PIPO-FE: AN UPDATED COMPUTER CODE TO EVALUATE HEAT EXCHANGER AND STEAM GENERATOR FLOW-INDUCED VIBRATION

*Y. Han, V.P. Janzen, J. Kim**

AECL – Chalk River Laboratories
Inspection, Monitoring and Dynamics Branch
Chalk River, Ontario, Canada, K0J 1J0
* Co-op Student from University of Toronto

ABSTRACT

PIPO-FE is the latest version of a computer code that has been developed over the past 30 years to assess the flow-induced vibration of heat exchanger and steam generator tubes due to fluidelastic instability, random-turbulence excitation and vortex shedding. The code has recently been updated to include design guidelines prepared by Chalk River Laboratories of Atomic Energy of Canada Ltd. and the Department of Mechanical Engineering of École Polytechnique, Montréal, and to estimate fretting-wear damage caused by random-turbulence vibration. This paper provides a general description of the updated code and includes examples to illustrate how the code can be used to support the design and modification of power plant heat exchangers and related research activities.

1. INTRODUCTION

The Vibration and Tribology Section (VTS) of Atomic Energy of Canada Limited's (AECL's) Inspection, Monitoring and Dynamics Branch has studied flow-induced vibration of nuclear piping systems for over 30 years. During that time, special attention has been paid to the vibration of heat exchanger and steam generator tubes. Because of the complicated nature of flow-induced vibration in heat exchangers and steam generators, experimental measurement has been the main approach used to characterize the underlying mechanisms. Based upon the experimental results obtained by VTS and others, empirical formulations have been developed to describe vibration excitation phenomena and used in computer codes developed by AECL and others in the nuclear and process industries.

PIPO-FE is one such computer code, developed over the past 30 years and maintained and updated over time in accordance with AECL and other industry design specifications and guidelines focused on flow-induced vibration of heat exchanger and steam generator tubes. Since its inception, the code has been used both internally for various R&D and commercial projects, and externally by AECL customers to assess flow-induced vibration in the heat exchangers and steam generators used in the nuclear energy industry, and also in other sectors, such as the Canadian oil industry.

1.1 Historical Background

The first version of the code, named PIPEAU, was developed in the early 1970's. Approximately 10 years later the code was renamed PIPO1 (Campagna et al., 1988) and became part of the Heat Transfer and Fluid Flow Service (HTFS) set of programs, complete with HTFS theory and user manuals. In the early 1990's the code, now referred to as PIPO, was transferred to personal computers with Windows operating systems. PIPO-FE Version 1.1, the FE indicating the use of a finite-element approach, was developed in the early 2000's in response to customer requirements, including the need to model essentially any tube geometry. Version 1.2 was produced to comply with the Software Quality Assurance (SQA) controls, including updated code documentation.

The current version of the code, PIPO-FE V1.3, includes recent experimental findings and updated design guidelines produced through collaboration between VTS and the Department of Mechanical Engineering of École Polytechnique, Montréal (Pettigrew and Taylor, 2003).

This paper describes PIPO-FE V1.3, with a brief summary of the main features of the code and updated algorithms. The paper includes four examples of FIV analyses to illustrate the updates that have been made, in particular the revised random turbulence calculations. These examples also demonstrate how the updated code can be used to support the design and modification of power plant heat exchangers or steam generators and related research activities.

2. MAIN FEATURES

PIPO-FE is designed to assess the vibration of heat exchanger tubes and steam generator tubes in cross flow. The three dominant vibration mechanisms (fluidelastic instability (FEI), random-turbulence excitation (RTE) and vortex shedding) known to exist in heat exchangers and steam generators can be assessed by the code. The tube fretting-wear damage caused by random-turbulence excitation can also be estimated with PIPO-FE V1.3 based on an energy approach proposed by Pettigrew et al. (2007). Due to the use of bounding values of various empirical parameters in the calculations of damping and the three vibration mechanisms, the code generally yields conservative predictions.

Figure 1 shows a typical tube model used by PIPO-FE. As indicated in this figure the code analyses one tube at a time. Because it uses a finite-element formulation, the code can be used to model a U-shaped tube, as shown in this figure, a straight tube, or a tube having any shape.

PIPO-FE is designed to calculate the forced vibration of a tube based on the corresponding modal information calculated by other programs. For example, the FREMOD module of AECL's VIBIC code (Han and Fisher, 2002) can be used to calculate tube natural frequencies and mode shapes. Alternatively, commercially-available general-purpose finite-element programs can be used to obtain this tube free-vibration information.

The motivation to separate the free-vibration analysis from the forced-vibration analysis reflects the fact that there are situations when several forced-vibration analyses need to be conducted based on the same modal-analysis results. A secondary reason is that it facilitates code maintenance, since the forced-vibration portion of the code requires more frequent updates than the free-vibration code. To ease the way for further development, the PIPO-FE code has been designed and maintained in a modular organization. Therefore, a PIPO-FE simulation requires two input files, one containing the information needed for the free-vibration analysis and the other containing the input data required for the forced-vibration calculation.

When the VIBIC module FREMOD is used to conduct a free-vibration analysis, its output file MODES.TXT contains the data that PIPO-FE needs and, hence, is one of the two required input files. The user creates the other input file, FIVIN.DAT, to conduct a forced-vibration analysis. This input file includes dynamic properties of the cross flow along the tube and all the remaining information required for calculating total damping, vibration response due to the three vibration excitation mechanisms and fretting-wear damage.

As a specialized analytical code, PIPO-FE has a fast calculation speed. For a typical CANDU-type heat exchanger or steam generator tube, the time needed to run a forced-vibration analysis is typically several minutes.

The vibration-analysis results may not be easily interpreted by a non-specialist, so the program checks the results of the flow-induced vibration analysis to determine if certain basic criteria have been exceeded for the three excitation mechanisms considered. The appropriate messages appear in the output as warnings if any criteria have been exceeded. The basic vibration criteria applied in the current PIPO-FE code are the same as those used in previous versions of the code (Campagna et al, 1988 and Pettigrew and Taylor, 2003), namely:

- Fluidelastic Instability
  - The ratio of tube effective velocity over critical velocity, i.e., the FEI ratio, should not exceed one (1).
- Random-Turbulence Excitation
  - The response should not exceed 100 μm in amplitude.
- Vortex Shedding
  - At tube frequencies below 100 Hz, the vortex-shedding amplitude should not exceed 2% of the tube diameter.
  - At tube frequencies above 100 Hz, the vortex-shedding amplitude should not exceed 2% of the diameter times the ratio (100 Hz / natural frequency in Hz).

The criterion for fretting-wear is taken from the commonly accepted guideline suggested by Pettigrew and Taylor (2003), namely:

- The tube-wall fretting-wear depth for the entire life of the heat exchanger shall be less than 40% of the nominal-design tube wall thickness.

3. RECENT CODE UPDATES

The most significant recent updates of the PIPO-FE code involved changes to the calculation of vibration induced by random-turbulence excitation due to both single- and two-phase cross flows, and the addition of a prediction of fretting-wear damage. In both instances, the updated code follows the design guidelines specified by Pettigrew and Taylor (2003). In their paper, Pettigrew and Taylor summarized the experimental studies on random-turbulence excitation of tubes to date, and showed that a normalized excitation forcing function can be used to compare results of experiments with a single-span tube or tube bundle under different geometrical conditions and flow properties. The power spectral densities of the excitation forces for both single- and two-phase conditions were formulated as a function of reduced frequency.

The comparison between experimental data and design guidelines included in that paper shows that the new design guidelines are conservative. Sections 3.1 and 3.2 explain how the updated code uses the normalized forcing functions in the tube vibration prediction and outline the fretting-wear calculations. For comparison, the random-turbulence excitation formulations used in the previous version of the code can be found in Pettigrew et al. (1998, 1991) for single- and two-phase flow conditions, respectively.

3.1 Calculation of Random-Turbulence Response

For a tube such as the example shown in Figure 1, when the coupling between the normal modes is small or negligible, the tube response amplitude, *y(x)*, due to random turbulence excitation can be calculated using the following general equation:

$$\overline{y^2(x)} = \sum_s \frac{\phi_s^{\,2}(x)}{16\pi^4 f_s^4} \int_o^\infty H_s^*(f) H_s(f) \left(\int_o^L \int_o^L \phi_s(x)\phi_s(x') R(x,x',f)\, dx\, dx' \right) df \qquad (1)$$

where,

$$H_s(f) = \frac{1}{M_n\left(1-(f/f_s)^2 + 2j\zeta f/f_s\right)}$$

is the modal transfer function, H_s^* is the complex conjugate of H, ϕ_s and f_s are the tube normalized mode shape and natural frequency, respectively, for mode s, L is the tube length, $R(x,x',f)$ is the spatial correlation density function of the excitation force, ζ is the damping ratio, j is the imaginary unit, and M_n is the generalized mass that is defined as the following for a given vibration mode,

$$M_n = \int_0^L \phi(x) M \phi(x)\, dx \qquad (2)$$

In this equation, M is the total mass per unit length. Note that with this type of mode-shape generalization, a tube with variable inside and outside diameters or non-uniform mass density can be simulated. An example is a leaking nuclear steam generator tube with a sleeve or a stabilizer (a rod or a cable) that seldom spans the entire length of the tube. Another example is a secondary-side fluid density that varies along the length of the tube, e.g., from the water phase to the steam phase in a steam generator.

Such cases give rise to a non-uniform hydrodynamic mass, which may be higher than the mass displaced by the tube.

A difficulty arises in using Equation 1 because the spatial correlation density function *R(x,x'f)* is usually difficult to determine. Various methods have been used in order to estimate the spatial correlation density functions. Previous versions of the PIPO-FE code handled the distributed random forces largely by implementing methods suggested by Pettigrew and Gorman (1973).

The present version of the code uses the following assumptions:

1) The spatial correlation density function can be expressed as the product of a spatial correlation factor, $\gamma(x,x',f)$, and the power spectral density function $S(x,x',f)$, and

$$R(x,x',f)=\gamma(x,x',f)S(x,x',f) \tag{3}$$

2) The flow is well separated by the support plates and the random forces are completely uncorrelated between spans. Therefore, based on the statistical concept for a random process the spatial correlation density function can be considered for each tube span separately and it contributes to the tube response only due to the flow crossing the tube region within that span.

With these assumptions, and keeping in mind the statistical concept for a random process, the response of the tube can be calculated as

$$\begin{aligned}\overline{y^2(x)}=\sum_S \frac{\phi_s^{\,2}(x)}{16\pi^4 f_s^4}\Bigg[&\int_o^\infty H_s^*(f)H_s(f)\left(\int_o^{L_{SP1}}\int_o^{L_{SP1}}\phi_s(x)\phi_s(x')S_{F_{SP1}}(x,x',f)\,dxdx'\right)df\\ &+\ldots+\int_o^\infty H_s^*(f)H_s(f)\left(\int_{L_{SPn-1}}^{L_{SPn}}\int_{L_{SPn-1}}^{L_{SPn}}\phi_s(x)\phi_s(x')S_{F_{SPn}}(x,x',f)\,dxdx'\right)df\Bigg]\end{aligned} \tag{4}$$

where subscripts *SP1…SPn* denote the first span … nth (last) span.

Note that $S_{F_{SPi}}(x,x',f)$ as defined in Pettigrew and Taylor (2003), for *i=1…n*, is the power spectral density calculated based on the excitation length, and is a function of frequency and also the location along the tube. In the above equation, we need to include the power spectral density inside the double integral to take care of the effect of the flow velocity variation. One can take the power spectral density out of the double integral only when the flow, density and tube outside diameter are uniform over the whole span.

Substituting $S_{F_{SPi}}(x,x',f)$ from Pettigrew and Taylor (2003) into the above equation yields

$$\begin{aligned}\overline{y^2(x)}=\sum_S \frac{\phi_s^{\,2}(x)}{16\pi^4 f_s^4}\Bigg[&\frac{1}{L_{e1}}\int_o^\infty H_s^*(f)H_s(f)\left(\int_o^{L_{SP1}}\int_o^{L_{SP1}}\phi_s(x)\phi_s(x')\left(\rho U^2 D/2\right)^2(D/U)\widetilde{S}_{F_{SP1}}(f_R)_{e@L=1m}\,dxdx'\right)df\\ &+\ldots+\frac{1}{L_{en}}\int_o^\infty H_s^*(f)H_s(f)\left(\int_{L_{Spn-1}}^{L_{SPn}}\int_{L_{SPn-1}}^{L_{SPn}}\phi_s(x)\phi_s(x')\left(\rho U^2 D/2\right)^2(D/U)\widetilde{S}_{F_{SPn}}(f_R)_{e@L=1m}\,dxdx'\right)df\Bigg]\end{aligned} \tag{5}$$

Where L_{ei} and $\widetilde{S}_{F_{SPi}}(f_R)_{e@L=1m}$, $i = 1 \ldots n$, are the excitation length within each span and the reference equivalent power spectral density (EPSD), respectively, ρ is the single-phase fluid density, U is the flow pitch velocity, f_R is the reduced frequency $(= fD/U)$, D is the tube outside diameter for single-phase and scaled diameter $(= 0.1D/\sqrt{1-\varepsilon_g})$ for two-phase case, and ε_g is the void fraction.

Note that this equation is applicable to the case when the tube mass is non-uniform, provided that the mass-normalized modes are used in the equation normalization process. This formulation of the new power spectral density allows the use of a scaling length (excitation length) to calculate the equivalent force spectrum and thus eliminates the need for a correlation length. The force correlation has essentially been included in the formulation by using power spectral densities based on experimental results.

The reference EPSD are defined as follows.

For the single-phase case:

Interior flow
$$\begin{aligned} \widetilde{S}_{S_{SPi}}(f_R)_{e@L=1m} &= 4\times10^{-4}(f_R)^{-0.5}, && 0.01 < f_R < 0.5 \\ \widetilde{S}_{S_{SPi}}(f_R)_{e@L=1m} &= 5\times10^{-5}(f_R)^{-3.5}, && 0.5 \le f_R \end{aligned} \tag{6}$$

Inlet flow
$$\begin{aligned} \widetilde{S}_{S_{SPi}}(f_R)_{e@L=1m} &= 1\times10^{-2}(f_R)^{-0.5}, && 0.01 < f_R < 0.5 \\ \widetilde{S}_{S_{SPi}}(f_R)_{e@L=1m} &= 1.25\times10^{-3}(f_R)^{-3.5}, && 0.5 \le f_R \end{aligned} \tag{7}$$

For the two-phase cases:

$$\begin{aligned} \widetilde{S}_{S_{SPi}}(f_R)_{e@L=1m} &= 16(f_R)^{-0.5}, && 0.001 < f_R < 0.05 \\ \widetilde{S}_{S_{SPi}}(f_R)_{e@L=1m} &= 2\times10^{-3}(f_R)^{-3.5}, && 0.05 \le f_R \end{aligned} \tag{8}$$

When the fluid density, flow velocity and tube outside diameter are constant along each span, Equation 5 may better be written as

$$\begin{aligned} \overline{y^2(x)} = \sum_S \frac{\phi_s^2(x)}{16\pi^4 f_s^4} \Bigg[& \frac{1}{L_{e1}}\left(\rho_{SP1}U_{SP1}^2 D/2\right)^2 \left(D/U_{SP1}\right) \int_o^{L_{SP1}}\int_o^{L_{SP1}} \phi_s(x)\phi_s(x')dxdx' \int_o^{\infty} H_s^*(f)H_s(f)\widetilde{S}_{F_{SP1}}(f_R)_{e@L=1m}\,df \\ & + \ldots + \frac{1}{L_{en}}\left(\rho_{SPn}U_{SPn}^2 D/2\right)^2 \left(D/U_{SPn}\right) \int_{L_{SPn-1}}^{L_{SPn}}\int_{L_{SPn-1}}^{L_{SPn}} \phi_s(x)\phi_s(x')dxdx' \int_o^{\infty} H_s^*(f)H_s(f)\widetilde{S}_{F_{SPn}}(f_R)_{e@L=1m}\,df \Bigg] \end{aligned} \tag{9}$$

3.2 Prediction of Fretting-Wear Damage

Earlier experimental and numerical studies by VTS have shown that there are two ways to predict fretting-wear damage: one calculates the fretting-wear in the time domain, the other in the frequency domain. A time-domain calculation requires detailed information on how the tube contacts the support and on the tube displacement

(Pettigrew et al., 1991); a frequency-domain calculation uses an energy approach that relates the rate of dissipation of vibration energy in a mechanical structure with its vibration amplitude, mode shape, mass and damping (Pettigrew et al., 2007). Verification and validation results have been included in Pettigrew et al. (2007) to show that the energy approach is appropriate.

PIPO-FE contains or provides all the information required to estimate tube fretting-wear damage using the energy approach. The normal work rate, $\dot{W}_N$, for the worst mode (defined as the mode which has the highest normal work rate) is calculated in the PIPO-FE code using the following equation

$$\dot{W}_N = 16\pi^3 f^3 Ml\,\overline{y_{\max}^2}\,\zeta_s \qquad (10)$$

where f is the natural frequency of the tube for the worst mode, l is the tube length of the span where the vibration amplitude is maximum, $\overline{y_{\max}^2}$ and ζ_s are, respectively, the corresponding maximum mean-square vibration amplitude calculated using Equation 5 or Equation 9 and the damping ratio attributed to the supports.

Once the work rate value has been calculated, the volume fretting-wear rate, $\dot{V}$, is estimated with the modified Archard equation,

$$\dot{V} = K_{FW}\dot{W}_N \qquad (11)$$

where K_{FW} is a fretting-wear coefficient, which is determined from experiments for material combinations of a tube and its support.

The tube-wall wear depth is then estimated based on the wear rate and the length of time the tube has been in service. The calculations for a tube within a support, which can be a drilled hole, or formed by scalloped bars, lattice bars, FURs and AVBs, are conducted using the geometrical information of the tube and the support.

4. APPLICATIONS

PIPO-FE may be used at the design stage to assess a new heat exchanger or steam generator design, or during the operational stage to investigate a tube failure and to determine if the damage is caused by vibration. If a vibration problem exists, the code can be used to assess the effectiveness of any proposed design modifications.

PIPO-FE has also been used to assist in laboratory measurements of tube vibration. Simulation results of a test setup prior to or during the test can help VTS to plan and conduct test programs. For example, simplified equations were often used to back out power spectral density functions of excitation forces when tube response amplitudes are measured through experiments. For a test tube with more than one span, a simplified equation is not always available. In that case, PIPO-FE could be a very useful tool to calculate the vibration response.

Compliant with AECL's SQA program, the PIPO-FE code has updated theory and user manuals, and validation/verification reports that include sample cases. As a result it is straightforward to create input files. The simulation results and comments included in each output file are designed to help the user to easily interpret the vibration properties of the analyzed tube.

A report is being prepared at AECL to document all the validation and verification cases conducted for the PIPO-FE code, including several new cases added specifically for the most recent update. Another report will be prepared to document a comparison of the available formulations for random-turbulence-induced excitation.

In Sections 4.1 to 4.4, four cases are presented to show that the code update has been done correctly and that the updated code can be used to assist laboratory research activities and to support the design and modification of power plant heat exchangers and steam generators. These cases also serve as validation or verification examples of the updated code. In Section 4.5, a more detailed listing of the code output is described that includes an estimation of the fretting-wear due to random-turbulence excitation.

Note that even though this paper focuses on the calculation of random turbulence-induced vibration and the related fretting-wear, the code output also includes the results for damping, fluidelastic instability ratio and vortex shedding for completeness. In many applications, the fluidelastic instability ratio is particularly significant in assessing the FIV performance of the design.

4.1 Case 1 – Single-Span Tube Subjected to Interior Uniform Cross Flow

Table 1 lists the input data for a single-span tube pinned at both ends. This is a particularly simple example used to test the computer algorithms that predict tube vibration response due to turbulence-induced excitation. Both hand calculation and PIPO-FE predictions were conducted. A total of 80 beam elements were employed for the tube discretization in the free-vibration analysis to create MODES.TXT.

The comparison of frequency and response to turbulence-induced excitation for the first vibration mode is shown in Table 2. The similarity of the tube responses due to the excitation verifies that the PIPO-FE code Version 1.3 correctly performs the numerical calculations. The small difference is due to the different methods used to evaluate Equation 5. In the hand calculation, the modal factor and mode-shape constant given by Pettigrew and Taylor (2003) were used to approximate the integral, while PIPO-FE V1.3 calculates the integral explicitly.

4.2 Case 2 – Single-Span Tube Subjected to Air-Water Cross Flow

An extensive experimental program has been carried out at AECL Chalk River Laboratories to study the vibration of cantilevered tube bundles subjected to air-water two-phase cross flow. Fluidelastic instability ratios and responses to turbulence-induced excitation were reported by Pettigrew et al. (1989) and Taylor et al. (1989). Four tube-bundle configurations were studied: normal triangular, rotated triangular, normal square and rotated square. The test section consisted of a tube bundle, an air/water mixer-homogeniser and the associated vessel and piping as shown in Figure 2.

The properties of the tube bundle are listed in Table 3. Four simulation conditions are considered based on the experimental results for an interior tube in the rotated triangular configuration. The cross-flow conditions are shown in Table 4.

Code predictions and experimental results are compared in Table 5. Experimental results are limited to the first mode, while code predictions for the second mode obtained with PIPO and PIPO-FE V1.2 and V1.3 are also compared. The predicted fluidelastic instability ratios are the same, as expected since the algorithm is identical in all three versions of the code.

The responses to turbulence-induced excitation predicted by Version 1.3 of the code for all four of the simulations (runs) are greater than the measured values and the predicted values from earlier versions of the code (note that PIPO and PIPO-FE V1.2 use the same algorithm and, thus, predict the same level of turbulence-induced vibration). In particular, the result for Run 4 with Version 1.3, at 90% void fraction, is greater than the measured value (and, thus, conservative) while results from earlier versions of the code are less than the measured value. This comparison suggests that PIPO-FE Version 1.3 may handle high void-fraction conditions better than previous code versions.

4.3 Case 3 – Multi-Span Tube Subjected to Non-Uniform Cross Flow

This case is an analysis of a tube failure in a process heat exchanger in the Glace Bay Heavy Water Plant, due to the combination of all three of the major flow-induced vibration mechanisms. This tube was analysed by Pettigrew et al. (1977 and 1978) with an early version of PIPO. Table 6 lists the input data for the simulation. The vibration model and some analysis results from the early PIPO analysis are shown in Figure 3. This case tests the formulations used in PIPO-FE for the three vibration mechanisms, for a multi-span straight tube.

Table 7 lists the simulation results. This case shows that while the modal frequencies, vortex-shedding amplitudes and FEI ratios for the three code versions agree very well with each other, the random-turbulence-excitation results from PIPO-FE Version 1.3 are about four to five times less conservative than those obtained with earlier versions of the code.

4.4 Case 4 – A CANDU Heat Exchanger Tube

Case 4 models a U-tube in the tube bundle of a steam generator in a CANDU NGS. The support locations and geometry of the tube are shown in Figure 4, while the input data are listed in Table 8. In the U-bend region, the tube model includes the full arch plus a straight-leg portion on each side spanning three lattice grids modeled as pinned supports. A rotational bending stiffness was applied in the Y- and Z-directions at the lowest lattice grid on each side of the U-bend, to represent the restraint imposed by the neglected tube straight section on the modeled region.

With two-phase steam-water cross flow, this case verifies the PIPO-FE predictions of tube response to random turbulence-induced excitation. The tube responses to random turbulence excitation from the new version of the code are about three times higher than the previous versions. As they were for Case 3, the responses due to fluidelastic instability and vortex shedding are also included in Table 9 to show that the results for those two mechanisms remain the same.

4.5 Sample Output for Case 3

Sample code output for Case 3 is shown in Table 10. The output file includes the predicted results for the three vibration mechanisms and also the fretting-wear predictions based on the user specified fretting-wear coefficient and tube service design life. The sections of the output containing modal analysis parameters and modal information are quite lengthy and are not included in Table 10.

With the results shown in Table 10 and the modal information, one can understand the predicted tube-vibration properties. As explained in Section 2, if the results conflict with vibration guidelines, PIPO-FE provides warnings following the predictions for each vibration mechanism. If any such warnings appear in the output file, one can check which mode may cause a problem and locate the corresponding span that is most affected.

Note that the user should treat the PIPO-FE warnings differently for the three vibration mechanisms. For the fluidelastic-instability analysis, a warning indicates a significant problem (the FEI ratio has exceeded one) that needs to be addressed. Further checks are needed for the vortex shedding and random turbulence analyses.

If excessive vibration is predicted in the vortex shedding analysis, the corresponding Strouhal number(s), $S = f_sD/U_p$, should be calculated. Here, f_s is the vortex shedding frequency, D is the tube diameter and U_p is the tube pitch velocity. If the Strouhal number is between 0.33 and 0.67 for tube bundles of pitch-to-diameter ratio between 1.23 and 1.57, vortex shedding is likely. For gas heat exchangers, excessive vortex shedding is likely for Strouhal number between 0.32 and 0.70 for tube bundles with the above range of pitch-to-diameter ratios.

For random turbulence excitation, the basic criterion given in Section 2 is conservative, since there is no simple way to accurately predict an allowable vibration level. The underlying rationale is that the vibration response to random turbulence shall be sufficiently low to prevent excessive tube wall reduction due to fretting-wear. An estimation of the fretting-wear due to random turbulence excitation is included as Section [9] of the output listing. For this case identical fretting-wear predictions have been obtained with the calculations in Microsoft Excel.

5. SUMMARY

The PIPO-FE computer code was designed to assess the vibration of heat exchanger and steam generator tubes. It has been updated to calculate the vibration response due to random turbulence-induced excitation based on the most recent design-guidelines available. A new capability to assess fretting-wear damage related to random-turbulence vibration has also been added to the updated code.

The PIPO-FE code is intended to provide reasonably conservative results, which can be used to assess the expected vibration and fretting-wear performance of heat exchangers and steam generators. The examples included in this paper indicate that, compared to the previous version, the updated code reduces the degree of conservatism in the random-turbulence excitation prediction for tubes in single-phase cross flow and predicts higher responses for tubes in two-phase cross flow.

The updated code retains the advantages of a specialized analytical approach, in that it is relatively easy to run and provides results quickly. The code is able to provide a first level of interpretation based on well-established acceptance criteria for heat exchanger and steam generator tube design. The user is able to use PIPO-FE efficiently and easily in assessing the vibration properties of a heat exchanger or steam generator.

6. REFERENCES

Campagna AO, McMonagle JJ and Pettigrew MJ. 1988. PIPO1: Heat Exchanger Tube Vibration, Parts 1 and 2. HTFS Design Report No. 25.

Han Y and Fisher NJ. 2002. Validation of Flow-Induced Vibration Prediction Codes - PIPO-FE and VIBIC versus Experimental Measurements. ASME International Mechanical Engineering Conference & Exposition IMECE2002-32548. November 17-22. New Orleans, Louisiana, USA.

Pettigrew MJ and Gorman DJ. 1973. Experimental Studies on Flow-induced Vibration to Support Steam Generator Design Part III: Vibration of Small Tube Bundles in Liquid and Two-Phase Cross-Flow. International Symposium on Vibration Problems in Industry. Keswick, England.

Pettigrew MJ, Sylvestre Y and Campagna AO. 1977. Flow-Induced Vibration Analysis of Heat Exchanger and Steam Generator Designs. Invited Paper F/1, presented at the 4th International Conference on Structural Mechanics in Reactor Technology. August 15-19, San Francisco, California, USA.

Pettigrew MJ, Sylvestre Y and Campagna AO. 1978. Vibration Analysis of Heat Exchanger and Steam Generator Designs. Nuclear Engineering and Design, Vol. 48, pp. 97-115.

Pettigrew MJ, Tromp JH, Taylor CE and Kim BS. 1989. Vibration of Tube Bundles in Two-Phase Cross-Flow: Part 2 – Fluid-Elastic Instability. ASME Journal of Pressure Vessel Technology, Vol. 111, pp. 458-487.

Pettigrew MJ, Carlucci LN, Taylor CE and Fisher NJ. 1991. Flow-Induced Vibration and Related Technologies in Nuclear Components. Nuclear Engineering and Design 131, pp. 81-100.

Pettigrew MJ, Taylor CE, Fisher NJ, Yetisir M and Smith BAW. 1998. Flow-Induced Vibration: Recent Findings and Open Questions. Nuclear Engineering and Design 185, pp. 249-276.

Pettigrew MJ and Taylor CE. 2003. Vibration Analysis of Shell-and-Tube Heat Exchangers: An Overview – Part 2: Vibration Response, Fretting-Wear, Guidelines. Journal of Fluids and Structures 18, pp. 485-500.

Pettigrew MJ, Yetisir M and Fisher NJ. 2007. A Simple Energy Approach to Assess Vibration and Fretting-Wear Damage in Process Equipment. ASME Pressure Vessels and Piping Division Conference, July 22-26, San Antonio, Texas, USA.

Taylor CE, Currie IG, Pettigrew MJ and Kim BS. 1989. Vibration of Tube Bundles in Two-Phase Cross-Flow: Part 3-Turbulence-Induced Excitation. ASME Journal of Pressure Vessel Technology, Vol. 111, November, pp. 488-500.

Table 1
Input Data for Case 1

Variable	Unit	Value	Comments
Tube outside diameter	mm	20	
Tube inside diameter	mm	16.59	
Modulus of Elasticity	Pa	2.00E+11	
Span Length	m	1	
Density of tube material	kg/m^3	8000	
Density of internal fluid	kg/m^3	0	
Density of external fluid	kg/m^3	1000	
Velocity of external fluid	m/s	1	
Void fraction of external fluid	%	0	Single phase
Damping ratio	-	1.5	Applied to all modes
Steam-water coefficient	-	1	
Boundary condition	-	-	Pinned-pinned
Total elements	-	80	

Table 2
Tube Response for Case 1

Analysis	Mode	Damping Ratio (%)	Natural Frequency [Hz]	Max. Turbulence-Induced Vibration Amplitude (rms) [μm]
Hand Calculation	1	1.5	43.11	11.34
PIPO-FE			43.11	10.65

Table 3
Input Data for Case 2

Variable	Unit	Value	Comments
Tube outside diameter	mm	13.00	
Tube inside diameter	mm	10.86	
Modulus of Elasticity	Pa	$2.0 \cdot 10^{11}$	
Span Length	m	0.60	
Density of tube material	kg/m^3	8000.0	
Density of internal fluid	kg/m^3	1.2	
Density of external fluid	kg/m^3	See Table 4	
Velocity of external fluid	m/s	See Table 4	
Void fraction of external fluid	%	See Table 4	
Damping ratio	%	See Table 4	
Lift coefficient	-	0.1	
Fluidelastic instability constant	-	3.0	
Steam-water coefficient	-	1.0	
Pitch-to-diameter ratio	-	1.5	
Total elements	-	20	

Table 4
Flow Conditions for the Four-Simulation Runs of Case 2

Run	Void Fraction (%)	Flow Velocity (m/s)	Water/Air-Water Density (kg/m^3)	Mass Quality (%)
1	25	0.5	750	0.04
2	50	0.5	501	0.12
3	75	0.5	251	0.36
4	90	0.5	101	1.07

Table 5
Tube Response for Case 2

Run	Mode	Code Prediction or Experiment	Natural Frequency (Hz)	Damping Ratio (%)	FEI Ratio	Max. Turbulence-Induced Vibration Amplitude (rms) (μm)
1	1	PIPO	26.9	2.03	0.68	124
		PIPO-FE V1.2	26.9	2.03	0.69	125
		PIPO-FE V1.3	26.9	2.03	0.69	202
		EXPERIMENTAL	27.0	2.0	0.50	48
	2	PIPO	168.8	1.75	0.12	3.07
		PIPO-FE V1.2	168.3	1.75	0.12	3.14
		PIPO-FE V1.3	168.3	1.75	0.12	2.36
2	1	PIPO	27.9	3.19	0.45	145
		PIPO-FE V1.2	27.8	3.19	0.45	146
		PIPO-FE V1.3	27.8	3.19	0.45	169
		EXPERIMENTAL	28.3	3.9	0.36	80
	2	PIPO	174.9	2.94	0.07	3.56
		PIPO-FE V1.2	174.4	2.94	0.07	3.64
		PIPO-FE V1.3	174.4	2.94	0.07	2.82
3	1	PIPO	29.0	2.91	0.33	165
		PIPO-FE V1.2	28.9	2.91	0.33	165
		PIPO-FE V1.3	28.9	2.91	0.33	188
		EXPERIMENTAL	29.5	3.40	0.25	120
	2	PIPO	181.8	2.73	0.05	4.00
		PIPO-FE V1.2	181.3	2.73	0.05	4.07
		PIPO-FE V1.3	181.3	2.73	0.05	3.93
4	1	PIPO	29.7	1.31	0.31	111
		PIPO-FE V1.2	29.7	1.31	0.31	112
		PIPO-FE V1.3	29.7	1.31	0.31	268
		EXPERIMENTAL	30.2	3.00	0.23	152
	2	PIPO	186.3	1.20	0.05	2.65
		PIPO-FE V1.2	185.8	1.20	0.05	2.70
		PIPO-FE V1.3	185.8	1.20	0.05	6.11

Table 6
Input Data for Case 3

Variable	Unit	Value	Comments
Tube outside diameter	mm	19.00	
Tube inside diameter	mm	15.40	
Modulus of Elasticity	Pa	$1.467 \cdot 10^{11}$	
Span Length	m	Shown in Fig. 3	
Density of tube material	kg/m^3	7750.00	
Density of internal fluid	kg/m^3	0.00	
Density of external fluid	kg/m^3	1000	
Velocity of external fluid	m/s	Shown in Fig. 3	
Void fraction of external fluid	%	0.0	Single phase
Viscous damping coefficient	kg/s/m	9.6	Applied to all modes
Lift coefficient	-	0.1	
Fluidelastic instability constant	-	3.3	
Steam-water coefficient	-	1	
Total elements	-	90	

Table 7
Tube Response for Case 3

Analysis	Mode	Damping Ratio (%)	Natural Frequency [Hz]	Max. Vortex Shedding Amplitude [µm]	FEI Ratio	Max. Turbulence-Induced Vibration Amplitude (rms) [µm]
PIPO	1	4.2	17.71	705	0.694	47.8
PIPO-FE V1.2			17.70	718	0.694	48.7
PIPO-FE V1.3			17.70	718	0.694	13.0
PIPO	2	3.7	19.80	691	0.675	48.8
PIPO-FE V1.2			19.79	703	0.676	49.6
PIPO-FE V1.3			19.79	703	0.676	11.6
PIPO	3	3.2	22.87	663	0.653	50.4
PIPO-FE V1.2			22.86	676	0.653	51.4
PIPO-FE V1.3			22.86	676	0.653	10.6

Table 8
Input Data for Case 4

Variable	Unit	Value	Comments
Tube outside diameter	mm	15.95	
Tube inside diameter	mm	13.69	
Modulus of Elasticity	Pa	$1.8006 \cdot 10^{11}$	
Span Length	m	Shown in Fig. 4	
Density of tube material	kg/m^3	7944.13	
Density of internal fluid	kg/m^3	819.10	
Average Density of external fluid	kg/m^3	164.89	
Rotational bending stiffness	Nm/rad	967.01	
Average Void fraction of external fluid	%	91.8	
Damping ratios	%	-	Calculated internally
Lift coefficient	-	0.1	
Fluidelastic instability constant	-	3.0	
Steam-water coefficient	-	0.0644	
Total elements	-	130	

Table 9
Comparison of Results for Case 4

Plane of Vibration	Analysis	Mode	Damping Ratio (%)	Natural Frequency [Hz]	Max. Vortex Shedding Amplitude [µm]	FEI Ratio	Max. Turbulence-Induced Vibration Amplitude (rms) [µm]
In-Plane	PIPO	1	3.0	43.68	9.07	0.086	2.97
	PIPO-FE V1.2			43.64	9.30	0.087	3.08
	PIPO-FE V1.3			43.64	9.30	0.087	10.20
	PIPO	2	3.0	43.68	9.07	0.086	2.97
	PIPO-FE V1.2			43.64	9.17	0.086	3.08
	PIPO-FE V1.3			43.64	9.17	0.086	10.24
	PIPO	3	2.0	71.67	8.12	0.075	1.46
	PIPO-FE V1.2			71.59	7.99	0.075	1.48
	PIPO-FE V1.3			71.59	7.99	0.075	3.61
Out-of-Plane	PIPO	1	3.0	43.61	17.2	0.088	3.00
	PIPO-FE V1.2			43.58	17.2	0.088	3.02
	PIPO-FE V1.3			43.58	17.2	0.088	9.49
	PIPO	2	3.0	43.61	17.0	0.088	3.00
	PIPO-FE V1.2			43.58	17.1	0.088	3.03
	PIPO-FE V1.3			43.58	17.1	0.088	9.49
	PIPO	3	2.0	71.34	23.7	0.082	1.58
	PIPO-FE V1.2			71.29	24.1	0.082	1.61
	PIPO-FE V1.3			71.29	24.1	0.082	3.60

Table 10
Vibration Response and Fretting-Wear Estimation for Case 3

[6] FLUIDELASTIC INSTABILITY ANALYSIS

Fluidelastic instability constant = 3.30

Mode	Frequency[Hz]	Damping Ratio	Critical Velocity[m/s]	FEI Ratio	Plane of Vibration
1	17.70	0.042	2.511	0.694	YZ-plane
2	17.70	0.042	2.511	0.694	YZ-plane
3	19.79	0.037	2.578	0.676	Z-plane
4	19.79	0.037	2.578	0.676	Y-plane
5	22.86	0.032	2.667	0.653	Z-plane
6	22.86	0.032	2.667	0.653	Y-plane
7	26.53	0.028	2.770	0.629	Z-plane
8	26.53	0.028	2.770	0.629	Y-plane
9	30.48	0.024	2.900	0.601	Z-plane
10	30.48	0.024	2.900	0.601	Y-plane
11	34.27	0.021	3.094	0.563	Z-plane
12	34.27	0.021	3.094	0.563	Y-plane
13	37.24	0.020	3.433	0.507	Z-plane
14	37.24	0.020	3.433	0.507	Y-plane
15	43.11	0.017	1.697	1.026	Z-plane
16	43.11	0.017	1.697	1.026	Y-plane
17	69.38	0.011	4.937	0.353	Z-plane
18	69.38	0.011	4.937	0.353	Y-plane
19	73.56	0.010	4.932	0.353	Z-plane
20	73.56	0.010	4.932	0.353	Y-plane

* * * * * * * * * * * * * *WARNING* * * * * * * * * * * * * *

FLUID-ELASTIC INSTABILITY IS PREDICTED FOR MODE 15

FLUID-ELASTIC INSTABILITY IS PREDICTED FOR MODE 16

[7] VORTEX SHEDDING ANALYSIS

Lift coefficient = 0.100

Mode number	Frequency [Hz]	Damping Ratio	Response to Vortex Shedding [µm]	[in/1000]	Resonance Ratio	Plane of Vibration
1	17.70	0.042	718.405	28.284	0.193	YZ-plane
2	17.70	0.042	718.405	28.284	0.193	YZ-plane
3	19.79	0.037	703.512	27.697	0.216	Z-plane
4	19.79	0.037	703.512	27.697	0.216	Y-plane
5	22.86	0.032	676.496	26.634	0.249	Z-plane
6	22.86	0.032	676.496	26.634	0.249	Y-plane
7	26.53	0.028	583.667	22.979	0.289	Z-plane
8	26.53	0.028	583.667	22.979	0.289	Y-plane

9	30.48	0.024	570.341	22.454	0.332	Z-plane
10	30.48	0.024	570.341	22.454	0.332	Y-plane
11	34.27	0.021	492.192	19.378	0.374	Z-plane
12	34.27	0.021	492.192	19.378	0.374	Y-plane
13	37.24	0.020	430.196	16.937	0.406	Z-plane
14	37.24	0.020	430.196	16.937	0.406	Y-plane
15	43.11	0.017	1274.885	50.192	0.470	Z-plane
16	43.11	0.017	1274.885	50.192	0.470	Y-plane
17	69.38	0.011	193.082	7.602	0.757	Z-plane
18	69.38	0.011	193.082	7.602	0.757	Y-plane
19	73.56	0.010	199.462	7.853	0.802	Z-plane
20	73.56	0.010	199.462	7.853	0.802	Y-plane

* * * * * * * * * * * * * * *WARNING* * * * * * * * * * * * * * *

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 1

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 2

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 3

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 4

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 5

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 6

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 7

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 8

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 9

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 10

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 11

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 12

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 13

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 14

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 15

EXCESSIVE VIBRATION DUE TO VORTEX SHEDDING IS PREDICTED FOR MODE 16

[8] RESPONSE TO TURBULENCE EXCITATION

Steam-water coefficient = 1.00000

Mode	Frequency	Damping	Response to Turbulent Excitation	Plane of Vibration

number	[Hz]	Ratio	[μm]	[in/1000]	
1	17.70	0.042	13.022	0.513	YZ-plane
2	17.70	0.042	13.022	0.513	YZ-plane
3	19.79	0.037	11.552	0.455	Z-plane
4	19.79	0.037	11.552	0.455	Y-plane
5	22.86	0.032	10.621	0.418	Z-plane
6	22.86	0.032	10.621	0.418	Y-plane
7	26.53	0.028	9.049	0.356	Z-plane
8	26.53	0.028	9.049	0.356	Y-plane
9	30.48	0.024	8.812	0.347	Z-plane
10	30.48	0.024	8.812	0.347	Y-plane
11	34.27	0.021	7.178	0.283	Z-plane
12	34.27	0.021	7.178	0.283	Y-plane
13	37.24	0.020	5.036	0.198	Z-plane
14	37.24	0.020	5.036	0.198	Y-plane
15	43.11	0.017	37.707	1.485	Z-plane
16	43.11	0.017	37.707	1.485	Y-plane
17	69.38	0.011	0.658	0.026	Z-plane
18	69.38	0.011	0.658	0.026	Y-plane
19	73.56	0.010	0.814	0.032	Z-plane
20	73.56	0.010	0.814	0.032	Y-plane

VIBRATION DUE TO RANDOM TURBULENCE IS NOT A PROBLEM

[9] FRETTING-WEAR ESTIMATION

```
Fretting-wear coefficient =   0.4000E-13 [1/Pa]
SUPPORT_TYPE  =  1      1-Circular hole or scalloped bar
                        2-Lattice bar, flat bar, FUR, or AVB
SUPPORT_THICKNESS = 25.00 [mm]
COMPONENT_LIFE = 30 [year]
```

Mode	Frequency[Hz]	Damping Ratio	Mass-per-unit-length[m/s]	Power [mW]	Plane of Vibration
1	17.70	0.042	1.037	0.026	YZ-plane
2	17.70	0.042	1.037	0.029	YZ-plane
3	19.79	0.037	1.037	0.029	Z-plane
4	19.79	0.037	1.037	0.029	Y-plane
5	22.86	0.032	1.037	0.032	Z-plane
6	22.86	0.032	1.037	0.032	Y-plane
7	26.53	0.028	1.037	0.032	Z-plane
8	26.53	0.028	1.037	0.032	Y-plane
9	30.48	0.024	1.037	0.010	Z-plane
10	30.48	0.024	1.037	0.005	Y-plane
11	34.27	0.021	1.037	0.000	Z-plane
12	34.27	0.021	1.037	0.000	Y-plane
13	37.24	0.020	1.037	0.000	Z-plane
14	37.24	0.020	1.037	0.000	Y-plane
15	43.11	0.017	1.037	0.000	Z-plane
16	43.11	0.017	1.037	0.000	Y-plane
17	69.38	0.011	1.037	0.000	Z-plane
18	69.38	0.011	1.037	0.000	Y-plane
19	73.56	0.010	1.037	0.000	Z-plane

20	73.56	0.010	1.037	0.000	Y-plane

A circular hole or a scalloped bar is used

MAX_POWER and MAXPOWER_MODE = 0.032 and 5

Calculated wear depth = 0.1643E-05[mm]

Percent through-wall wear for the worst mode = 0.9126E-01[%]

Total rate of energy dissipation is estimated to be 0.256 mW

Total rate of tube material loss is estimated to be 0.323 mm³/year

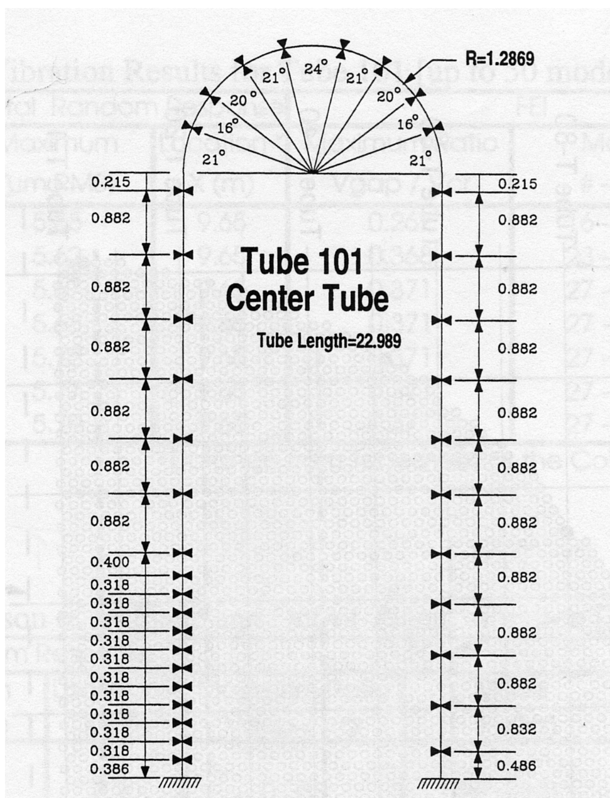

Figure 1 A Typical CANDU Steam Generator Tube Model (Dimensions in Metres)

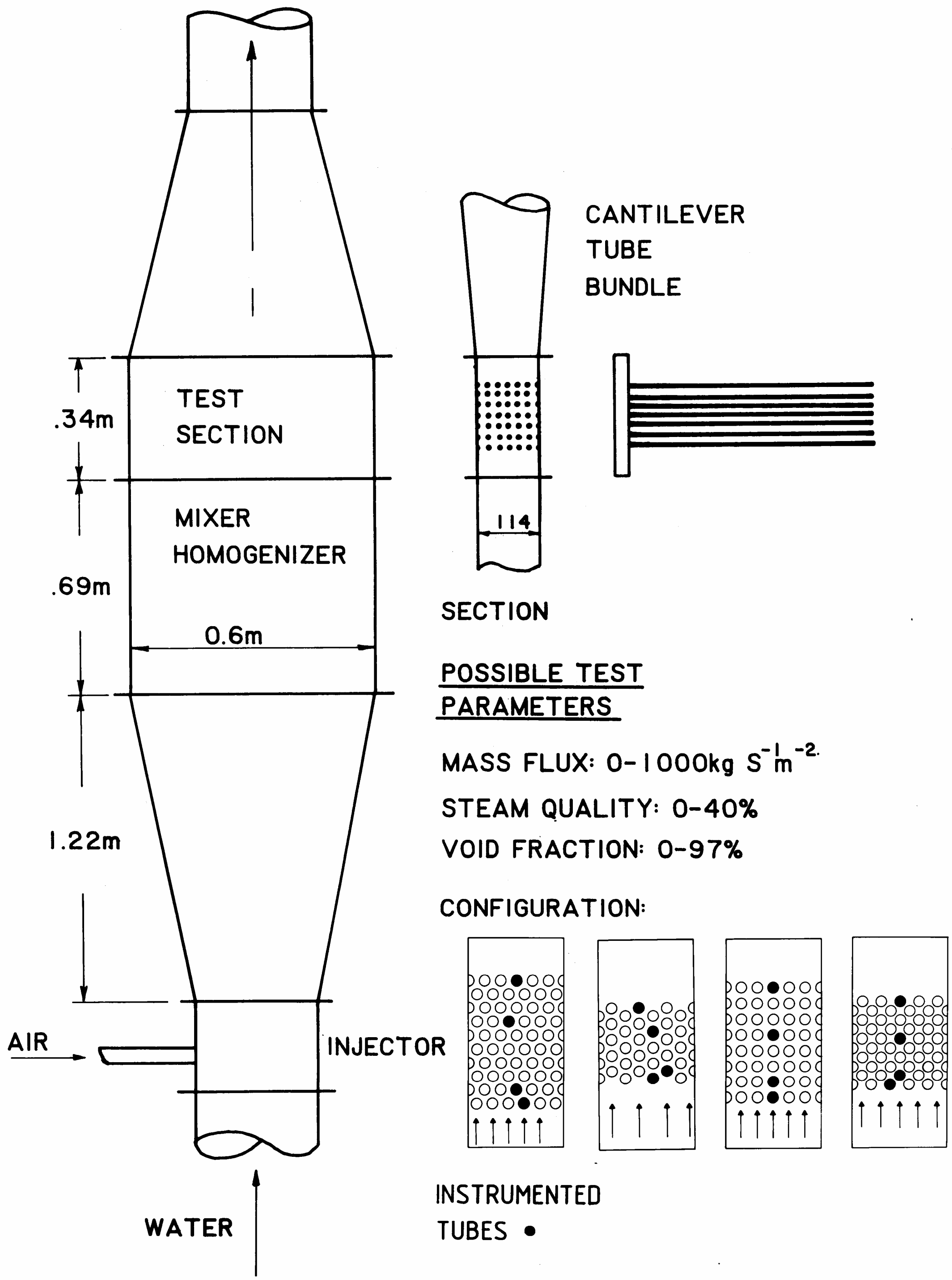

Figure 2 Sketch of Test Section and Bundle Configurations Showing Location of Instrumented Tubes (Case 2)

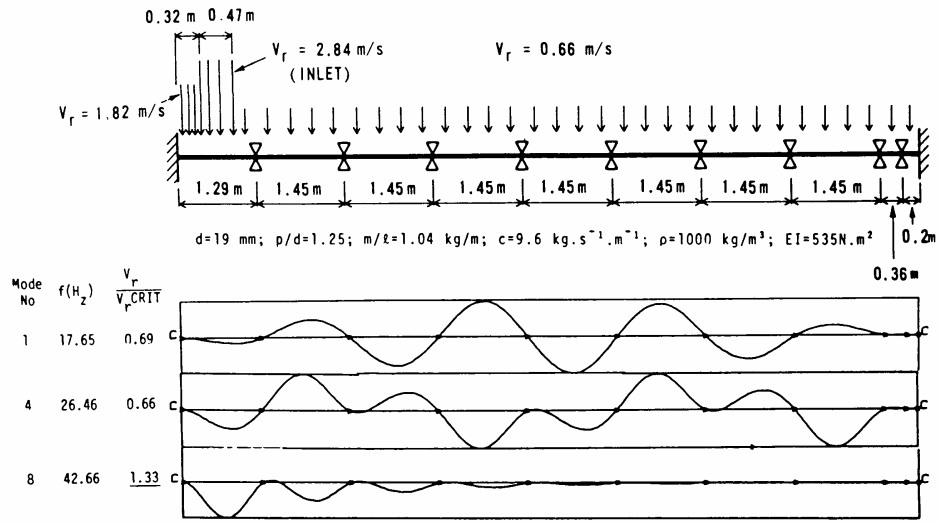

Figure 3 Tube Model, Flow Conditions, Vibration Modes and Fluidelastic-Instability Ratios for Case 3

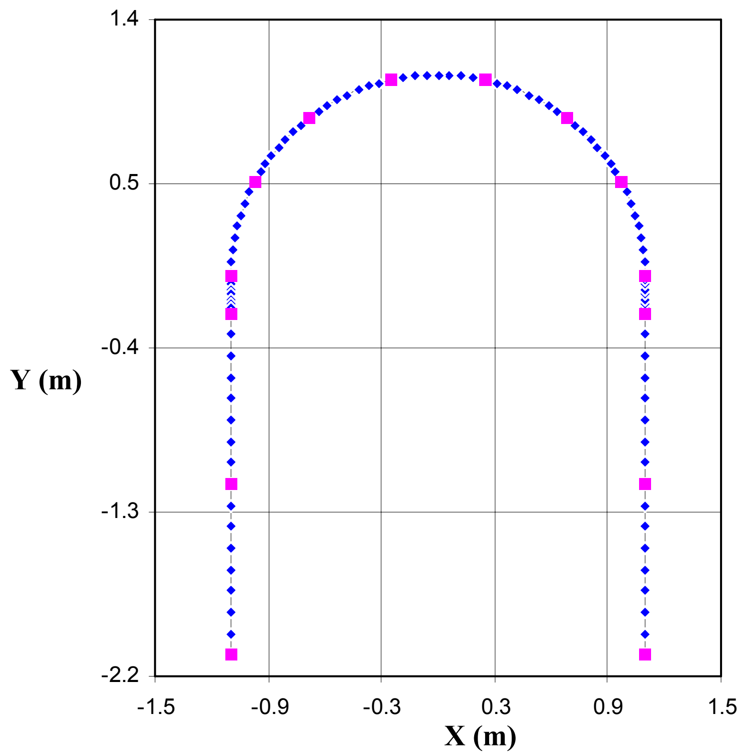

Figure 4 Tube Model for Case 4

■ = Support Location