GUIDELINES FOR RANDOM EXCITATION FORCES DUE TO CROSS FLOW IN STEAM GENERATORS

C.E. Taylor and M.J. Pettigrew

ABSTRACT

Random excitation forces can cause low-amplitude tube motion that will result in long-term fretting-wear or fatigue. To prevent these tube failures in steam generators and other heat exchangers, designers and trouble-shooters must have guidelines that incorporate random or turbulent fluid forces. Experiments designed to measure fluid forces have been carried out at Chalk River Laboratories and at other labs around the world. The data from these experiments have been studied and collated to determine suitable guidelines for random excitation forces.

In this paper, a guideline for random excitation forces in single-phase cross flow is presented in the form of normalised spectra that are applicable to a wide range of flow conditions and tube frequencies. In particular, the experimental results used in this study were carried out over the full range of flow conditions found in a nuclear steam generator.

The proposed guidelines are applicable to steam generators, condensers, reheaters and other shell-and-tube heat exchangers. They may be used for flow-induced vibration analysis of new or existing components, as input to vibration analysis computer codes and as specifications in procurement documents.

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1. INTRODUCTION

Many heat exchangers operate with a single-phase fluid. As well, tubes near the inlet region and preheater of two-phase heat exchangers, such as a nuclear steam generator, are exposed to single-phase cross flow. In single-phase cross flow, three major mechanisms can lead to tube failure in a shell-and-tube heat exchanger: fluidelastic instability, periodic wake shedding and random excitation forces. Both fluidelastic instability and periodic wake shedding can cause large amplitude vibrations in a tube bundle and quickly lead to catastrophic tube failure. However, random excitation forces cause comparatively small amplitude vibrations, which will not lead to short-term failure. These vibrations do lead to continuous rubbing of a tube against its supports resulting in progressive damage to the tube due to fretting-wear. As existing reactors age, it is increasingly important to be able to assess the effects of random excitation vibration to determine the maximum safe operating life for each component. As well, the production of more reliable and longer lasting steam generators requires an accurate assessment of tube response to random excitation during the design process. To this end, it is necessary to have accurate design guidelines.

While a great deal of experimental and theoretical work on flow-induced vibration is available, it is widely agreed that design-oriented information focusing on random excitation vibration is insufficient. In preparation for this paper, a database containing most of the available single-phase random-excitation data for tube bundles was compiled. Based on the available data, this paper presents a guideline for determining the random excitation forces in tube bundles subjected to single-phase cross flow.

2. THEORETICAL BACKGROUND

2.1 Random Vibration Theory

The equation for the mean square tube deflection has been derived from the equation of motion for the forced vibration of a beam. The initial stages in the development of this equation can be found in advanced vibration texts, but the final form, applied to cross flow, was first presented by Pettigrew and Gorman (1973). They show that the mean square of tube response for mode 1 can be expressed as follows:

$$\overline{y^2(x)_1} = \frac{\phi_1^2(x) S_F(f) J_1^2}{64\pi^3 f_1^3 M_1^2 \zeta_1} \tag{1}$$

where, ϕ_1 is the normalised mode shape for mode 1, $S_F(f)$ is the auto-power spectral density of the input force per unit length, J_1^2 is the joint acceptance for mode 1, f_1 is the tube natural frequency for mode 1, M_1 is the generalised mass for mode 1, and ζ_1 is the generalised damping ratio for mode 1.

It should be noted that, for most applications, the generalised mass, M_1, and the generalised damping, ζ_1, can be assumed to be constant along the tube length. Under this assumption, the generalised mass is equal to the tube mass per unit length. The total mass per unit length, m, is equal to the tube mass, m_t, plus the hydrodynamic mass, m_h. The joint acceptance, J_1^2, can be calculated if the correlation factor is known. The correlation factor is difficult to measure and values have been reported in very few papers in the literature.

Antunes (1986) has shown that for values of λ_c/L <<1, the joint acceptance is proportional to the correlation length divided by the excited tube length, λ_c/L_e, if the correlation factor is represented by a function which is exponentially decreasing with separation distance along the tube, such as $\gamma(x,x') = exp(-|x-x'|/\lambda_c)$. An expression for the joint acceptance can then be written using a proportionality coefficient, a_i, as $J_i^2 = a_i\, \lambda_c / L_e$. Then, the expression defining the first mode mean square tube response, $\overline{y^2(x)_1}$, in terms of the power spectral density of the excitation force, $S_F(f)$, can be written as follows:

$$\overline{y^2(x)_1} = \frac{\phi_1^2 S_F(f)\, a_1\, {}^{\lambda_c}\!/_{L_e}}{64\pi^3 f_1^3 m^2 \zeta_1} \qquad (2)$$

Values for $\phi_1(x)$ and a_1 for a variety of end conditions are given in Table 1.

2.2 Nature of the Flow

Inside the tube bundle, the pitch velocity, V_p, is defined as $V_p = V_\infty\, P/(P-D)$ where, P is the pitch between tubes, D is the tube diameter and V_∞ is the free stream (or upstream) velocity. A highly turbulent upstream flow can cause a significant increase in tube response in a tube bundle. This effect is discussed further in Section 5.4.

3. LITERATURE SEARCH

A great deal of research on flow-induced vibrations has taken place in the past 25 years. However, much of this work has focused on fluidelastic instability and vortex-shedding, since these mechanisms can quickly lead to catastrophic failures. It is comparatively recently that random excitation vibration has been widely studied. As well, much random excitation research has been conducted on single tubes and tube rows, rather than tube bundles.

One of the earliest sources of data on single-phase random excitation in tube bundles was presented by Pettigrew and Gorman (1978). This work covered a variety of bundle orientations, though only in a narrow reduced frequency band. Chen and Jendrzejczyk (1987) gave the results of tests on a normal square bundle over a wide range of reduced frequencies. Axisa et al. (1990) presented new tests on normal triangular, square and rectangular bundles in air and water. Their results were compared with row data from Taylor et al. (1988) and an upper boundary was proposed. Oengören and Ziada (1992) proposed a boundary based on tests on a normal square bundle in air cross flow as well as bundle data from Chen and Jendrzejczyk (1987) and row data from Taylor et al. (1988). Blevins (1990) presented another bounding spectrum based on data from Axisa et al. (1990), Taylor et al. (1988) and Chen and Jendrzejczyk (1987). It should be noted that the boundary spectrum given in Axisa et al. (1990) was presented incorrectly in both Blevins (1990) and in Oengören and Ziada (1992). Tests on a normal square bundle in water are reported in Taylor et al. (1996). Results from a normal triangular bundle in air are provided in Oengören and Ziada (1992). Wolgemuth (1994), while focused primarily on the effect of tube support clearances, gave valuable data on the effect of highly turbulent flow on random excitation forces in a tube bundle. The geometric characteristics for the tube bundles used in each of these studies are summarised in Table 2

4. APPROACH TAKEN

Axisa et al. (1990), Blevins (1990) and Oengören and Ziada (1992) have compared their data with that of other researchers and proposed bound spectra. In this paper, the effect of many different parameters such as bundle geometry, upstream turbulence, and fluid properties will be considered as new bound spectra are developed. These upper bounds can be used as guidelines by heat exchanger designers, trouble shooters and analysts.

The experimental studies examined in this paper have collected random excitation data using one of two methods:

1) Random excitation determined from reaction forces. These tests used force transducers to directly measure the reaction forces.
2) Random excitation determined from tube response. In these tests, strain gauges were used to measure tube displacement. In this case, tube displacement was used to calculate excitation force *PSD*s, using Equation 6.

To be able to compare the data and find an upper bound, the results must be presented as a normalised excitation force spectrum. Researchers in this field have used various methods of normalising their results. Therefore, it was necessary to select one means of normalisation and apply it all of the data. The method adopted was the "equivalent power spectral density" (*EPSD*) described by Axisa et al. (1990).

The power spectral density, $S_F(f_R)$, can be rendered dimensionless using the dynamic pressure head, as follows:

$$\tilde{S}_F(f_R) = \frac{S_F(f)}{\left(\frac{1}{2}\rho V_p^2 D\right)^2} \frac{V_p}{D} \tag{3}$$

where, f_R is the reduced frequency, defined as fD/V_p , ρ is the fluid density, V_p is the pitch velocity, and D is the tube diameter.

A difficulty arises in the calculation of $S_F(f)$ since the correlation length, λ_c, is rarely known. Axisa et al. (1990) present an "equivalent power spectral density" (*EPSD*), $\tilde{S}_F(f_R)_e$, defined as:

$$\tilde{S}_F(f_R)_e = \frac{\lambda_c}{L_e}\,\tilde{S}_F(f_R) \tag{4}$$

Substituting Equations 2 and 3 into Equation 4, one obtains:

$$\tilde{S}_F(f_R)_e = \frac{\overline{y^2(x)}_1 64\pi^3 f_1^3 m^2 \zeta_1}{\phi^2 a_1} \frac{1}{\left(\frac{1}{2}\rho V_p^2 D\right)^2} \frac{V_p}{D} \tag{5}$$

Using Equation 5, the mean square tube displacement can be found without knowledge of the correlation length. Instead, a small correlation length has been assumed, as discussed in Section 2.1. To correctly compare spectra obtained using experimental rigs with varying geometries, it is necessary to reference a single excited tube length, L_e. In this paper, a reference length of 1 metre is applied, as follows:

$$\tilde{S}_F(f_R)_{e\ L=1m} = \tilde{S}_F(f_R)_e \times \frac{L_e}{1} \tag{6}$$

5. DISCUSSION OF PARAMETERS

5.1 Directional Dependence (Lift vs. Drag)

Figure 1 gives three examples of lift and drag results from different papers in the literature. Based on this figure, it can be said that the excitation forces in the lift direction are either greater than or equal to the forces in the drag direction. Using this fact, the remainder of this paper deals only with lift direction results.

5.2 Bundle Orientation

The standard tube bundle orientations are normal square, rotated square, normal triangular and rotated triangular. In a real heat exchanger, both normal and rotated flow orientations are possible within a given tube bundle. Therefore, it is necessary to consider the bundle orientation with the highest excitation force spectrum for design purposes. The bounds for a normal square and a normal triangular bundle of similar *P/D* are shown in Figure 2. The differences in excitation force magnitude between the bundles are not significant for *P/D* < 3.0. Similarly, Pettigrew and Gorman (1978) (1.23 < *P/D* < 1.57) did not find any significant differences between the various bundle orientations.

5.3 Pitch-to-Diameter Ratio (*P/D*)

Oengören and Ziada (1995) tested normal triangular tube bundles with a large range of pitch-to-diameter ratios. Oengören and Ziada (1992) tested normal square bundles over a smaller range of *P/D*. The results of these tests are shown in Figures 3a and 3b, respectively. The data does not reveal a clear trend. From *P/D* = 1.26 to *P/D* = 1.95, there is an increase in *EPSD* magnitude with pitch-to-diameter ratio. However, at *P/D* = 2.08 and greater, there is no such trend.

Most heat exchangers are designed with pitch-to-diameter ratios between 1.2 and 1.6. The proposed guideline in this paper will be based on data from *P/D*s of 1.5 to 1.75. The designer of tube bundles with smaller *P/D*s may wish to use the trends shown in Figure 3 to appropriately reduce the proposed upper bound that is presented in Section 5.6.

5.4 Upstream Turbulence

Highly turbulent flow, such as that found in the inlet region of a steam generator, is observed to have significant effect on tube response. Figure 4 (Wolgemuth 1994) shows the tube responses for the first two rows and two interior rows when the bundle was exposed to highly turbulent flow.

A high upstream turbulence can cause a significant increase in tube response. However, this effect is observed only in the first few rows of the bundle. Wolgemuth (1994) also showed that by the fifth row, tube response is similar to that caused by uniform flow.

Since turbulence can cause a significant increase in tube response, a separate bound will be proposed for use as a guideline when designing tube bundles which will be exposed to highly turbulent upstream flow.

5.5 Fluid Density (Gas vs Liquid)

Figure 5 shows *EPSD* results for air and water data. Most heat exchangers using gas as a secondary fluid will have very high flow rates and consequently, low reduced frequencies. Examination of Figure 5 shows that most air data fall below $f_R = 0.2$. Very few water heat exchangers have flow rates that result in reduced frequencies below 0.2. Thus, the proposed guidelines will place a greater emphasis on air results for $f_R < 0.2$.

5.6 Summary

The proposed boundary spectra of equivalent power spectral density of the excitation force per unit length (*EPSD*) resulting from single-phase cross flow for bundles with *P/D*s between 1.5 and 1.75 are shown in Figure 6. The general collapse of the data is surprisingly good. An upper bound is drawn over most of the data and then a second boundary is provided for tubes that are subjected to highly turbulent inlet flows.

A comparison with previously proposed guidelines is presented in Figure 7. The guideline proposed in this paper is based on the largest amount of data from the literature and therefore provides an upper bound. The bound from Blevins (1990) includes the effects of vortex shedding.

6. DESIGN GUIDELINES

This section will provide the designer with the necessary methodology to estimate the random excitation forces for specified flow conditions. These guidelines are based on a limited database and should not be applied outside of the limitations given.

The lower bound in Figure 6 should be used when the upstream turbulence is less than or equal to the turbulence within the tube bundle. The upper bound in Figure 6 should be used if the upstream turbulence exceeds the turbulence inside the tube bundle.

Using the following expression, the random excitation power spectral density times the correlation length, $S_F(f)\lambda_c$, can be calculated from the equivalent power spectral density.

$$S_F(f)\lambda_c = \tilde{S}_F(f_R)_e \text{ x } \frac{\left(\frac{1}{2}\rho V_p^2 D\right)^2 D}{V_p} \tag{7}$$

where ρ is the fluid density [kg/m^3], V_p is the pitch velocity [m/s] and D is the tube diameter [m].

For a single-span tube, the maximum mean square vibration response , $\overline{y^2}$, can be calculated from the power spectral density of the excitation force as follows:

$$\overline{y^2(x)_1} = \frac{[S_F(f)\lambda_c]\,\phi_1^2(x)a_1}{64\pi^3 f_1^3 m^2 \zeta_1 L_e} \tag{8}$$

where, $\phi_1(x_1)$ is the normalised mode shape for the 1st mode, a_1 is the numerical coefficient for the 1st mode, f is the tube natural frequency in single-phase flow, m is the total tube mass (tube mass + hydrodynamic mass), ζ_1 is the damping ratio for the 1st mode, and L_e is the excited tube length (the length of the portion of the tube that is subjected to flow). Table 1 contains values of ϕ_1 and a_1 for various end conditions.

Once $S_F(f)$ is known, the tube response for multispan heat exchanger tubes can be calculated using computer codes such as PIPO and VIBIC.

7. ACKNOWLEDGEMENTS

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Table 1: Modal Factor (a_1) and Mode Shape $(\phi_{1\,max})^2$ Constants for Mode 1

	Rigid Tube	Clamped-Clamped	Clamped-Pinned	Clamped-Free	Pinned-Pinned
Modal Factor	2	0.8	0.9	0.5	1.1
Mode Shape	Translation	2.522	2.278	4.0	2.0

Table 2: Summary of Bundle Geometries

Reference	Bundle Orientation	Tube Diameter (mm)	P/D (P/D drag)	Excited Length (m)	Total Mass (kg/m)	Natural Freq. (Hz)	End Conditions	Notes
Taylor (1996)	90	30	1.5	0.3	-	268	Clamped-Clamped	Water flow
Pettigrew & Gorman (1978)	30	19	1.33	0.051	1.19	40	Clamped-Free	Water flow
	30	13	1.36, 1.54	0.051	0.52	30	Clamped-Free	"
	30	19	1.57	0.051	1.19	40	Clamped-Free	"
	45	13	1.3	0.051	0.52	30	Clamped-Free	"
	60	19	1.33, 1.57	0.051	1.19	40	Clamped-Free	"
	60	13	1.36	0.051	0.52	30	Clamped-Free	"
Oengoren & Ziada (1992)	90	20	1.26, 1.5, 1.95, 3.0	0.2	-	900	Clamped-Clamped	Air flow
Oengoren & Ziada (1995)	30	31	1.61	0.2	-	>1000	Clamped-Clamped	Air flow
	30	18	2.08	0.2	-	>1000	Clamped-Clamped	
	30	22	3.41	0.2	-	>1000	Clamped-Clamped	
Chen & Jendrezjczyk (1987)	90	25.4	1.75	0.3	-	-	Clamped-Clamped	Water flow
Axisa et al. (1990)	90	24	1.25 (2.16), 1.25(1.44)	0.25	0.624	78.8	Clamped-Free	#1 Air flow
	90	20	1.5(2.6), 1.5(1.73)	0.25	0.433	68.6	Clamped-Free	"
	90	20	1.5	0.25	0.244	57.8	Clamped-Free	"
	30	24	1.25(1.08)	0.25	0.624	78.8	Clamped-Free	"
	30	20	1.5(1.3)	0.25	0.433	68.6	Clamped-Free	"
	30	15	2.0(1.73), 2.0(1.15)	0.25	0.244	60.4	Clamped-Free	"
	90	25	1.5	0.48	0.3	33	Clamped-Free	#2 Air Flow
	60	38	1.18	0.43	1.71	109	Clamped-Free	#3 Water Flow
	90	19.05	1.44 (80 U-tubes)	0.3 (average)	-	-	Clamped-Free	#4 Water Flow
Wolgemuth (1994)	60	16	1.38	0.87	0.6093	36	Clamped-Pinned	Water flow

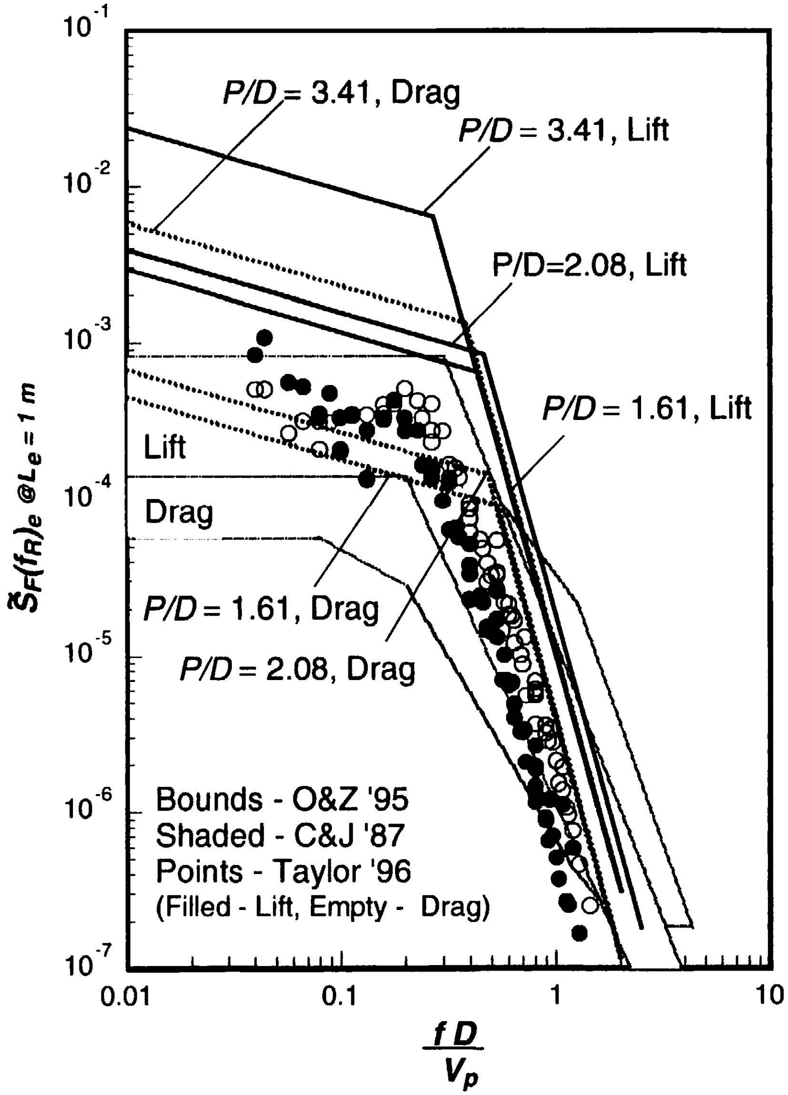

Figure 1: Directional Dependence (Lift vs. Drag)

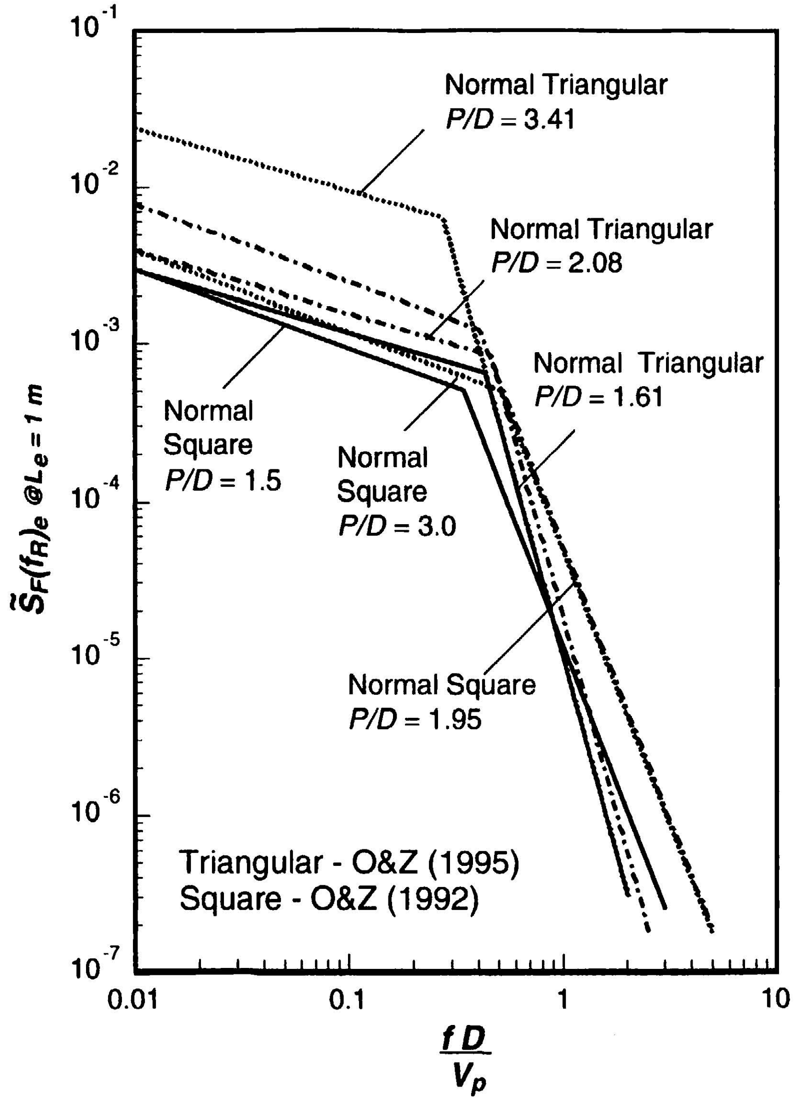

Figure 2: Effect of Tube Bundle Orientation

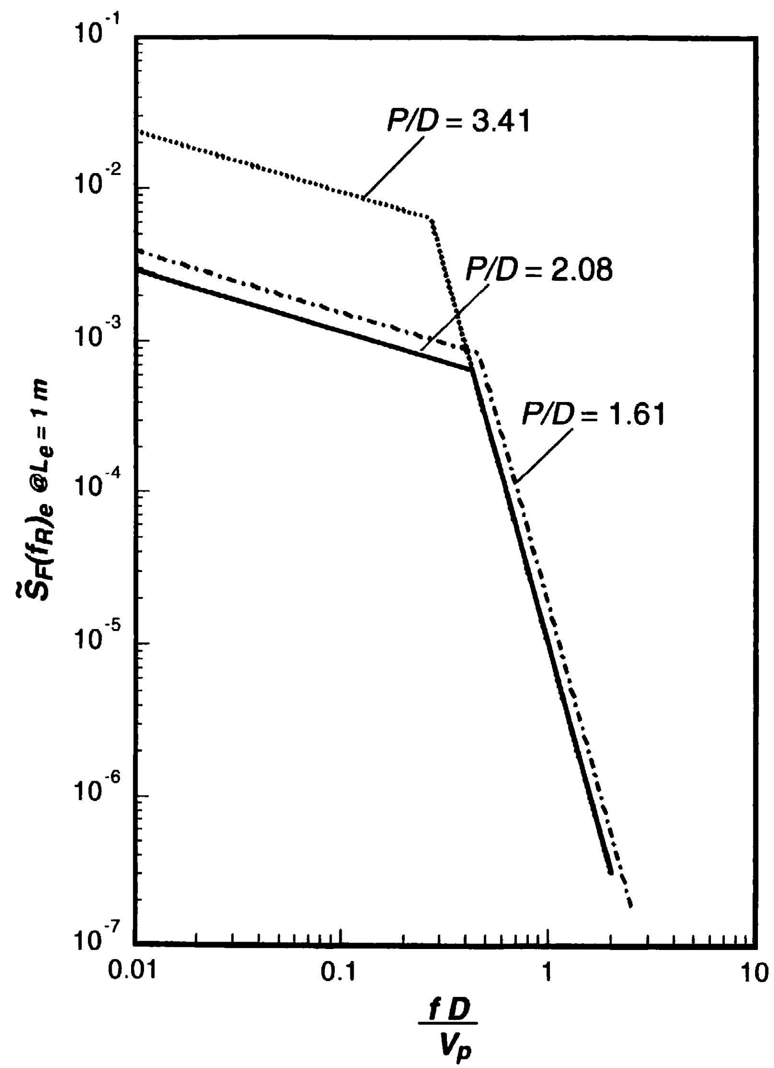

Figure 3: Effect of Pitch to Diameter Ratio
(a) Normal triangular tube bundle in air

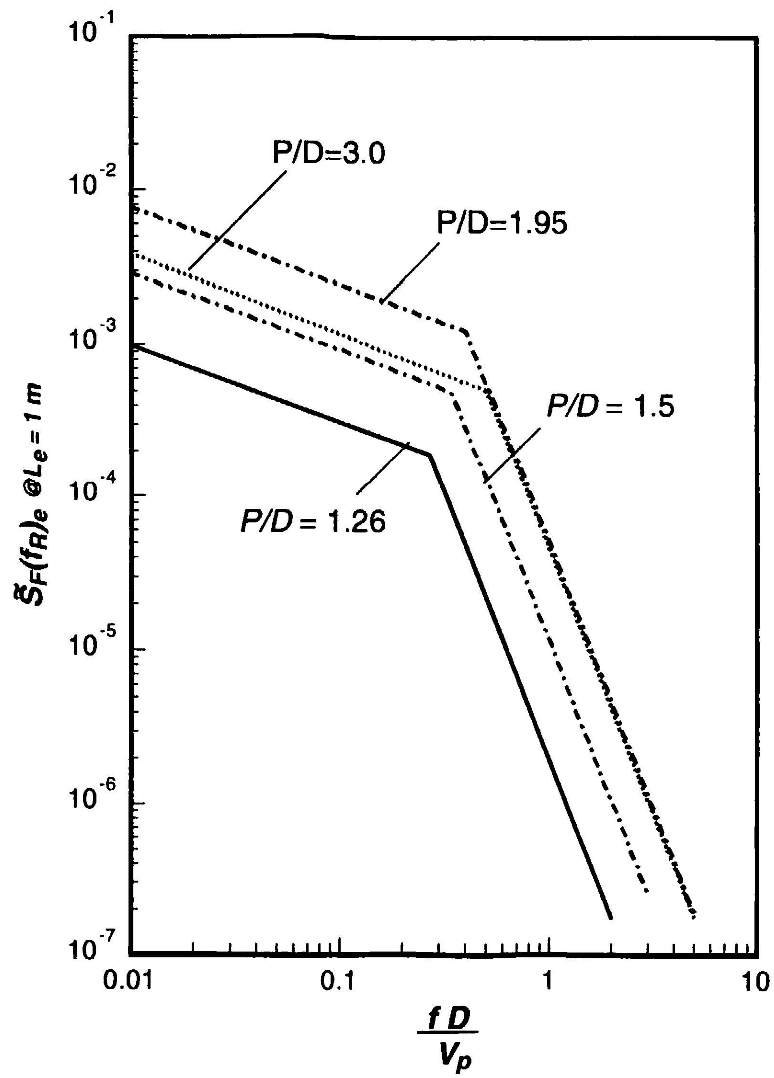

Figure 3: Effect of Pitch to Diameter Ratio
(b) Normal square tube bundle in air

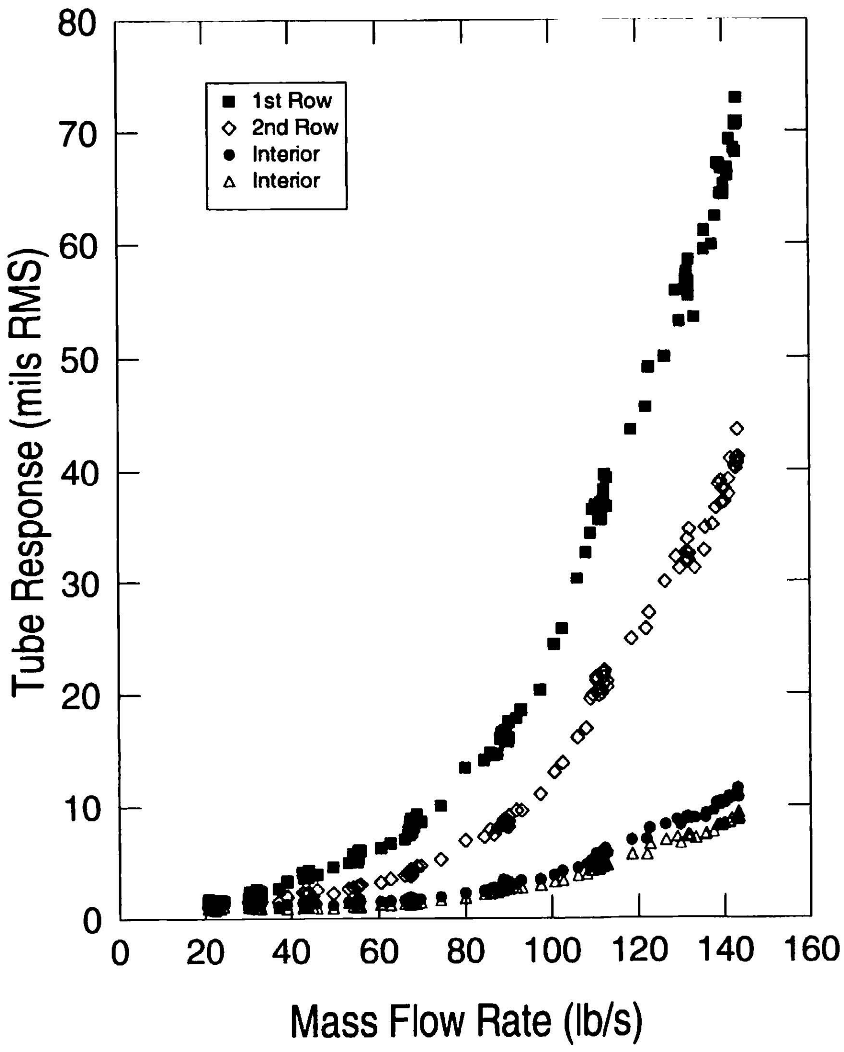

Figure 4: Effect of Upstream Turbulence

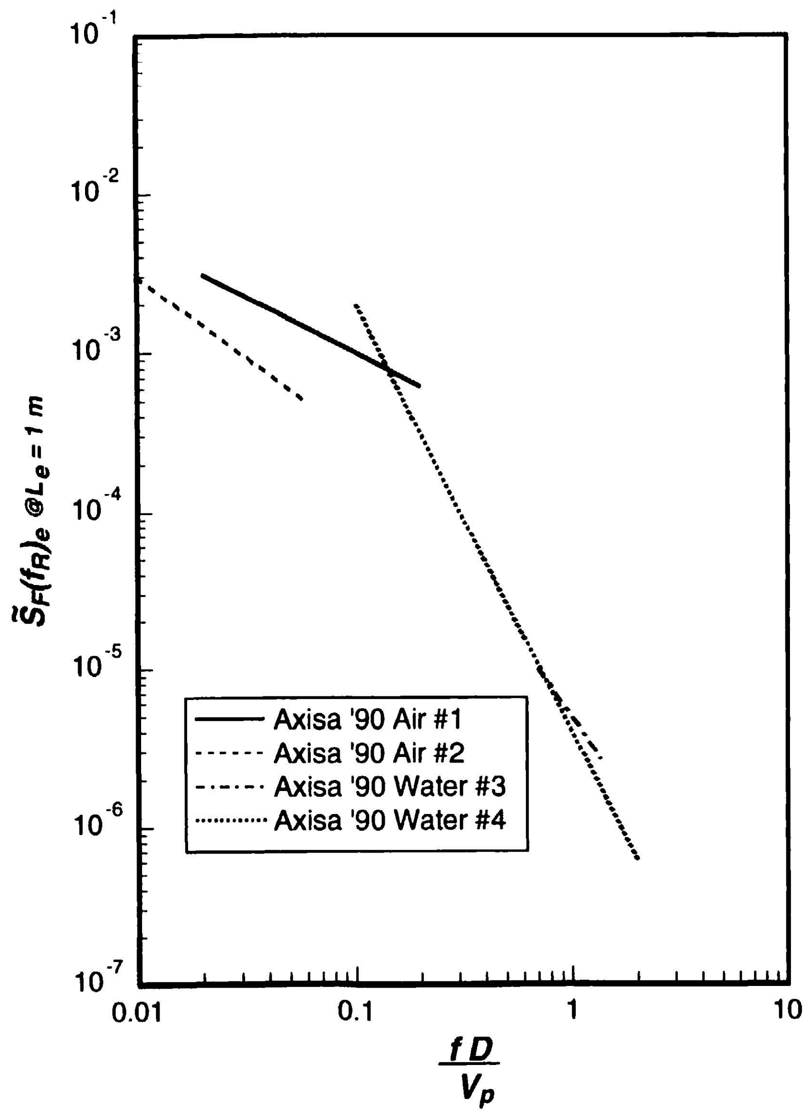

Figure 5: Effect of Fluid Density (Gas vs Liquid)

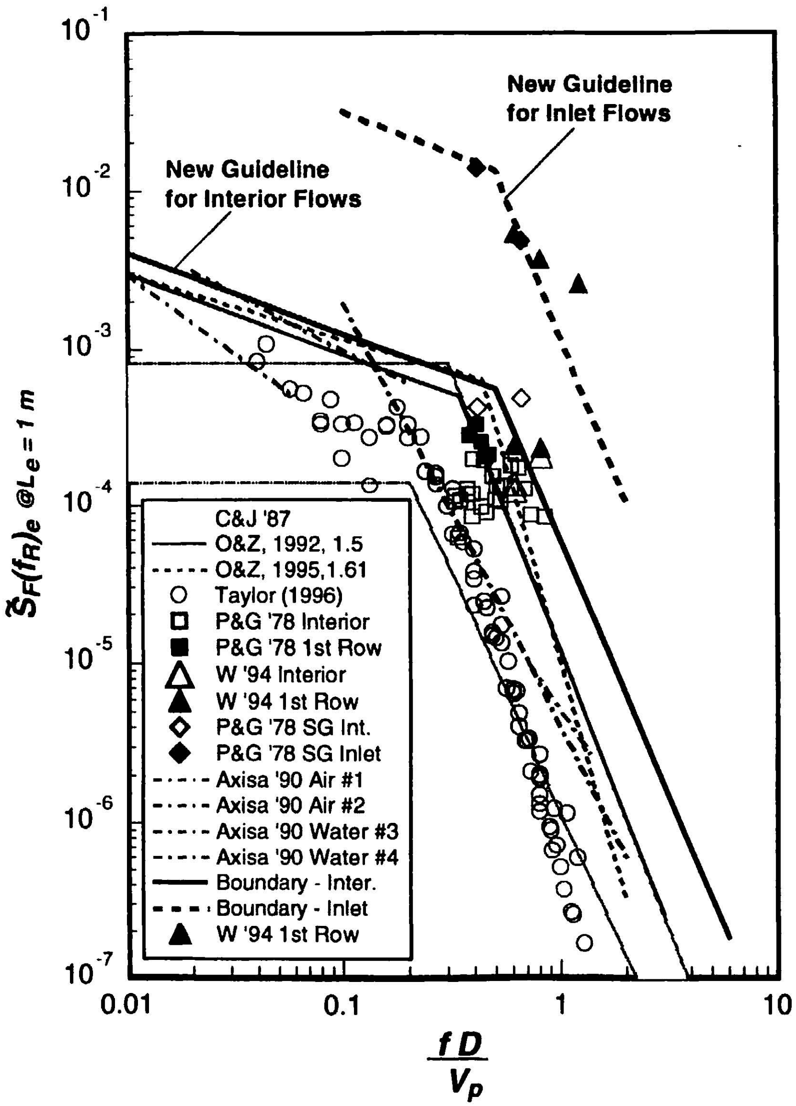

Figure 6: Proposed Guideline for Excitation Forces

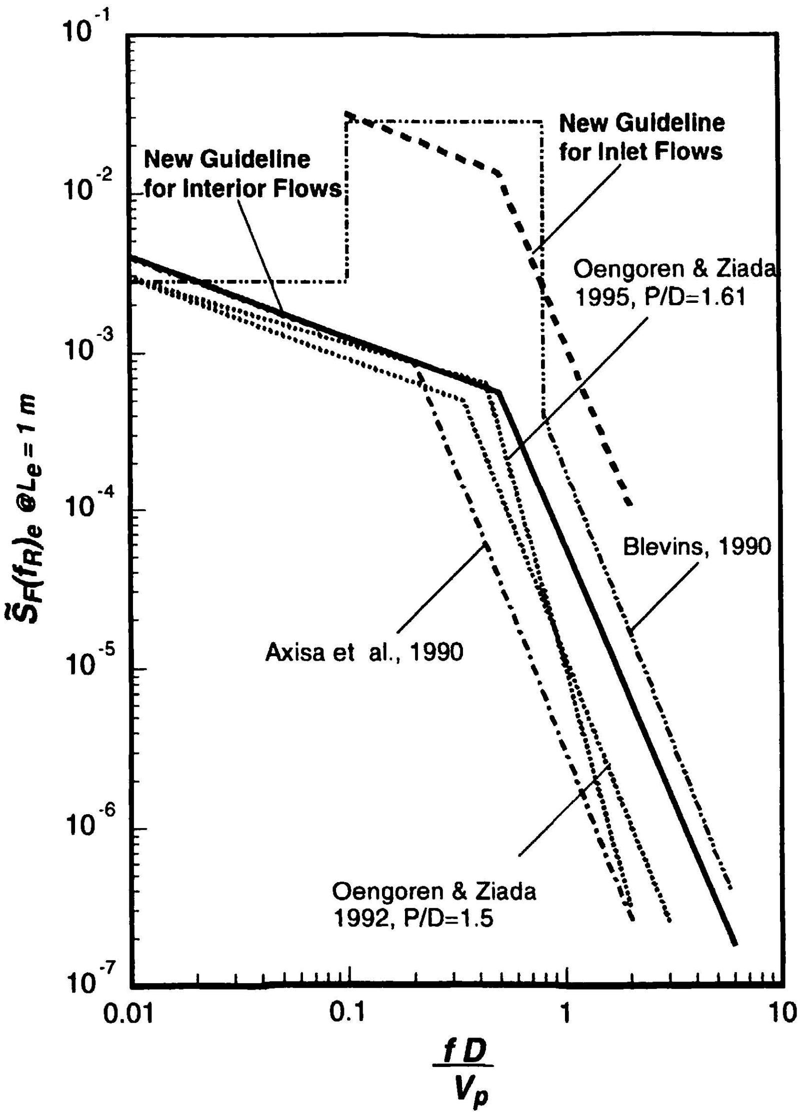

Figure 7: Comparison with Previous Guidelines