

FLOW-INDUCED VIBRATION AND IMPACT FRETTING-WEAR IN NUCLEAR STEAM GENERATORS AND HEAT EXCHANGERS¹

by

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ABSTRACT

Mechanical damage due to excessive vibration may limit the life of steam generators and heat exchangers and may cause costly station shutdowns. It is important to understand flow-induced vibration and damage mechanisms to prevent problems at the design stage and to assist station owners in predicting the life of heat exchange components. This paper outlines state-of-the-art technologies in the related areas of thermal-hydraulic analysis, flow-induced vibration and mechanical damage prediction.

Vibration problems often occur locally in areas where excessive flow velocities exist, such as inlet regions and around sealing strips in heat exchangers. Detailed three-dimensional thermal-hydraulic analyses are required to obtain flow velocity distribution in components. Recent development work to predict flow velocity distribution around sealing strips and in the U-bend tube region of steam generators is described.

Vibration excitation mechanisms must be formulated to predict the vibration response of heat exchange components. Turbulence-induced excitation is important in axial-flow. In cross-flow, fluidelastic instability is usually the cause of excessive vibration. While a reasonable amount of information is available in liquid and gas flow, very little work has been done in two-phase flow. The results of recent experimental work on vibration of tube bundles in two-phase cross-flow is presented. Damping, fluidelastic instability and turbulence-induced excitation are discussed. Design guidelines are suggested.

Damage mechanisms must be understood to predict component life. Studies on fretting-wear damage of steam generator materials under simulated operating environments are discussed. Techniques to model the dynamic interaction between vibrating tubes and their supports are outlined.

As an example, the effect of increased tube-to-support clearances due to chemical cleaning on fretting-wear damage will be discussed.

INTRODUCTION

High component reliability is crucial to the successful operation of nuclear power stations. Component failures are very undesirable in terms of safety and lost production. This paper presents an overview of the technology required to avoid component failures due to flow-induced vibration. Nuclear component includes steam generators, heat exchangers, condensers, piping systems, reactor internals and nuclear fuels.

¹ Submitted for presentation at the Canadian Nuclear Society Conference on Steam Generators and Heat Exchangers, Toronto, Ontario, 1990 April.

Fig. 1: Tube-to-Tube Fretting-Wear Damage in Early Steam Generator Design

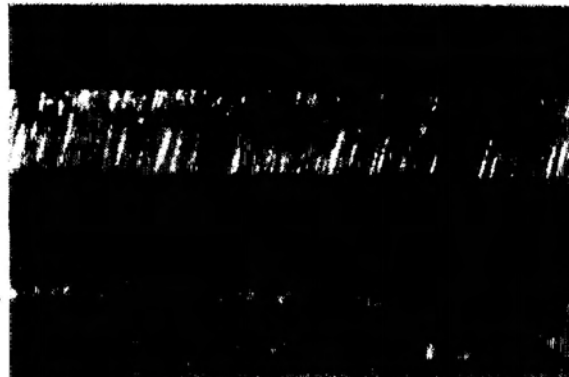
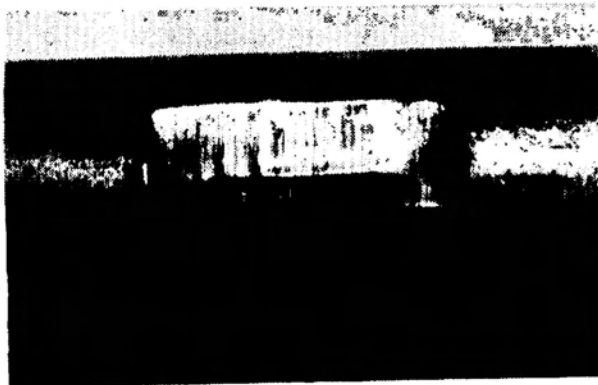


Fig. 2: Heat Exchanger Tube Failure due to Fretting-Wear at Support Locations



Fig. 3: Fatigue Failure of Titanium Condenser Tube

Nuclear components are often subjected to high flow velocities to increase heat transfer performance or reduce component size. As well, it is sometimes desirable to minimize structural support to reduce pressure drop or improve neutron economy. The combination of high flow velocity and reduced structural support could lead to excessive flow-induced vibration.

Typical problems related to excessive vibration are impact-fretting-wear, fatigue, operational difficulties and acoustic noise. For example, in shell-and-tube heat exchangers, vibration may cause tube failures by fatigue or by fretting-wear. Fig. 1 shows tube-to-support fretting-wear damage. This occurred in one steam generator of a prototype nuclear power station. The tubes in the U-bend region vibrated with sufficient amplitude to cause tube-to-tube fretting-wear. The problem was attributed to inadequate support combined with high two-phase cross-flow velocities. A first tube failure occurred in 1971 and a second in 1977, some ten years after the initial start-up of the station. This shows that fretting-wear can be a slow but continuous damage mechanism.

An example of tube-to-tube support fretting-wear is illustrated in Fig. 2. This failure took place in the inlet region of a process heat exchanger. Obviously, the tubes were inadequately supported for the high flow velocities in the inlet region.

Fig. 3 shows a classical fatigue failure of a titanium condenser tube. This problem was largely due to abnormal operation of the condenser, resulting in higher than normal steam flow velocities.

Many other problems have been encountered such as excessive vibration of control absorber guide tubes, fatigue failures of PWR¹ core barrel tie rods and in-core instrumentation nozzles,

¹ PWR: Pressurized Water Reactor

excessive acoustic noise due to control valve dynamics and mechanical damage resulting from acoustic resonance. Although most vibration problems have very costly consequences, they are usually solved by simple design modifications or changes in operating conditions. After the fact, it is easy to see that most problems could have been avoided by proper understanding of flow-induced vibration phenomena. Thus, it is important to understand flow-induced vibration and damage mechanisms to prevent problems at the design stage and to assist station operators in predicting the life of nuclear components. This paper outlines state-of-the-art technologies in the related areas of thermal-hydraulic analysis, flow-induced vibration and mechanical damage prediction.

FLOW CONSIDERATIONS

Definition of the Problem

Many flow situations are possible in CANDU¹ or PWR nuclear power stations. This is illustrated by a few examples. In the reactor core, axial flow generally prevails along the fuel elements. In PWR and earlier CANDU stations, the flow remains liquid throughout the fuel channels. In BWR² and later CANDU stations, the coolant is allowed to boil, and downstream fuel elements and outlet piping are subjected to two-phase flow.

Several flow situations are possible in recirculating-type steam generators. On the shell side, the tubes are subjected to liquid cross-flow in the preheater region and in the recirculated water entrance region near the tubesheet. Within the tube bundle, the shell side flow is mostly axial. It is liquid at the bottom and gradually becomes two-phase as boiling takes place. Two-phase cross-flow is predominant in the U-bend region where the steam quality reaches 15-25% or even higher in some designs. This corresponds to void fractions in excess of 80%.

The high pressure steam produced by the steam generators flows at high velocity in steam lines to reach the turbine and eventually the condenser, which operates at subatmospheric pressures. The condenser is essentially an enormous heat exchanger whose tubes are exposed to high velocity steam cross-flow. There are many other heat exchangers and reactor internals in nuclear stations which are subjected to cross-flow, particularly near inlets and outlets where flow velocities are generally high.

From a flow-induced vibration point of view, nuclear components are essentially cylindrical structures (i.e., piping, fuel channels) or bundles of cylinders (i.e., fuel bundles, tube bundles) subjected to axial or cross-flow. The flow may be internal or external to the cylinders. It may be confined in narrow annuli or in close-packed bundles. The flow may be adiabatic or diabatic where boiling or condensing takes place. Liquid, two-phase (steam-water), and vapour or gas flows are possible. These cylindrical structures are often multispans and supported in several places. For example, heat exchanger tubes are supported by baffle-plates, fuel elements by bearing pads and support grids, and piping systems by piping supports.

Flow Calculations

The first step in understanding flow-induced vibration is to assess the flow conditions. When the flow paths are well defined, this is usually simple. However, the geometry of many nuclear components is complicated. A detailed knowledge of local flow conditions is required for flow-induced vibration analysis since problems occur mostly in regions where high flow velocities exist. Examples of such regions are component inlets and outlets, tube-free lanes and gaps between sealing devices and tube bundles in heat exchangers, and U-bend regions in steam generators.

¹CANDU: CANada Deuterium Uranium (Registered Trademark)

²BWR: Boiling Water Reactor

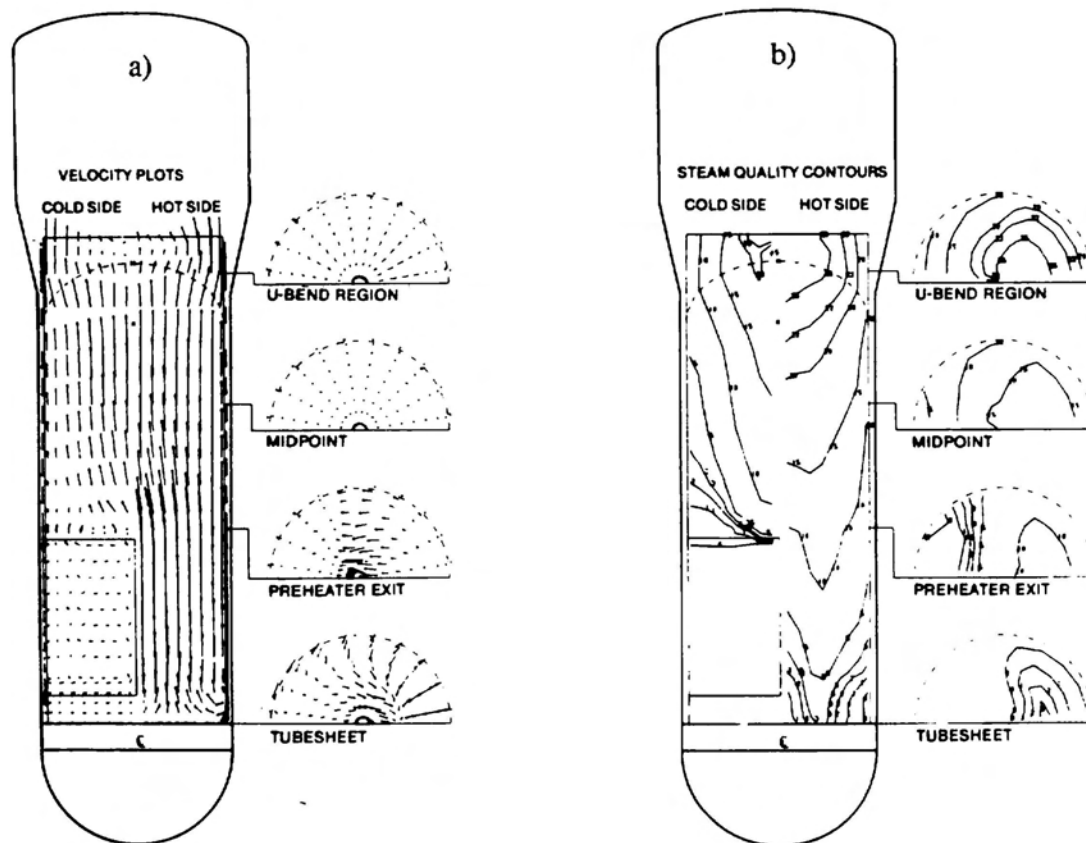


Fig. 4: Steam Generator Thermal Hydraulics Analysis

Conventional procedures for the assessment of the thermal and hydraulic performance of heat exchange components are usually based on simple methods such as the stream analysis technique. While such methods are sufficiently accurate for the evaluation of overall parameters such as heat transfer and pressure drop, they are inadequate for the prediction of local flow conditions. Thus, computer-based multi-dimensional calculation techniques are required to model the thermal-hydraulic behaviour of nuclear components to provide detailed flow information in complicated geometries. This information is also needed to identify stagnant zones, which are prone to corrosion and fouling, and contribute little to heat transfer performance. Consequently, we have developed numerical models to predict the thermal-hydraulic behaviour of steam generators [1,2], heat exchangers [3], condensers [4] and the CANDU reactor core vessel [5].

The computer code THIRST [2] was developed to analyze recirculating steam generators. This code solves the steady, three-dimensional shell-side equations of momentum, continuity and energy together with the tube-side energy equation. The code employs the homogeneous as well as the drift-flux models of two-phase flow. Tubes, baffles and other internal obstacles are treated using porous media concepts. These boundaries are described by a local isotropic porosity factor and fluid-solid source terms. The governing equations are solved numerically by iteration using an implicit finite control volume formulation.

THIRST predicts three-dimensional distributions of velocities, fluid and wall temperatures, heat fluxes, steam qualities, etc. The code has been used to assess the thermal-hydraulic performance of a wide range of steam generator designs used in nuclear power plants (for example, see [6] and [7]). Typical velocity and steam quality distributions predicted for a steam generator with a built-in economizer are shown in Fig. 4.

Thermal-hydraulic analyses using THIRST give an overall representation of the flow conditions in steam generators. These results are used as input to flow-induced vibration analyses.

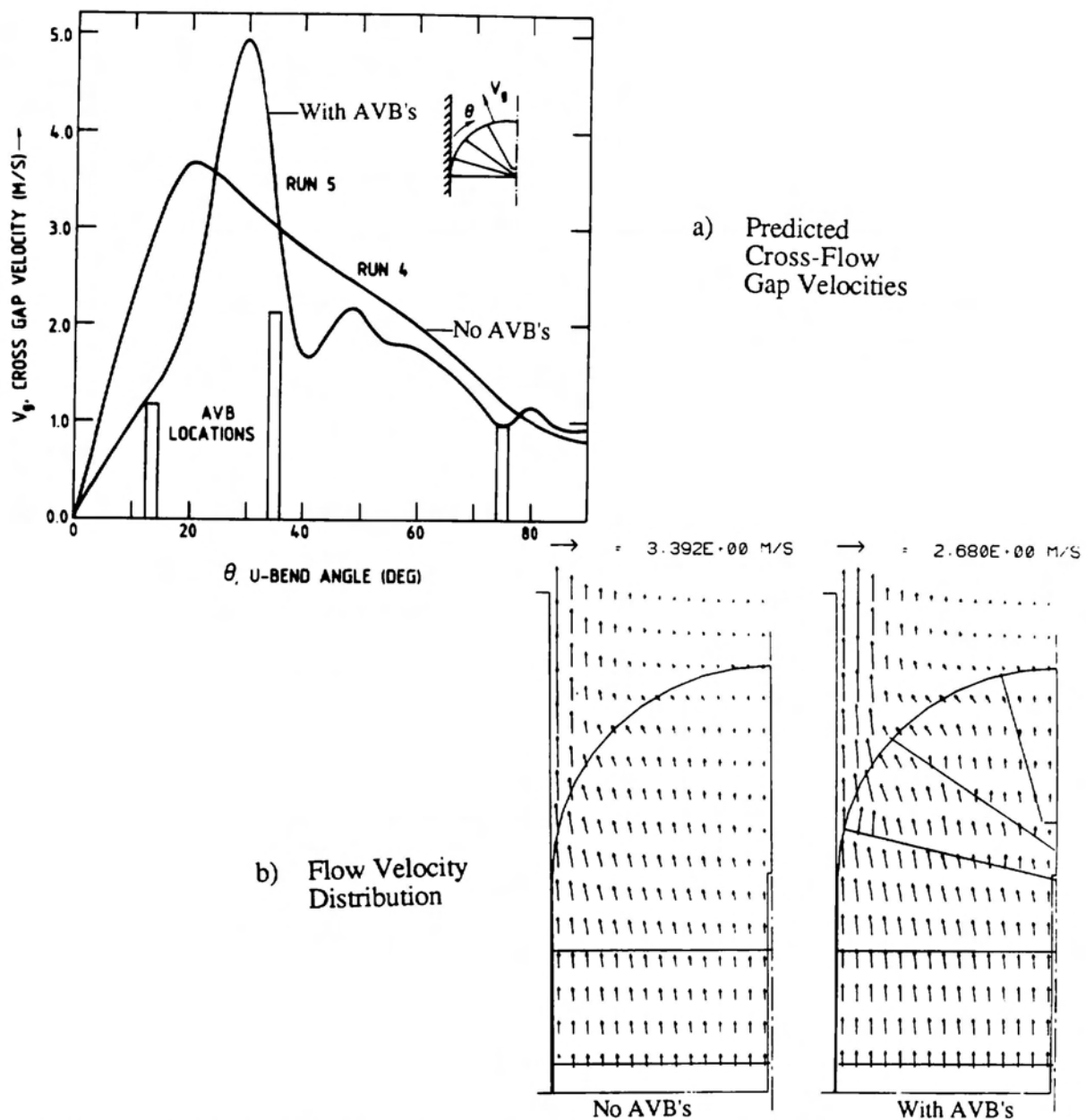


Fig. 5: Effect of Anti-Vibration Bars on Flow Velocity Distribution in U-bend Region of a Typical Steam Generator

Sometimes, it is desirable to focus on a particular region of a steam generator to get a more detailed understanding of the flow in that region. An example is the U-bend region of steam generators where the flow is generally quite non-uniform. This is due to non-uniform heat loading between hot and cold side, the anisotropic hydraulic resistance characteristics of the tube bundle and the presence of tube supports. These so-called anti-vibration bars or AVBs are a source of relatively high hydraulic resistance.

The U-bend region of a typical steam generator was analyzed in detail to investigate in particular the effect of AVBs on flow distribution [8]. For this analysis, the two-phase mixture is modelled as a homogeneous pseudo-fluid with mixture properties based on the mean mixture composition in the U-bend region. To improve the geometrical modelling of the AVBs, and to obtain greater detail in the flow distribution, only a vertical slice through the U-bend region of the steam generator was considered. Typical results are outlined in Fig. 5a and 5b. They show that the AVBs have a significant effect on flow distribution.

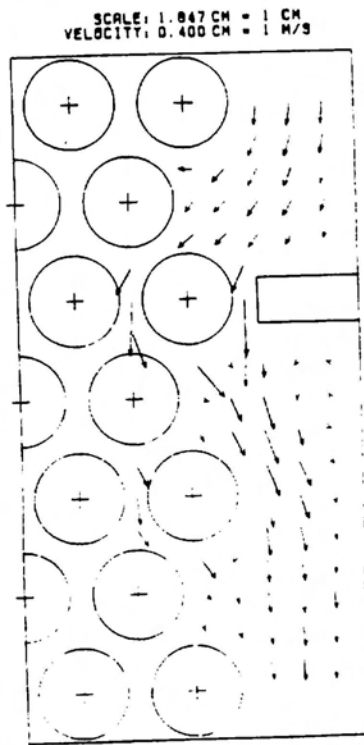


Fig. 6: Maximum Flow Velocity near Sealing Strips in Hypothetical Heat Exchanger Configuration

The above thermal-hydraulic analysis techniques have been applied to other heat exchanger problems [3, 4, 9]. Particular attention has been given to modelling complicated tube bundle geometries, inlet and outlet flow distribution, and the effect of sealing devices. Sealing devices such as sealing rods and sealing strips are often used in tube-free lanes to force the flow through the tube bundle. High flow velocities are usually found near sealing devices. This situation is illustrated in Fig. 6, which shows detailed flow velocity calculations around a typical sealing strip configuration. The much higher flow velocities are very undesirable from a vibration point of view.

Such a problem occurred a few years ago in a power station. High flow velocities around sealing strips limited a moderator heat exchanger to 60% of its design rating because of the possibility of excessive vibration. Two-dimensional flow calculations were carried out with the computer code STEFAN [9] to optimize the location of the sealing strips in an attempt to solve this problem. This is shown in Fig. 7. The optimized design allows nearly 100% heat transfer duty within recommended vibration guidelines. The improved sealing strip configuration (Fig. 7b) forces more flow through the tube bundle while reducing the maximum flow velocities around the sealing strips.

VIBRATION EXCITATION MECHANISMS

Overview

Dynamic forces are generated by fluid flow in nuclear components causing them to vibrate. Generally, four flow-induced vibration excitation mechanisms are relevant in nuclear components, namely: 1) fluidelastic instability, 2) periodic shedding, 3) turbulence-induced excitation, and 4) acoustic resonance. The relative importance of these mechanisms for different flow situations is outlined in Table 1.

Fluidelastic instabilities result from coupling between fluid-induced dynamic forces and the motion of structures. Instability occurs when the flow velocity is sufficiently high so that the energy absorbed from the fluid forces exceeds the energy dissipated by damping. Fluidelastic instability usually leads to excessive vibration amplitudes. The minimum velocity at which instability occurs is called the critical velocity for fluidelastic instability.

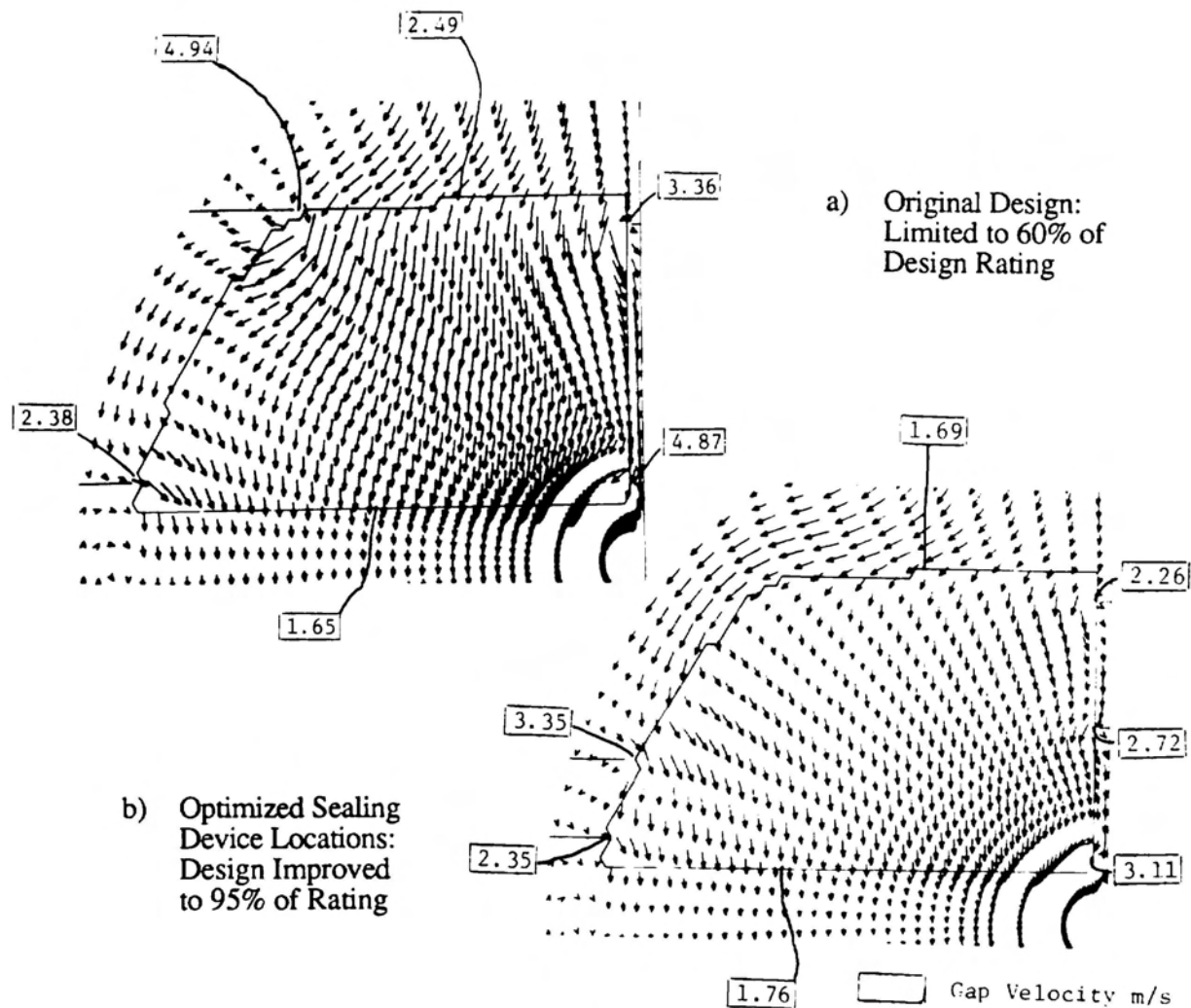


Fig. 7: Detailed Flow Velocity Distribution in Quarter Section of Moderator Heat Exchanger

Fluidelastic instability is usually not a problem for nuclear components in axial flow [10]. The flexural rigidity of components such as piping and fuel elements is relatively large. Thus, flow velocities are normally much lower than those required for instability.

On the other hand, fluidelastic instability is the most important vibration excitation mechanism for tube bundles in cross-flow. It is equally important for liquid, gas and two-phase cross-flow. The latter will be discussed in more detail further in this paper.

Periodic wake shedding often occurs immediately downstream of structures subjected to cross-flow. Periodic wake shedding generates periodic fluid forces. If the shedding frequency coincides with a natural frequency of the structure, resonance may occur. This may be a problem if the vibration response is large enough to control the mechanism of wake shedding. Then the periodic forces become spatially correlated to the mode shape and large vibration amplitude may result.

For an isolated cylinder in cross-flow, periodic wake shedding is a classical phenomenon called Karman vortex shedding. What happens in a closely packed bundle of cylinders is not so well understood. However, some periodic wake shedding resonances are possible in tube bundles [11]. In either case, excessive vibration amplitude can occur, particularly in liquid flows where the periodic forces generated are relatively large due to the high density of liquids. Conversely, large amplitudes due to wake shedding resonance are seldom a problem in gas cross-flow.

TABLE 1: VIBRATION EXCITATION MECHANISM

FLOW SITUATION	FLUIDELASTIC INSTABILITY	PERIODIC SHEDDING	TURBULENCE EXCITATION	ACOUSTIC RESONANCE
AXIAL-FLOW				
<u>Internal</u>				
Liquid	°	°	*	√
Gas	°	°	√	*
Two-Phase	°	°	*	°
<u>External</u>				
Liquid	°	°	*	°
Gas	°	°	√	√
Two-Phase	°	°	*	°
CROSS-FLOW				
<u>Single Cylinders</u>				
Liquid	°	*	*	°
Gas	°	√	√	°
Two-Phase	°	°	*	°
<u>Tube Bundle</u>				
Liquid	*	√	√	°
Gas	*	°	√	*
Two-Phase	*	°	*	°

° Unlikely √ Possible * Most important

Vibration excitation may be induced by turbulence. Turbulence can be generated locally by the fluid as it flows around the component of interest. This is called near-field excitation. Alternately, far-field excitation can be generated by upstream components such as valves, elbows and other piping elements. Turbulence-induced excitation generates random pressure fluctuations around the surface of components forcing them to vibrate. Turbulence-induced excitation is the principal vibration excitation mechanism in axial flow.

Turbulence-induced excitation is also important in cross-flow. While fluidelastic instability and periodic wake shedding may cause failures in a very short time, turbulence excitation may induce enough vibration response to cause long-term fretting-wear damage. Turbulence-induced excitation should be considered in both liquid and two-phase cross-flow.

Acoustic resonance is possible in tube bundles subjected to gas cross-flow. It occurs when the period wake shedding frequency coincides with the natural frequency of the acoustic cavity formed by the structures surrounding the tube bundle. The acoustic cavity resonance correlates shedding and causes intense acoustic noise, which can lead to severe structural damage.

Acoustic resonance can also occur in axial gas flow, for example, in the main steam lines of nuclear stations. Acoustic noise generated by piping elements such as valves can promote acoustic standing wave resonance in a receptive section of the piping system. If the acoustic resonance frequency is close to piping natural frequencies, large vibration amplitudes of the piping may occur.

It is beyond the scope of this paper to discuss all the above vibration excitation mechanisms in detail. We shall concentrate instead on recent work done on vibration of tube bundles subjected to two-phase cross-flow. Although little work has been done so far in this area, it is particularly relevant to nuclear steam generators.

Vibration in Two-Phase Cross-Flow

An understanding of damping and flow-induced vibration excitation mechanisms is necessary to avoid vibration problems in two-phase cross-flow. Accordingly, we have undertaken a comprehensive experimental program to study the vibration behaviour of tube bundles in two-phase cross-flows. Four cantilever tube bundles of different configurations were tested in support of our CANDU nuclear steam generator technology development program, namely: a normal-triangular (30°), a parallel-triangular (60°), a normal-square (90°) tube bundle, all of pitch over diameter ratio, p/d , of 1.47 and a normal-triangular tube bundle of $p/d = 1.32$ [12, 13, 14, 15]. These bundles were subjected to air-water mixtures to simulate realistic vapour qualities over a practical range of mass fluxes or flow velocities. The test section is shown in Fig. 8. Void fractions from 5% to 99% were simulated. The tubes were stainless steel, 13 mm diameter and had a natural frequency in air of 33 Hz, which is typical for steam generator tubes.

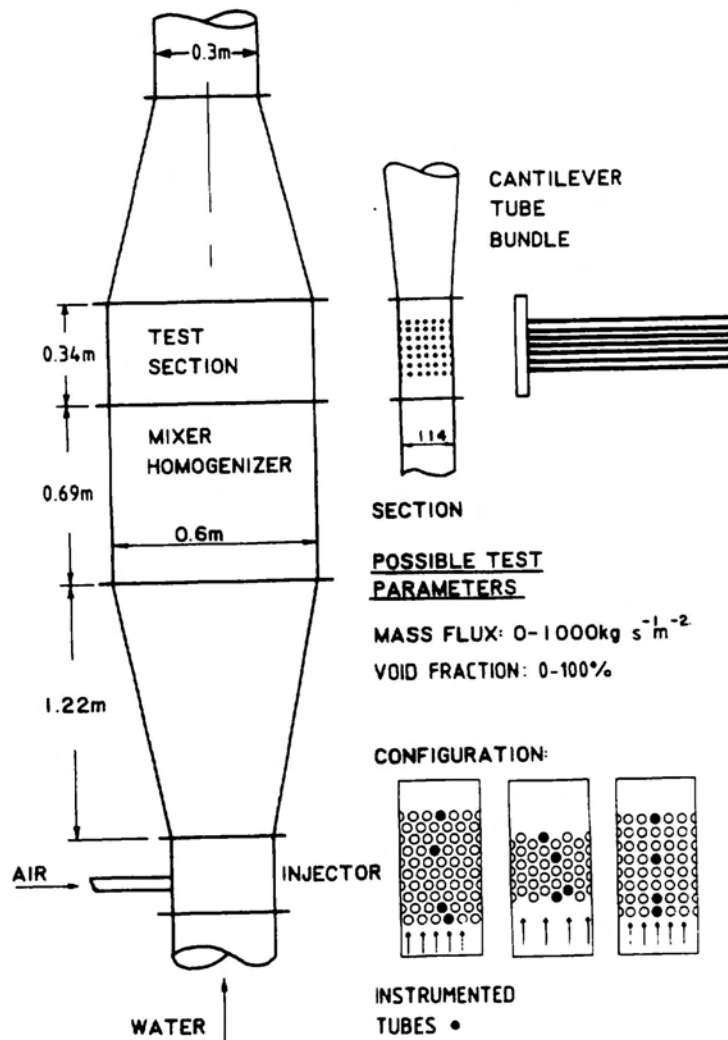


Fig. 8: Sketch of Test Section and Tube Bundle Configurations for Vibration Experiments in Two-Phase Cross-Flow

Upstream, interior and downstream tubes were instrumented. The tests were done with all the tubes in the bundle free to vibrate to study in particular fluidelastic instability. The tests were repeated with one instrumented tube surrounded by rigid tubes to obtain more reliable damping measurements.

Vibration excitation mechanisms and damping information were simply deduced from the vibration response. Each tube bundle was subjected to increasing mass fluxes until the vibration amplitudes were sufficiently high to indicate fluidelastic instability. During a given test, the void fraction was kept constant. We measured the tube vibration response to turbulence-induced excitation at mass fluxes below instability.

Fluidelastic instability may be formulated in terms of a dimensionless velocity, V_r/fd , and a dimensionless mass-damping term, $2\pi\zeta m/\rho d^2$ [16], such that:

$$V_r/fd = K(2\pi\zeta m/\rho d^2)^n \quad (1)$$

where f is the tube natural frequency in the fluid, d is the tube diameter, ζ is the total damping ratio, m is the mass per unit length including the hydrodynamic mass, and ρ is the homogeneous fluid density. The homogeneous flow velocity, V_r , is the so-called "pitch velocity" which is defined as $V_r = V_\infty p/(p-d)$ where V_∞ is the free stream flow velocity. It is also related to the pitch mass flux, $\dot{m}_r$, as $V_r = \dot{m}_r/\rho$. Knowing damping, the instability coefficient K and the exponent n can be deduced from the measured value of critical velocity for fluidelastic instability, V_{rc} .

The threshold for fluidelastic instability was often very clear, being characterized by a dramatic increase in vibration amplitude, by significant hydrodynamic coupling and by motion largely in the lift direction. The results for tube bundles of $p/d = 1.47$ are presented in Fig. 9. It shows two regions of instability. A region where the exponent n of the mass-damping parameter is

roughly 0.5, as expected from previous work. This takes place at void fraction, ϵ_g , below roughly 80-90%. There appears to be another instability region at void fractions above 80-90% where the exponent n is much lower, say around 0.1.

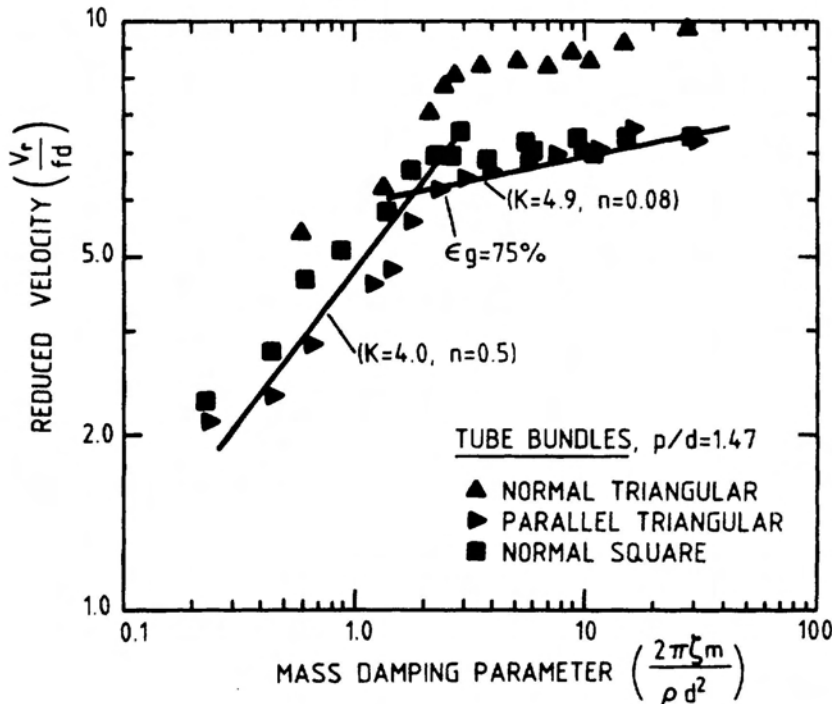


Fig. 9: Fluidelastic Instability of Tube Bundles of $p/d=1.47$ in Two-Phase Cross-Flow Showing Two Flow Regime Dependent Regions

The above behaviour is explained in terms of flow regime changes. The transition between the two fluidelastic instability regions corresponds to the transition between continuous flow and intermittent flow. In the continuous (bubbly or froth) flow region below 80-90% void fraction, the formulation of the fluidelastic instability mechanism is as expected, with a mass damping parameter exponent, n , around 0.5. Not surprisingly, the behaviour in the intermittent flow regime is quite different and instability may occur at much lower flow velocities. Thus, it is important to consider flow regime for flow-induced vibration studies in two-phase flow.

The above results, for continuous flows only, are compared with the data of Axisa et al. [17] for steam-water in Fig. 10. The agreement is remarkably good, particularly for the normal square bundle data. This suggests that dimensionless velocity and dimensionless mass-damping are reasonable parameters to formulate fluidelastic instability in two-phase cross-flow. It appears that this formulation is equally valid for air-water, steam-water and probably other liquid-vapour mixtures. Fig. 10 leads to a design recommendation for fluidelastic instability of the form

$$V_{rc} / fd = K \left(2\pi \zeta m / \rho d^2 \right)^{1/2} \quad (2)$$

where a minimum value of $K=3.3$ may be used as a general criterion and $K=4$ may be used for square bundles. This applies to tube bundle of p/d between 1.4 and 1.5 in continuous two-phase cross-flow (bubbly or froth liquid-vapour mixtures).

Information about turbulence-induced excitation may be deduced from the vibration response of the tube bundles at flow velocities below instability. It may be shown from random vibration theory that the mean square response, $\overline{y^2(x)}$, of a single span tube in the first mode of vibration excited by spacially correlated random forces, $S_F(f)$, may be expressed by:

$$\overline{y^2(x)} = \frac{\phi_1^2(x) S_F(f) J_1^2}{64\pi^3 f_1^3 m^2 \zeta_1} \quad (3)$$

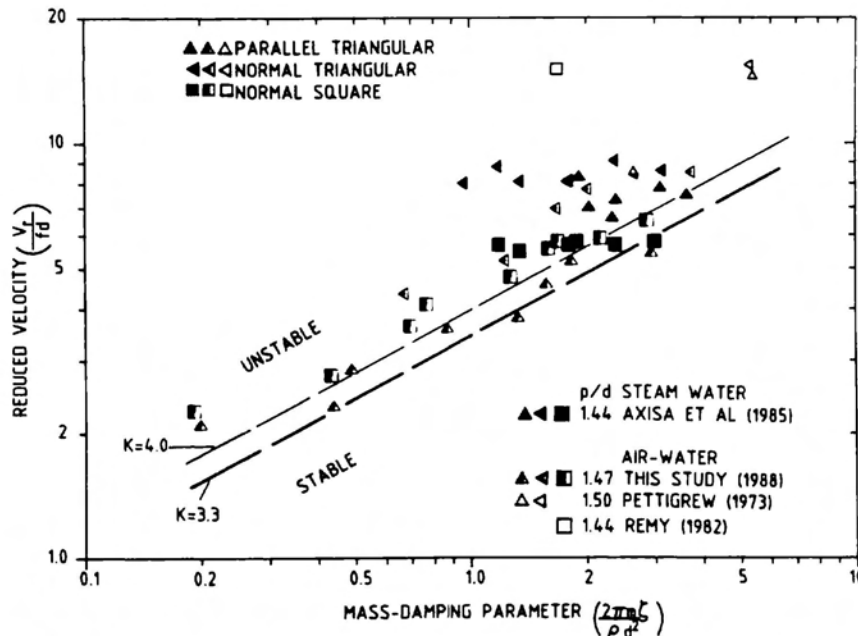


Fig. 10: Dimensionless Presentation of Fluidelastic Instability Results in Two-Phase Cross-Flow

where $\Phi_1(x)$ is the normalized mode shape of the first mode, $S_F(f)$ is the power spectral density per unit length of the excitation forces, and J_1^2 is the joint acceptance for the first mode. For a cantilever tube $J_1^2 = 0.613$. Thus, Eq. (3) may be used to deduce $S_F(f)$ from the measured vibration response, $\overline{y^2}(x)$.

The results of the above experiments and those of Taylor et al. [18], Axisa et al. [19] and Pettigrew and Gorman [20] were analyzed to identify the most important parameters governing turbulence-induced excitation in two-phase cross-flow. We found [14] that $S_F(f)$ 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 . Thus,

$$S_F(f) \propto \frac{\dot{m}_r^2 d^2}{L_e f^{1.25}} \quad (4)$$

While the other relationships are entirely empirical, Antunes [21] has analytically shown that, if the spatial correlation factor along the tube length is much less than unity, $S_F(f)$ will be inversely proportional to L_e . Eq. (4) was used to define a normalized power spectral density (Normalized PSD) such that:

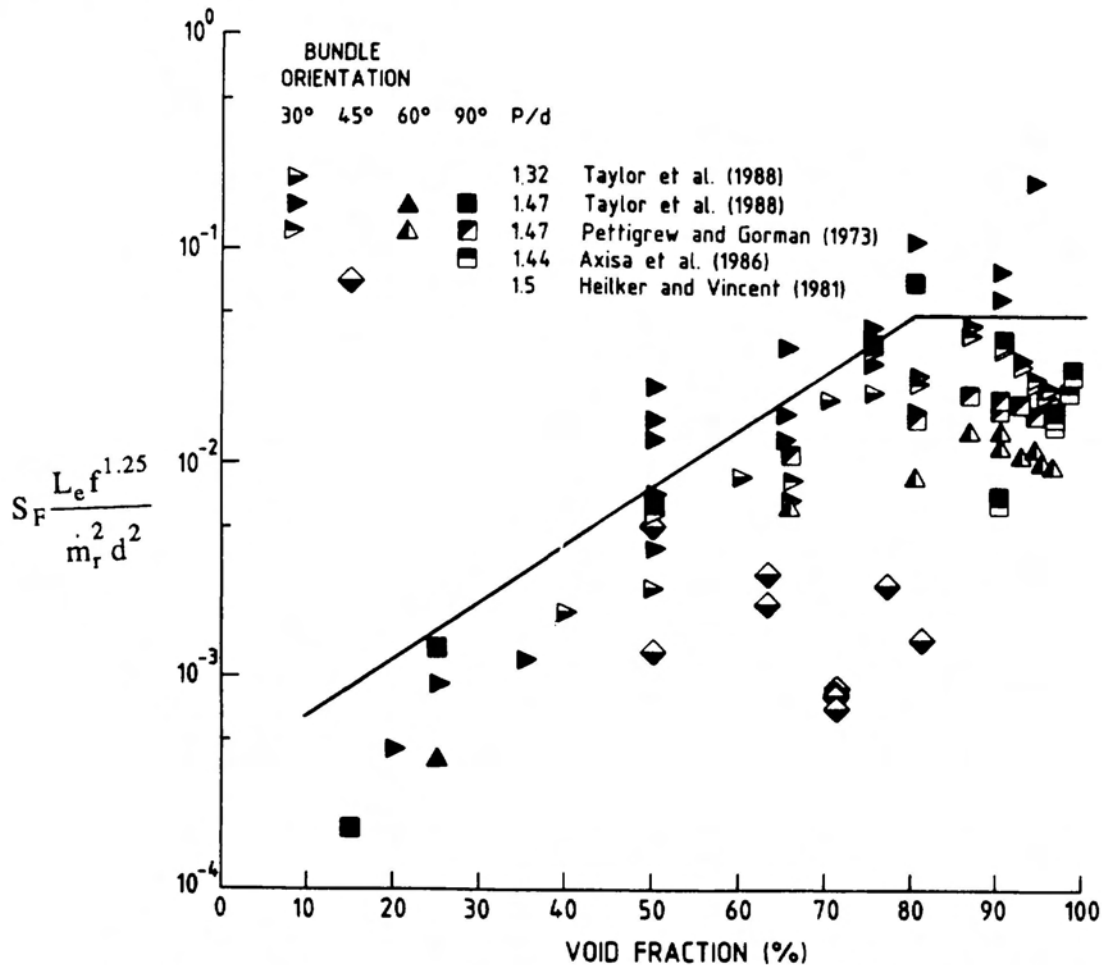


Fig. 11: Normalized Power Spectral Density of Turbulence-Induced Excitation for Tube Bundles in Two-Phase Cross-Flow

$$\text{Normalized PSD} = \frac{S_F(f) \cdot L_e f^{1.25}}{\dot{m}_r^2 d^2} \quad (4a)$$

The turbulence-induced excitation data in the form of the Normalized PSD is presented in Fig. 11 as a function of void fraction. Note that void fraction is a dominant parameter. The enveloping line of Fig. 11 may be used as a design guideline to calculate $S_F(f)$ with the help of Eq. (4a). This applies to air-water or to ambient temperature steam-water mixtures. Turbulence-induced excitation forces have been observed to be less in higher temperature steam-water mixtures.

DAMPING MECHANISMS

The understanding of damping mechanisms is as important as the knowledge of vibration excitation mechanisms. Unfortunately, the study of damping mechanisms in nuclear structures has been rather neglected until very recently. Damping is difficult to study. For example, damping of multispan heat exchanger tubes depends on both deterministic parameters such as tube frequency, tube-support thickness and fluid viscosity, and statistical parameters such as tube straightness and tube-support alignment. Damping is further complicated by non-linear effects and the multiplicity of energy dissipation mechanisms. This may explain why information on damping is still scarce.

There are several possible energy dissipation mechanisms that may contribute to damping of nuclear structures. These are: 1) internal or material damping, 2) viscous damping between the structure and the fluid, 3) squeeze-film damping between the structure and its supports, 4) flow dependent damping due to fluid flow around the structure, 5) damping due to internal flow, 6) friction damping between the structure and its supports, 7) energy dissipated by impact of the structure on its support and the resulting travelling waves, and 8) damping due to two-phase mixtures.

As an example, we shall discuss damping of multispan heat exchangers and steam generator tubes. We have recently conducted a review of heat exchanger tube damping in gases [22] and in liquids [23]. We found that the main damping mechanism in gas heat exchangers is friction between tube and tube-supports. Damping is strongly related to support thickness whereas diametral clearance between tube and support is much less important. The effect of other parameters such as tube frequency and diameter does not appear relevant. Damping of heat exchanger tubes in liquids and in two-phase flows is discussed in more detail below.

Damping of Heat Exchanger Tubes in Liquids

There are three important energy dissipation mechanisms that contribute to damping of heat exchanger tubes in liquids [23]. These are: viscous damping between tube and liquid, squeeze-film damping between tube and support and friction damping at the support. Thus, the total damping ratio, ζ_T , is:

$$\zeta_T = \zeta_V + \zeta_{SF} + \zeta_F = \zeta_V + \zeta_S \quad (5)$$

Where ζ_V is the viscous, ζ_{SF} the squeeze-film, and ζ_F the friction damping ratio. Both squeeze-film and friction damping take place at the support. Thus, support damping, $\zeta_S = \zeta_{SF} + \zeta_F$.

Tube to fluid viscous damping is reasonably well understood. Chen et al. [24] developed a comprehensive theory for viscous damping of a cylinder in confined viscous fluid. Rogers et al. [25] developed a simplified version which is valid for most heat exchanger parameters. This

simplified expression for ζ_v (in percent) is:

$$\zeta_v = \frac{100\pi}{\sqrt{8}} \left(\frac{\rho d^2}{m} \right) \left(\frac{2\nu}{\pi f d^2} \right)^{1/2} \left\{ \frac{[1 + (d/D_d)^3]}{[1 - (d/D_d)^2]^2} \right\} \quad (6)$$

where ν is the kinematic viscosity and D_e is the effective diameter of the surrounding tubes. D_e is related to p/d as discussed in [23] and [25]. Measured viscous damping values are compared to calculated values using Eq. (6) in Fig. 12.

The problem is to formulate support damping, ζ_s . Most of the available data on damping of multispan heat exchanger tubes in liquids were compiled in the form of a database [23]. These data were used to deduce values of support damping by subtracting calculated viscous damping from the total damping data. The resulting support damping values were analyzed to find trends. We found that support damping is dominant. Roughly 75% of the damping energy is dissipated at the support. It is attributed mostly to squeeze-film damping which is related to support thickness and is inversely proportional to tube frequency.

With the help of Mulcahy's theory [26], we have developed a semi-empirical expression to formulate support damping, ζ_s :

$$\zeta_s = \left(\frac{N-1}{N} \right) \left(\frac{2200}{f} \right) \left(\frac{\rho d^2}{m} \right) \left(\frac{L}{\ell_m} \right)^{0.6} \quad (7)$$

where N is the number of spans, L is the support thickness and ℓ_m is a characteristic span length which is usually the average of the three longest spans. This expression is compared to the available support damping data in Fig. 13. It shows that Eq. (7) is a reasonable formulation for the minimum support damping values. This is appropriate for design purposes. Thus, damping of multispan heat exchanger tubes in liquids may be estimated with Eqs. (5), (6) and (7).

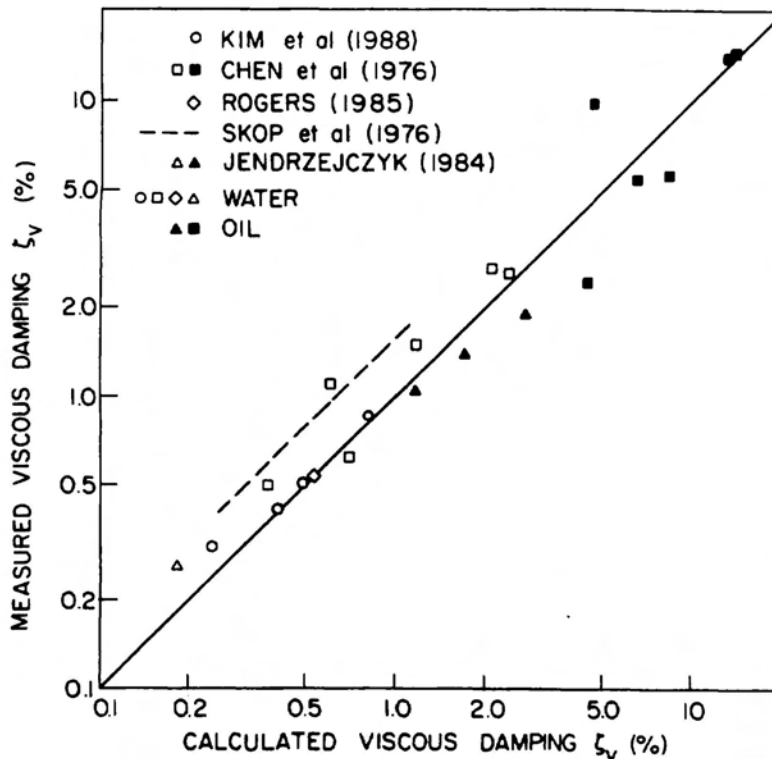


Fig. 12: Viscous Damping of a Cylinder in Unconfined and Confined Liquids: Comparison between Theory and Experiments

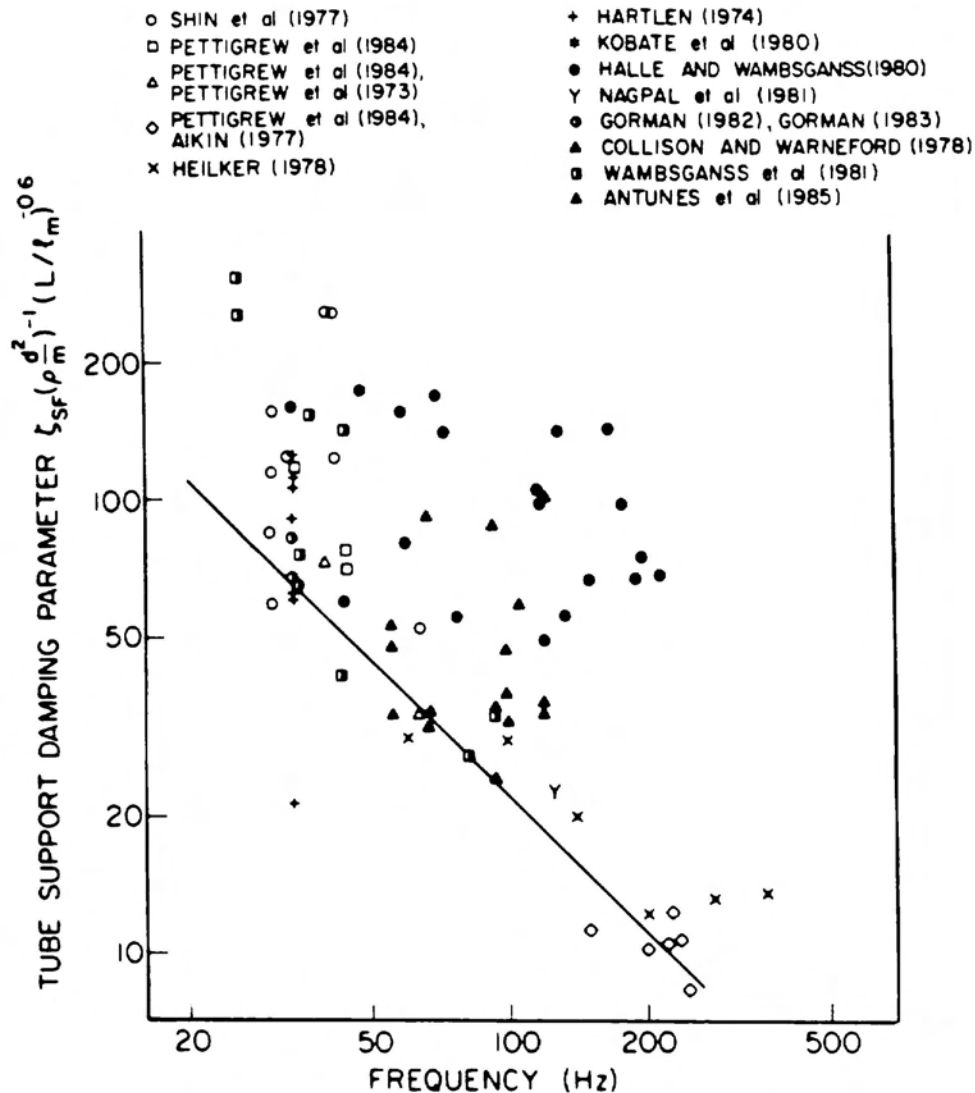


Fig. 13: Comparison between Tube Support Damping Parameter and Experimental Data for Multispan Heat Exchanger Tube Damping in Water: Lower Envelope Line Represents Design Guidelines

Damping of Heat Exchanger Tubes in Two-Phase Flow

The study of damping in two-phase flow is difficult from an experimental point of view. It is not possible to maintain a stagnant two-phase mixture. Thus, damping measurements must be taken under some flow. On the other hand, the flow must be lower than that required for fluidelastic instability to avoid the effect of so-called "negative damping". Damping in two-phase flow is further complicated by the various possible flow regimes and by an additional parameter which is void fraction.

We have investigated damping at the same time as vibration excitation mechanisms in the test section shown in Fig. 8 [12]. The measurements were taken on tube bundles with one instrumented flexible tube surrounded by rigid tubes. This was necessary to avoid hydrodynamic coupling between the tubes which tends to make the interpretation of the results difficult. We used the random vibration method to obtain damping values. This assumes that the turbulence excitation due to two-phase flow is essentially broadband random. Under these conditions, damping can be readily deduced from the frequency response spectrum.

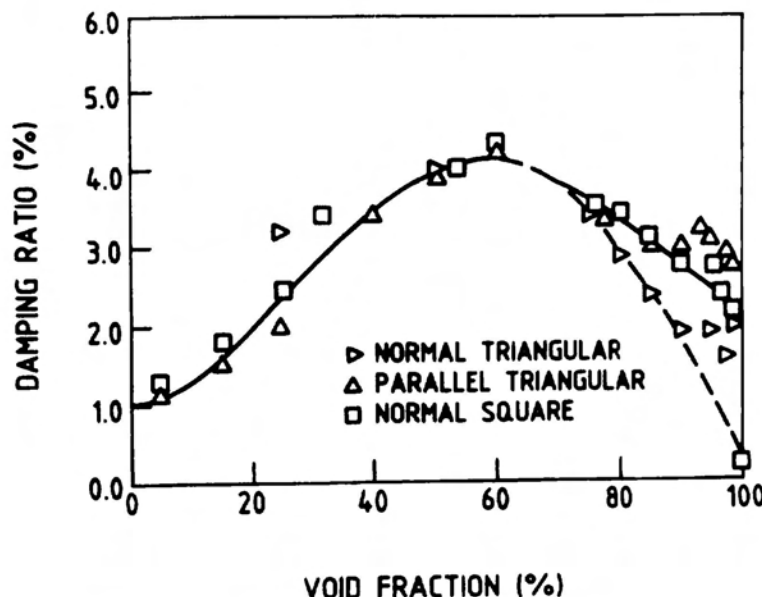


Fig. 14: Damping for Tube Bundles of $p/d=1.47$ in Air-Water Mixtures

Damping results for three tube bundles of $p/d = 1.47$ are shown in Fig. 14. Damping is relatively high in two-phase flow. The damping characteristics of all tube bundles are generally similar. The effect of void fraction is dominant. Damping is maximum between 40 and 80% void fraction where the damping ratio reaches about 4%. We found the effect of confinement (i.e.: p/d) and the effect of mass flux to be generally weak. It appears that tube frequency is not a dominant parameter.

Damping in two-phase mixtures is very different than in liquids. It should be taken into account in two-phase flow situations such as in the U-bend region of steam generators.

VIBRATION ANALYSIS

Procedures and calculation techniques have been developed to analyze nuclear components to avoid flow-induced vibration problems. Generically, these analyses consist of the following steps: (1) flow calculations, (2) formulation of flow-induced excitation forces, (3) definition of dynamic parameters (stiffness or flexural rigidity, mass including hydrodynamic mass and damping), (4) calculation of natural frequencies and mode shapes, (5) prediction of vibration response and of critical velocities for fluidelastic instabilities, (6) assessment of fatigue or fretting-wear damage and (7) acceptance or modification of design, if necessary.

For example, for nuclear steam generators and heat exchangers, we have developed an analysis method which is outlined in Fig. 15 [27]. From a mechanical dynamics point of view, heat exchanger tubes are simply multi-span beams clamped at the tubesheet and held at the supports with varying degree of constraints. The latter is dependent on the support geometry and particularly on the tube-to-support clearance. To predict the tube vibration response, we assume the intermediate supports to be hinged to keep the analysis linear. However, we consider the non-linear dynamics of tube to tube-support interactions to predict fretting-wear damage, as discussed later. The dynamics of multi-span tubes and the formulation of vibration excitation mechanisms form the basis of a computer code called PIPO, to predict the vibration response of heat exchanger tubes [28]. The program first calculates the mode shapes $\phi_i(x)$ and the natural frequencies f_i (i.e., eigenvalue solution). Then the vibration response and the critical velocities for fluidelastic instabilities are calculated.

The vibration analysis of the U-bend region of a typical steam generator is illustrated in Figs. 16 and 17. The U-bend region is subjected to two-phase cross-flow and supported by AVBs at several locations as shown in Fig. 16a. The flow calculations indicate a somewhat non-uniform

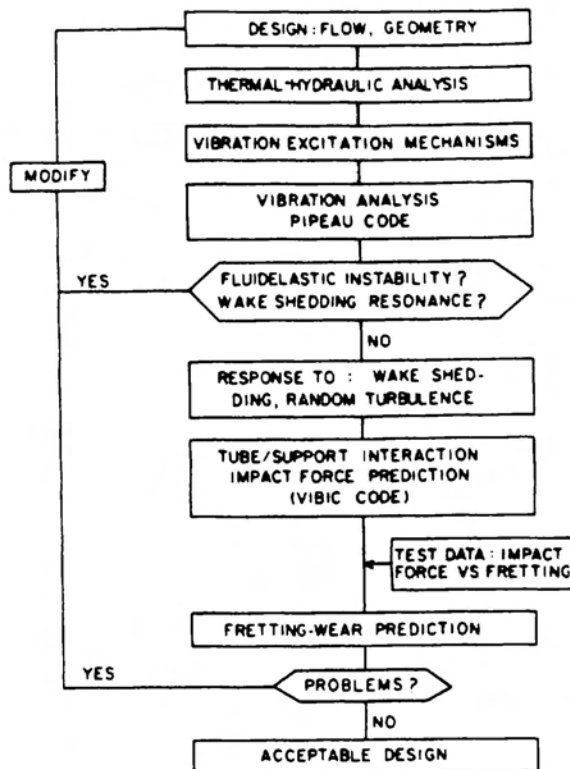
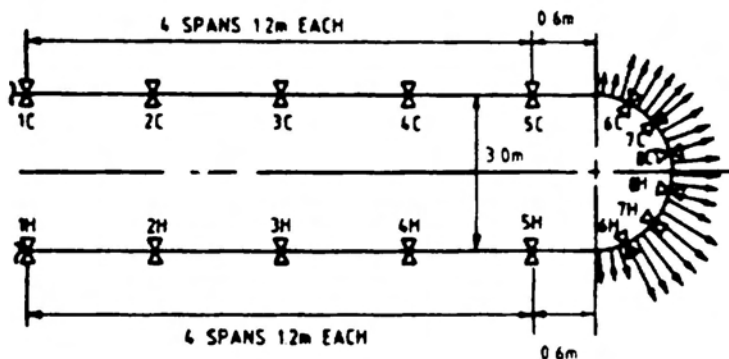


Fig. 15: Comprehensive Approach to Avoid Vibration Problems in Heat Exchangers

flow distribution in the U-bend region as expected because of thermal hydraulic interaction between hot and cold side and because of the presence of the AVBs (see Fig. 16)

The vibration analysis outlined in Fig. 17 shows that fluidelastic instability is not a problem in the U-bend region if all the AVBs provide effective support. The actual flow velocity is below the critical velocity for instability (i.e.: $V_r/V_{rc} = 0.37 < 1$, Fig. 17a). On the other hand, the same U tube would probably undergo fluidelastic instability if one support is ineffective.



(a) Tube Layout and Flow Velocities

(b) Computer Input Data:
Mixture Velocity and Density

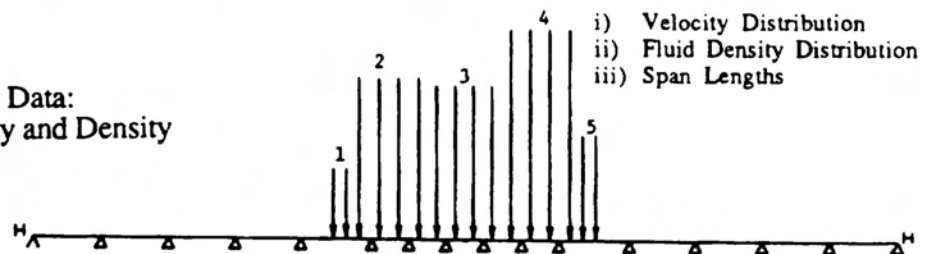


Fig. 16: Vibration Analysis of Typical Steam Generator U-bend Tube

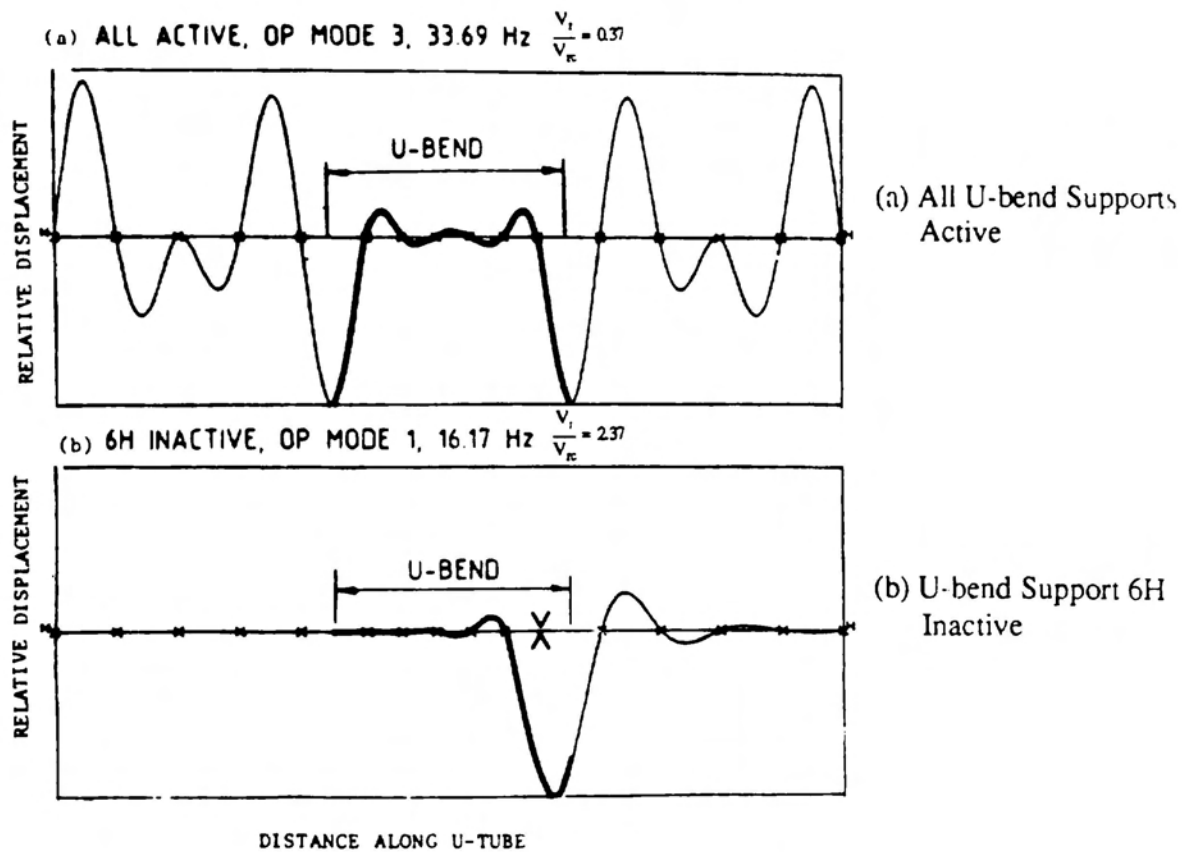


Fig. 17: Out-of-Plane Mode Shapes and Natural Frequencies Corresponding to Worst Predicted Instability Ratios

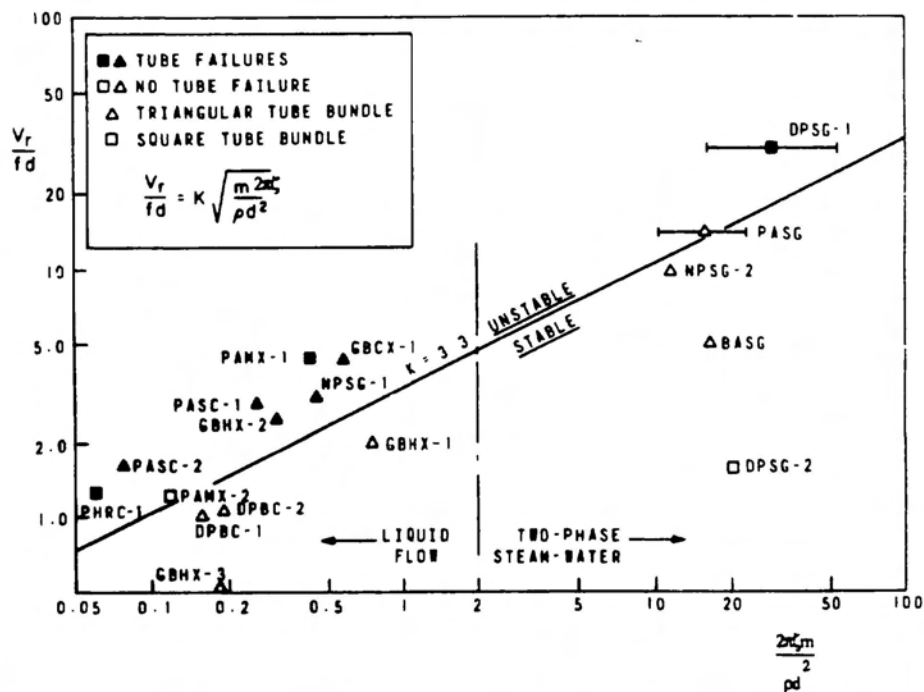


Fig. 18: Fluidelastic Instability Analysis of Field Experiences with Heat Exchangers: Comparison against Recommended Design Guideline $K=3.3$

(i.e., $V_r/V_{rc} = 2.37 > 1$, Fig. 17b). This example emphasizes the importance of providing adequate support in the U-bend region of steam generators [29].

A few years ago [30], we reviewed the operating performance of ten heat exchangers from a fluidelastic instability point of view. Both failures and satisfactory performances were considered. The results are presented in terms of dimensionless parameters in Fig. 18. All the tube failures occurred for values of fluidelastic instability constant K larger than 4.4, which is the equivalent of an actual velocity/critical ratio $V_r/V_{rc} = 1.33$. Thus, our recommended guideline of $K = 3.3$ appears very realistic in practice, as it gives a reasonable safety margin. The credibility of this criterion is further enhanced by the wide variety of components considered (i.e., heat exchangers, coolers and steam generators) and by the range of operating conditions encountered (i.e., liquid and two-phase flow). Some of the above tube failures could also be explained in terms of vibration excitation mechanisms other than fluidelastic instability. In most cases, it was not possible to identify positively the excitation mechanism from the field data. However, the main objective here was to validate design guidelines, not to obtain fundamental information about vibration excitation mechanisms.

By now, hundreds of heat exchangers have been operating for more than a decade in CANDU nuclear stations. To our knowledge, none of the heat exchangers that were designed according to the above guidelines have experienced vibration problems. On the other hand, all reported vibration related tube failures could be attributed to non-compliance with the recommended design guidelines.

IMPACT FRETTING-WEAR DAMAGE

Unfortunately, flow-induced vibrations cannot be eliminated altogether; vibration due to turbulence induced excitation always exists. "How long will a component survive?" or "How much vibration can be tolerated without excessive damage?" are relevant questions. To address such questions, predictions of impact fretting-wear damage are often necessary.

Impact fretting-wear damage results from heat exchanger tubes impacting and rubbing on tube supports or against adjacent tubes. The prediction of fretting-wear damage requires: 1) an understanding of the dynamic interaction between the tube and support, 2) experimental correlations of fretting-wear damage for relevant environments and material combinations, and 3) a realistic formulation of impact fretting-wear mechanisms.

Simulation of Dynamic Interaction Between Tube and Support

Analytical techniques are required to predict the dynamic interaction between a tube and its tube supports. Thus, we have developed a computer code called VIBIC to simulate the motion of a multispan tube considering the non-linearities due to clearances between tubes and tube-supports [31]. It is essentially a time domain numerical simulation of the dynamics of heat exchanger tube-to-support interactions. The equations of motion, written in matrix form, are

$$[M] \{\ddot{u}\} + [c] \{\dot{u}\} + [k] \{u\} = \{F(t)\} \quad (8)$$

where the number of equations equals the number of unconstrained degrees of freedom. The mass matrix, $[M]$, which includes rotary inertia terms, and the stiffness matrix $[k]$ which includes shear deformation are banded and symmetrical. The column matrix $\{u\}$ represents all the possible displacements and rotations, and $\{F(t)\}$ represents the known forcing functions. At each node the axial and torsional degrees of freedom are constrained.

During a vibration simulation, the modes (obtained from a modal analysis of the tube constrained only at the end) are numerically integrated and superimposed to provide the tube response. Each support is centered at a finite-element node. When contact at a support occurs, the normal force and sliding friction force are generated and applied to the tube. The procedure is equivalent to superimposing the constrained modes (where the tube is assumed hinged at the intermediate supports), and is valid for both no-clearance and clearance cases [32].

The added mass and damping of the fluid between the supports is also included in the finite-element model. The added mass is lumped at the nodes. Fluid damping can either be calculated and applied as a modal factor or modelled as a squeeze-film damping force and applied at the nodes.

Several experiments were conducted to verify this computer code. One of these experiments was carried out on a four-span heat exchanger tube [33]. Good agreement was obtained between calculated and measured dynamic forces at the supports.

Recently [34], we have studied in detail the motion of a tube within a tube support as shown in Fig. 19. Measurements and predictions show very similar features. They are not identical as expected, since this is not a deterministic phenomenon.

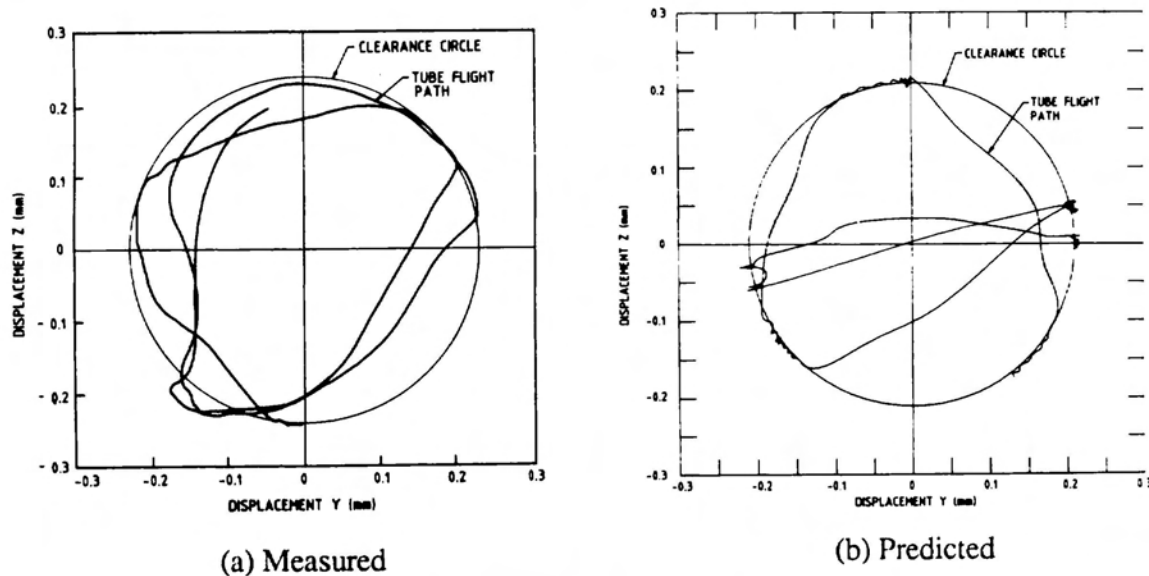


Fig. 19: Orbital Plots of Tube Motion at the Support

Impact Fretting-Wear Correlations

Correlations between the dynamic interaction at the support and impact fretting-wear are needed to predict damage. This is done experimentally in fretting-wear test rigs. A test rig consists of a single-span cantilevered tube which is vibrated against a tube support specimen (Fig. 20). Some test rigs are enclosed in autoclaves for operation at high temperature under controlled environments. Others are open to atmosphere for operation at room temperature. They are used for simple tests in air or water. Vibration amplitudes, frequency and impact forces are monitored during the tests. The fretting-wear damage is assessed by weight loss or wear depth measurements. We have conducted a large number of fretting-wear tests on different tube and tube support materials in the room temperature rigs. The results have provided some insight into fretting mechanisms as discussed in [35].

It is often necessary to know the fretting-wear behaviour under realistic operating conditions to make valid fretting damage predictions. We have seven fretting-wear autoclaves in operation for such tests (see Fig. 21). The autoclaves are connected to a pressurized loop to simulate and control realistic operating environments. The effects of temperature, oxygen content, pH, chemistry control additives and crud deposition on fretting-wear rates can be studied. The tests are done for different tube and tube support material combinations, tube-to-support clearances, tube support geometries, force levels, etc. For example, Fig. 22 shows the effect of temperature on fretting-wear rate.

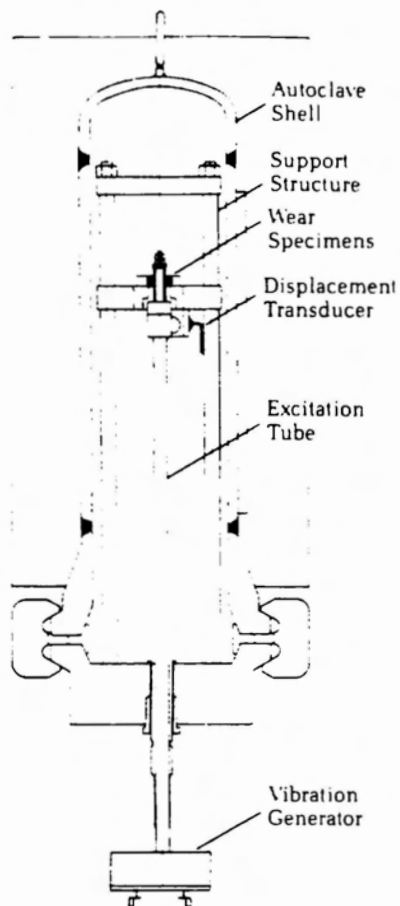


Fig. 20: Schematic of Impact Fretting-Wear Test Rig

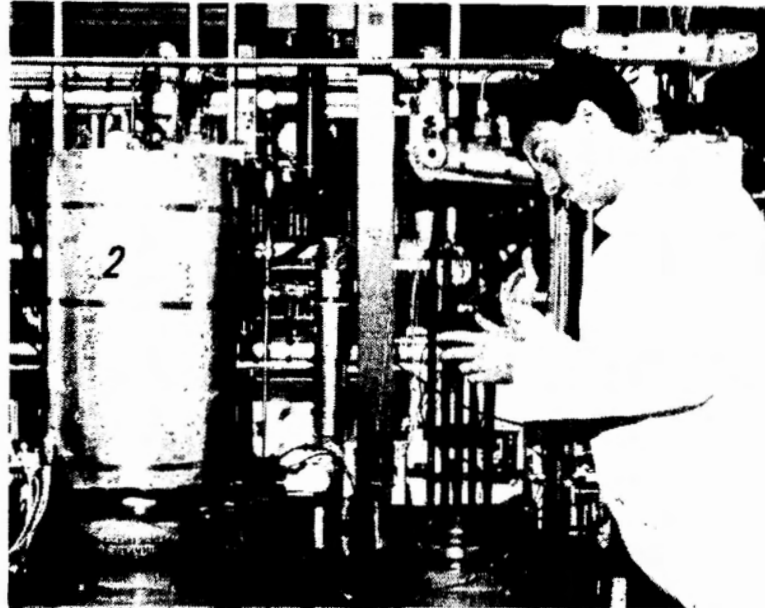


Fig. 21: High Temperature Autoclaves for Impact Fretting-Wear Tests

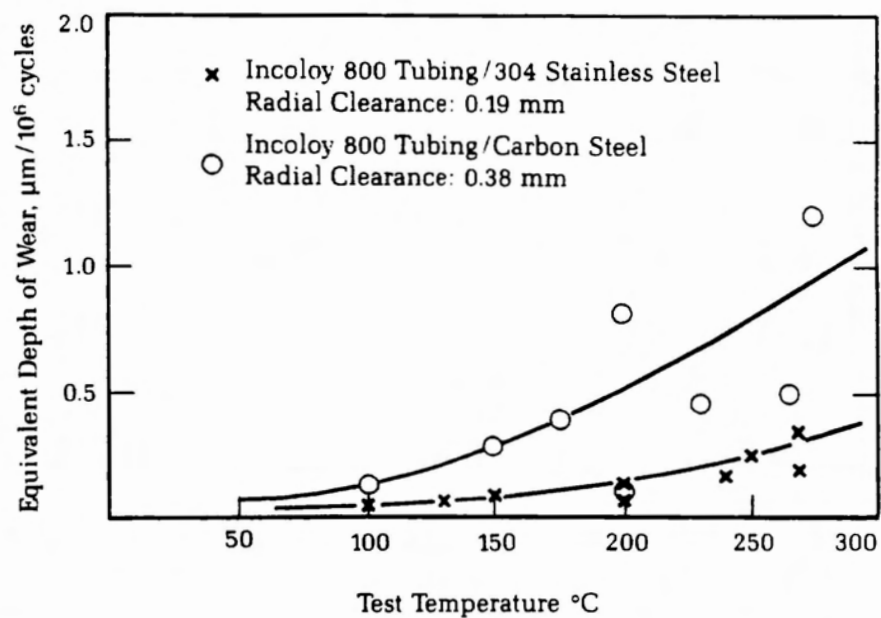


Fig. 22: Effect of Temperature on the Fretting-Wear Rate of Incoloy 800 Tubing in Realistic Steam Generator Environment

Formulation of Impact Fretting-Wear Mechanisms

A realistic formulation of impact fretting-wear mechanisms is needed to predict fretting-wear damage. As already stated, fretting-wear is related to the dynamic interaction between a tube and its supports. This is characterized by normal and oblique forces, by sliding forces and by sliding distances. Normal impact forces cause little sliding, and consequently, not much damage. Most of the fretting-wear damage is due to sliding of the tube against its supports. Unfortunately, it is difficult to distinguish between impact and sliding force during measurements. Thus, in earlier work fretting-wear was often correlated against the resultant rms (root-mean-square) forces at the supports [36].

Ko and Basista [37] showed that higher force levels correspond more to normal impact forces, whereas lower force levels are more indicative of sliding forces. Thus, it is possible by statistical analysis of the resultant forces to define a weighted forcing function which would correlate well with fretting-wear damage. Fig. 23 shows such a correlation for Incoloy 800 tubing against Inconel 600 supports [37]. The correlation is of the form

$$\dot{w} = aF^b \quad (9)$$

where $\dot{w}$ is the fretting-wear damage rate in mg/10⁶ cycles, F is the weighted forcing function, a is a constant equal to 6.8×10^{-16} and b is an exponent equal to 4.75 for this particular material combination. Thus, fretting-wear damage can be predicted with this type of correlation. The forcing function F may be evaluated by numerical simulation of the dynamic interaction (e.g., VIBIC).

Fretting-wear damage may be formulated more directly using a parameter called the work-rate [38], $\dot{W}$, which is defined as the normal component of contact force, F_n , integrated over the sliding distance, s ;

$$\dot{W} = \frac{1}{t} \int F_n ds \quad (10)$$

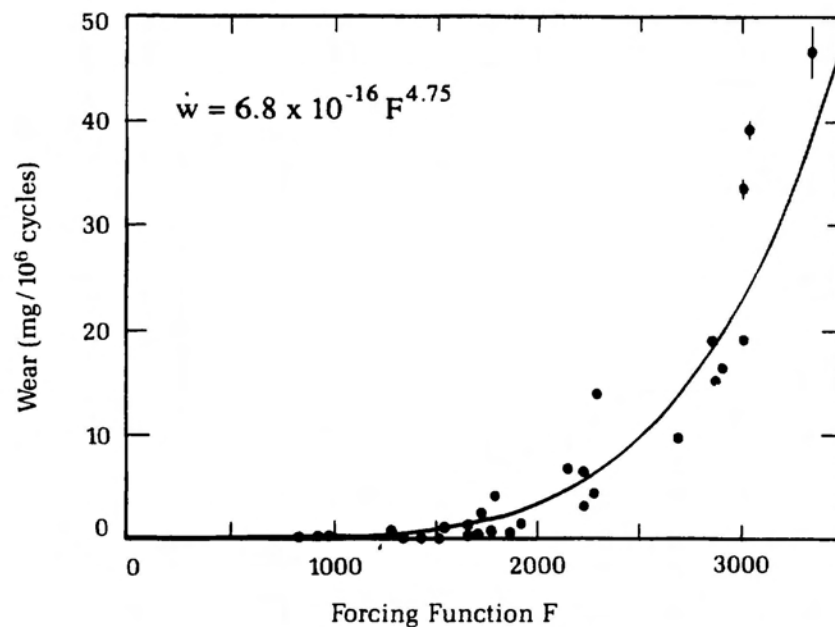


Fig. 23: Force-Wear Correlation Curve for Incoloy 800 Tubing vs Inconel 600 Supports

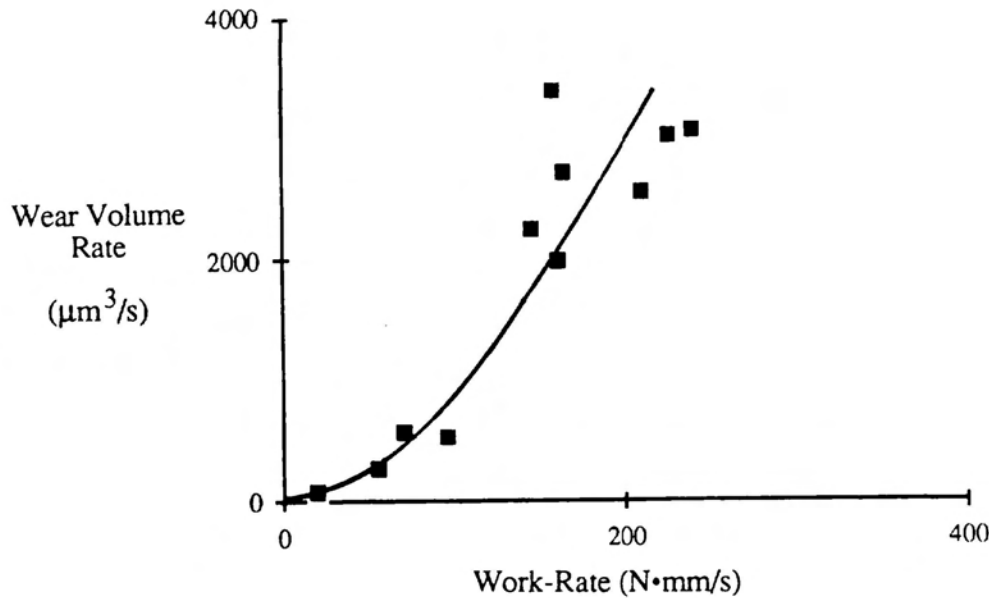


Fig. 24: Wear Rate - Work Rate Correlation for Incoloy 800 Tubing vs Inconel 600 Supports

where t is time. Work rate is directly related to wear rate according to Archard's wear equation, which states that wear damage is linearly related to the product of normal contact force and sliding distance. Thus, wear rate, $\dot{w}$, may be expressed as

$$\dot{w} = \frac{A}{t} \int F_n ds \quad (11)$$

where A is the fretting-wear coefficient and, ideally, is a constant for a given material and environment. The coefficient A is obtained experimentally from fretting-wear tests.

Fig. 24 shows a fretting-wear correlation based on the work-rate for an Incoloy 800 tube vibrating against an Inconel 600 support. The relationship is not quite linear. In fact, a power relationship of the work-rate would be more appropriate in this case, thus:

$$\dot{w} = A \left[\frac{1}{t} \int F_n ds \right]^b \quad (12)$$

Equation (12) is very similar to (9) and would also be a very appropriate damage mechanism formulation to predict fretting-wear damage.

CONCLUDING REMARKS

It is concluded that most flow-induced vibration problems can be avoided by thorough analyses at the design stage. This requires thermal-hydraulics modelling, a good understanding of vibration excitation and damping mechanisms and techniques to predict fatigue and fretting-wear damage. To our knowledge, nuclear components that have been properly analysed and designed do not normally suffer from vibration problems.

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DISCUSSION

Author: M.J. Pettigrew, L.N. Carlucci, C.E. Taylor and N.J. Fisher.

Paper: Flow Induced Vibration and Fretting Wear in Nuclear Steam Generators and Heat Exchangers.

Questioner: S. Odar, Siemens AG KWU

Question/Comment: Is there an enveloping value for the gap size (tube to tube support plate) to avoid fretting?

Response: No. There is no maximum value of gap size to avoid fretting. It depends on a number of parameters such as natural frequency, span length, type of support, etc. It is a statistical problem. One should do a parameter analysis, i.e., look at the effect of clearance, eccentricity, etc. on fretting-wear rate. Then a reasonable clearance gap can be recommended for a given component.

Questioner: James Gutzwiller, B & W Nuclear Services Co.

Question/Comment: What correction is made, if any, for differences in slip velocity when correlating steam/water and air/water two-phase flow vibration?

Response: We use homogeneous two-phase flow parameters for two-phase flow vibration. This is a complex problem. Slip velocities as defined for thermal-hydraulics are not necessarily appropriate for vibration. So far we have found reasonably good correlation for vibration parameters using homogeneous two-phase flow parameters in both air/water and steam/water flows.