FLOW-INDUCED VIBRATION & FRETTING-WEAR SPECIFICATIONS TO ENSURE STEAM-GENERATOR AND HEAT EXCHANGER LIFETIME PERFORMANCE

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ABSTRACT

The current interest in refurbishment, life extension and new-build activity has meant a renewed emphasis on technical specifications that will ensure improved reliability and longer life. Preventing vibration and fretting-wear problems in steam generators and heat exchangers requires design specifications that bring together specific guidelines, analysis methods, requirements and appropriate performance criteria. The specifications must be firmly based on experimental data and field inspections. In addition, the specifications must be supported by theoretical analyses and fundamental scaling correlations, to cover conditions and geometries over the wide range applicable to existing components and probable future designs. The specifications are expected to evolve to meet changing industry requirements.

This paper outlines the steps required to generate and support design specifications, and relates them to typical steam-generator design features and computer modeling capabilities. It also describes current issues that are driving changes to flow-induced vibration and fretting-wear specifications that can be applied to the design process for component refurbishment, replacement or new designs. These issues include recent experimental or field evidence for new excitation mechanisms, e.g., the possibility of in-plane fluidelastic instability of U-tubes, the demand for longer reactor and component lifetimes, the need for better predictions of dynamic properties and vibration response, e.g., two-phase random-turbulence excitation, and requirements to consider system "excursions" or abnormal scenarios, e.g., a main steam line break in the case of steam generators. The paper describes steps being taken to resolve these issues.

1. INTRODUCTION

Flow-induced vibration (FIV) in process equipment continues to attract considerable attention, mainly because of its potential to cause significant damage to components under adverse conditions. This is particularly so in the case of nuclear steam generators (SG) and liquid heat exchangers (HX), where the dominant excitation mechanisms are fluidelastic instability and random-turbulence excitation. Of the two, fluidelastic instability has received wider attention, due to its potential for extremely large vibration amplitudes and resulting risk to the integrity of steam-generator and heat-exchanger tubing. While the buffeting due to random turbulence is in principle less destructive, even in a well-designed unit it also can degrade tube integrity due to long-term fretting-wear (FW). In a poorly designed unit, or if conditions such as tube-support effectiveness do not meet the design intent or they change with time, long-term fretting-wear can be a life-limiting problem.

For CANDU®[1] nuclear power plants, Atomic Energy of Canada Limited (AECL) is the design authority and establishes criteria for plant and component performance. AECL has developed the Advanced CANDU Reactor™-1000 (ACR-1000™)[2] to meet customer needs for reduced cost, shorter construction schedule, high plant capacity factor, improved operations and maintenance, increased operating life, and enhanced safety features. Minimizing damage due to flow-induced vibration is one of the primary issues given particular attention in specifying design requirements (Subash and Hau, 2006).

A typical requirement is that the tube bundle be designed and constructed, and the tubes supported, in such a way that damaging vibrations will not occur during normal service. In addition the SG manufacturer is required to provide an assessment of how the design meets the reliability requirement for tube survival for the specified lifetime (60 years, in the case of the ACR-1000 steam generators).

The necessity to avoid vibration and fretting-wear problems requires design specifications that bring together specific guidelines, analysis methods, requirements and appropriate performance criteria. The specifications must be firmly based on experimental data and field inspections. In addition, the specifications must be supported by theoretical analyses and fundamental scaling correlations, to cover conditions and geometries over the wide range applicable to existing components and probable future designs. In the end, the designer or supplier of the steam generator or heat exchanger must be able to show by analysis or calculations that tube-vibration and fretting-wear levels will be below allowable levels, and that unacceptable resonances and fluidelastic instabilities will be avoided.

AECL has accumulated an extensive knowledge base in flow-induced vibration and fretting-wear technologies from R&D programs that date back to the early 1970's, and is considered a world leader in this area. Based on that experience, AECL has developed methodologies to assess the progression of tube wear damage during component life (Fisher et al. 2002, 2005). AECL's approach includes the use of

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[1] CANDU® (Canada Deuterium Uranium) is a registered trademark of Atomic Energy of Canada Limited (AECL).

[2] ACR-1000™ (Advanced CANDU Reactor™-1000) is a trademark of Atomic Energy of Canada Limited (AECL).

in-house codes PIPO-FE (Han et al., 2008), VIBIC (Han and Fisher, 2002) and H3DMAP (Morandin and Sauvé, 2002) to predict tube vibration and fretting-wear. The methodology described by Fisher et al. (2002, 2005), validated by observed fretting-wear damage in CANDU nuclear steam generators, has been used to produce vibration and wear specifications aimed at preventing tube failures due to fretting-wear associated with flow-induced vibration. The specifications are updated and then made available as the technology and designs change and as more information from experiments and from the field is made known. The two most recent versions that are widely available were produced by Pettigrew, Taylor and Subash (1995) as an AECL / COG report, and Pettigrew and Taylor (2003) as a pair of published papers. Some of the issues discussed in this paper are addressed in the FIV Specifications produced recently by Janzen and Pettigrew (2007) as an AECL report.

As outlined by Pettigrew and Taylor (2003), a vibration and fretting-wear analysis of a steam generator or a tube-and-shell heat exchanger consists of the following steps:

1. flow-distribution calculations,
2. dynamic parameter evaluation (damping, effective tube mass, dynamic stiffness, effectiveness of tube supports),
3. formulation of vibration excitation mechanisms,
4. prediction of vibration response, and
5. resulting damage assessment (including a comparison against allowable limits).

The present paper reviews these steps and relates them to typical steam-generator design features and computer modeling capabilities. However the focus of the paper is on current issues that are driving changes to design specifications, such as:

- Recent experimental or field evidence for new excitation mechanisms, e.g., the possibilities of in-plane fluidelastic instability and quasi-periodic forces in tube bundles.

- A demand for increased thermalhydraulic performance at a lower cost, and longer reactor and component lifetimes. Increased performance and cost-efficiency typically translate into larger size, lower pitch-to-diameter ratios, higher pressure and temperature and consequently higher maximum secondary-side velocities and steam qualities. Component lifetimes of 60 years are now being considered. These changes require accurate predictions of vibration response and fretting-wear trends over longer periods of time and over larger ranges of thermalhydraulic conditions.

- An improved understanding of dynamic properties, e.g., two-phase damping, and excitation formulation, e.g., two-phase random-turbulence excitation.

- Requirements to consider system “excursions” or abnormal scenarios, e.g., a main steam-line break. Although these events tend to be short-lived, they can lead to conditions that are significantly outside a component’s normal operating envelope, e.g., pressure drops and flow velocities that are a factor of two or more higher than normal, and the consequences of these events may need to be addressed at the design stage.

In a separate paper in these proceedings, Han et al. (2008) describe the implementation of updated sections of the vibration and fretting-wear analysis in

AECL's PIPO-FE code. They also list the basic vibration acceptance criteria and illustrate how the code can be used to support the design and modification of power plant heat exchangers and related research activities.

2. VIBRATION / FRETTING-WEAR ANALYSES AND EMERGING ISSUES

2.1 Flow-Distribution Calculations

For relatively simple components, where the flow paths are well defined, local flow velocities are often calculated with a flow-path approach. For complex components such as nuclear steam generators, a comprehensive three-dimensional thermalhydraulic analysis is required. The power and process industries have developed several codes that are mathematically robust and sufficiently well validated to reliably predict thermalhydraulic performance and steady-state flow properties. To predict the performance and flow properties of steam generators AECL uses its THIRST code (e.g., Heppner et al., 2006), which not only models two-phase flow within a recirculating SG but also assesses SG bulk water chemistry and the effect of fouling on SG thermalhydraulic conditions (Turner et al., 2006).

Emerging issues related to flow-distribution calculations are as follows:

- The demand for more detailed and realistic flow calculations, e.g., flow through complicated geometries such as Tube Support Plates (TSP's) with flow openings that become partially or fully plugged. These details can be addressed with codes such as THIRST on a distributed scale, e.g., across a horizontal plane of a steam generator, or with Computational Fluid Dynamics (CFD) codes on a finer scale.
- Transient situations, e.g., Main Steam Line Break (MSLB) scenarios. Characterizing such events requires either sophisticated numerical codes or an appropriate conservative approach to bounding their effect in terms of FIV and FW (see Section 2.4).
- In both of these cases, there is the associated issue of validating these code predictions.

2.2 Dynamic Parameters

Flow-induced vibration is a specific manifestation of the general phenomenon of fluid-structure interaction. The dynamic variables that need to be considered are the added (hydrodynamic) mass, the dynamic stiffness, the damping due to energy exchange between fluid and structure, and boundary conditions.

In principle, the dynamic behaviour of fluid-structure systems can be calculated with coupled equations of fluid and structural dynamics. In engineering practice, flow-induced vibration is sufficiently specialized, and the underlying physics sufficiently complex, that a separate set of practical tools has evolved. AECL's approach to calculating these parameters is described most recently by Pettigrew and Taylor (2003). The hydrodynamic mass is easily calculated while the dynamic stiffness, for heat-exchanger and steam-generator tubes, is simply the flexural rigidity, *EI*. Damping, particularly in two-phase flow, and the accurate description of boundary (tube-support) conditions are more complicated issues.

Damping in Two-Phase Flow

The total damping ratio, ζ_T, of a multispan heat exchanger tube in two-phase flow includes support damping, ζ_S, viscous damping, ζ_V, and two-phase damping, ζ_{TP}; thus

$$\zeta_T = \zeta_S + \zeta_V + \zeta_{TP} \tag{1}$$

Pettigrew and Taylor (2004) have reviewed the available information and outlined the development of a semi-empirical model to formulate damping of heat-exchanger tube bundles in two-phase cross-flow. This model is incorporated in AECL's current FIV specifications.

Figure 1 shows the experimental information compiled for tube bundles of various geometries in two-phase cross-flow. The two-phase component is very dependent on void fraction, and reaches a maximum between 40 and 70% void fraction. The current two-phase damping formulation is shown as a dashed line in Figure 1, and is an appropriate (conservative) design guideline.

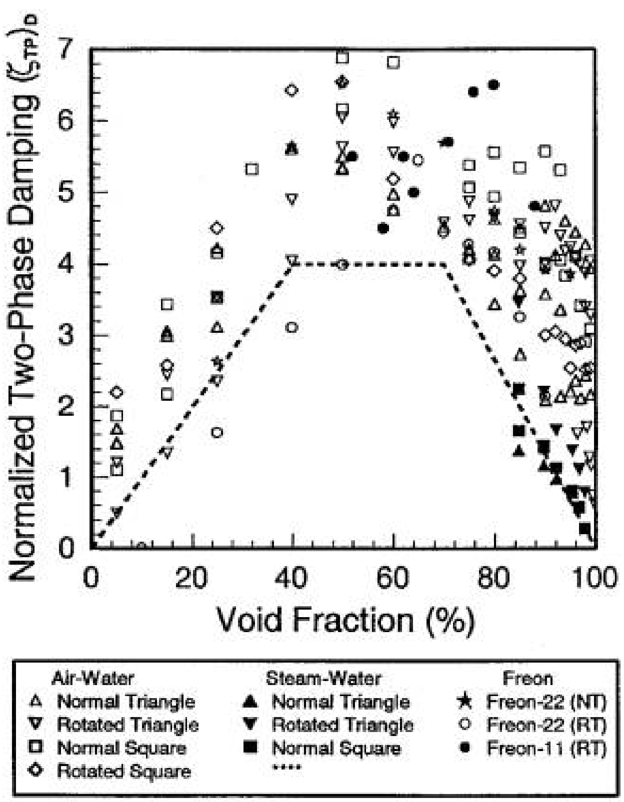


Figure 1 Two-phase damping component in air-water, Freon and steam-water two-phase flow (Pettigrew and Taylor, 2003).

Nevertheless, it should be pointed out that the formulation illustrated in Figure 1 is intended as a preliminary guideline for heat exchanger and steam generator design.

The basic energy dissipation mechanism responsible for two-phase damping is not yet understood. In addition, the guideline is based on somewhat limited information.

In particular, additional data are required on the effect of fluid properties and flow regime. Experiments are underway to investigate the dependence of two-phase damping on fluid properties such as phase density differences and interfacial surface area in bubbly flow, and to the transition between bubbly and slug-churn flow regimes (e.g., Béguin et al., 2008). Further studies are underway to investigate damping in slug-churn flow. The results will be incorporated in updated FIV specifications.

Boundary Conditions / Effective Tube Supports

The remaining dynamic parameters are the boundary conditions, which, for multi-span heat-exchanger or steam-generator tubes, are essentially the tube-support conditions. From a mechanical point of view, the tubes are simply multi-span beams, clamped at the tubesheet and held at the supports with varying degrees of constraint. The issue of how best to specify a support design that eliminates excessive vibration, fretting-wear and impact abrasion, while at the same time allowing thermal expansion and offering low hydraulic resistance, has been the subject of many theoretical analyses and experimental tests. Information relevant to CANDU steam generators and heat exchangers can be found in, for example, publications by Weaver and Schneider (1983), Taylor et al. (1991), Boucher and Taylor (1996), Janzen et al. (2005) and Dyke and Garland (2006).

For flatbar type U-bend supports, AECL's specifications currently prescribe clearances less than 0.1 mm / 0.004 in. (diametral) on average. Industry now claims to be capable of achieving a nominal tube to flatbar diametral gap of 0.05 mm / 0.002 in. (Klarner et al., 2006), in an effort to further reduce the possibility of damaging flow-induced vibration. AECL specifications also require some support redundancy; any flow-induced vibration analysis of a design must satisfy standard criteria for flow-induced vibration while assuming that any one support in the U-bend region may not be effective.

The relevant issue currently facing HX and SG designers is how best to determine whether or not a given support design is effective in preventing excessive flow-induced vibration and fretting-wear. That is, will the supports act to constrain the motion of the tubes in such a way as to prevent both (i) excessive impacts of the tube against the support and (ii) excessive fretting-wear due to motion of the tube along the support while the two are in contact. This issue is discussed in Section 3.2.

2.3 Excitation Mechanisms

In nuclear steam generators and liquid heat exchangers, the dominant flow-induced-vibration excitation mechanisms are fluidelastic instability and random-turbulence excitation. Recent experimental evidence has shown that widely used formulations of both of those excitation mechanisms need to be updated.

2.3.1 Fluidelastic Instability

Fluidelastic instability (FEI) corresponds to a sudden rapid increase in tube response with increasing cross-flow velocity, and occurs when the excitation energy imparted to the tube by flow is greater than the energy dissipated by damping mechanisms. For the industrial designer, FEI is easily the most important vibration excitation mechanism in steam-generator and heat-exchanger tube bundles. The requirement that none of the thousands of tubes are subjected to such uncontrollable vibration levels is paramount, since even short-term excursions into the FEI regime can represent an unacceptable risk of damage and potential loss of tube integrity.

Fluidelastic instability has been studied extensively, particularly with respect to SG and HX tube FIV. Nevertheless, although there are promising methods being developed to predict the response of a tube excited by FEI (e.g., see Section 2.4.1), current design specifications merely place limits on the design parameters that are known from experiment to affect the flow velocity at which FEI appears. The onset of instability is typically compared to the Connors formulation in terms of a dimensionless flow velocity, U_p / fD, and a dimensionless mass-damping parameter, $2\pi\zeta m / \rho D^2$, such that the critical dimensionless flow velocity is given by

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

where f is the tube natural frequency, ρ is the fluid density, m is the mass per unit length, including both tube and hydrodynamic mass, ζ is the total damping ratio, U_p is the pitch flow velocity (flow velocity in the tube bundle), U_{pc} is the threshold, or critical, pitch flow velocity at which fluidelastic instability occurs, D is the tube diameter, and K is a fluidelastic-instability coefficient that can be taken to be constant for a given tube-bundle pitch-over-diameter ratio.

Two issues related to fluidelastic instability have recently emerged. Firstly, it has long been assumed that for U-shaped tubes such instability would only practically occur in the “out-of-plane” direction, associated with the bending modes perpendicular to the plane of the U-bend. For CANDU-type tube bundles, with $P/D \simeq 1.5$, the fluidelastic-instability constant of $K = 3.0$ specified by AECL has proven to be a reliable lower bound for out-of-plane motion. For U-tubes supported by flatbars, the value of K is reduced by a factor of 0.75 to provide some support redundancy, i.e., $K = 2.25$ is specified for tube bundles with $P/D \simeq 1.5$.

Recent experimental evidence has shown that “in-plane” fluidelastic instability is also possible, and needs to be considered in any FIV analysis. This issue is presented as a “case history” in Section 3.1, to illustrate how initial experimental evidence led to tests intended to isolate and quantify the effect on SG / HX tubes. Results of those tests were eventually incorporated into a revised FIV specification for fluidelastic instability.

The second issue related to fluidelastic instability concerns the parameterization of the fluid force in such a way that the tube response and associated fretting-wear can be calculated for circumstances where the flow is unusually high, e.g., transient scenarios where the tubes are subject to flow velocities above the FEI critical value. This issue is discussed in Section 2.4.

2.3.2 Random Turbulence Excitation

Random turbulence is a significant vibration excitation mechanism in both liquid and two-phase cross-flow. The excitation force is typically parameterized by a (dimensionless) power spectral density and is a function of the tube vibration frequency, tube diameter, flow velocity, homogeneous fluid density, void fraction, and length of tube exposed to cross flow.

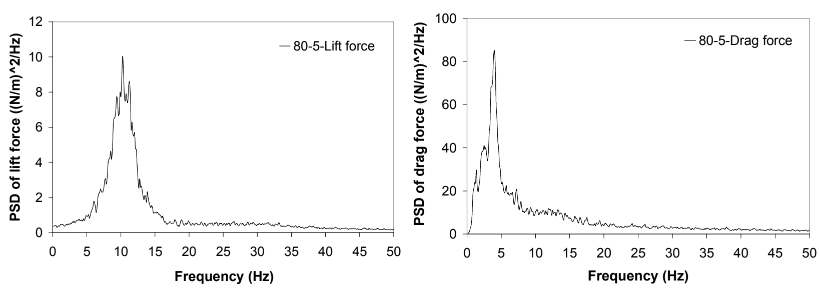
A current issue is the choice of an appropriate functional form of the power spectral density that can a) be properly scaled, to apply to different tube-bundle geometries and flow conditions, and b) provide an accurate, conservative representation of experimental data. This issue, and the related issue of how best to predict the vibration response due to Random Turbulence Excitation (RTE) based on these excitation forces, are discussed in some detail by Han et al. (2008) and summarized in Section 2.4.

2.3.3 Periodic and Quasi-Periodic Forces

Periodic wake shedding, or vortex shedding, may lead to resonance and large vibration amplitudes when the shedding frequency coincides with a tube natural frequency. It may be of concern in liquid cross flow in cases where the flow is relatively uniform. Periodic wake shedding is generally not a problem in two-phase flow, except at low void fractions (less than 15%).

For steam generators or other components operating at moderate-to-high void fractions, there are recent experimental results that point to a different, quasi-periodic excitation mechanism. In tests with bundles of straight cantilevered tubes in a rotated-triangular array with a P/D ratio of 1.5, a geometry typically found in CANDU steam generators, Pettigrew, Zhang and co-workers performed detailed flow and vibration-excitation force measurements in two-phase (air-water) flow (e.g., Pettigrew et al., 2005, Zhang et al., 2008). Distributions of both void fraction and bubble velocity were measured, along with dynamic forces in the lift and drag direction.

Somewhat unexpectedly, significant levels of quasi-periodic excitation forces were observed in both the drag and lift directions (Figure 2). The forces were well correlated along the tube. In the lift direction, the quasi-periodic forces were related to local void fraction properties in the unsteady wake area between cylinders. In the drag direction, the quasi-periodic forces appeared related to void-fraction fluctuations in the main flow path between the cylinders.



(a) (b)

Figure 2 (bottom of previous page): Typical dynamic force spectra for 80% void fraction at 5 m/s pitch flow velocity; (a) Lift force spectra, (b) Drag force spectra (Zhang et al., 2008).

These quasi-periodic drag and lift forces can be compared to measurements of the flow characteristics made with bi-optical void-fraction probes. The forces appear to be generated by mechanisms related specifically to the dynamic properties of two-phase flow in periodic tube arrays (see Figure 3).

The following specific technical issues are associated with these new excitation mechanisms:

- How do the two-phase forces scale with tube bundle size and geometry, and with thermalhydraulic properties (e.g., steam-water c.f. air-water)?
- What is the effect of bundle anisotropy (e.g., in-plane vs. out-of-plane motion in U-tubes)?
- What is the best approach to formulating a mathematical model of the force power spectral densities, validating the model and implementing it in FIV codes?

This type of excitation mechanism has not been observed before and is, therefore, not presently included in FIV specifications for CANDU steam generators and heat exchangers. Work is underway to include quasi-periodic forcing functions in AECL's FIV codes.

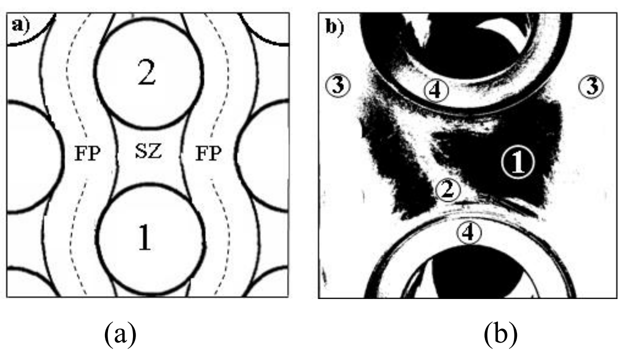


Figure 3 Two-phase flow structure in a rotated triangular tube bundle: a) simplified figure (FP: flow path, SZ: Stagnation zone), b) flow picture (1: low void fraction mixture belonging to the stagnation zone, 2: oscillating high void fraction mixture in stagnation zone, 3: flow path, 4: rigid tubes) (Zhang et al., 2008).

2.4 Vibration Response Prediction

Because of the complexity of a multispan heat-exchanger tube, a computer code must be used to predict vibration response, and to determine the susceptibility of a tube to fluidelastic instability, random turbulence forces and periodic / quasiperiodic forces.

The computer code must also be capable of calculating vibration modeshapes and natural frequencies.

To predict tube response, it is convenient and often appropriate to assume that the intermediate supports are pinned. With this assumption, the analysis is linear and either a finite-element code or an analytical code can be used. AECL uses its PIPO-FE finite-element code (Han et al., 2008) to assess the vibration response of HX or SG tubes with the assumption of pinned supports.

To calculate the detailed motion of a tube for a specific HX or SG design with clearance-type supports, a non-linear analysis that takes into account dynamics of the interaction between tube and tube support is required. AECL uses the VIBIC code (Han and Fisher, 2002) or the H3DMAP code (e.g., Morandin and Sauvé, 2002) to accomplish this.

Two issues have recently emerged that have challenged the existing methodologies for predicting tube-vibration response in two-phase flow. The first is related to requirements in some jurisdictions, for either refurbished or new components, to consider transient scenarios involving very high flow rates, e.g., a main steam-line break scenario with flow rates two-to-three times normal operating values. The second is related to inconsistencies in the methods used to calculate the response to random-turbulence excitation under normal operating conditions.

These issues have most recently arisen when considering refurbishment or replacement of CANDU SG or HX components, but in the future will likely be required for new designs as well.

2.4.1 Predictions for "Off-Normal" or Transient Conditions

For steady-state conditions, fluidelastic instability is the excitation mechanism with the potential to cause the most damage to HX or SG tubes. Vibration analyses typically predict threshold (critical) conditions for the onset of fluidelastic instability with the use of empirical constants obtained from steady-state tests, and include an appropriate safety margin. Historically, as long as the components were operated below this critical threshold, and mechanical parameters such as design support conditions were adhered to, FEI was effectively a non-issue.

Nevertheless, there are instances in which a plant licensee has been asked to assess the consequences of fluidelastic instability for short-term operation under "off-nominal" conditions. Examples include clearance supports becoming clamped supports due to tubes unexpectedly swelling (NRC, 2002) and transient analyses such as a blowdown analysis of a postulated steam main line break (e.g., Sauvé and Anderson, 1994), commissioning tests required to establish operating limits, and operation under "stretch-out" conditions. Particularly in nuclear steam generators, maintaining tube integrity throughout such transients is vital. In some of these scenarios, fatigue is a possible tube failure mechanism that also needs to be assessed.

Prerequisites to making these assessments are a thermalhydraulic code capable of analyzing two-phase, transient flow in piping systems and steam generators, a realistic, numerically robust FEI model implemented in one of the existing non-linear FIV/FW codes, and test results that can be used to validate these numerical calculations.

For transient events, codes such as CATHENA (Hanna and Arsenault, 2001) can provide the necessary thermalhydraulic conditions. Alternatively conservative upper bounds can be used, e.g., increases in the velocity loading by factors of two or three are commonly used for main steam-line break scenarios. While the density profile may be expected to change in reality, keeping it unchanged is a typical, conservative assumption.

Another possible approach to estimating the degree of fretting-wear expected in the “post-critical” flow regime, i.e., if the conditions are beyond the FEI critical velocity, is to use the simple energy approach of Yetisir et al. (1998) and Pettigrew et al. (1999) (see also Section 2.5), combined with some bounding assumptions. If the onset of fluidelastic instability can be associated with tube-to-tube clashing at the predicted critical flow velocity, the energy available in the dynamic interaction between tube and support can be scaled for higher flow velocities. This scaled energy can then be used to estimate the fretting-wear expected under those conditions. Complications are the need to consider a) which vibration modes/frequencies dominate at high velocity, and b) changes in thermalhydraulic conditions, e.g., pressure and temperature changes.

A mathematical calculation to a) accurately predict the stability boundaries for excitation in two-phase flow, and b) assess tube dynamics and related tube wear in the “post-critical” flow regime would be preferable to using these approximations. There are mathematical models that attempt to solve coupled fluid-structural dynamic equations in the time domain in order to predict tube motion under the influence of fluidelastic instability, the most widely recognized being the model of Chen (1983). However, such calculations are typically numerically difficult, model-dependent, and difficult to validate. In practice these models have not been able to provide reliable predictions, either due to such difficulties or because they require extensive (and expensive) experimental measurements of the fluid forces under a wide range of flow conditions and geometries.

Mureithi et al. (2008) have recently proposed a model that has the potential to estimate fluid dynamic loads under realistic conditions including the effects of fluidelastic instability. Their approach relies on measurements in flow of the quasi-static fluid force field in a tube bundle, for a series of void fractions and flow velocities, to parameterize the fluid force field. Hence, quasi-steady lift and drag forces with a time delay are used, instead of a Connors-based negative damping formulation.

In this model the difficulty and number of experimental measurements is less, and the force-field data are more readily scaled, than in earlier models. In principle the results can be applied to time-domain computations for tube wear, e.g., the numerical simulations performed by AECL’s VIBIC code. Development is continuing along these lines.

2.4.2 Vibration Response due to Random Turbulence Excitation

AECL's current specification provides that in two-phase cross flow, the power spectral density (PSD) of the random excitation force, $S_F(f)$ is, for a given void fraction, directly related to the mass flux squared and the tube diameter squared, and inversely related to the frequency to the exponent 1.25 and to the length of tube exposed to cross flow, L_e (Pettigrew et al., 1991). Thus

$$S_F(f) \; \alpha \; \frac{\dot{m}_p^2 D^2}{L_e f^{1.25}} \tag{3}$$

Equation (3) can be used to define a normalized power spectral density (*NPSD* in $\mathrm{m}^3.\mathrm{s}^{-2.25}$) of the random excitation forces such that

$$NPSD = \frac{S_F(f) L_e f^{1.25}}{\dot{m}_p^2 D^2} \tag{4}$$

The following expressions can be used to calculate $S_F(f)$ for tube bundles subjected to air-water or to steam-water mixtures:

$$\begin{aligned} NPSD &= 10^{\left(0.027\varepsilon_g - 3.45\right)} && \text{for } 10\% < \varepsilon_g < 80\% \\ NPSD &= 0.05 && \text{for } 80\% < \varepsilon_g < 99\% \end{aligned} \tag{5}$$

where ε_g is the two-phase void fraction.

Note that the above expressions are not a best fit to the experimental data, but are chosen to provide a conservative upper bound to the RTE force. The upper bound incorporates data spanning more than thirty years of testing under a wide variety of conditions. However, in the form of Equation (4) the PSD does not scale with reduced frequency $f_R = fD/U_p$ as would be expected for a mathematically ideal, dimensionless treatment. The selection of an appropriate upper bound, and the issue of how best to define a dimensionless effective power-spectral density that can be scaled with reduced frequency rather than simply frequency, are currently under review. One possible form of an Effective Power Spectral Density (EPSD) is described by Han et al. (2008).

Following the definition of the RTE force, the next step is prediction of the tube vibration response. AECL's specifications state that the response shall be determined using random vibration theory, and they go on to provide general equations to obtain the mean square amplitude response, $\overline{y^2(x)}$, of a cylindrical tube to distributed random forces. The formulation of those equations requires a number of assumptions discussed by Taylor and Pettigrew (1998). More details are given by Pettigrew et al. (1978).

There are currently a number of different approaches available to calculate the RTE response:

- Previous versions of AECL's PIPO-FE code treated distributed random-turbulence forces largely by implementing methods suggested by Pettigrew and Gorman (1973) and then by Taylor et al. (1996).

- The more recent approach of Pettigrew and Taylor (2003) uses some methods suggested by de Langre and Villard (1998) and yields conservative upper bounds on the magnitude of the random-turbulence forces. However, the treatment of correlation lengths, and the relation between equivalent spectra obtained with different tube geometries, in particular different reference lengths, is inconsistent with the approaches used in some other industry codes. In addition, as is the case with all of the currently available methods, the methodology does not take into account the different flow regimes that are possible in two-phase flow.
- PIPO-FE is currently being upgraded to include some recent experimental findings and updated design guidelines produced through collaboration with the BWC/AECL/NSERC Fluid-Structure Interaction Program in the Department of Mechanical Engineering at École Polytechnique, Montréal. This process is described in a separate paper in these proceedings (Han et al., 2008).

There is no real controversy concerning the appropriateness of the general equation for the tube response given by Pettigrew and Taylor (2003). The problem lies in its implementation in the various codes used by industry. The physical phenomenon is sufficiently complex that there is no consensus on a mathematical formulation of the RTE vibration response that properly takes into account dimensionally correct scaling of tube-bundle and hydraulic parameters and the possibility of different flow regimes. Neither is there agreement on the mathematical implementation of force correlations within and between spans.

The practical effect is that calculations with different codes can lead to different vibration responses, depending on the choices of parameters and the assumptions made. When this happens, the choice could be to use the most conservative result to make decisions related to design or to refurbishment of SG and HX components. In some cases this may be an overly conservative approach. To ensure that SG and HX components are appropriately specified and designed, the random turbulence excitation formulation needs to be reviewed and agreement reached on its implementation in the codes used by industry.

For each implementation, there are the associated issues of verifying the code and validating the code predictions. As a general statement, and for the issue of RTE formulation in particular, the industry would benefit from a move towards a common set of verification and validation cases that could be applied to codes that are in common use.

2.5 Fretting-Wear Damage Considerations

To calculate the degree of fretting-wear expected for a specific design, a non-linear code may be required that models the tube-to-support interaction in terms of dynamics and in terms of impact-wear and tribological wear. AECL's VIBIC and H3DMAP codes both include this capability.

Note that even though a linear analysis does not predict fretting-wear, a fretting-wear assessment can still be made with a suitable estimate of the work-rate obtained from the mechanical energy available in the dynamic interaction between tube and support, as outlined by Yetisir et al. (1998) and Pettigrew et al. (1999). AECL currently specifies this energy approach be used to estimate a tube wall fretting-wear depth over the lifetime of the component, and recommends that estimated depth should be less than 10% through-wall (TW).

This approach is conservative and if followed will have the intended effect of mitigating if not virtually eliminating tube fretting-wear. Nevertheless, there are a number of issues that arise in practical applications:

Design / Inspection Strategy – Tube-inspection strategies and wear-indication disposition methods are well developed for nuclear steam generators, less so for heat exchangers. The AECL specification of maximum 10% TW is, therefore, appropriate to heat exchangers but may be somewhat conservative for steam generators. For example, a 40% TW plugging criterion is sometimes used for steam generators when accompanied by an extensive tube-inspection strategy.

Longer Component Lifetimes / Life Extension – Traditionally, the most important element of an FIV and FW analysis was to ensure the design prevented the damaging effects of fluidelastic instability. Traditionally, CANDU steam generators and heat exchangers have been designed to have maximum lifetimes of 40 years. The design lifetime for the ACR-1000 SG is 60 years. In terms of preventing life-limiting fretting-wear damage, this increase in expected lifetime places relatively more emphasis on assessing long-term fretting-wear due to random turbulence. Producing a practical, cost-effective, design for long life requires that we refine the RTE formulation as discussed in Section 2.4.

The requirement for accurate long-term fretting-wear predictions also puts an increased emphasis on the choice of fretting-wear coefficients, which affects the fretting-wear calculation in the Archard equation:

$$\dot{V} = K_{FW} \dot{W}_N \qquad (6)$$

where $\dot{V}$ is the volume fretting-wear rate, K_{FW} is the fretting-wear coefficient, and $\dot{W}_N$ is the normal work-rate.

Fretting-wear coefficients are sensitive to the choice of tube and tube-support material, to chemistry and to temperature. Based on extensive testing, AECL recommends that a fretting-wear coefficient of $K_{FW} = 20 \times 10^{-15}$ m^2/N be used for the following suitable materials: Incoloy-800 (I800), Inconel-690 (I690) and I600 tubes with 410s, 304L, 316L, 321 SS or carbon steel supports (Pettigrew and Taylor, 2003). Other material combinations may be used providing that a reliable and documented fretting-wear coefficient is available. However, fretting-wear coefficients are not available for all tube and support material combinations being considered for new designs or for life extensions. Particularly when considering a 60-year lifetime, accurate knowledge of the fretting-wear coefficient could prevent overly conservative decisions concerning other aspects of the design.

3. EXAMPLES OF CURRENT ISSUES DRIVING CHANGES TO DESIGN SPECIFICATIONS

3.1 In-Plane Fluidelastic Instability – A Case Study

Since the early 1970's, when Connors formulated his simple equation to predict the flow velocity at which instability would occur, it has been assumed that for U-shaped tubes such instability would only practically occur in the "out-of-plane" direction, associated with the bending modes perpendicular to the plane of the U-bend (see Figure 4). A fluidelastic-instability constant of $K = 3.0$ has proven to be a reliable lower bound for out-of-plane motion in CANDU-type tube bundles with pitch-over-diameter ratios $P/D \cong 1.5$.

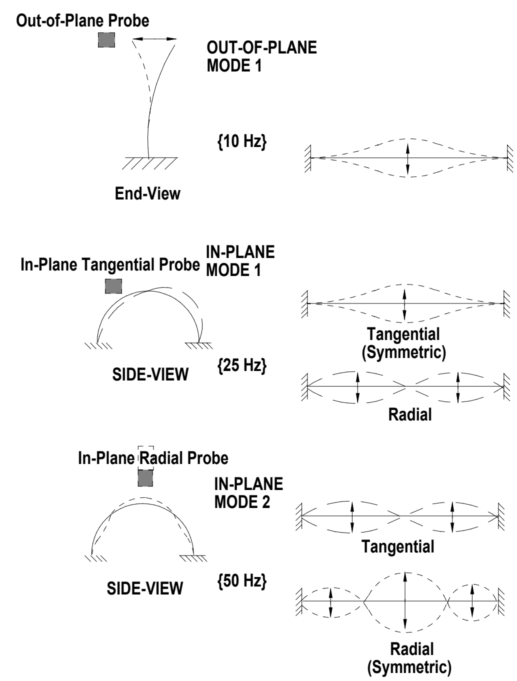


Figure 4 Lowest vibration modes for a simple unsupported U-tube (Boucher and Taylor (1996) and Janzen et al. (2005)).

The assumption that only out-of-plane FEI could occur was partly a result of the vast majority of experiments being performed with straight tubes, thus having no intrinsic geometry to define in- or out-of-plane, and partly a result of difficulties disentangling competing vibration modes once unstable large-amplitude vibration sets in. This situation persisted despite some indications that in-plane modes needed to be considered in the design of steam-generator tube supports, e.g., Weaver and Schneider (1983). Tests were primarily focused on fluidelastic instability in the out-of-plane direction, and related issues such as amplitude-limited instability (see Figure 5). Au-Yang (2001) reported that in-plane instability had not been observed in tests with U-bend tubes in liquid or two-phase flow.

The picture fundamentally changed with the observation of in-plane fluidelastically unstable vibration of semi-circular U-tubes in water and in air-water cross-flows at low void fraction (Janzen et al. (2005), see Figure 6). This observation was soon followed by investigations of fluidelastic instability in arrays of tubes preferentially flexible in the flow direction (the straight-tube equivalent of in-plane instability) in air (Mureithi et al., 2005) and in air-water cross-flow at medium-to-high void fraction (Violette et al., 2006). These results led AECL to expand its specifications for fluidelastic instability in steam generators and liquid heat exchangers, to include the possibility of in-plane FEI (Janzen and Pettigrew, 2007).

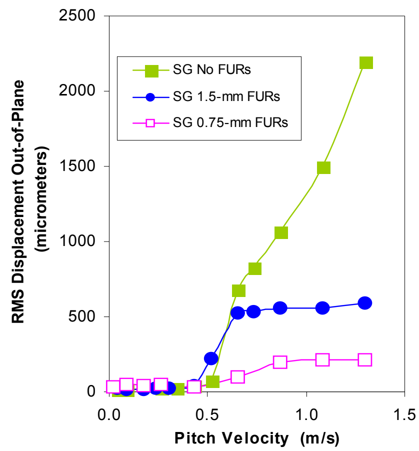


Figure 5 Out-of-plane vibration amplitudes for a semi-circular U-tube, in liquid flow, for no tube supports (No FURs) and flat U-bend restraints (FURs) with two different tube/support clearances (Janzen et al., 2005).

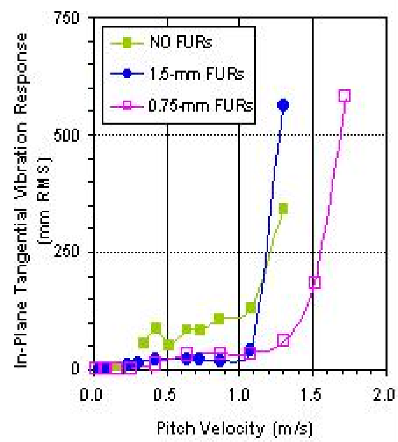


Figure 6 In-plane vibration amplitudes measured for a semi-circular U-tube, in liquid flow (Janzen et al., 2005).

Data from Mureithi et al. (2005) and Violette et al. (2006) are shown in Figure 7, along with historical data for the onset of out-of-plane FEI with axisymmetrically flexible tube bundles. Although an FEI constant of $K = 3.0$ is an appropriate lower bound for out-of-plane motion, a value of $K = 8.0$ better represents the data for in-plane motion. To ensure a conservative design guideline, a value of $K = 6.0$ is used as a lower bound for in-plane fluidelastic instability in AECL's most recent flow-induced vibration specifications.

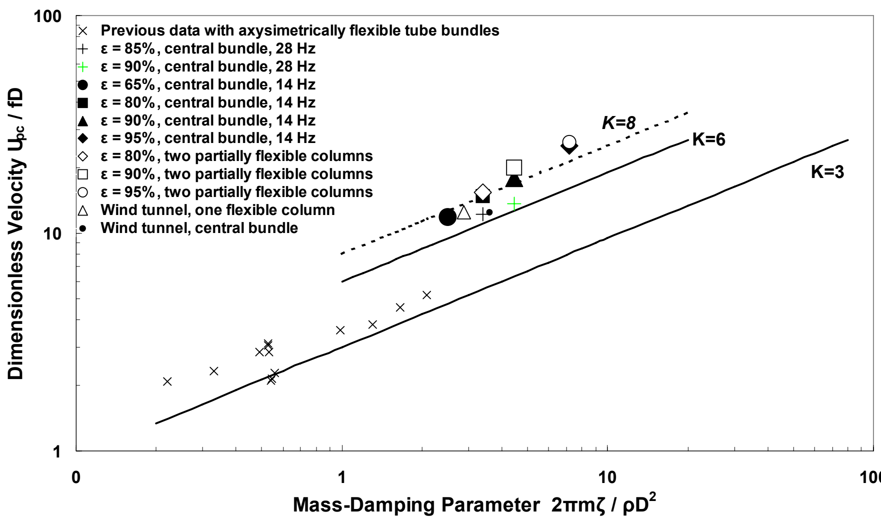


Figure 7 Fluidelastic-instability map for rotated triangular tube bundles (after Violette et al., 2006). The symbol X corresponds to data obtained with axisymmetrically flexible tube bundles (Pettigrew et al., 1989). All other symbols correspond to data obtained with tubes preferentially flexible in the flow direction.

3.2 Improved Assessment of Tube-Support Effectiveness

In many nuclear heat exchangers and steam generators, such as the proposed ACR-1000 steam generator (Klarner et al., 2006), the U-tubes are supported by flat bars in the plane of the U-bend. The diametral clearance between tube and intermediate support, necessary for assembly and thermal expansion, is typically 0.25 to 0.80 mm (for many nuclear heat exchangers, the diametral clearance is specified to be 0.38 mm, or 0.015 in.). In steam generators, industry now claims to be capable of achieving a nominal tube to flatbar diametral gap of 0.05 mm / 0.002 in. (Klarner et al., 2006), in an effort to further reduce the possibility of damaging flow-induced vibration. This trend towards smaller clearances is predicated on the assumption that this will lead to lower levels of vibration, in turn leading to lower levels of fretting-wear.

The flatbar design restrains excessive motion in the "out-of-plane" direction, i.e., the direction perpendicular to the plane of the flat surface of the bars, offers low hydraulic resistance, and has "line contact support" to prevent entrapment of vapour and minimize build-up of contaminants and fouling. However, the design does allow some motion in the out-of-plane direction, particularly if there are locations where the clearances are larger than specified, and offers relatively low resistance to movement in the so-called "in-plane" direction along the flat surface of the bars.

With this in mind, when assessing the effectiveness of flatbar-type tube supports, two issues emerge. First, is out-of-plane vibrational motion, which usually occurs at relatively low frequency and is more susceptible to fluidelastic instability, sufficiently arrested so that tubes with one ineffective support will not become unstable under normal operating conditions? Second, is in-plane vibrational motion sufficiently resisted that the tube will not undergo excessive fretting wear when in contact with flatbar supports?

The interaction between tube and supports is complex and non-linear, and makes the development of realistic numerical models difficult. Improvements to tube-to-support impact models are particularly needed, and there is some recent progress in this area (e.g., Hassan et al, 2005). Several other models have been developed to calculate the dynamic response of tubes with clearance supports, but few experimental results are available to compare to their predictions. AECL's early results with their fretting-wear methodology using the VIBIC code still represent the state of the art (Fisher et al., 1989) but do not represent a comprehensive solution to the general issue of effective supports under a wide range of geometries and conditions.

This situation suggests the need for further modeling and, in particular, further experimental work aimed at understanding the fundamental properties of tube-to-support interactions. Nowlan et al. (2008) describe a set of experiments that has recently begun with a single tube clamped at both ends and with a clearance support. Figure 8 shows the test rig including top and bottom supports (Items 1 and 4), an electromagnetic shaker assembly (2) and a flatbar displacement assembly (3), mounted on an I-beam (5). Random turbulence and quasi-periodic excitation forces are simulated with non-contacting electromagnets. With this experimental setup, the clearance, alignment, preload and orientation of the support can be accurately adjusted and measured to determine the effect of these parameters on tube-support effectiveness.

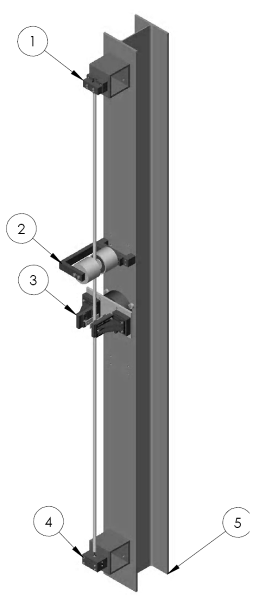


Figure 8 Experimental setup to measure the dynamic properties of a single tube supported by flatbars (Nowlan et al., 2008).

Preliminary results show that the flatbar supports redistribute the dynamic energy to higher vibration modes. Negative clearance (addition of a pre-load) increases the contact forces and decreases the acceleration amplitude, and positive clearance decreases the contact forces and increases the acceleration.

These results can be compared to measurements of tube-to-support work-rate with semi-circular U-tubes in air-water cross flow (Janzen et al., 2005). In those tests, at void fractions 50% and higher, random turbulence largely dictated the tube response and the work-rate was influenced more by impact rate than by impact-force magnitude. When random-turbulence effects dominated, larger clearances actually led to lower work-rates. When fluidelastic effects were clearly present, larger clearances led to higher impact forces and higher work-rates.

Although Nowlan et al. (2008) did not calculate the work-rate during contact between tube and support, their observed decrease in measured contact force for increased clearance may be consistent with the results of Janzen et al. (2005), for situations in which random-turbulence effects dominated the tube FIV response. The two experiments are not directly comparable since one is a bench-top test with a well-defined forcing function and zero or small clearance (0.045 mm / 0.002 in.), the other a flow-induced vibration test with relatively large clearances (0.75 mm / 0.030 in. and 1.5 mm / 0.060 in.).

A further complication in industrial heat exchangers and steam generators is that the issue is likely not so much the work-rate at a particular support (e.g., with larger than average clearances), as it is the work-rate at neighbouring supports. These neighbouring supports may bear the brunt of the enhanced tube-vibration amplitudes caused by one or more overly loose supports. From a numerical modelling point of view, this makes the problem difficult to address.

Experimentally, this issue needs to be addressed by two complementary types of measurements. First, tests that probe the fundamental characteristics of tube-to-support interaction, and allow fine control over variables such as clearance, alignment, preload and orientation of the support. Along these lines, further tests are planned that will build on the work of Nowlan et al. (2008) to fully assess the effectiveness of tube supports when varying these experimental parameters.

Second, tests in flow under more realistic conditions are needed. AECL's early U-bend tests (Janzen et al., 2005) used air-water, which is much more turbulent than steam-water or Freon, relatively large tube-to-support clearances (0.75 mm / 0.030 in. and 1.5 mm / 0.060 in. diametric), and a single mid-span support. Verification of an effective support design can only be addressed by vibration and work-rate measurements in a more realistic U-bend test section, with multiple supports and representative flow conditions. Such tests with a U-tube bundle in two-phase Freon flow are in the preparation phase.

4. CONCLUDING REMARKS

The necessity to avoid vibration and fretting-wear problems requires design specifications that bring together specific guidelines, analysis methods, requirements and appropriate performance criteria. In this paper, we have outlined the steps required to generate, support and improve flow-induced vibration specifications, and summarized issues that need to be addressed in order to improve existing specifications.

AECL's FIV specifications are updated on a regular basis, as more information becomes available from the field or predictive models improve, to meet the needs of the nuclear power industry. Past recent changes included improved flow calculation methods, more accurate boundaries on fluidelastic instability, and more realistic bounds on two-phase damping.

The present paper has focused on current issues that are driving changes to FIV specifications, changes that will improve the design process for component refurbishment, replacement or new designs. The current cycle of changes reflects expected improvements in computational (modelling) capabilities and recent experimental information. Specific issues discussed in this paper include:

- Evidence for new excitation mechanisms, e.g., the possibility of in-plane fluidelastic instability of U-tubes, presented as an example of how the FIV specifications are updated with new information, and quasi-periodic forces in tube bundles.
- The demand for longer reactor and component lifetimes.
- A need for better predictions of dynamic properties and vibration response, e.g., two-phase damping and two-phase random-turbulence excitation.
- Requirements to consider system "excursions" or abnormal scenarios, e.g., a main steam line break in the case of steam generators, and the accompanying demands on predictive computer codes.

In each case, the paper has listed steps being taken by AECL and its partners to resolve these issues. These steps include improvements in computational (modelling) capabilities, e.g., upgrades to AECL's PIPO-FE and VIBIC codes. For these upgrades, there are the associated issues of verifying the code and validating the code predictions. As a general statement, and for the issue of RTE formulation in particular, the industry would benefit from a move towards a common set of verification and validation cases that can be used to benchmark commonly used codes.

AECL and its partners are also taking steps to obtain additional experimental information of two types: fundamental bench tests of tube-to-support dynamics, and tests in two-phase flow with a realistic multi-span U-bend tube bundle. Apart from providing parameters for the FIV specifications, the results will be used to validate updated FIV models.

In the end, the goal is to provide designers or suppliers of CANDU heat exchangers and steam generators with design specifications ensuring that tube-vibration levels will be below allowable levels, and that the life of the component will not be limited by excessive fretting-wear due to flow-induced vibration.

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