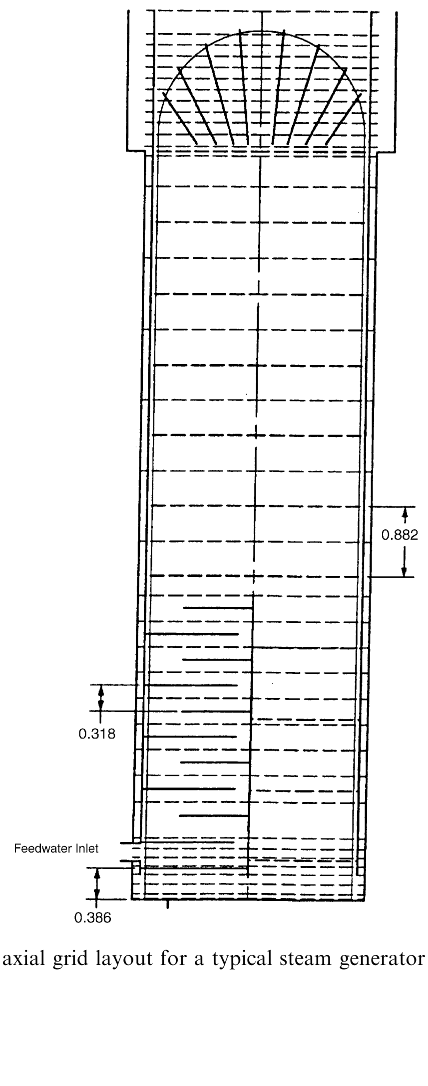


Fig. 2. Thermalhydraulic analysis: axial grid layout for a typical steam generator with preheater; distances in metres.

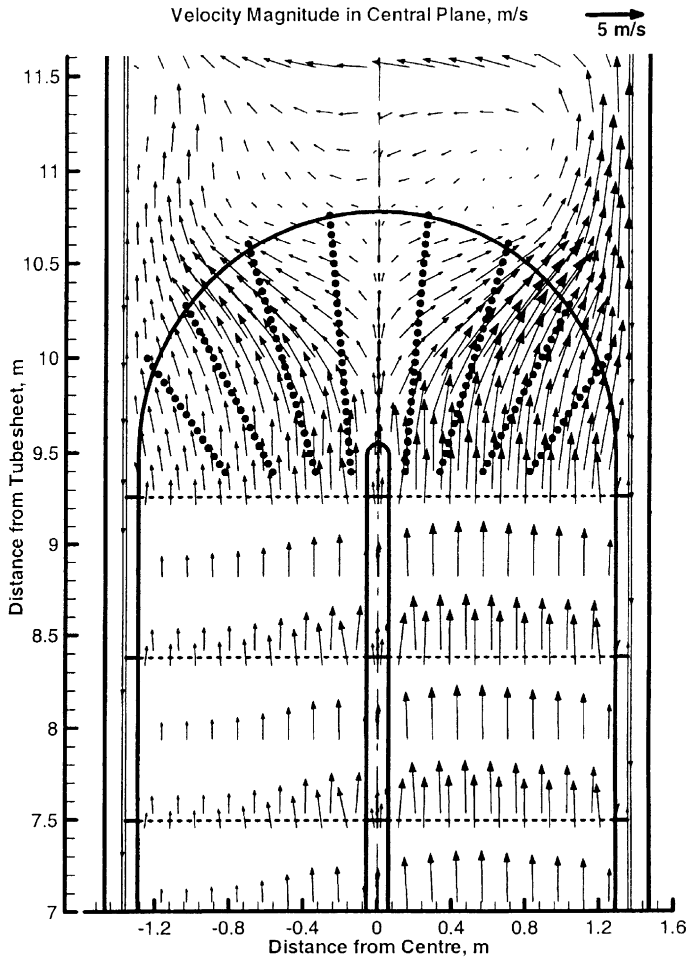
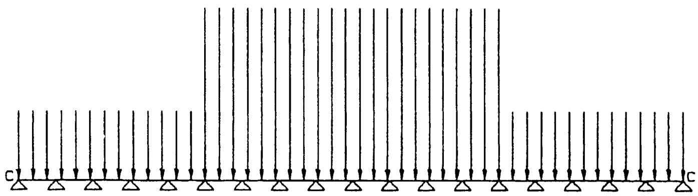


Fig. 3. Flow velocity vectors in the central plane of a typical steam generator U-bend region.



VELOCITY DISTRIBUTION

REGION	START IN. (CM	END IN. CM	VELOCITY FT./S M/S)
1.	0.00	157.50	508.000
	(0.00	400.05	154.838)
2.	157.50	409.50	
	(400.05	1040.13	388.620)
3.	409.50	567.00	508.000
	(1040.13	1440.18	154.838)

E = 15.0E+06 PSI
(10.3E+07 KPA)
OUT. DIAM.= .875 IN.
(2.223 CM)
IN. DIAM.= .819 IN.
(2.080 CM)
DAMPING R= .0070

CL = .100
M/L = .0312 LB./IN.
(.5566 KG/M)
RO = .00 LB./CU.FT.
(.02 KG/M3)
INST. COEFF.= 3.30
PITCH = 1.1250 IN.
(2.8575 CM)

L(1.)= 31.50 IN. (80.01 CM.)
L(2.)= 31.50 IN. (80.01 CM.)
L(3.)= 31.50 IN. (80.01 CM.)
L(4.)= 31.50 IN. (80.01 CM.)
L(5.)= 31.50 IN. (80.01 CM.)
L(6.)= 31.50 IN. (80.01 CM.)
L(7.)= 31.50 IN. (80.01 CM.)
L(8.)= 31.50 IN. (80.01 CM.)

Fig. 4. Gap cross flow distribution along typical condenser tube.

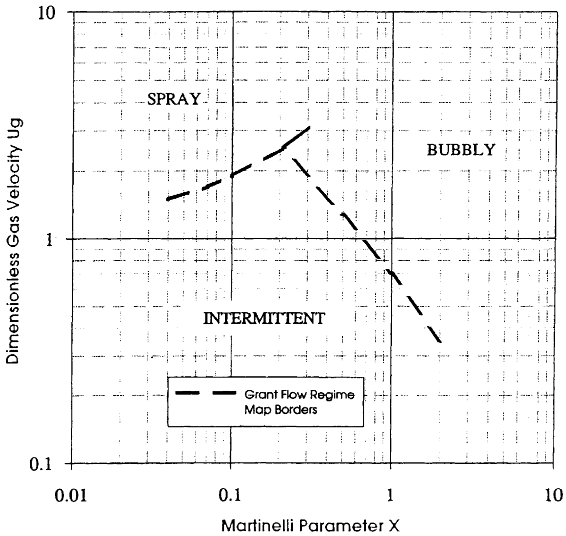


Fig. 5. Grant flow regime map for two-phase cross flow (Pettigrew and Taylor, 1994).

where μ_ℓ and μ_g are the dynamic viscosity of water and gas, respectively. The dimensionless gas velocity is defined as follows:

$$U_g = \frac{\dot{m}_{pg}}{[d_e g \rho_g(\rho_\ell - \rho_g)]^{0.5}}, \tag{8}$$

where $\dot{m}_{pg}$ is the pitch mass flux of the gas phase, $d_e \cong 2(P - D)$ is the equivalent hydraulic diameter and g is the acceleration due to gravity.

The Grant map of Fig. 5 shows three flow regimes: spray, bubbly and intermittent. The terms spray and bubbly are used loosely here. Perhaps they would be more appropriately defined as "continuous flow" covering the whole range from true bubbly flow to wall-type flow to spray flow. Intermittent flow is characterized by periods of flooding (mostly liquid) followed by bursts of mostly gas flow. As discussed by Pettigrew et al. (1989a, 1995, 2002) and Pettigrew and Taylor (1994), this is an undesirable flow regime from a vibration point-of-view. Thus, intermittent flow should be avoided in two-phase heat exchange components and particularly in the U-bend region of steam generators.

3. Dynamic parameters

The relevant dynamics parameters for multispan heat exchanger tubes are mass, flexural rigidity and damping. For finned tubes, the effect of the fins on mass per unit length, m, flexural rigidity, EI, and hydraulic diameter, D_h, of the tube should be evaluated as discussed below.

3.1. Hydraulic diameter

For unfinned tubes the hydraulic diameter is simply the tube outside diameter, D. From a hydraulic viewpoint, the fins may be approximated by an unfinned tube of equivalent diameter, D_h, as explained in Halle et al. (1984) and Mair et al. (1975). The hydraulic diameter is based on the ratio, R_F, of the area occupied by the fins over the available area between the root diameter, D_r, and the outer diameter of the fins, D_o:

$$D_h = D_r + R_F\,(D_o - D_r). \tag{9}$$

This diameter should be used in the calculation of vibration excitation parameters such as Strouhal number, dimensionless flow velocity, random excitation, etc. The equivalent pitch over diameter P/D becomes P/D_h for finned tubes.

3.2. Hydrodynamic mass

The hydrodynamic mass is the equivalent dynamic mass of external fluid vibrating with a tube. In liquid flow, the hydrodynamic mass per unit length of a tube confined within a tube bundle may be expressed by

$$m_h = \left(\frac{\pi}{4}\rho D^2\right)\left[\frac{(D_e/D)^2+1}{(D_e/D)^2-1}\right], \tag{10}$$

where D_e is the equivalent diameter of the surrounding tubes and the ratio D_e/D is a measure of confinement. The effect of confinement is formulated by

$$D_e/D = (0.96 + 0.5P/D)P/D \tag{11}$$

for a tube inside a triangular tube bundle (Rogers et al., 1984). Similarly, for a square tube bundle confinement may be approximated by

$$D_e/D = (1.07 + 0.56P/D)P/D. \tag{12}$$

The hydrodynamic mass of tube bundles subjected to two-phase cross flow may also be calculated with Eq. (10) providing that the homogeneous density of the two-phase mixture, ρ_{TP}, is used in the formulation (Pettigrew et al., 1989a). Fig. 6 compares Eq. (10) to experimental data.

The total dynamic mass of the tube per unit length, m, comprises the hydrodynamic mass per unit length, m_h, the tube mass per unit length, m_t, and the mass per unit length of the fluid inside the tube, m_i:

$$m = m_h + m_t + m_i. \tag{13}$$

3.3. Damping

3.3.1. Heat exchanger tube in gases

As discussed in Pettigrew et al. (1986a) the dominant damping mechanism in heat exchangers with gas on the shell-side is friction between tubes and tube-supports. All the available information on damping of heat exchanger tubes in gases has recently been reviewed. Most of these data pertain to unfinned tubes. This work yielded the following design recommendation for estimating the friction damping ratio, ζ_F, in percent:

$$\zeta_F = 5\left(\frac{N-1}{N}\right)\left(\frac{L}{\ell_m}\right)^{1/2}, \tag{14}$$

which takes into account the effect of support thickness, L, span length, ℓ_m, and number of spans, N.

Hartlen and Barnstaple (1971) reported the only relevant damping data for finned tubes. When their minimum damping values are generalized following the formulation of Eq. (14) we obtain the following:

$$\zeta_F = 4\left(\frac{N-1}{N}\right)\left(\frac{L}{\ell_m}\right)^{1/2}; \tag{15}$$

Eq. (15) is the recommended design guideline for finned tubes.

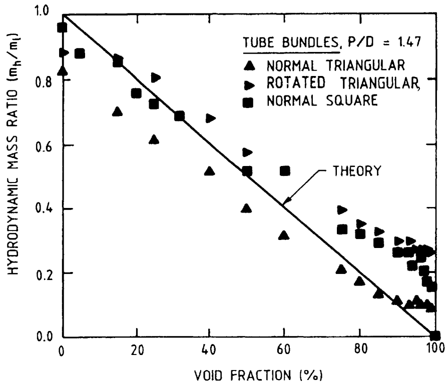


Fig. 6. Hydrodynamic mass in air–water two-phase cross flow: comparison between theory and experiments (Pettigrew et al., 1989a).

3.3.2. Heat exchanger tubes in liquids

As discussed by Pettigrew et al. (1986b), there are three important energy dissipation mechanisms that contribute to damping of multispan heat exchanger tubes with liquids on the shell-side. These are viscous damping between tube and liquid, squeeze-film damping in the clearance between tube and tube-support and friction damping at the support. Thus, the total tube damping, ζ_T, which we define as the ratio of actual to critical damping in per cent, is expressed by

$$\zeta_T = \zeta_V + \zeta_{SF} + \zeta_F, \tag{16}$$

where ζ_V, ζ_{SF} and ζ_F are respectively the viscous, squeeze-film and friction damping ratios.

The tube to fluid viscous damping is related to the Stokes Number, $\pi f D^2/2v$, and to the degree of confinement of the heat exchanger tube within the tube bundle or D/D_e. Rogers et al. (1989) developed a simplified formulation for viscous damping, valid for $\pi f D^2/2v > 3300$ and $D/D_e < 0.5$ which covers most heat exchangers. His simplified expression for ζ_v (in per cent) is

$$\zeta_v = \frac{100\pi}{\sqrt{8}}\left(\frac{\rho D^2}{m}\right)\left(\frac{2v}{\pi f D^2}\right)^{1/2}\left\{\frac{[1+(D/D_e)^3]}{[1-(D/D_e)^2]^2}\right\}, \tag{17}$$

where v is the kinematic viscosity of the fluid and f is the natural frequency of the tube for the mode being analysed. Clearly, viscous damping is frequency dependent. Calculated values of damping using Eq. (17) are compared against experimental data in Fig. 7. It shows reasonable agreement.

Squeeze-film damping, ζ_{SF}, and friction damping, ζ_F, take place at the supports. Based on the available experimental data for total tube damping shown in Fig. 8, semi-empirical expressions were developed, to formulate friction and squeeze-film damping as discussed by Pettigrew et al. (1986b). It shows that squeeze-film damping may be formulated by

$$\zeta_{SF} = \left(\frac{N-1}{N}\right)\left[\frac{1460}{f}\left(\frac{\rho D^2}{m}\right)\left(\frac{L}{\ell_m}\right)^{1/2}\right], \tag{18}$$

and friction damping by

$$\zeta_F = \left(\frac{N-1}{N}\right)\left[0.5\left(\frac{L}{\ell_m}\right)^{1/2}\right], \tag{19}$$

where $(N-1)/N$ takes into account the ratio of the number of supports over the number of spans, L is the support thickness and ℓ_m is a characteristic tube length. The latter is defined as the average of the three longest spans when the lowest modes and the longest spans dominate the vibration response. This is usually the case. When higher modes and shorter spans govern the vibration response, then, the characteristic span length should be based on these shorter spans. This could happen when there are high flow velocities locally such as in entrance or exit regions.

Interestingly, friction damping is roughly ten times less in liquids than in gases (Eq. (19) versus (14)). This may be due to liquid lubrication reducing friction forces and to some friction damping being included in the squeeze-film damping term. In practice, it is very difficult to separate friction and squeeze-film damping terms as explained in (Pettigrew et al., 1986b).

The above formulation for squeeze-film damping is compared against the available experimental data in Fig. 9. There is a large scatter in the data, which is mostly due to the very nature of the problem. Heat exchanger tube damping depends on parameters that are difficult to control, such as support alignment, tube straightness, relative position of tube within the support and side loads. These parameters are statistical in nature and probably contribute to much of the scatter in the damping data. A conservative but realistic criterion for squeeze-film damping is obtained by taking the coefficient in Eq. (18) to be 1460, which roughly corresponds to the lower decile of the available data as shown in Fig. 9.

Substituting Eqs. (17), (18) and (19) in Eq. (16):

$$\zeta_T = \frac{100\pi}{\sqrt{8}}\left\{\frac{[1+(D/D_e)^3]}{[1-(D/D_e)^2]^2}\right\}\left(\frac{\rho D^2}{m}\right)\left(\frac{2v}{\pi f D^2}\right)^{1/2} + \left(\frac{N-1}{N}\right)\left[\frac{1460}{f}\left(\frac{\rho D^2}{m}\right) + 0.5\right]\left(\frac{L}{\ell_m}\right)^{1/2}. \tag{20}$$

Although somewhat speculative, Eq. (20) formulates all important energy dissipation mechanisms and fits the data best. Thus, it is our recommendation as a damping criterion for design purposes. However, if the damping ratio predicted by this equation is less than 0.6%, we recommend taking a minimum value of 0.6%. As shown in Fig. 8, a minimum damping of 0.6% appears reasonable.

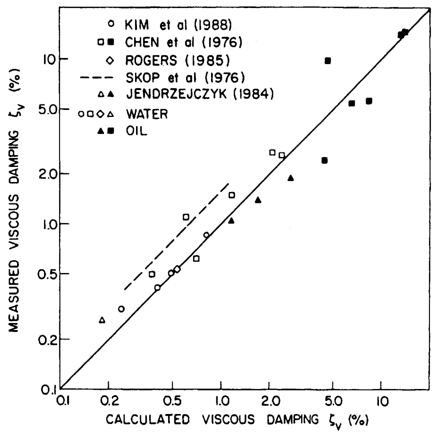


Fig. 7. Viscous damping data for a cylinder in unconfined and confined liquids: comparison between theory and experiment.

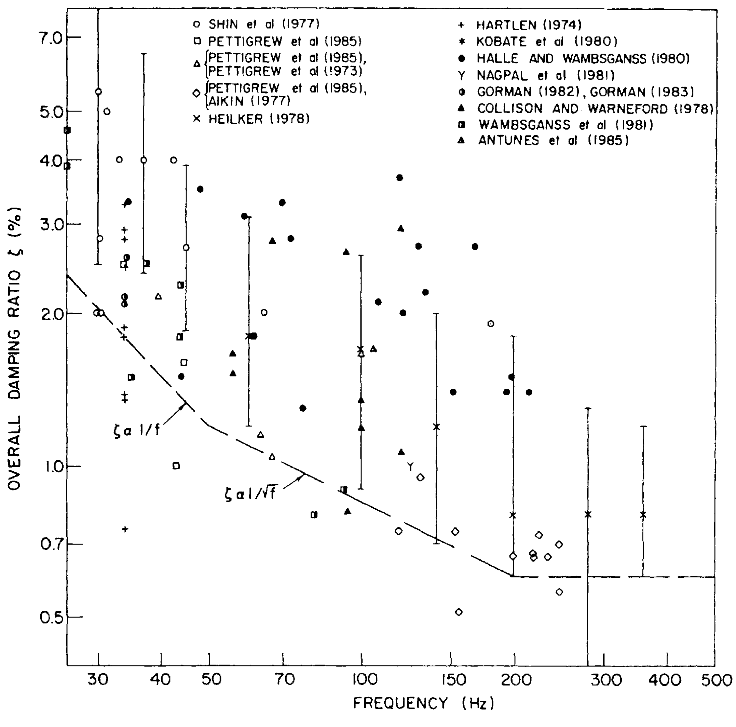


Fig. 8. Damping data for multispan heat exchanger tubes in water (references provided in the legend can be found in (Pettigrew et al., 1986b)).

3.3.3. Damping in two-phase flow

The subject of heat exchanger tube damping in two-phase flow was reviewed recently by Pettigrew and Taylor (1997). The total damping ratio, ζ_T, of a multispan heat exchanger tube in two-phase flow comprises support damping, ζ_S, viscous damping, ζ_V, and two-phase damping, ζ_{TP}:

$$\zeta_T = \zeta_S + \zeta_V + \zeta_{TP}. \tag{21}$$

Depending on the thermalhydraulic conditions (i.e., heat flux, void fraction, flow, etc.), the supports may be dry or wet. If the supports are dry, which is more likely for high heat flux and very high void fraction, only friction damping takes place. Support damping in this case is analogous to damping of heat exchanger tubes in gases (Pettigrew et al., 1986a).

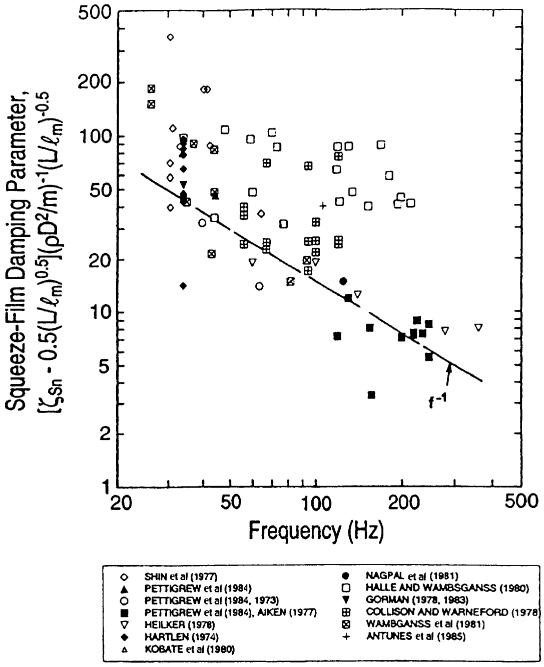


Fig. 9. Comparison between squeeze-film damping model (Eq. (18)) and experimental data.

Damping due to friction (and impact) in dry supports may be expressed by Eq. (14). The above situation is unlikely in well-designed recirculating steam generators. However, it could exist in once-through steam generators.

When there is liquid between the tube and the support, support damping includes both squeeze-film damping, ζ_{SF}, and friction damping, ζ_F. This situation is analogous to heat exchanger tubes in liquids and the support damping may be evaluated with Eqs. (18) and (19) above as follows:

$$\zeta_S = \zeta_{SF} + \zeta_F = \left(\frac{N-1}{N}\right)\left[\frac{1460}{f}\left(\frac{\rho_\ell D^2}{m}\right) + 0.5\right]\left(\frac{L}{\ell_m}\right)^{1/2}. \tag{22}$$

Note that in Eq. (22), ρ_ℓ is the density of the liquid within the tube-support, whereas m is the total mass per unit length of the tube including the hydrodynamic mass calculated with the two-phase homogeneous density.

Viscous damping in two-phase mixtures is analogous to viscous damping in single-phase fluids (Pettigrew and Taylor, 1997). Homogeneous properties of the two-phase mixture are used in its formulation, as follows:

$$\zeta_V = \frac{100\pi}{\sqrt{8}}\left(\frac{\rho_{TP} D^2}{m}\right)\left(\frac{2\nu_{TP}}{\pi f D^2}\right)^{1/2}\left\{\frac{[1+(D/D_e)^3]}{[1-(D/D_e)^2]^2}\right\}, \tag{23}$$

where ν_{TP} is the equivalent two-phase kinematic viscosity as per McAdams et al. (1942),

$$\nu_{TP} = \nu_l/[1+\varepsilon_g(\nu_l/\nu_g - 1)]. \tag{24}$$

Above 40% void fraction, viscous damping is generally small and could be neglected for the U-bend region of steam generators. However, it is significant for lower void fractions.

There is a two-phase component of damping in addition to viscous damping. As discussed by Pettigrew and Taylor (1997), two-phase damping is strongly dependent on void fraction, fluid properties and flow regimes, directly related to confinement and to the ratio of hydrodynamic mass over tube mass, and weakly related to frequency, mass flux or flow velocity, and tube bundle configuration. A semi-empirical expression was developed from the available experimental data to formulate the two-phase component of damping, ζ_{TP}, in percent:

$$\zeta_{TP} = 4.0\left(\frac{\rho_\ell D^2}{m}\right)[f(\varepsilon_g)]\left\{\frac{[1+(D/D_e)^3]}{[1-(D/D_e)^2]^2}\right\}. \tag{25}$$

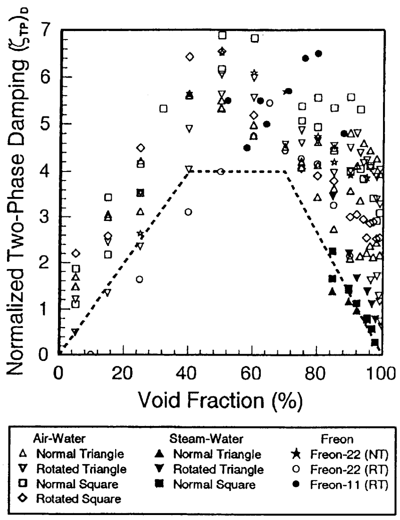


Fig. 10. Comparison between proposed design guideline and available damping data. Normalized two-phase damping ratio $(\zeta_{TP})_D$: $(\zeta_{TP})_D = \zeta_{TP}(\rho_\ell D^2/m)^{-1}\{[1+(D/D_e)^3]/[1-(D/D_e)^2]^2\}^{-1}$.

As shown in Fig. 10, the void fraction function, $f(\varepsilon_g)$, may be approximated by taking the envelope through the lower decile of the data:

$$
\begin{aligned}
&f(\varepsilon_g) = \varepsilon_g/40 \quad \text{for } \varepsilon_g < 40\%,\\
&f(\varepsilon_g) = 1 \quad \text{for } 40\% \leqslant \varepsilon_g \leqslant 70\%,\\
&f(\varepsilon_g) = 1 - (\varepsilon_g - 70)/30 \quad \text{for } \varepsilon_g > 70\%.
\end{aligned} \tag{26}
$$

The fluid properties/flow regime dependence is difficult to assess in the absence of a broad range of damping data for different two-phase mixtures. Flow regime effects are partly taken care of by the void-fraction function.

Tube fouling is not expected to contribute much to damping. However, support damping is considerably reduced by crudding within the support. At the limit, when tubes are jammed in the support by severe crud deposition, a support damping value of $\zeta_s = 0.2\%$ should be used in the analysis.

3.4. Dynamic stiffness and support effectiveness

For unfinned tubes, the dynamic stiffness of multispan heat exchanger tubes is simply the flexural rigidity, EI.

For finned tubes, Bolleter and Blevins (1982) found that the increase in flexural rigidity due to the fins is equivalent to an increase in wall thickness equal to 1/4 of the fin width, a, at the root of the fin over the length of the fin. Thus the equivalent stiffness, EI_{es}, is found using

$$
\frac{1}{EI_{es}} = \left(\frac{a}{a+b}\right)\frac{1}{EI_a} + \left(\frac{b}{a+b}\right)\frac{1}{EI_b}, \tag{27}
$$

where I is the moment of inertia and b is the tube length between fins (i.e., $a+b$ corresponds to the fin pitch).

The boundary conditions, that is the support conditions, are somewhat more complex. The tubes are effectively clamped at the tubesheet. To facilitate assembly and to allow for thermal expansion, there is a clearance between tubes and tube-supports. The diametral clearance between tube and intermediate support is typically 0.25–0.80 mm (for many nuclear heat exchangers, the diametral clearance is specified to be 0.38 mm or 0.015 in). Thus, the dynamic interaction between tube and tube-support is inherently nonlinear. In well-designed heat exchangers, the tube vibration response at mid-span is mostly less than 100 μm r.m.s. and much less at the supports (usually less than 25 mm r.m.s.). This is significantly less than the available diametral clearance. Thus, the tubes do not generally vibrate back and forth across the available clearance. Instead, most tubes are not centered within the supports and are, therefore, contacting or vibrating very close to one side of the supports. It is difficult to imagine that many tubes of typically one metre span length, would be located concentrically within a 0.38 mm diametral clearance without touching the support. The chances of a tube not touching a support are probably much less than 1%. Thus, it is reasonable to assume pinned

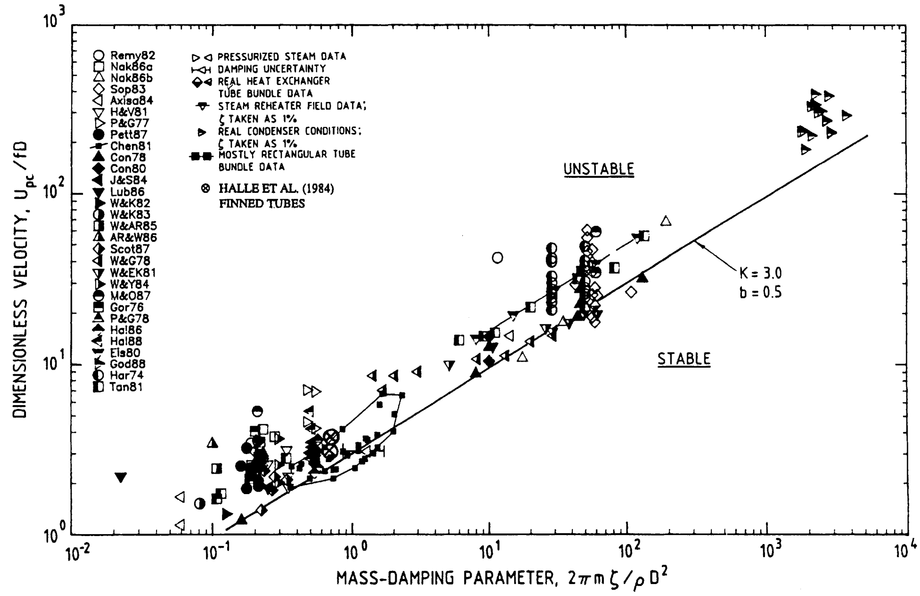


Fig. 11. Summary of fluidelastic instability data for single-phase cross flow: recommended design guidelines.

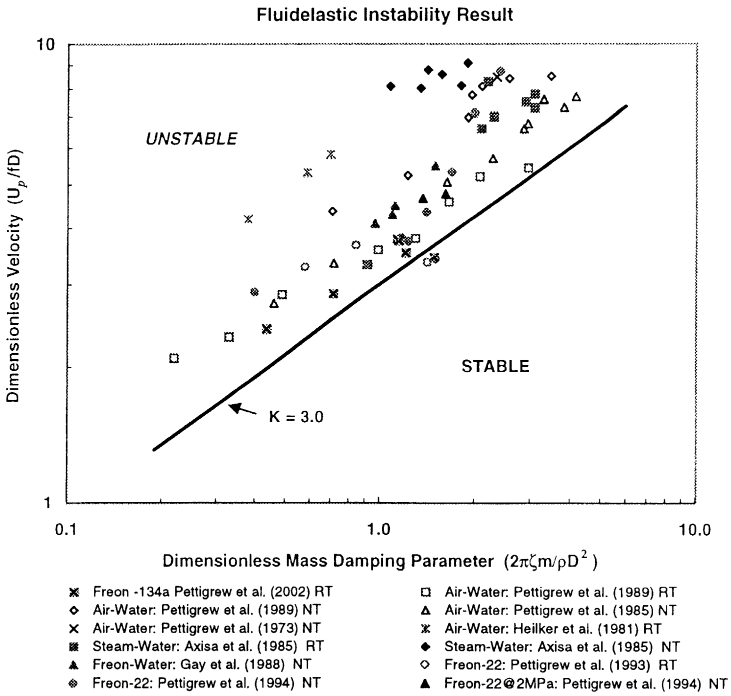


Fig. 12. Fluidelastic instability data in two-phase cross flow.

conditions at the supports to allow for a quasi-linear vibration analysis. For the purpose of vibration analysis, drilled hole, broached hole, and egg-crate type supports may be taken as pinned supports.

Antivibration bars (AVBs) or flat vertical U-bend restraints (FURs) are often used in the U-bend region of nuclear steam generators. FURs are reasonably effective in the out-of-plane direction, but offer little restraint in the in-plane (plane of the U-bend) direction (Taylor et al., 1995). Fortunately, the vibration response is usually much less in the in-plane direction than in the out-of-plane direction. Nevertheless, FURs are less effective than other clearance supports. For steam generator vibration analyses, it is recommended that one (any one) FUR in the U-bend region be assumed ineffective. The possibility that two adjacent FURs be ineffective at the same time is considered too improbable to be of concern. For example, if we assume that the probability of one FUR being ineffective is less than one percent, then the probability of two adjacent FURs being ineffective is less than 0.01% or less than one tube in 10 000, which is less than one tube per steam generator and thus insignificant.

4. Fluidelastic instability

4.1. Single-phase flow (gas or liquid)

Fluidelastic instability is the most important vibration excitation mechanism in heat exchanger tube bundles. This topic was reviewed by Pettigrew and Taylor (1991) for tube bundles subjected to single-phase cross flow. Fluidelastic instability is formulated in terms of a dimensionless flow velocity, U_p/fD, and a dimensionless mass-damping parameter, $2\pi\zeta m/\rho D^2$:

$$\frac{U_{pc}}{fD} = K\left(\frac{2\pi\zeta m}{\rho D^2}\right)^{1/2}, \tag{28}$$

where f is the tube natural frequency, ρ is the fluid density, m is the tube mass per unit length and U_{pc} is the threshold, or critical, flow velocity for fluidelastic instability. A fluidelastic instability constant $K = 3.0$ is recommended for all tube bundle configurations as shown in Fig. 11. The damping ratio, ζ, is the total damping ratio as defined in Eq. (16). The above formulation applies to all cross flow regions such as steam generator preheater regions, heat exchanger tube bundles between baffle plates, and outlet and inlet regions.

The limited data on fluidelastic instability for finned tubes were reviewed. The finned tube data were added to Fig. 11. It shows that fluidelastic instability is also possible for finned tubes and that a similar design guideline would be appropriate. Thus, $K = 3.0$ is also the recommended design guideline for finned tube bundles.

4.2. Two-phase flow

We have found that the fluidelastic instability behaviour is somewhat similar in continuous two-phase cross flow (Pettigrew et al., 1989b). As explained in earlier, continuous flow means two-phase flow regions of a continuous nature such as bubbly, spray, fog and wall flows as opposed to intermittent flow regimes leading to bundle reflooding and large flow oscillations. Such oscillations can lead to much lower critical velocity for fluidelastic instability (Pettigrew and Taylor 1994; Pettigrew et al., 1989b, 1995, 2002). Thus, intermittent flow regimes should be avoided in two-phase cross flow.

Fluidelastic instability in two-phase cross flow can also be formulated using Eq. (28). As shown in Fig. 12, a fluidelastic instability constant $K = 3.0$ is also recommended for two-phase cross flow, but only for tube bundles of

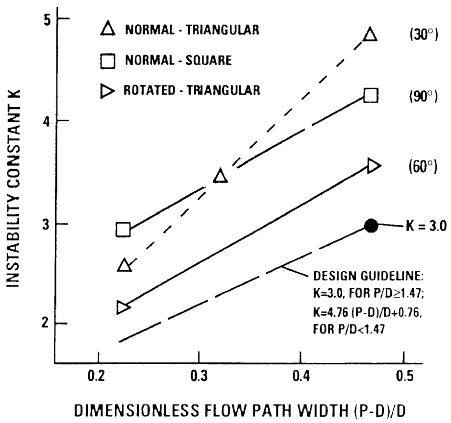


Fig. 13. Effect of P/D on fluidelastic instability constant in two-phase cross flow.

$P/D > 1.47$ which is the case for many steam generators. Lower values of K must be used for bundles of P/D ratios lower than 1.47, as discussed by Pettigrew and Taylor (1994). As shown in Fig. 13, the expression

$$K = 4.76(P - D)/D + 0.76 \tag{29}$$

would be a reasonable design guideline for $P/D < 1.47$.

5. Concluding remarks

Design guidelines to avoid flow-induced vibration problems in shell-and-tube heat exchangers are outlined in this paper. It presents an overview of the basic vibration excitation mechanisms and required steps to carry out a flow-induced vibration analysis.

Part 1 of this paper covers flow calculations, damping estimates and fluidelastic instability. This is required to avoid the most serious vibration problems due to fluidelastic instability. However to assure long-term performance, it is often necessary to look at vortex-shedding resonance, response to turbulence excitation and fretting-wear damage. This is outlined in the following companion paper (Pettigrew and Taylor, 2003, Part 2).

Acknowledgements

This document is based on some 30 years of research and development conducted at the Chalk River Laboratories in the area of flow-induced vibration. This technology development effort was largely supported by Atomic Energy of Canada Limited (AECL) and by the CANDU Owners Group (COG). It also received support from the Heat Transfer and Fluid Flow Service (HTFS), the Centre d'Etudes Nucléaires de Saclay (CEN-Saclay), the Pressure Vessel Research Council (PVRC) and the Washington Public Power Supply System (WPPSS). The support of all these organizations is very gratefully acknowledged. Many people have contributed to this effort including colleagues from industry and universities.

References

Bolleter, U., Blevins, R.D., 1982. Natural frequencies of finned heat exchanger tubes. Journal of Sound and Vibration 80 (2), 367–371.

Grant, I.D.R., 1975. Flow and pressure drop with single phase and two-phase flow in the shell-side of segmentally baffled shell-and-tube heat exchangers. NEL Report No. 590, National Engineering Laboratory, Glasgow, pp. 1–22.

Grant, I.D.R., Chisholm, D., 1977. Two-phase flow on the shell-side of a segmentally baffled shell-and-tube heat exchanger. Paper 77-WA/HT-22, ASME Winter Annual Meeting, Atlanta, USA, November.

Halle, H., Chenoweth, J.M., Wambsganss, M.W., 1984. Flow-induced vibration thresholds in heat exchangers from shellside water tests. ASME Symposium on Flow-Induced Vibration, Vibration in Heat Exchangers, Vol. 3. New Orleans.

Hartlen, R.T., Barnstaple, S.G., 1971. Unpublished report.

Mair, W.A., Jones, P.D.F., Palmer, R.K.W., 1975. Vortex shedding from finned tubes. Journal of Sound and Vibration 39, 293–296.

McAdams, W.H., Woods, W.K., Herman, L.C., 1942. Vaporization inside horizontal tubes-II-benzene-oil mixtures. Transactions of ASME 64, 193–200.

Pettigrew, M.J., Taylor, C.E., 1991. Fluidelastic instability of heat exchanger tube bundles: review and design recommendations. ASME Journal of Pressure Vessel Technology 113 (2), 242–256.

Pettigrew, M.J., Taylor, C.E., 1994. Two-phase flow-induced vibration: an overview. ASME Journal of Pressure Vessel Technology 116 (3), 233–253.

Pettigrew, M.J., Taylor, C.E., 1997. Damping of heat exchanger tubes in two-phase flow. Proceedings of the Fourth International Symposium on FSI, AE & FIV + N, ASME International Congress, ASME Publication AD-Vol 53.2, Dallas, Texas, November 16–21, pp. 407–418.

Pettigrew, M.J., Taylor, C.E., 2003. Vibration analysis of shell-and-tube heat exchangers: an overview; Part 2: vibration response, fretting-wear, guidelines. Journal of Fluids and Structures 18, 485–500.

Pettigrew, M.J., Goyder, H.G.D., Qiao, Z.L. Axisa, F., 1986a. Damping of multispan heat exchanger tubes, Part 1: in gases. Symposium on Special Topics of Structural Vibration, Vol. 104, ASME PVP Conference, Chicago, pp. 81–88.

Pettigrew, M.J., Rogers, R.J., Axisa, F., 1986b. Damping of multispan heat exchanger tubes—Part 2: in liquids. Proceedings of the ASME Pressure Vessels and Piping Conference, ASME Publication PVP-Vol. 104, Chicago, IL, July, pp. 89–98.

Pettigrew, M.J., Taylor, C.E., Kim, B.S., 1989a. Vibration of tube bundles in two-phase cross-flow: Part 1, hydrodynamic mass and damping. ASME Journal of Pressure Vessel Technology 111 (4), 466–477.

Pettigrew, M.J., Tromp, J.H., Taylor, C.E., Kim, B.S., 1989b. Vibration of tube bundles in two-phase cross-flow: Part 2, Fluidelastic instability. ASME Journal of Pressure Vessel Technology 111 (4), 478–487.

Pettigrew, M.J., Taylor, C.E., Jong, J.H, Currie, I.G., 1995. Vibration of a tube bundle in two-phase freon cross flow. ASME Journal of Pressure Vessel Technology, 117(4).

Pettigrew, M.J., Taylor, C.E., Janzen, V.P., Whan, T., 2002. Vibration behavior of rotated tube bundles in two-phase cross-flows. ASME Journal of Pressure Vessel Technology 124 (2), 144–153.

Pietralik, J.M., 1995. Thermal-hydraulic analysis of steam generators: application of the THIRST code to vibration analysis. Session 3.1. Proceedings of the 16th Annual Conference of the Canadian Nuclear Society, Vol. II, Saskatoon, Saskatchewan, June 4–7.

Rogers, R.J., Taylor, C., Pettigrew, M.J., 1984. Fluid effects on multispan heat exchanger tube vibration. Proceedings of the ASME Pressure Vessels and Piping Conference, San Antonio, TX, June, ASME Publication H00316, Topics in Fluid Structure Interaction, pp. 17–26.

Taylor, C.E., Boucher, K.M., Yetisir, M., 1995. Vibration and impact forces due to two-phase cross flow in U-bend regions of nuclear steam generators. Proceedings Sixth International Conference on Flow-Induced Vibration, London, UK, April 10–12, pp. 401–411.

Ulbrick, R., Mewes, D., 1994. Vertical, upward gas–liquid two-phase flow across a tube bundle. International Journal of Multiphase Flow 20 (2), 249–272.