VIBRATION ANALYSIS AND VIBRATION DAMAGE ASSESSMENT IN NUCLEAR AND PROCESS EQUIPMENT

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Component failures due to excessive flow-induced vibration are still affecting the performance and reliability of process and nuclear components. The purpose of this paper is to discuss flow-induced vibration analysis and vibration damage prediction. Vibration excitation mechanisms are described with particular emphasis on fluidelastic instability.

The dynamic characteristics of process and power equipment are explained. The statistical nature of some parameters, in particular support conditions, is discussed. The prediction of fretting-wear damage is approached from several points-of-view. An energy approach to formulate fretting-wear damage is proposed.

Introduction

Component failures due to excessive flow-induced vibrations continue to affect the performance of process and power plants. Such failures are very costly in terms of repairs and lost production. Generally, the problems caused by excessive vibration are fatigue cracks and fretting-wear damage. Tube failures due to fretting-wear in shell-and-tube heat exchangers are of particular concern.

Considerable progress has been made in the area of flow-induced vibration since the early seventies. Vibration excitation mechanisms in single-phase (liquid or gas) flows are now reasonably well understood. Much work remains to be done in two-phase (liquid/gas) flows, although some very relevant studies have been conducted since the mid-eighties. It is now possible to make some prediction of fretting-wear damage due to vibration. Twenty years ago this was a far away dream. The purpose of this paper is to outline some of the more relevant findings and to discuss their applications to current problems.

Definition of the Problem and Flow Considerations

The study of flow-induced vibration obviously requires a good understanding of prevailing flow conditions. For example, several flow situations are possible in recirculating-type nuclear steam generators (see Figure 1). 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 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, process and nuclear components are essentially cylindrical structures (i.e., piping) or bundles of cylinders (i.e., tube bundles) subjected to axial or cross flow. The flow may be internal or external. It may be confined to 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 multispan and supported in several places. For example, heat exchanger tubes are supported by baffle-plates, and piping systems by piping supports.

In summary, it is necessary to understand flow-induced vibration excitation mechanisms and damping mechanisms for all possible flow situations.

Vibration Excitation Mechanisms

Dynamic forces are generated by fluid flow in components causing them to vibrate. Generally, four flow-induced vibration excitation mechanisms are relevant, namely: (1) fluidelastic instability, (2) periodic wake 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.

Fluidelastic instability is usually not a problem in axial flow. The flexural rigidity of components such as piping 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. This 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 amplitudes 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, Pettigrew and Gorman (1981). 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 except for high density gases. Periodic wake shedding is generally not a problem in two-phase flow except at very low void fractions (i.e., $\varepsilon_g < 15\%$), Pettigrew and Taylor (1994).

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. Alternatively, far-field excitation can be generated by upstream components such as inlet nozzles, 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 failure 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. Turbulence excitation can be very important near fluidelastic instability when the apparent damping becomes very small.

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

Acoustic resonance can also occur in axial flow. Acoustic pressure pulsations originating from the pumps or acoustic noise generated by piping elements such as valves can promote acoustic resonance in a receptive section of the piping system. If the acoustic resonance frequency is close to that of the structure, large vibration amplitudes and damage 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 fluidelastic instability.

Fluidelastic Instability in Cross Flow

Fluidelastic instability is by far the most important vibration mechanism for tube bundles subjected to cross flow. In our experience, most heat exchanger and steam generator tube vibration problems are related to fluidelastic instability. The topic of fluidelastic instability of tube bundles in single-phase cross flow has been reviewed fairly recently by Pettigrew and Taylor (1991). Fluidelastic instability is formulated in terms of a dimensionless flow velocity, Up/fD, and a dimensionless mass-damping parameter, $2\pi\zeta m/\rho D^2$. For the simple case of a tube bundle subjected to uniform flow over its entire length:

$$U_{pc}/fD = K\,(2\pi\zeta m/\rho D^2)^{1/2} \qquad (1)$$

where f is the tube frequency in the fluid, m is the mass per unit length including the hydrodynamic mass, ζ is the total damping ratio (structural and fluid damping), ρ is the fluid density, D is the tube diameter and U_{pc} is the threshold, or critical, velocity for fluidelastic instability. The instability constant, K, is obtained from the available experimental data as shown in Figure 2. As a simple practical design guideline, an instability constant K = 3.0 is recommended for all tube bundle configurations in single-phase cross flow.

Fluidelastic instability is equally important in two-phase cross flow and can be similarly formulated. However, in two-phase the nature of the flow regime can be very important. Flow regimes can be continuous (i.e., bubbly, spray, fog) or intermittant (i.e., slug, wavy). Flow regimes can drastically affect dynamic paramters such as hydrodynamics mass and damping. For example, Figure 3 shows the effect of void fraction on damping in two-phase cross-flow. An overview of the topic of two-phase flow-induced vibration was prepared recently by Pettigrew and Taylor (1994).

Dynamic Characteristics of Process and Nuclear Structures

The dynamic behaviour of process and nuclear structures is generally very complex. These structures often have ill-defined boundary conditions, and are non-linear because of necessary clearances at the supports. They are sometimes non-stationary and subjected to random excitation forces. They can only be defined in terms of both deterministic parameters such as diameter, length and fluid properties and statistical parameters such as straightness, alignment and contact forces at the supports.

For example, from a dynamic point-of-view, a nuclear steam generator is essentially a multitude (1000's) of multispan U-bend tubes supported by clearance-supports of more-or-less complex geometries. This is effectively a highly non-linear system because of the clearance between the tube and tube-supports. At a given support location, the tube may be typically: not touching the support, contacting lightly, or in contact with a significant preload depending of tube straightness, support alignment and hydraulic drag forces. Thus, the system is considerably ill-defined.

Furthermore, the tube contact loading at the support may change with time since hydrodynamic forces may vary due to thermal power changes, and tube straightness and alignment may be affected by thermal expansion. Crudding over time can also affect the tube loading conditions at the support. Thus, the system is also non-stationary.

The geometry of the support can be a further complication. For example, in a tri-lobar broached hole there are twelve principal contact points, as shown in Figure 4. In a lattice bar type of support there are eight principal contact points. A large number of contact points increases the number of possible boundary conditions dramatically. Considering that there are some twenty supports along a steam generator tube, there may be some 10^{30} different combinations of possible contact situations in a given steam generator, as shown in Figure 5. It is obviously impossible to analyse all of these tube contact combinations. The approach taken is to model a statistically representative sample of the population of contact combinations. This is done for example to predict the effect on fretting-wear damage of increasing the clearance between tube and tube-supports. Enlarged diametrical clearances may result from chemical cleaning or from manufacturing difficulties.

The dynamic behaviour of several other components such as piping systems could be similarly described. These components can either be analysed using a statistical approach or by understanding the dynamic behaviour within the bounding limits of the dynamic parameters as discussed later.

Fretting-Wear Damage Prediction

Fretting-wear damage usually occurs between a vibrating structure and its supports. It is related to the dynamic interaction within the supports. The dynamic interaction between structure and supports is conveniently formulated in terms of a parameter called "work-rate", $\dot{W}$. The work-rate is simply the integral of the product of the contact force, F(t), times the sliding distance, S, per unit time, or :

$$\dot{W} = \frac{1}{T}\int_0^T F(t)\cdot|S|dt \qquad (2)$$

The fretting-wear damage volume rate, $\dot{V}$, can be calculated from:

$$\dot{V} = K_W \times \dot{W} \qquad (3)$$

where K_W is a wear coefficient obtained experimentally.

To predict fretting-wear damage it is necessary to evaluate the work-rate at every support location. The work-rate can be measured experimentally or calculated with a time domain computer model such as the VIBIC code, Fisher et al. (1992). VIBIC simulates the vibration response of multispan structures, and, in particular, the dynamic interaction between structure and support which yields the work-rate. VIBIC can model deterministic fluid forces and random turbulence excitation, and realistic support geometry with appropriate contact forces.

The approach has been validated in several benchmark tests where the work-rate was measured experimentally and compared against model predictions. For example, the dynamic interaction between a nuclear fuel element and a section of fuel channel was measured for the configuration shown in Figure 6, Fisher et al. (1996). The work-rate measurements are compared to prediction in Figure 7. The agreement between measured and predicted work-rate is reasonably good. In this figure the work-rate results are presented as a function of gap or preload at the point of contact. Obviously when the clearance gap is larger than the maximum vibration amplitude, there is no interaction and the work-rate is equal to zero. At the other extreme when the friction force due to the preload is sufficient to prevent motion, there is little work-rate. Work-rate appears maximum when the preload or clearance gap is very small. The above is illustrated in terms of the fuel element motion relative to the fuel channel in Figure 8. For this particular case, which simulated turbulence excitation under normal condition, the work-rate was very low being less than 0.1 mW. Thus, little fretting-wear damage would occur for this fuel element configuration. We found that excitation forces four times larger, resulting in work-rates in excess of 1 mW, would be required to cause unacceptable fretting-wear damage in these fuel channels based on a fretting-wear coefficient $K_w \sim 2000\times10^{-15}(1/Pa)$ for zirconium alloys.

Fretting-wear coefficients for heat exchanger materials are generally much lower, being typically 40 x 10-15(1/Pa), Fisher et al. (1994). For example, we can predict tube wear depth quite simply for a well designed heat exchanger where the work-rate is not expected to exceed 5 mW. Assuming continuous and uniform wear around a tube of diameter, D = 20 mm, wall thickness, w = 1.0 mm and a support of thickness, L = 15 mm, we can show that the fretting-wear damage would amount to a ~20% wall reduction after 40 years of operation. Of course, this is a conservative estimate, since fretting-wear damage does not necessarily occur continuously. However, the above calculations show that a maximum of 5 mW would be a reasonable design guideline to prevent fretting-wear problems in many heat exchangers.

Work-rate is effectively mechanical energy at the support or more precisely mechanical power dissipated by the dynamic interaction between structure and support. In summary, an energy approach to understand and quantify fretting-wear is emerging.

Fretting-Wear: An Energy Approach

As already discussed, the dynamic interaction between a structure and its supports and, consequently, fretting-wear damage are conveniently formulated in terms of the work-rate parameter. The work-rate, being simply the integral of the contact force times the sliding distance per unit time, is essentially a measure of mechanical energy or power dissipated at the supports.

In air or gases, the energy absorbed by the structure from the fluid, the structure vibration energy, vibration damping energy, the energy dissipated at the supports and fretting-wear damage are all directly related. In liquids, the vibration energy is also dissipated by viscous damping between structure and fluid and by squeeze-film damping at the supports. Only the vibration energy dissipated through contact forces and sliding causes fretting-wear damage at the supports. On the whole there is an energy balance and fretting-wear damage should be related to the vibration energy in the structure.

An experimental program was conducted to understand damping of multispan heat exchanger tubes, Taylor et al. (1997). A single heat exchanger tube, clamped at one end and supported at intermediate locations by three realistic supports, was used in these experiments. The supports were fully instrumented to measure work-rate. The tube was excited by random vibration and the energy input was also measured.

An energy approach was used to investigate damping in this program. In these experiments, the energy dissipated at each support was measured in the form of a shear work-rate. The shear work-rate is defined as the integral of the shear or sliding force times the sliding distance per unit time. The so-defined shear work-rate is identical to the mechanical energy dissipated by friction at the supports. The vibration excitation energy was similarly measured in the form of an input work-rate. In this case, the work-rate is defined as the average mechanical power the shaker provides to the test tube.

In air, we found that the excitation energy was approximately equal to the sum of the mechanical energies dissipated at the supports as shown in Figure 9a. We found similar results in water. However, the energy dissipated through viscous damping between tube and fluid had to be taken into account separately since it is not measured as work-rate at the supports. The total dissipated work-rate is remarkably close to the input work-rate as shown in Figure 9b. Although energy conservation is not surprising, it was very interesting to observe it probably for the first time, in multispan heat exchanger tubes.

In summary, vibration excitation energy, work-rate dissipated at the supports and fretting-wear damage are all related because there must be an overall energy balance. Thus, it may be possible to predict fretting-wear damage directly from energy considerations.

Proposed Fretting-Wear Damage Criterion

A simplified fretting-wear damage criterion based on heat exchanger tube vibration response is discussed in this section. Normally, multispan heat exchanger tubes are clamped at the tubesheet and supported at several locations by clearance baffle-supports. As we have already discussed, this type of problem requires a time domain non-linear simulation to model the details of the dynamic interaction between tube and support. Many of the parameters governing the vibration response of multi-span tubes are well defined such as tube diameter, span length, flexural rigidity and vibration excitation forces. On the other hand, the boundary conditions at the supports are ill-defined. The tube can contact at a number of possible locations around a support. The contact force or preload depends on some deterministic factors such as thermal distortion and hydraulic drag forces, but also on statistical parameters such as tube straightness and support alignment. The point is that there are many possible boundary conditions at the supports ranging from maximum preload to a variety of clearances. This type of problem can be handled statistically by doing an ensemble of calculations on a sample of the population of possible boundary conditions.

Fortunately, the boundary conditions vary within a limited range of preloads, typically 0 to 10 N, and clearances, say 0 to 0.2 mm. The possible contact locations at the support are also within the range allowed by support geometry. The end result is that the ensemble of simulations predicting the dynamic interaction and work-rate between tubes and supports are also within a reasonably well-defined range for a given multispan heat exchanger tube. The range is largely dependent on the statistical part of the problem, that is the boundary conditions at the support. On the other hand, the average value of work-rate is largely dependent on the deterministic parameters. This suggests the possibility of predicting the average value of work-rate directly from the deterministic parameters. As already stated, work-rate is contact force times sliding distance. Sliding distance must be related to vibration amplitude and frequency, and contact forces must also be related to amplitude and frequency and also tube mass.

The above idea is further supported by the energy balance concept discussed in the previous section. Work-rate and fretting-wear damage are directly related to the mechanical vibration energy in the structure.

Yetisir et al. (1997) show that the vibration energy per unit time or mechanical power, P, of a one-span uniform structure vibrating in the i^{th} mode may be formulated by:

$$P = 16\pi^3 m L f_i^3 \overline{Y^2} \zeta_i \qquad (4)$$

where m is the mass per unit length, L is the structure length, $\overline{Y^2}$ is the mean-square maximum vibration response, and f_i and ζ_i are the natural frequency and damping ratio in the i^{th} mode respectively. For a multispan heat exchanger tube with N_s spans vibrating in two orthogonal directions, the total mechanical power, P_T, would become:

$$P_T = 32\pi^3 N_s m L f_i^3 \overline{Y^2} \zeta_i \qquad (5)$$

Equation 5 shows that the total power due to vibration response is governed by frequency, mass, span length, number of spans and vibration response. Generally, we find that the dominant vibration frequency of a tube with clearance-supports is roughly the same as that with idealized pin-supports. Thus, Equation 5 could be applied in an approximate way to clearance-supports. The tube vibration energy is dissipated partly by viscous damping but mostly at the support in the form of work-rate. Thus, for a typical multi-span heat exchanger tube with N_s spans and N_s-1 supports, an upper bound expression for the work-rate, W_u, at one support may be written as:

$$\dot{W}_u = 32\pi^3 \frac{N_s}{(N_s - 1)} m L f_i^3 \overline{Y^2} \zeta_i \qquad (6)$$

Based on Equation 6 a simplified expression to predict work-rate should emerge.

Yetisir et al. (1997) conducted thousands of work-rate calculations on realistic heat exchanger tube configurations to confirm the form of Equation 6. The simulations were done using the VIBIC code. The parameters that were investigated were, number of spans, excitation levels, support clearances, preload at the supports, tube flexural rigidity, mass per unit length, span length, modal damping, etc. The results were compared to those obtained with Equation 6 in Figures 10a and 10b for two- and five-span heat exchanger tubes respectively. The agreement between VIBIC simulations and the simple calculations using the proposed criterion was remarkably good being within a factor of ±2. The range reflects the statistical aspect of this problem and, in particular, the effect of the statistical parameters governing the boundary conditions at the supports. The exponents of the parameters in Equation 6 were found to be a best fit to the data from the simulations. The proposed criterion was tested for a frequency range of 8 to 136 Hz, maximum vibration amplitude range of 7 to 1860 μm, span-length range of 0.5 to 2.0 m, mass per unit length range of 0.3 to 1.2 kg/m and a damping ratio range of 0.01 to 0.05. The calculated work-rate varied from 0.08 to 65 mW, which encompasses most heat exchanger situations.

In summary, the proposed criterion to predict fretting-wear damage appears very promising. It may provide an alternative to comprehensive time-domain simulations for simple cases of multispan heat exchanger tubes.

Concluding Remarks

Most of the current knowledge in flow-induced vibration was developed in the last thirty years. While much progress has been accomplished to understand flow-induced vibration mechanisms, there are still some important areas requiring further attention. One such area is two-phase flow-induced vibration and in particular the effect of two-phase flow regime. Another promising area is the prediction of fretting-damage. However, most flow-induced vibration problems can now be avoided by proper analysis at the design stage.

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TABLE 1: VIBRATION EXCITATION MECHANISMS

Flow situation	Fluidelastic instability	Periodic shedding	Turbulence excitation	Acoustic resonance
Axial flow				
Internal				
Liquid	*	-	**	***
Gas	*	-	*	***
Two-phase	*	-	**	*
External				
Liquid	**	-	**	***
Gas	*	-	*	***
Two-phase	*	-	**	*
Cross flow				
Single Cylinders				
Liquid	-	***	**	*
Gas	-	**	*	*
Two-Phase	-	*	**	-
Tube Bundle				
Liquid	***	**	**	*
Gas	***	*	*	***
Two-phase	***	*	**	-

***Most important **Should be considered *Less likely
-Does not apply

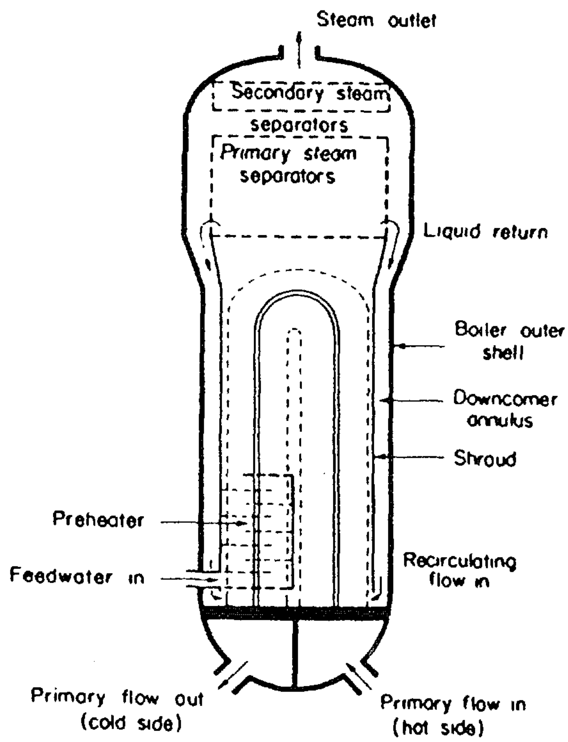


Figure 1: Typical Nuclear Steam Generator

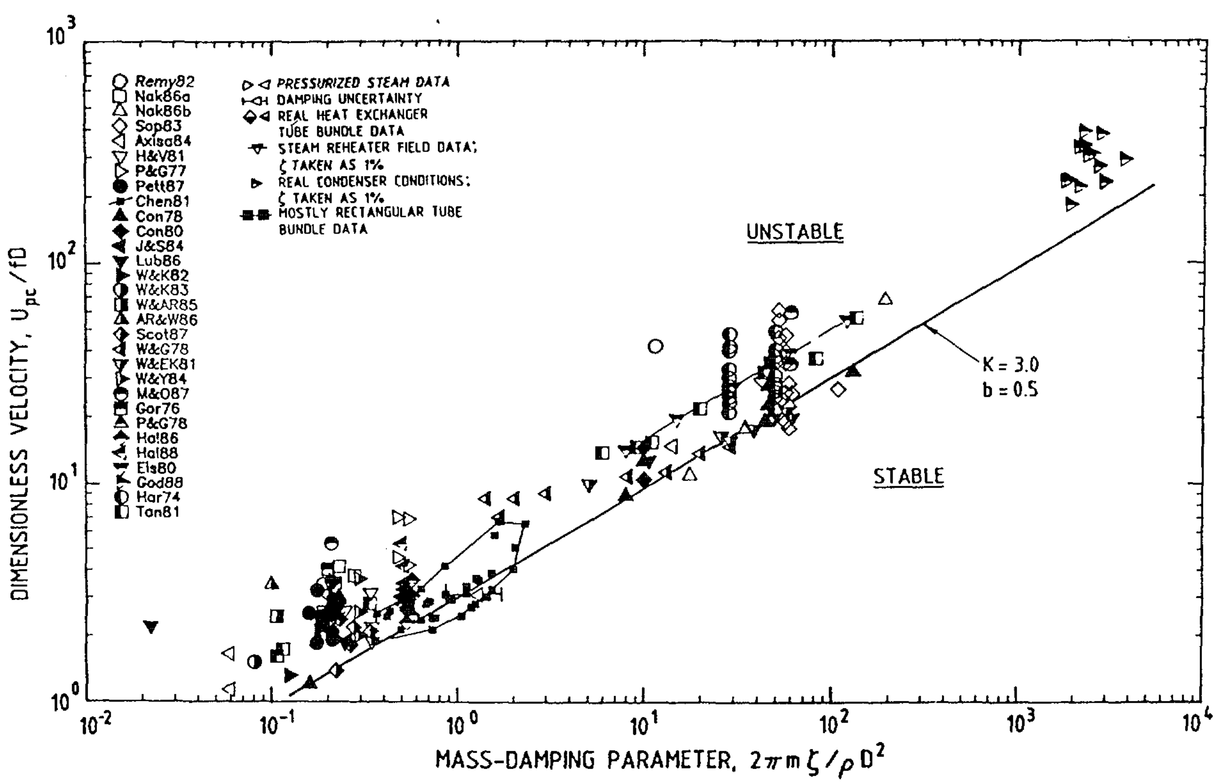


Figure 2: Summary of Fluidelastic Instability Data in Single-Phase Cross Flow

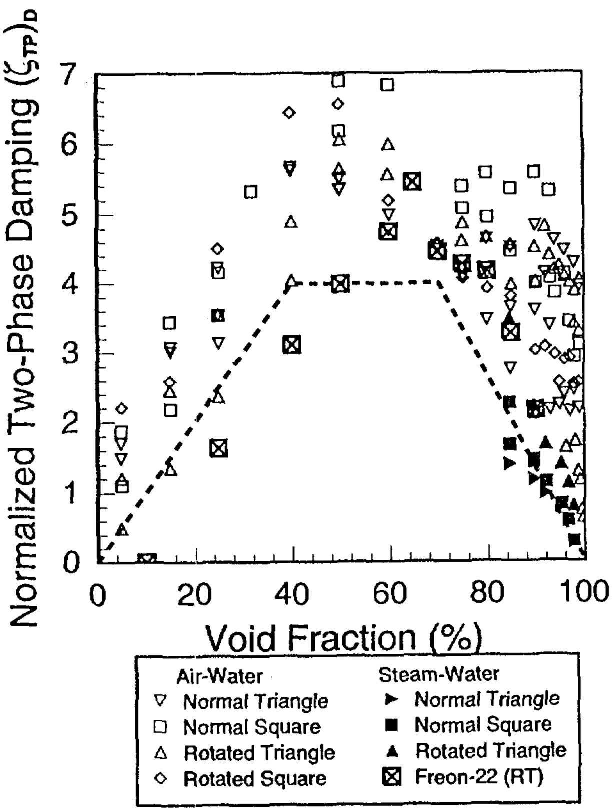


Figure 3: Available Damping Data in the form of Normalized Two-Phase Damping Ratios Showing the Effect of Void Fraction

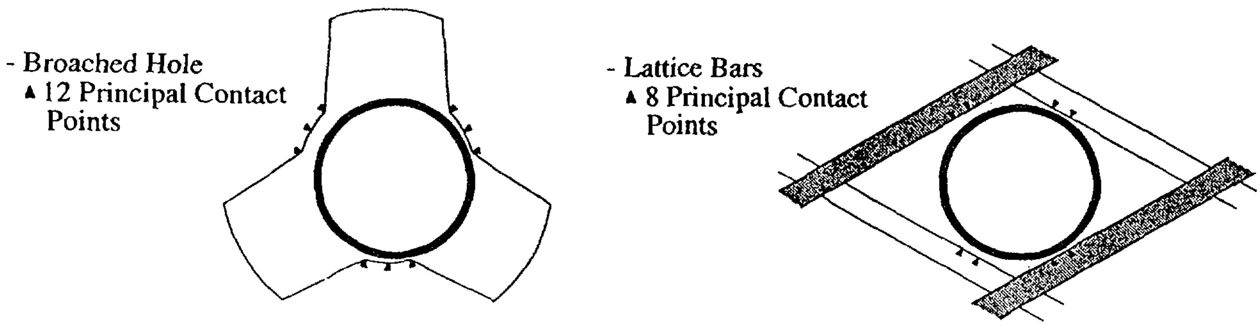


Figure 4: Complex Tube-Support Geometry: Possible Contact Points

- Contact Points:
  12 possibilities

- Eccentricity/Preload:
  3 possibilities

- 20 Supports

  TOTAL:

  $(12 \times 3)^{20} = 13 \times 10^{30}$
  possibilities

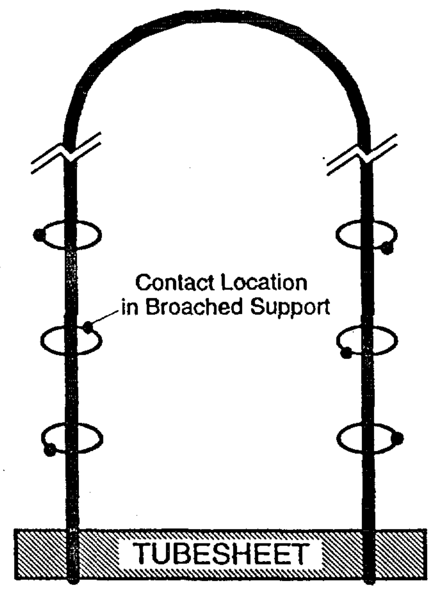


Figure 5: Steam Generator Tube and Support Contact Combinations

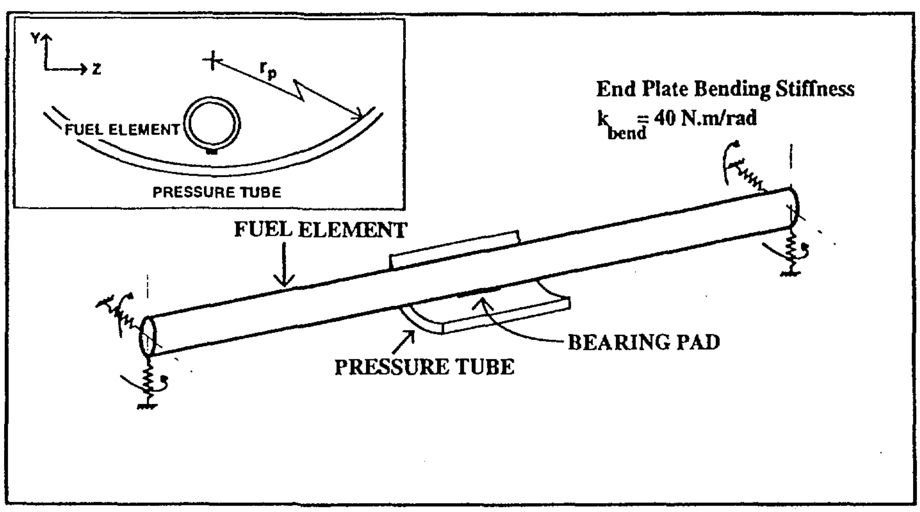


Figure 6: Model of Fuel Element Bearing Pad and Fuel Channel Contact

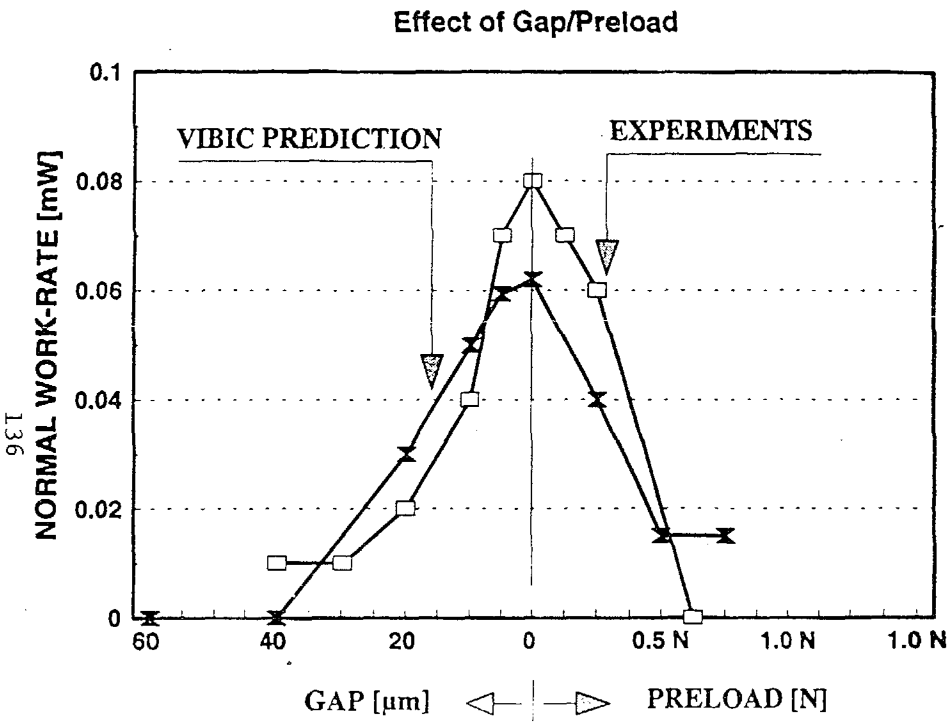


Figure 7: Work-Rate Versus Gap/Preload for a Fuel Element Vibration Response of 25 µm RMS

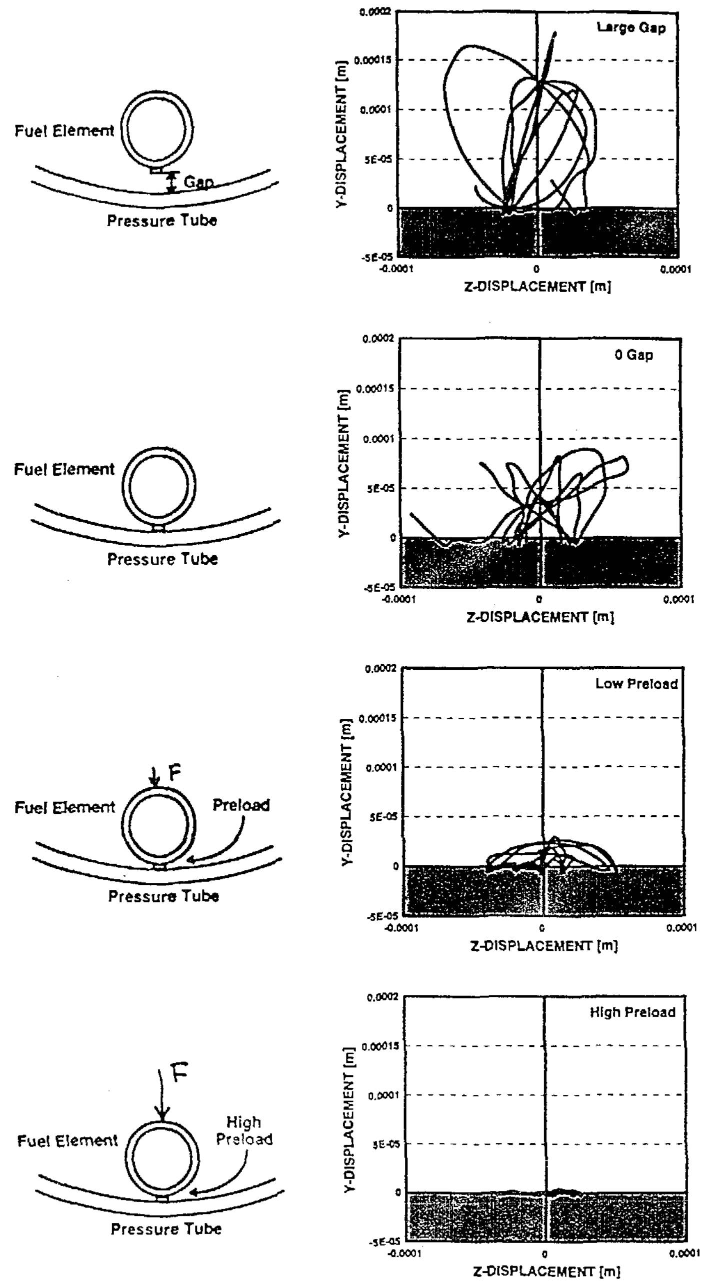
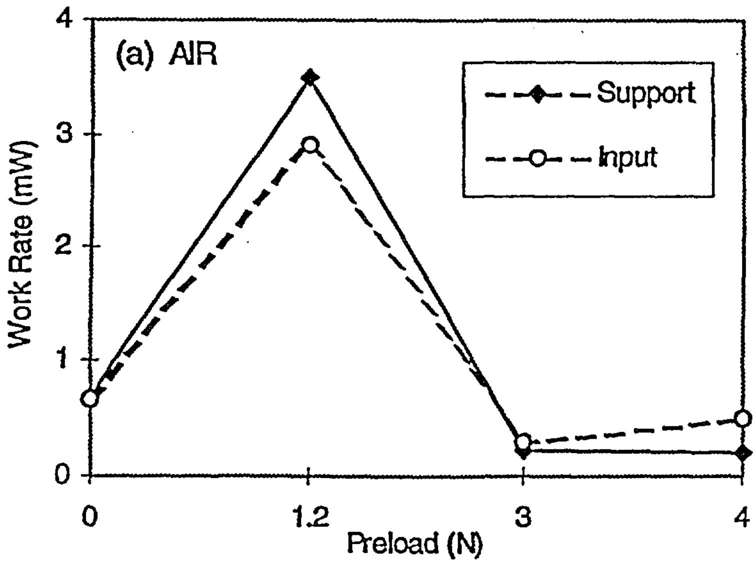


Figure 8: Effect of Clearance or Preload on Dynamic Interaction Between Fuel Element and Fuel Channel



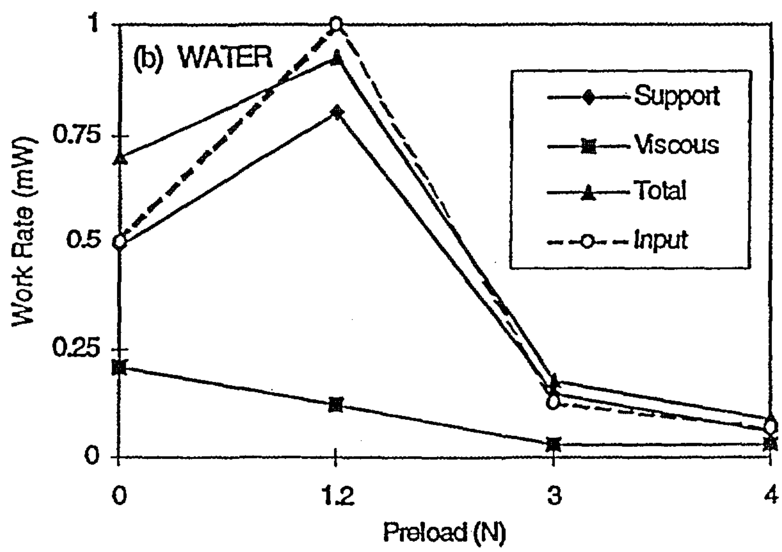


Figure 9: Input and Dissipated Work-Rates for Multispan Heat Exchanger Tube Tests

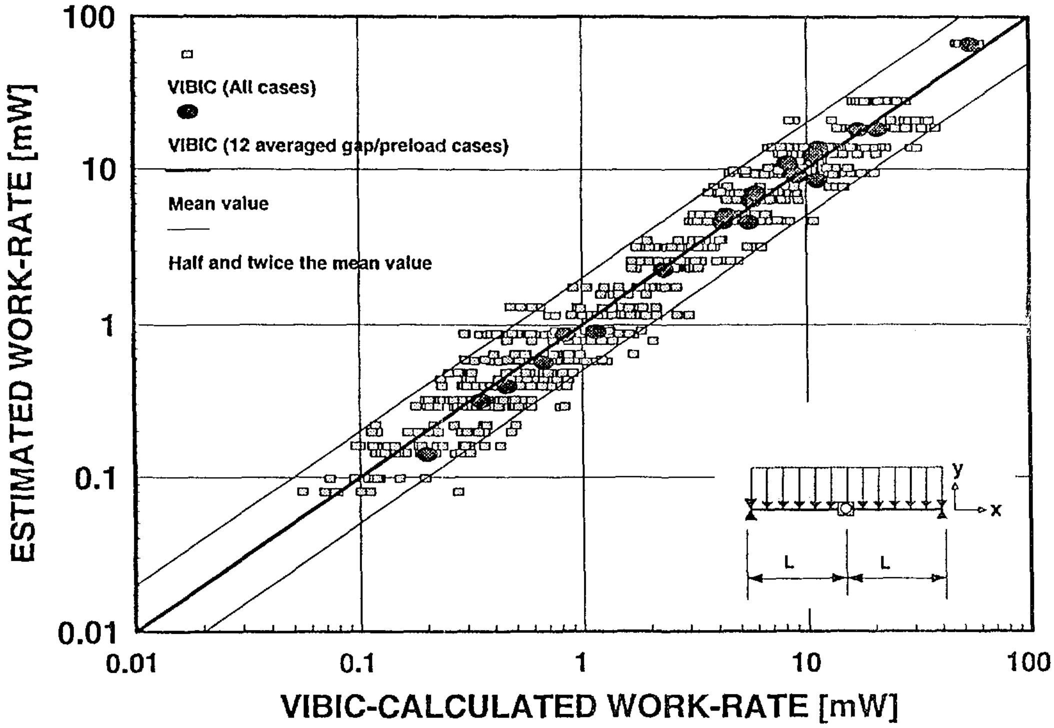


Figure 10a: *Estimated and VIBIC-Calculated Work-Rates for all Two-Span Simulations. (Total number of simulations is 1144. Only 27 (3%) data points are outside the factor-of-two bounds.)*

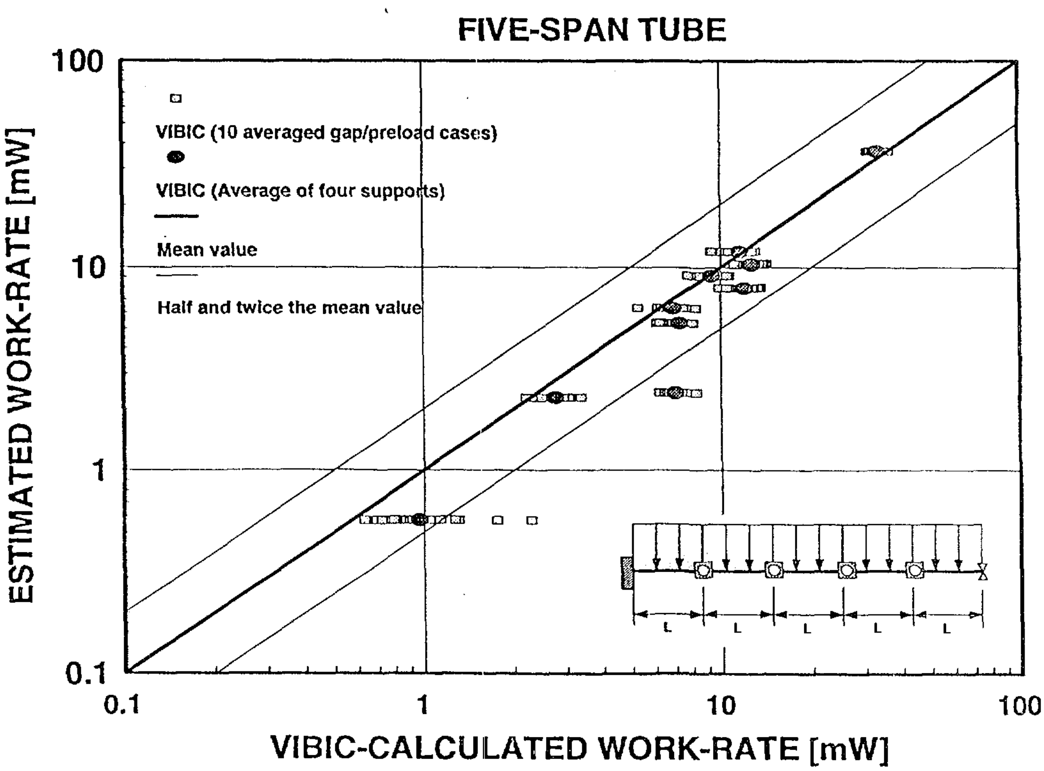


Figure 10b: *Estimated and VIBIC-Calculated Work-Rates for all Five-Span Simulations. (Total number of simulations is 275. Twenty nine (10.5%) data points are outside the factor-of-two bounds.)*