

FLOW-INDUCED VIBRATION ANALYSIS OF HEAT EXCHANGER AND STEAM GENERATOR DESIGNS

M. J. PETTIGREW, Y. SYLVESTRE, A. O. CAMPAGNA

*Engineering Research Branch, Atomic Energy of Canada Ltd.,
Chalk River Nuclear Laboratories, Chalk River, Ontario KOJ 1J0, Canada*

SUMMARY

Tube and shell heat exchange components such as steam generators, heat exchangers and condensers are essential parts of CANDU (CANadian Deuterium Uranium) nuclear power stations. Excessive flow-induced vibration may cause tube failures by fatigue or more likely by fretting-wear. Such failures may lead to station shutdowns that are very undesirable in terms of lost production. Hence good performance and reliability dictate a thorough flow-induced vibration analysis at the design stage. This paper presents our approach and techniques in this respect.

In a recirculating type steam generator for example all flow situations are possible. Liquid cross-flow exists at the entrance region. Within the tube bundle the flow is mostly axial. It is liquid at the bottom and gradually becomes two-phase to reach 15–20% steam quality at the top. Two-phase cross-flow is predominant in the "U" bend tube region.

In cross-flow three basic flow-induced vibration excitation mechanisms are considered, namely: 1) fluidelastic instability, 2) forced vibration response due to random flow turbulence and 3) periodic wake shedding. We have observed the first two mechanisms in both liquid and two-phase cross-flow. Periodic wake shedding has not been detected in two-phase flow but is possible in liquid flow. It is only significant for upstream tube rows. Random flow turbulence is the dominant excitation in both liquid and two-phase axial flow.

These vibration excitation mechanisms and the dynamics of multispan tubes are formulated in a computer model. The model predicts tube vibration response and critical velocities for fluidelastic instability. A description of the model is given. The vibration analysis of a steam generator is outlined as an example. We find that higher than fundamental modes are sometimes critical.

The parameters required to formulate the vibration excitation mechanisms are discussed. Periodic wake shedding excitation is formulated in terms of a Strouhal No. and a lift coefficient which is generally less than unity. Fluidelastic instability thresholds are related to dimensionless flow velocity V/fd and dimensionless damping $m\delta/\rho d^2$ for both liquid and two-phase cross-flow. Some statistical parameters to describe random flow turbulence excitation are deduced from our experimental data. The power spectral density of the latter is related to a power of the flow velocity. The velocity exponent is roughly two for liquid flow and near unity for two-phase flow. For a given mass flux, the random excitation reaches a maximum at a steam quality of roughly 15%. Damping in two-phase flow is found to be at least four times greater than in liquid flow.

Our design specifications from a vibration point of view are: 1) avoid fluidelastic instabilities; 2) make sure the tube response to random excitation is low enough to avoid problems; and 3) avoid periodic wake shedding resonance or demonstrate it is not a problem. When our response calculation technique is not sufficient to satisfy the above specification, we use it to compare the design under study to that of our existing satisfactory designs. The computer model is then used as a normalization tool.

It is concluded that most flow-induced vibration problems may be avoided by proper analysis at the design stage.

1. INTRODUCTION

Tube and shell heat exchange components such as steam generators, heat exchangers and condensers are essential parts of CANDU* nuclear power stations. Relatively high flow velocities exist in some of these components. Thus flow-induced vibration problems are possible. Excessive flow-induced vibration may cause tube failures by fatigue or more likely by fretting wear. A typical example of tube fretting wear is shown on Fig. 1. Such failures may lead to station shutdowns that are very undesirable in terms of lost production and maintenance personnel irradiation. Small tubes, typically 12.5 - 16 mm OD, are used in nuclear components to minimize heavy water inventory. These tubes are flexible and thus prone to vibration. Also some components such as the steam generators are relatively inaccessible after manufacturing making subsequent modifications difficult. Hence good performance and reliability dictate a thorough flow-induced vibration analysis at the design stage. This paper presents our approach and techniques in this respect.

2. FLOW CONSIDERATIONS

Consider for example the recirculating type steam generator shown on Fig. 2. Several flow situations are possible in this component. Heavy water flow inside the tubes varies from 5% steam quality to subcooled liquid. The tubes are subjected to liquid cross-flow in the preheater section 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 to reach 15 - 20% steam quality at the top. Two-phase cross-flow is predominant in the "U" bend tube region.

In heat exchangers the tubes are often subjected to cross-flow particularly near inlets and outlets where flow velocities are generally high. Obviously the first step in the vibration analysis is to evaluate operating conditions and flow velocities.

3. TUBE DYNAMICS AND DAMPING

From a mechanical dynamics point of view the tubes are simply multi-span beams clamped at the tubesheet and held at the baffle-supports with varying degree of constraint. The latter is dependent on support geometry and particularly tube to support clearance. To be conservative we assume the intermediate supports to be hinged. Although we are currently supporting the development of an analytical model to consider the dynamics of tube to tube support interactions [1], we do not yet take it into account to keep the analysis linear.

The tube response $y(x,t)$ at any point x and at any time t may be expressed as a normal mode expansion in terms of the generalized coordinates $q_i(t)$ as follows

$$y(x,t) = \sum_{i=1}^{\alpha} \phi_i(x) q_i(t) \tag{1}$$

Using Lagrange's equation and assuming that the damping is small and that it does not introduce coupling between modes, the equation of motion for the i^{th} mode is:

$$\ddot{q}_i(t) + c/m \dot{q}_i(t) + \omega_1^2 q_i(t) = \int_0^{\ell} g(x,t) \phi_i(x) dx \tag{2}$$

where c is the damping coefficient per unit length, ℓ is the tube length and ω_1 and $\phi_1(x)$ are respectively the angular frequency and the mode shape of the i^{th} natural mode. The natural

* CANDU (CANada Uranium Deuterium)

modes are normalized so that

$$\int_0^{\ell} m \phi_i^2(x) dx = 1 \tag{3}$$

where m is the mass per unit length and includes the hydrodynamic mass due to the inertia of the fluid around the tube and the fluid inside the tube when applicable. Knowing m , c , ℓ , the support locations and the flexural rigidity (EI) of the tube, the response to different types of forcing functions $g(x,t)$ may be obtained by solving eq. (1) and (2).

The method used to obtain the natural frequencies and mode shapes is essentially an extension of the single span beam theory to multispan beams as suggested by Darnley [2]. The main assumptions are: uniform mass distribution and flexural rigidity, negligible effect of shear and rotary inertia, no axial motion of any point along the tube, homogeneous boundary conditions and continuity at the intermediate supports. The method yields an algebraic formulation of the natural modes. This is very convenient for the subsequent computation of the vibration response.

Structural damping expressed as the ratio of critical damping ζ is usually considered independent of frequency. Preliminary information from damping tests in water [3] and earlier measurements on a steam generator model [4] suggest on the other hand that the fluid contribution to the damping ratio decreases with frequency. Knowing that $c=4\pi m\zeta f$, the damping coefficient c instead of the ratio ζ appears nearly frequency-independent for fluid damping. Damping in heat exchanger tubes is probably between the above two limiting cases since both structural and fluid damping exist.

Some damping values measured in both water and two-phase mixtures are given in Table I. They are expressed in the form of a normalized damping coefficient $c_n = c/d$. Note that damping in two-phase mixtures is much larger than in water. For vibration analyses we normally take $c_n = 400$ for water and $c_n = 1400 \text{ kg}\cdot\text{s}^{-1}\cdot\text{m}^{-2}$ for two-phase mixtures. The latter was obtained on a steam generator model under realistic steam-water flow conditions.

TABLE I: TUBE DAMPING INFORMATION

Component Description	Environment	Diam. (mm)	Material	f (Hz)	m (kg/m)	ζ	c_n ($\text{kg}\cdot\text{s}^{-1}\cdot\text{m}^{-2}$)
Process Head Exchanger	Water	19.0	Stainless	44.4	1.04	0.0160	504
			Steel	33.7		0.0246	574
Steam Generator Model [4]	Water	13.0	Stainless	39.8	0.525	0.0216	438
			Steel	63.5		0.0115	372
Fuel Element[5]	Air-Water	19.0	Zircaloy-2	45.0	1.79	0.060	3181
Steam Generator Model	Steam-Water	13.00	Stainless	38.0	0.472	0.081	1407
			Steel				

4. VIBRATION EXCITATION MECHANISMS AND RESPONSE ANALYSIS

Generally in cross-flow, we consider three basic flow-induced vibration excitation mechanisms, namely: 1) fluidelastic instability, 2) random excitation due to flow turbulence and 3) periodic wake shedding. We have observed the first two mechanisms in both liquid and two-phase cross-flow [6]. Periodic wake shedding resonance is possible in liquid flow but has not been observed in two-phase flow. Either it does not exist or it is dominated by the response to random flow turbulence. We do not consider it in two-phase flow.

Response to random flow turbulence is the dominant excitation mechanism in both liquid and two-phase axial flow. It is treated in the same manner as cross-flow. Fluidelastic

instabilities are possible in liquid flow and could occur in two-phase flow. However the critical velocities for axial flow instability are much higher than those encountered in heat exchange components.

Vibration Response to Flow Turbulence Excitation

The mean square amplitude response $\bar{y}^2(x)$ of a continuous uniform cylindrical structure to distributed random forces $g(x,t)$ may be expressed by:

$$\bar{y}^2(x) = \sum_r \sum_s \frac{\phi_r(x) \phi_s(x)}{16\pi^4 f_r^2 f_s^2} \int_0^\infty |H_r(f)| |H_s(f)| \cos[\theta_r(f) - \theta_s(f)] \dots \dots \int_0^l \int_0^l \phi_r(x) \phi_s(x') R(x,x',f) dx dx' df \quad (4)$$

where: 1) the spatial correlation density function $R(x,x',f)$ is defined by

$$R(x,x',f) = 2 \int_{-\alpha}^{\alpha} \left[\lim_{T \rightarrow \infty} \frac{1}{2T} \int_{-T}^T g(x,t) g(x',t+\tau) dt \right] e^{-j(2\pi f)\tau} d\tau \quad (5)$$

2) the frequency response function is $H_r(f) = \frac{1}{(1-f^2/f_r^2) + jc f/(2\pi m f_r^2)}$ (6)

and θ_r is the argument of $H_r(f)$ for the r^{th} mode.

3) $\phi_r(x)$ and $\phi_s(x)$ represent the normal mode of vibration of the structure for the r^{th} and s^{th} mode, and

4) x and x' are points on the structure and τ is a difference in time t .

The main difficulty here is to obtain the statistical properties of the forcing function, e.g., its spatial correlation density function. We have done this in some cases by deduction from vibration response measurements [6]. This requires that some simplifying assumptions be made. The most important one is that the random force field be homogeneous and spatially correlated, that is $R(x,x',f) = S(g)$ where $S(g)$ is the power spectral density. In reality this is not quite correct. However the random forces were probably much better correlated over the length of the idealized tube bundles from which we deduced our data than over the greater lengths of multispan heat exchangers. Thus vibration response analyses based on the above would yield conservative results when applied to actual heat exchanger designs providing the forces are also assumed to be correlated in the analyses.

In liquid flow, the vibration response and consequently the spectral density $[S(g)]^{1/2}$ of the random forces is roughly related to flow velocity squared. By analogy to the formulation of lift forces, $[S(g)]^{1/2}$ may be expressed as

$$[S(g)]^{1/2} = C_R \rho V_r^2 d/2 \quad (7)$$

where C_R is called the random turbulence excitation coefficient, ρ is the density of the fluid and V_r is the reference gap velocity defined in terms of the free stream velocity V_α as

$$V_r = V_\alpha p/(p-d) \quad (8)$$

Values of C_R deduced from the results of several and quite different experiments [4,6,7] are shown in Figure 3. Interestingly, they are in the same range. Values for upstream tubes are somewhat larger as one might expect.

In two-phase cross-flow the random turbulence forces are approximately related to the mass flux G_r [6]. For a given mass flux the vibration response reaches a maximum at a steam quality of roughly 15% as shown on Figure 4. Deduced normalized power spectral densi-

ties are presented on Figure 5.

Although the above deductions are somewhat contentious, they nevertheless give the designer an idea of the magnitude of the flow turbulence forces.

Fluidelastic Instability

Fluidelastic instabilities are possible in a tube bundle subjected to cross-flow when the interaction between the motions of the individual tubes is such that it results in fluid force components that are both proportional to tube displacements and in-phase with tube velocities. Instability occurs when during one vibration cycle the energy absorbed from the fluid forces exceeds the energy dissipated by damping. It is shown [6] that, for a tube bundle subjected to the non-uniform flow velocity $V_r(x) = V_r \psi(x)$, the critical velocity at which instability occurs in the i^{th} mode may be expressed by:

$$V_r / f_{i1} d = K \left\{ c / \left[2 f_{i1} d^2 m \rho \int_0^l \psi^2(x) \phi_1^2(x) dx \right] \right\}^{1/2} \quad (9)$$

where $\psi(x)$ is a flow velocity distribution function and the instability factor K is determined experimentally. The above relationship is a generalization of Connors' formulation [8]. For the special case where the flow is uniform over the whole bundle length eq.(9) reduces to

$$V_r / f_{i1} d = K (\delta m / \rho d^2)^{1/2} \quad (10)$$

where the logarithmic decrement $\delta = c / (2m f_{i1})$.

We have found in a number of experiments [6,7] that for tube bundles in liquid flow $K \approx 6.6$ as shown on Figure 6. Note that the same fluidelastic criterion appears to apply for both liquid and two-phase flow. To allow for a realistic safety margin, a fluidelastic instability factor of 3.3 is recommended in vibration analyses of heat exchange components.

Periodic Wake Shedding

Periodic wake shedding would generate periodic forces in tube bundles. From eqs. (1) and (2) the vibration response $y(x,t)$ of a multispan uniform tube to a periodic forcing function $g(x,t) = g'(x)e^{j\omega t}$ may be shown to be expressed by:

$$y(x,t) = e^{j\omega t} \sum_{i=1}^{\infty} \phi_i(x) \frac{H_i(\omega)}{\omega_i^2 - \omega^2} \int_0^l g'(x') \phi_i(x') dx' \quad (11)$$

where $g'(x')$ could be a complex function formulating the spatial correlation along the tube. When $\omega = \omega_i$ resonance occurs in the i^{th} mode. Resonance may be a problem if the vibration response is large enough to control the mechanism of periodic wake shedding. Then the periodic forces become spatially correlated to the mode shape.

Assuming that damping is small, the peak vibration amplitude $Y(x)$ at resonance in the i^{th} mode may be expressed as:

$$Y(x) = \frac{m \phi_i(x)}{2\pi f_{i1} c} \int_0^l F_L(x') \phi_i(x') dx' \quad (12)$$

Under these conditions it may be shown that the contribution to the response of modes other than the i^{th} mode is negligible. The distributed periodic force $F_L(x)$ is formulated by:

$$F_L(x) = C_L \rho d V_r^2 \psi^2(x) / 2 \quad (13)$$

where C_L is the dynamic lift coefficient based on the reference gap velocity V_r . The wake shedding frequency is usually expressed in terms of a Strouhal Number $S = fd/V_r$. If the flow velocity distribution function $\psi(x)$ varies considerably, the frequency of wake shedding will tend to vary along the tube. Then the tube motion would not control periodic wake

shedding at the resonant frequency everywhere along the tube and periodic forces would be less correlated.

Strouhal Numbers extracted from Gorman's results [7] and from data published by other authors including those reviewed by Fitz-Hugh [9] are presented in Figure 7 for tube bundles of pitch over diameter ratios $p/d \leq 2$. Gorman's results were obtained in a water tunnel whereas most of the others came from wind tunnel tests. Unfortunately little information is available on the magnitude of the periodic wake shedding forces. Dynamic lift coefficients were extracted from Gorman's results as shown in Table II. Only the Strouhal Numbers corresponding to dynamic lift coefficients larger than 0.01 are considered. The scatter of the data on Figure 7 shows that there is no well defined criterion to predict periodic wake shedding frequencies except that the Strouhal Numbers are generally lower than 0.6. In practice resonance may be avoided if Strouhal Numbers for actual tube bundles are kept above unity for adequate separation of tube and wake-shedding frequencies. However this is an overly conservative criterion in many cases particularly where localized high flow velocities exist. Hence for conservatism we assume resonance whenever fd/V_r is less than one. No significant periodic wake shedding resonance has been observed for tubes inside a tube bundle in liquid flow. Resonance is more likely in upstream tubes, e.g., in the second and mostly the first tube row. Except for one case, dynamic lift coefficients C_L based on the reference gap velocity are generally less than 0.1.

An upper estimate of the vibration response at resonance may be obtained by assuming $C_L = 0.1$. In extreme cases we use the values of C_L given in Table II for the tube bundle configuration considered. As a design guideline we try to keep the resonant amplitude below 250 μm for tube frequencies below 100 Hz and even less for higher frequencies.

5. DESIGN ANALYSIS: APPROACH AND TECHNIQUES

Eqs. (1), (2), (4), (9) and (11) above are the basis of a computer program called PIPEAU to predict the vibration response of multispan tube bundles. The program first calculates the mode shapes $\phi_1(x)$ and the natural frequencies f_1 (i.e., eigenvalue solution) and then the response. It is best illustrated by examples as shown on Figure 8. Vibration response analyses of steam generator "U" bend tube regions are outlined on Figure 8a. A tube failed due to fretting-wear in one of the earlier Douglas Point steam generators for which the analysis indicates the likelihood of fluidelastic instability. The ratio of actual to critical flow velocity is much larger than recommended (for $K=3.3$). The range in the values is limited by whether the damping coefficient c or the damping ratio ζ is assumed independent of frequency. Subsequently a support was added at the top of the "U" bend. This solved the problem as predicted. In the Pickering nuclear power station, the steam generator "U" tubes are near the recommended vibration criteria. No problems have been experienced in five years of operation. In the Bruce station the steam generators "U" tubes were more conservatively designed as shown on Figure 8a. Steam generator designs are now vibration analyzed to be as good or better than Pickering.

Figure 8b illustrates the analysis of a process heat exchanger which experienced fluid-elastic instabilities in the 8th natural mode of vibration. This is explained by the shape of the 8th mode which is largest near the inlet where high flow velocities exist. Since the Strouhal Number $f_8 d/V_r = 0.29$ periodic wake shedding resonance could have also been a problem. Based on $C_L = 0.1$ the resonant amplitude is 1850 μm . The problem was solved by adding tube supports. Calculations of the random turbulence response of a hypothetical tube located in

the entrance region of a steam generator are shown on Figure 8c. The excitation was obtained from Figure 3. The first four modes are considered in the analysis. The maximum random vibration response is 29 μm RMS.

Ideally our approach to the vibration analysis of heat exchanger and steam generator designs should be that outlined on Figure 9. That is: starting from an initial design, given the flow conditions (1); the flow distribution and velocities are calculated (2); this indicates the excitation mechanisms and permits the formulation of the forcing function $g(x,t)$ (3); the latter is the input to the system (tube bundle) which needs to be defined in terms of f_1 , $\phi_1(x)$, ζ (4); then the response is calculated in the form of $y(x,f)$, the dynamic stresses $\sigma(x,f)$ and the forces at the supports $F(x,f)$ (5); the next step is to predict fatigue and fretting damage; (6) this leads to the last step which is to either accept (7) or modify the design depending on whether or not there are problems. The response calculation technique described earlier essentially links (3) to (5). We are not yet at this ideal stage. It is sometimes difficult to determine flow velocities in complex three-dimensional flow paths particularly in two-phase flow. We do not yet have enough information to formulate the forcing function in all cases. More tube damping information is required. It is desirable to express the tube to tube support dynamics in terms of the statistical properties of the impact forces [1]. This will likely prove to be the criterion governing the vibration-fretting relationship. Finally we need to understand better the vibration-fretting relationship for different materials in various relevant environments [10]. We are currently doing work in all these areas.

Our design specifications from a vibration point of view are: 1) avoid fluidelastic instabilities; 2) make sure the tube response to random excitation is low enough to avoid fretting or fatigue problems; and, 3) avoid periodic wake shedding resonance or demonstrate it is not a problem.

When our response calculation technique is not sufficient to satisfy the above specifications we can still use it to compare the design under study with existing satisfactory designs as discussed earlier for steam generator "U" tubes. The calculation technique is then used as a normalization tool. Alternatively we can test a model of the region in doubt [4]. It is also possible to conduct a fretting endurance test on a single tube subjected to the vibration response we estimate using our response calculation technique. If the heat exchanger component is easily accessible after installation in the reactor system, we can measure its vibration behaviour and take corrective action subsequently if necessary. For this purpose we have developed in collaboration with a manufacturer a very sensitive biaxial accelerometer probe that can be inserted in the tubes during operation (see Figure 10).

6. CONCLUDING REMARKS

It is concluded that, although there are still areas of uncertainty, most flow-induced vibration problems can be avoided. This requires that components be properly analyzed at the design stage and that the analyses be supported by adequate testing and development work.

ACKNOWLEDGEMENT

Extensive discussions with D.J. Gorman of the University of Ottawa, R.T. Hartlen of Ontario Hydro Research and R.I. Hodge of Atomic Energy of Canada Limited have greatly contributed to this paper.

REFERENCES

- [1] ROGERS, R.J., PICK, R.J., "On the Dynamic Spatial Response of a Heat Exchanger Tube with Intermittent Baffle Contacts", Nuclear Engineering and Design 36, 81-90 (1976).
- [2] DARNLEY, E.R., "The Transverse Vibration of Beams and the Whirling of Shafts Supported at Intermediate Points", Phil. Mag. S.6, Vol. 41, No. 241, Jan. 1921.
- [3] PETTIGREW, M.J., "Flow-Induced Vibration of Nuclear Reactor Power Station Components", presented at the 90th Annual Congress of the Engineering Institute of Canada, Halifax, Nova Scotia, October 1976. (to be published).
- [4] PETTIGREW, M.J., PLATTEN, J.L., SILVESTRE, Y., "Experimental Studies on Flow Induced Vibration to Support Steam Generator Design; Part II - Tube Vibration Induced by Liquid Cross-Flow in the Entrance Region of a Steam Generator", Paper No. 424, International Symposium Vibration Problems in Industry, Keswick, U.K., (1973) also Atomic Energy of Canada Limited, Report AECL-4515.
- [5] GORMAN, D.J., "An Analytical and Experimental Investigation of the Vibration of Cylindrical Reactor Fuel Elements in Two-Phase Parallel Flow", Nuc. Sci. Engrg. 44, 277-290 (1971).
- [6] PETTIGREW, M.J., GORMAN, D.J., "Experimental Studies on Flow Induced Vibration to Support Steam Generator Design, Part III - Vibration of a Small Tube Bundle in Liquid and Two-Phase Cross Flow", Paper No. 424, International Symposium Vibration Problems in Industry, Keswick, U.K. (1973), also Atomic Energy of Canada Limited, Report AECL-4804.
- [7] GORMAN, D.J., "Experimental Development of Design Criteria to Limit Liquid Cross-Flow - Induced Vibration in Nuclear Reactor Heat Exchange Equipment", Journal of Nuclear Science and Engineering, 61, 324-336 (1976).
- [8] CONNORS, H.J. JR., "Fluidelastic Vibration of Tube Arrays Excited by Cross Flow", Proceedings of the Symposium on Flow-Induced Vibration in Heat Exchangers, ASME Winter Annual Meeting, New York, Dec. 1, 1970, pp. 42-56.
- [9] FITZ-HUGH, J.S., "Flow Induced Vibration in Heat Exchangers", Paper No. 427, Int. Symp. on Vibration Problems in Industry, Keswick, U.K. (1973).
- [10] KO, P.L., "Impact Pretting of Heat Exchanger Tubes", Atomic Energy of Canada Limited, Report AECL-4653, October 1973.

TABLE II: PERIODIC WAKE SHEDDING: DYNAMIC LIFT COEFFICIENT AND STROUHAL NUMBERS EXTRACTED FROM GORMAN'S RESULTS (BASED ON THE REFERENCE GAP VELOCITY $V_r = V_\infty p / (p-d)$)

Tube Configuration	p/d	d (mm)	Tube Location	Direction* of Resonant Vibration	$S = \frac{fd}{V_r}$	C_L
Normal Triangle	1.33	19.0	1st row	normal	0.44	0.065
			2nd row	in	0.42	0.028
			2nd row	normal	0.45	0.016
	1.57	19.0	1st row	in	0.57	0.038
			1st row	normal	0.42	0.027
			2nd row	in	0.55	0.040
			2nd row	normal	0.42	0.023
	1.36	13.0	1st row	normal	0.56	0.183
			1st row	in	0.36	(0.026)**
	Parallel Triangle	1.33	19.0	N O N E		
1st row				in	0.42	0.081
1.57		19.0	1st row	normal	0.40	0.029
			2nd row	in	0.52	0.032
			interior	normal	0.52	(0.010)
			downstream	in	0.45	0.020
			downstream	normal	0.45	0.014
			interior	normal	0.20	(0.016)
1.36		13.0	N O N E			
			1st row	in	0.45	0.047
1.54	13.0	1st row	normal	0.45	(0.026)	
Normal Square	1.30	13.0	N O N E		*Direction: in flow direction; normal to flow direction.	
Rotated Square	1.30	13.0	N O N E		**Parenthesis indicate less prominent resonance peak (i.e., Resonance peak < 2 x Random turbulence response).	



FIG. 1: Tube Failure Due to Fretting Wear

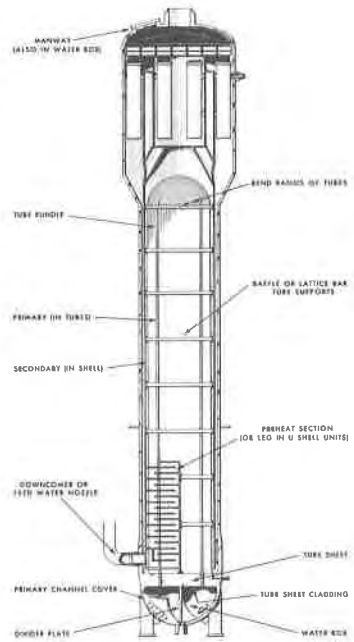


FIG. 2: Typical Nuclear Steam Generator

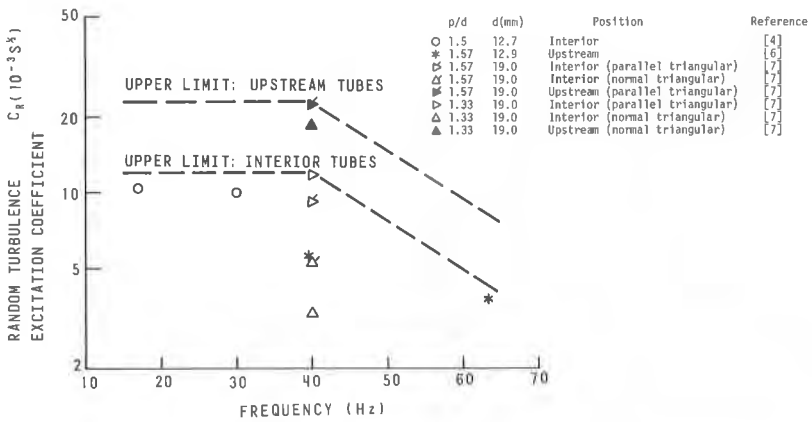


FIG. 3: Normalized Power Spectral Density of Random Turbulence Excitation for Triangular Tube Bundles in Liquid Flow Expressed as $C_R = \frac{2(S_R)^{1/2}g}{\rho V^2 d}$

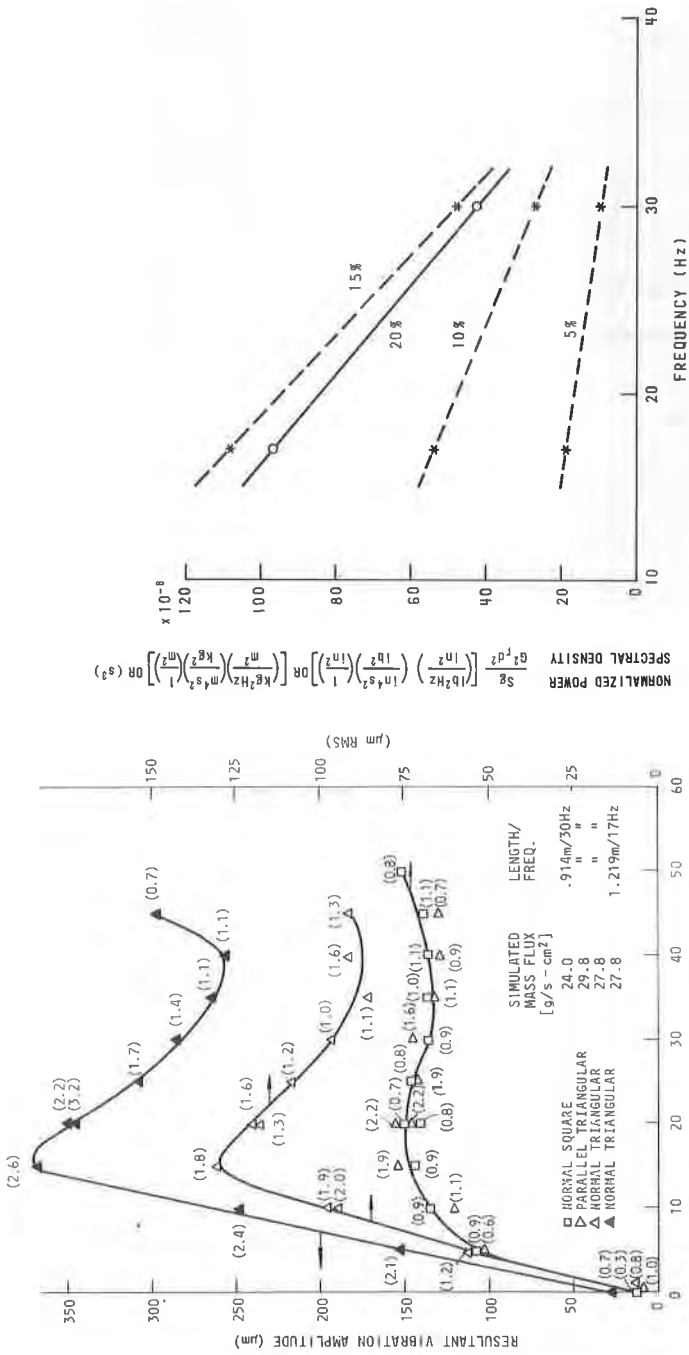


FIG. 5: Deduced Normalized Power Spectral Density of Random Turbulence Excitation in Two-Phase Flow [6]

FIG. 4: Effect of Steam Quality on Vibration Response [6]

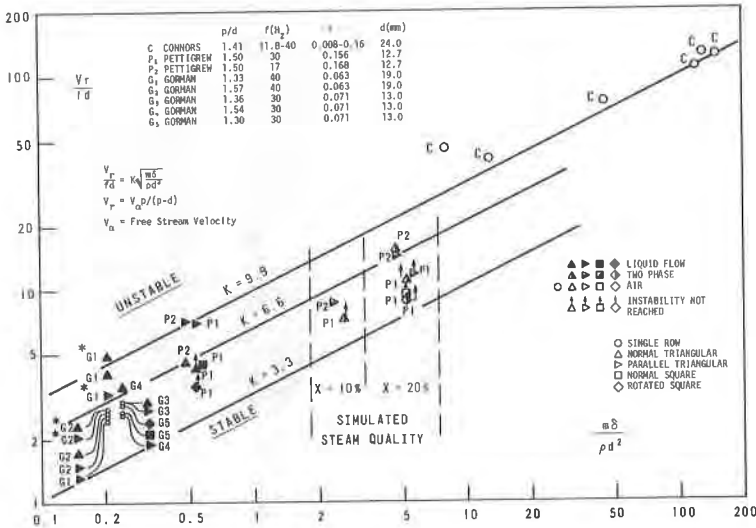


FIG. 6
Non-Dimensional
Presentation of
Experimental Data
on Fluid Instabilities
in
Cross-Flow

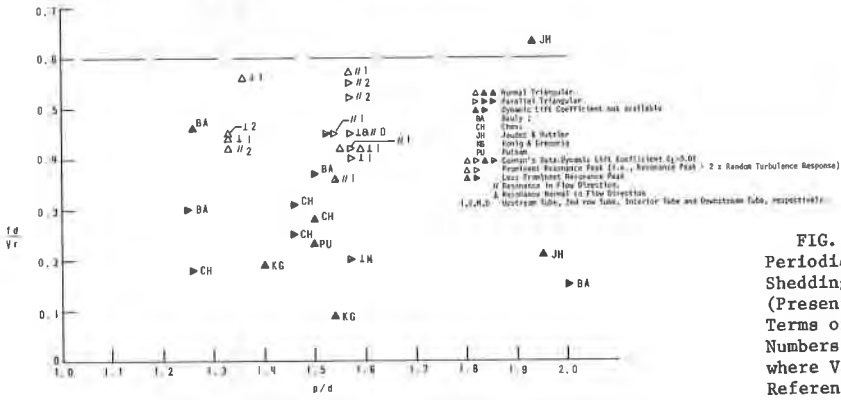


FIG. 7
Periodic Wake
Shedding Data
(Presented in
Terms of Strouhal
Numbers: fd/V_r
where V_r is the
Reference Gap
Velocity Defined
as $V_r = V_\alpha p/(p-d)$)

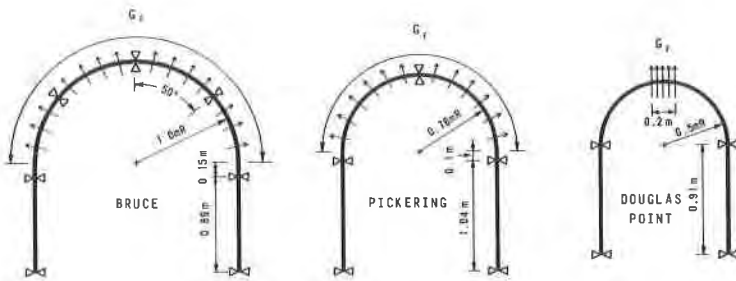


FIG. 8a
Vibration Analysis
of Steam Generator
'U' Bend Tube
Region (Fluid-
elastic Instability
in Two-Phase
Cross-Flow)

	G_p (kg m ⁻¹ s ⁻¹)	X (%)	d (mm)	f_s (Hz)	σ (kg m ⁻¹ s ⁻¹)	ω (rad/s)	V_r V _{ZRIT}
DOUGLAS POINT (* top support)	853	21.2	12.7	17.0	5.4-17.9 17.9	16.3-54.2	2.26-1.24 0.11
PICKERING	446	12.5	12.7	17.0	8.0-17.9	10.5-23.3	1.3-0.87
BRUCE	328	20	13.0	35.5	18.2	15.9	0.49

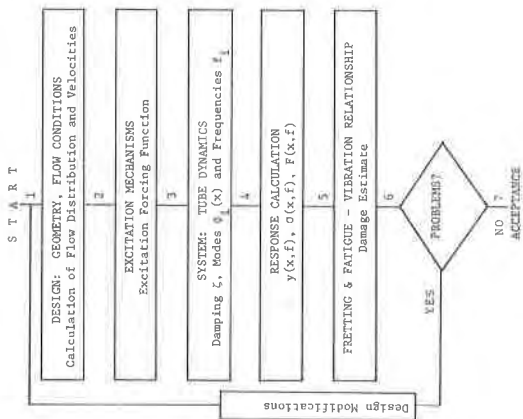


FIG. 9: Ideal Approach to Design Analysis

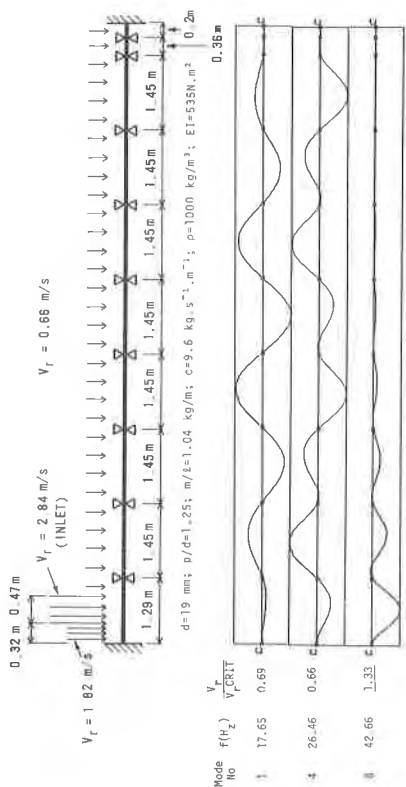


FIG. 8b: Vibration of Process Heat Exchanger Tubes in Liquid Flow and Selected Natural Modes

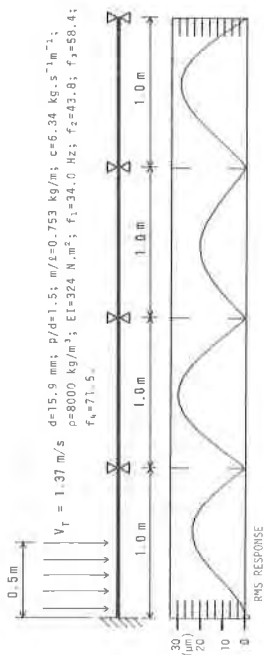


FIG. 8c: Vibration Response to Random Turbulence in Entrance Region of a Steam Generator



FIG. 10: Biaxial Accelerometer Probe for Tube Vibration Measurements