1-2006

Damping Measurements In Tube Bundles Subjected To Two-phase Cross Flow

Joaquin Moran
Sheridan College, joaquin.moran@sheridancollege.ca

David S. Weaver
McMaster University

Proceedings of PVP2006/ICPVT-11
ASME 2006 Pressure Vessels and Piping/ICPVT Conference
July 23-27, 2006, Vancouver, Canada

PVP2006-ICPVT11-93431

DAMPING MEASUREMENTS IN TUBE BUNDLES SUBJECTED TO TWO-PHASE CROSS FLOW

Joaquin E. Moran
Department of Mechanical Engineering
McMaster University
Hamilton, Canada L8S 4L7
E-mail: moranje@mcmaster.ca

David S. Weaver
Department of Mechanical Engineering
McMaster University
Hamilton, Canada L8S 4L7
E-mail: weaverds@mcmaster.ca

ABSTRACT

An experimental study was conducted to investigate two-phase damping in tube arrays. The objective was to compare different measurement methodologies in order to obtain a more reliable damping estimate. This will allow for improved guidelines related to failures due to fluidelastic instability in tube bundles. The methods compared were the traditionally used half-power bandwidth, the logarithmic decrement and an exponential fitting to the tube decay response. The working fluid used was Refrigerant 11 (Freon), which better models the real steam-water problem, as it allows for phase change. The void fraction was measured using a gamma densitometer, introducing an improvement over the traditional Homogeneous Equilibrium Model (HEM) in terms of velocity and density predictions. The results obtained by using the half-power bandwidth method agree with data previously reported for two-phase flow. The experiments showed that the half-power bandwidth produces higher damping values than the other two, but only up to a certain void fraction. After that point, the results obtained from the three methods are very similar. The exponential fitting proved to be more consistent than the logarithmic decrement, and it is not as sensitive as the half-power bandwidth to the frequency shifting caused by the change in added mass around the tube. By plotting the damping ratio as a function of void fraction, pitch mass flux and flow regime, we were able to verify that damping is more dependent on void fraction and flow regime than on mass flux.

NOMENCLATURE

A_{max}	Maximum amplitude of the frequency spectrum
D	Tube diameter [m]
D_e	Effective tube diameter [m]
D_{ref}	Reference diameter, related to confinement [m]
f	Frequency of vibration [Hz]
f_n	Natural frequency [Hz]
HEM	Homogeneous Equilibrium Model
k	Stiffness [N/m]
m	Lineal mass, including fluid added mass [kg/m]
m_s	Mass per unit length of the structure [kg/m]
m_a	Fluid added mass [kg/m]
m_R	Hydrodynamic mass ratio
$m_{R,theor.}$	Theoretical hydrodynamic mass ratio
RAD	Radiation Attenuation Determination
δ	Logarithmic decrement damping
ρ_l	Liquid density [kg/m^3]
ρ_{tp}	Mixture density [kg/m^3]
ω	Frequency of vibration [Hz]
ω_n	Natural frequency [Hz]
ζ_T	Total damping ratio
ζ_v	Viscous damping component
ζ_s	Structural damping component
ζ_{tp}	Two-Phase damping component
$(\zeta_{tp})_D$	Normalized two-phase damping
ζ_n^f	Normalized fluid damping
ν	Kinematic viscosity [m^2 s]

INTRODUCTION

In the power and process industries, the use of two-phase shell-and-tube heat exchangers is a common practice due to their thermal capabilities. However, the increasing demand in the performance of these devices requires higher flow rates and lower pressure drops. These factors make the heat exchanger prone to vibration problems, leading to fretting wear at the tube supports and possibly tube-to-tube clashing. Although the nature of the excitation mechanisms is relatively well understood in single-phase flows, some difficulties arise in two-phase problems, mainly related to the flow characterization. The mechanisms of excitation in two-phase flow have been described by Pettigrew and Taylor [1]. Fluidelastic instability, which can produce large oscillations and early failure of the tubes, has been the main subject of attention. From a practical point of view, it has been postulated that the occurrence of fluidelastic instability can be predicted using the two non-dimensional parameters used for single-phase flows: the reduced velocity and the mass-damping parameter. However, in two-phase flows, the determination of damping is not straightforward because it is highly dependant on void fraction and flow regime. Therefore, measuring two-phase damping of tube bundles is more difficult than for single-phase flows.

Carlucci [2] and Carlucci and Brown [3] found that the damping in two-phase air-water flow was much higher than for single-phase flows, either liquid or gas. The relationship between damping and void fraction (or gas fraction in the mixture) was studied by Hara [4], who established that the maximum damping ratio corresponded to a certain void fraction (in the 30 to 60 % range). Axisa *et al.* [5] presented experiments using steam-water mixtures. They found that damping is indeed related to void fraction, and that the values obtained for steam-water were considerably lower than those for air-water mixtures, at least for void fractions over 80% (based on the Homogeneous Equilibrium Model, HEM). This is an indicator of the significant differences between experiments carried out with air-water and those using one-component mixtures.

More recently, in an attempt to find a relationship between damping and other flow-related parameters, Pettigrew and Knowles [6] reported results for a single cylinder subjected to two-phase axial flow. They concluded that the effect of surface tension on two-phase damping was strong, and the effect of frequency was not very important. For the analysis of flow-induced vibration problems, Pettigrew *et al.* [7] proposed to express the total damping ratio as follows,

$$\zeta_T = \zeta_s + \zeta_v + \zeta_{tp}, \tag{1}$$

where ζ_s is the structural damping of the system, ζ_v is the viscous component due to the presence of the fluid, ζ_{tp} is the two-phase contribution and ζ_T represents the total damping ratio. The structural component can be measured in air, and the viscous damping is evaluated using equation 2 proposed by Rogers *et al.* [8] for single-phase flows

$$\zeta_v = \frac{\pi}{\sqrt{8}} \left(\frac{\rho D^2}{m}\right) \left(\frac{2\nu}{\pi f D^2}\right)^{0.5} \left\{ \frac{[1+(D/D_e)^3]}{[1-(D/D_e)^2]^2} \right\}. \tag{2}$$

In equation 2, ρ represents the liquid density, D is the tube diameter, m is the mass per unit length (including the fluid added mass), ν is the kinematic viscosity, f is the tube frequency and the bracketed term on the right side represents a confinement function. The two-phase component is calculated by subtracting the sum of ζ_s and ζ_v from the total damping ratio ζ_T, which is measured using the half-power bandwidth method. Equation 2 was originally formulated to estimate the "added damping" provided by a single-phase stagnant fluid. Pettigrew *et al.* [7] however, assumed that an analogous expression could be used in two-phase flow problems. They replaced the density and viscosity by two-phase quantities, with the density defined according to the HEM and the equivalent two-phase viscosity given by an expression proposed by McAdams in 1942 (see Pettigrew and Taylor [9]). In order to compare damping results from different researchers, Pettigrew and Taylor [1] proposed a normalized form of the two-phase component, given by:

$$(\zeta_{tp})_D = \zeta_{tp} \left(\frac{m}{\rho_l D^2}\right) \left\{ \frac{[1-(D/D_e)^2]^2}{[1+(D/D_e)^3]} \right\}, \tag{3}$$

where ρ_l is the density of the liquid phase. Baj and de Langre [10] used a different approach. They considered the total damping ratio as comprised for only two components: structural and fluid. In order to improve the stagnant-flow defined damping, a new normalization procedure was introduced, in which the liquid density was displaced by a two-phase density. The resultant normalized fluid damping ratio is:

$$\zeta_n^f = \zeta^f \left(\frac{m}{\rho_{tp} D^2}\right) \left(\gamma \frac{D}{D_{ref}}\right). \tag{4}$$

In equation 4, ζ_n^f is the normalized fluid damping, ρ_{tp} is the density of the mixture and the terms D_{ref}/D and γ take into account the confinement effects. This equation assumes that the fluid damping is linearly dependent on the tube diameter, as in single-phase flow. By extrapolating the values of the normalized fluid damping for low flow velocities, Baj and de Langre [10] introduced the concept of "quiescent fluid damping" in two-phase mixtures.

Nakamura *et al.* [11] carried out experiments to measure two-phase damping and added mass in two-phase flows. They used steam-water mixtures at high pressure (up to 5.8 MPa), and implemented an electromagnetic device to excite a metal block mounted on the tube support wires. They found that the damping ratio changes depending on the location of the tube within the array, and the inclination of the bundle respect to the direction of the flow is also a factor to be considered. They also found that damping was larger in the lift direction than in the drag direction. In order to analyze and compare their results, they used the two-phase damping formulation proposed by Axisa *et al.* [5].

In addition to the difficulties that arise from different normalization procedures, the way that damping is estimated from experimental data is also an important issue. Janzen *et al.* [12] proved that there are differences in the damping values obtained by different measurement methods. They compared the logarithmic decrement, decay trace fitting and half-power bandwidth methods. It was observed that the half-power method produced higher damping values than the other two. Although the predictions using the decay trace fitting and the logarithmic decrement were similar, the former proved to be more consistent. Chandler [13] presented results of a series of experiments aimed to compare different damping measurement methodologies. A single tube was impacted with a calibrated force transducer hammer, and the responses were recorded and analyzed. The trend obtained by Janzen [12] was also observed in these experiments.

Moran *et al.* [14] proposed an alternative damping measurement technique that can be used in experiments with two-phase flow across a tube array, which is confined in a test section. The device can produce a steady and controlled oscillation of the tube in a non-intrusive fashion. A pair of electromagnetic coils was used to achieve this purpose. Once the electromagnets are shut off, the decay trace of the tube can be captured using strain gauges. This allowed a comparison between the traditional half-power bandwidth method against the logarithmic decrement and the curve fitting approaches for tube bundle vibrations in two-phase flow.

Previous studies have typically used the Homogeneous Equilibrium Model (HEM) to analyze two-phase flow data. HEM assumes that the liquid and gas phases are uniformly distributed and move with the same velocity (no slip). In general, this is not a valid assumption and it certainly does not provide for void segregation and different flow regimes, including intermittent flows. It also distorts the values assumed for effective flow velocity and two-phase density. This paper presents the results of an experimental program to measure the damping of a tube array in two-phase flow using the three different measurement techniques. These results are compared and analyzed using void fraction measurements determined by a gamma densitometer. Such an approach provides a more reliable measure of void fraction.

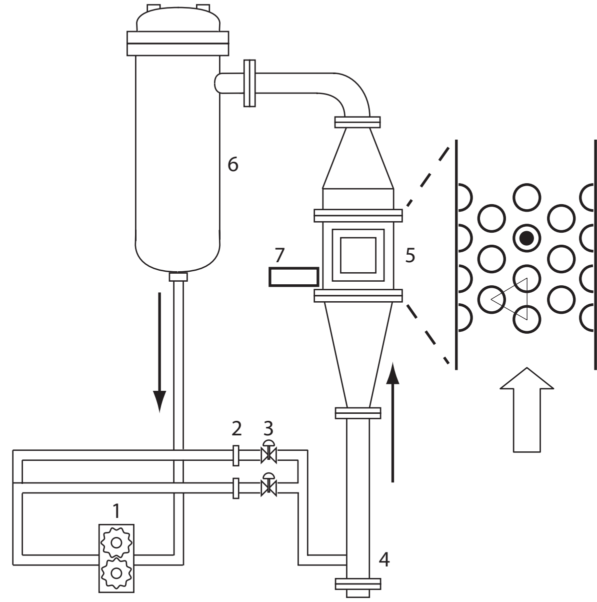

Figure 1. Diagram of the two-phase flow loop and tube array geometry: (1) Main pump, (2) orifice plates for mass flux measurement, (3) flow control valves, (4) heaters, (5) test section, (6) shell-and-tube condenser, (7) gamma densitometer.

EXPERIMENTAL SET-UP

Two-Phase Flow Loop

A diagram of the two-phase flow loop is shown in Figure 1. The working fluid is Refrigerant 11 (Trichlorofluoromethane, CCl_3F), and is circulated through the loop by means of a gear pump. The heaters are located below the test section, and they are capable of transmitting up to 46 kW of power to the fluid. The condenser located downstream of the model tube bundle allows for a better control of the thermodynamic parameters inside the test section, and can remove 48 kW from the Freon while operating at a pressure around 173 kPa (25 psi). The maximum flow rate attained by the main pump is 2.2 L/s, equivalent to a single-phase pitch mass flux of 1000 kg/m^2 s. A more detailed description of the flow loop can be found in Feenstra *et al.* [15, 16].

Test Section and Model Tube Array

The section of the flow loop where the model bundle is located, has a rectangular cross-section of 49.2 mm by 197 mm. It has two glass windows on the sides for observation, as well as a frontal window. The test section has half-tubes attached to the sides, in order to minimize the effect of the flat walls on the flow configuration.

The array consists of ten cantilever-mounted brass tubes, with an external diameter of 9.525 mm (0.375 in). The geometric pattern of the bundle is a parallel triangle, with a pitch-over-diameter ratio of 1.49, similar to that in CANDU steam genera-

Table 1. Natural Frequencies and Damping Ratio for the Monitored Tube

Medium	Frequency (Hz)	Damping Ratio (%)
Gaseous R11	50.0	0.1
Liquid R11	41.8	0.5

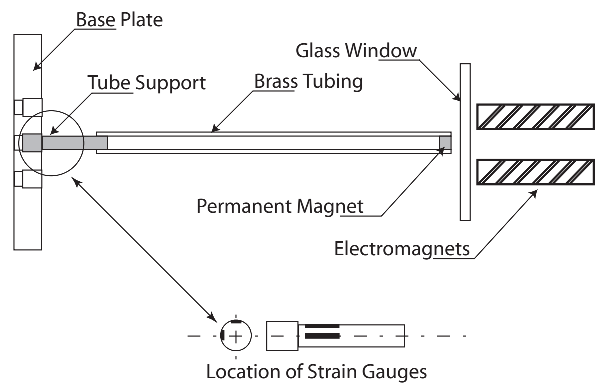

Figure 2. Mounted tube, strain gauges and electromagnetic device.

tors. The tubes were tuned to within ± 1 % of the average natural frequency measured in air. For the monitored tube, the natural frequency and damping ratio in both liquid and gas freon are shown in Table 1.

The vibratory response of the tube was measured by two strain gauges, located on the cylindrical support between the tubes and the base plate of the array (see Figure 2). These strain gauges were positioned at 90 degrees from each other, allowing for the measurement of displacement in both the transverse and stream-wise directions. Although the response in both directions is recorded, the tube is excited to vibrate only in the lift direction. Hence, the damping reported in this study corresponds only to the transverse plane. The output signal was collected using a data acquisition card and a dynamic analyzer (HP 35670A). The data acquisition card provided the time history information of the monitored tube, whereas the dynamic analyzer provided the averaged frequency response, based on the flow excitation and used as input for the half-power bandwidth damping measurements.

Electromagnetic Excitation Device

Figure 2 shows a diagram of the electromagnetic coils in position. The end-cap of the monitored tube has a permanent magnet attached, which improves the response of the tube to the electromagnetic field imposed. The polarity of these temporal magnets is changed at a frequency equal to the resonant frequency of the tube, in such a way that the force required for a given tip displacement is much less than for the case of static deflection. Since the excitation comes from outside the test section, the method is non-intrusive, that is, it does not interfere with the two-phase flow distribution across the bundle or around the monitored tube.

Experimental Procedure

For each experiment, the mass flux was held constant while the void fraction was increased by means of the heaters. The temperature of the Freon was measured at several points along the loop, including locations upstream and downstream of the test section, at the heaters and downstream of the condenser. When the temperatures had remained constant for a certain period of time, then the void fraction and tube response were recorded. This procedure of waiting for a "steady state" ensures that the flow regime and void fraction will not change while the measurements are performed. First, the averaged frequency spectra of the tube was captured, after which the void fraction was measured directly using the gamma densitometer. The latter was located upstream of the model tube bundle, as shown in Figure 1. After these two parameters had been recorded, the electromagnets were turned on. They remained working for a period of 5 to 10 seconds, to ensure that the tube was vibrating at a constant amplitude. Then, the power was shut-off, and the tube decay response was captured by the strain gauges. This procedure was repeated three times for each value of heat flux, and then averaged to obtain a representative damping ratio.

DAMPING MEASUREMENT METHODS

As mentioned above, the logarithmic decrement and the exponential fitting were both based on the decay trace of the monitored tube. On the other hand, the averaged frequency spectra was used as input for the half-power bandwidth method. The number of averages taken was 100, with a resolution of 0.25 Hz from 0 to 100 Hz.

Half-Power Bandwidth Method

This method assumes that the turbulent excitation due to the flow is broadband and random in the frequency range close to the natural frequency of the tube. Then, it can be considered that the frequency response of the tube is equivalent to the vibration response obtained from a frequency sweep. Once the averaged frequency spectrum of the tube is captured, a FORTRAN program was used to fit a curve to the frequency response by using a least-square regression technique. The program fits a frequency response of a single-degree of freedom system to the frequency data gathered during the experiment. Initially, estimates of the damping ratio and the natural frequency are provided. The program then iterates to find the combination that will produce the best fitting to the experimental data. The value of the damping

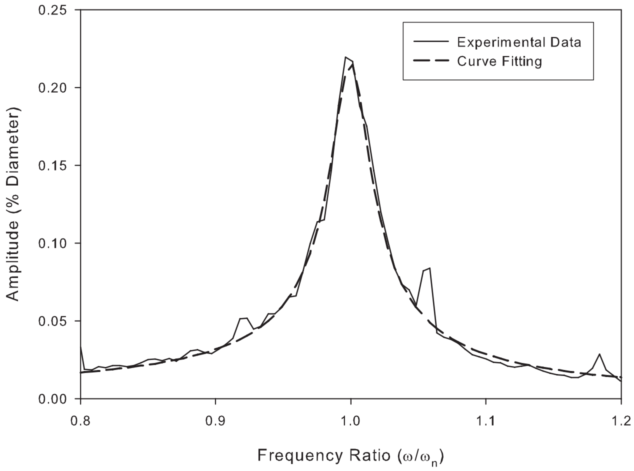

Figure 3. Evaluation of damping using the frequency response.

ratio at which the program converges is equivalent to that obtained by the half-power method. In this case, the damping ratio is a function of the width of the peak at an amplitude equal to $A_{max}/\sqrt{2}$, were A_{max} is the peak amplitude (see Figure 3). For small damping ratios, we may write

$$\zeta = \frac{\Delta f}{2 f_n}. \tag{5}$$

The main disadvantage of this method is that, in two-phase flow, the continuous change in the average fluid density around the monitored tube produces a continually shifting frequency, artificially broadening the resulting peak. This can potentially lead to an over-estimation of the damping ratio.

Logarithmic Decrement

For this procedure, it is assumed that the damping present in the system is purely viscous and linear. In this study, although the tube is forced to vibrate at about 10% of its diameter by the electromagnets, only the portion of the decay trace below 3% is used. By doing this, we ensure that nonlinear damping behaviour will be minimized in our results and analysis. Once the time response of the tube was captured, it was divided in various sections, and the logarithmic decrement was computed for each one of them. The resulting values of damping ratio were averaged along the time interval considered. For damping ratios below 4%, we may write

$$\zeta \approx \frac{\delta}{2\pi}, \text{ where } \delta = \frac{1}{n}\ln\left(\frac{x_1}{x_n}\right). \tag{6}$$

In equation 6, n is the number of cycles between the peaks considered and x_1, x_n represent the peak amplitudes.

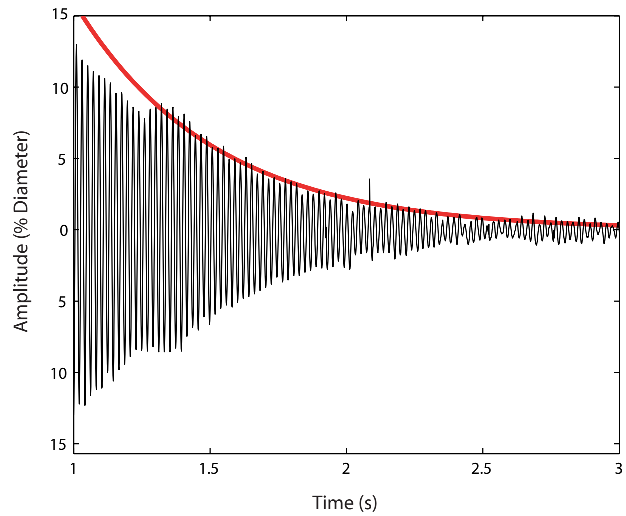

Figure 4. Exponential Fitting method

Exponential Curve Fitting

Also based on the decay response of the tube, this procedure simply fits an exponential function to the decay trace. The fitting is less sensitive than the logarithmic decrement to the unsteadiness in the signal caused by the two-phase mixture (see Figure 4). After the response is recorded, a program in MATLAB performs a least squares fitting using the peaks maxima. The function used was

$$y = Ae^{-Bt}, \text{ where } B = \zeta\omega_n. \tag{7}$$

A true complex exponential was not fitted to the frequency spectra obtained in the experiments because we wanted to avoid the noise produced by the shifting frequency. This is why we proposed this approach in the first place, to try to get a better estimate of damping by reducing the noise in the data.

Two-Phase Flow Modelling

In two-phase flows, experience has shown that damping is strongly dependent on void fraction, and possibly on flow regime. As a result, the reliable measurement of void fraction is a key element in the analysis of two-phase damping. Traditionally, the Homogeneous Equilibrium Model (HEM) has been used for the estimation of void. This model assumes that there is no slip between the liquid and gas phases, that is, they flow together at the same velocity. If this approach is taken, determining the flow density and velocity is a very simple task. However, for the case of vertical-upwards gas-liquid flows, the buoyancy contributes to accelerate the gas phase, causing a velocity difference that should not be neglected. Moreover, as the normalized

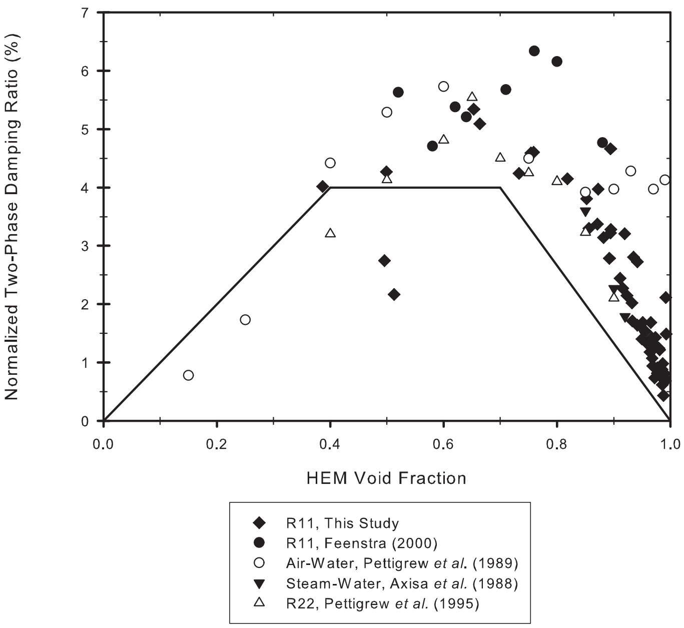

Figure 5. Normalized two-phase damping ratio vs. HEM Void fraction. The continuous line shows the design guideline suggested by Pettigrew *et al.* [9].

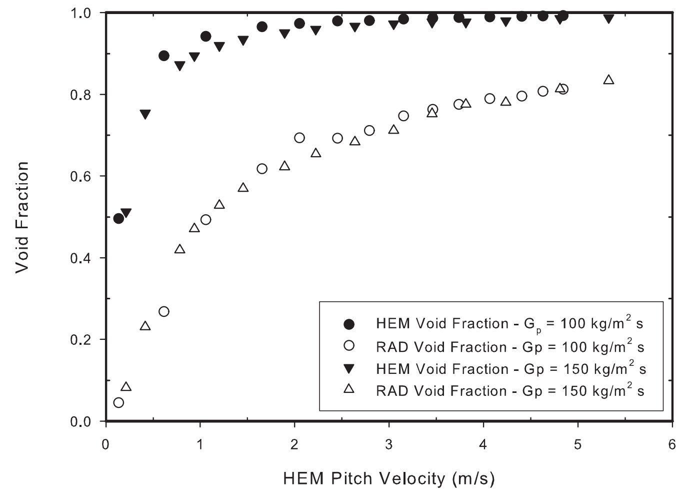

Figure 6. Comparison between the void fraction predicted by the HEM and the gamma densitometer measurements, as a function of HEM pitch velocity.

component of damping (which is used as a design guideline) depends upon the two-phase density, an important deviation from the real void fraction affects the results and analysis involved.

As an alternative, Feenstra *et al.* [15] used gamma densitometry in order to improve the two-phase density calculation. This is especially important when the flow regime changes from bubbly to intermittent, because the unsteadiness and turbulence present in the flow departs from the physical behaviour assumed in the HEM. In this study, the void fraction (RAD Void - Radiation Attenuation Determination) was measured using a gamma densitometer, but also the HEM void was calculated for comparison purposes.

EXPERIMENTAL RESULTS

As the first step, the half-power bandwidth data obtained from the experiments was compared to the results previously presented by Axisa *et al.* [5] and Pettigrew and Taylor [1]. This was carried out by normalizing the two-phase component of damping using the equation 3. Figure 5 shows the relationship between the normalized two-phase damping ratio and the flow pitch velocity based on the homogeneous equilibrium model. The continuous line represents the design guideline suggested by Pettigrew *et al.* [9]. It can be seen that the results obtained in this study agree with those presented by previous researchers.

There are two points at an HEM void fraction of about 0.5 in Figure 5, that do not follow the general trend. The reason for this is related to the methodology employed for the void fraction estimation. These points correspond to the experiments where the pitch mass flux was lower, between 100 and 150 kg/m^2 s. Figure 6 shows the HEM and RAD void fractions as a function of the HEM pitch velocity. It can be observed that at the lowest velocities, the HEM predictions are about 10 times larger than the gamma densitometer measurements. As the flow velocity increases, the two methods get closer to each other, with the HEM about 20% above the RAD void fraction at the highest measured HEM flow velocity. The HEM substantially over-predicts the void fraction over the entire pitch velocity range.

Frequency Shifting

It was mentioned before that the change in added mass around the monitored tube produces a shifting frequency. This phenomenon may artificially broaden the peak observed in the frequency spectrum, leading to unreliable damping results. This effect can be quantified by using equation 8

$$\omega_n = \sqrt{\frac{k}{m_s + m_a}}, \tag{8}$$

where ω_n is the natural frequency of the tube, k is the stiffness, m_s represents the mass per unit length of the tube and m_a is the hydrodynamic mass or added mass. The sum of m_s and m_a is the total mass that has inertial effect on the dynamics of the tube. If the tube is surrounded by vapour or air, the effect of m_a is minimum, and the total mass approaches that of the tube alone. When the tube is surrounded by liquid, the total mass is larger, contributing to a decrease in natural frequency. For the

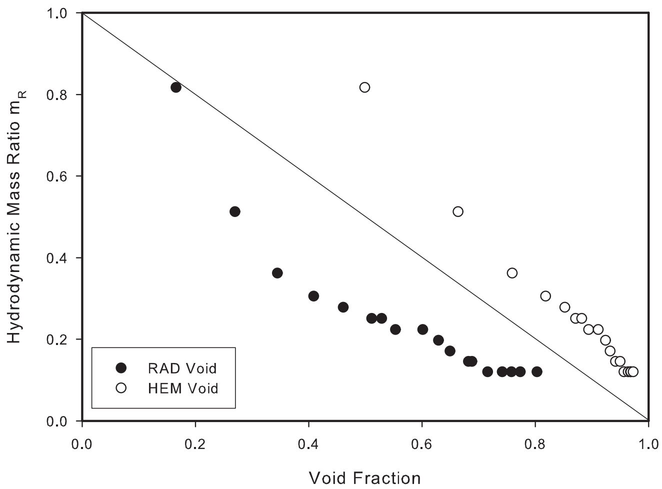

Figure 7. Hydrodynamic mass ratio vs. void fraction for a mass flux of 250 kg/m^2 s. The continuous line represents the theoretical hydrodynamic mass ratio.

particular case of this model tube bundle, the difference between the frequencies in vapour and liquid Freon is about 15% as seen in Table 1.

In two-phase flow problems, the unsteadiness of the flow will contribute to changing the amount of liquid and gas that is in contact with the tube at a given time. The frequency of the tube will then fluctuate around a mean value, which is determined through an averaging process during a certain period of time. The distribution of the phases around the tube is related to the flow regime and void fraction. For different values of void fraction, the effect of frequency shifting changes. Figure 7 shows the relationship between the void fraction and the hydrodynamic mass ratio (m_R) for both the HEM and RAD void estimations. The quantity m_R is equal to the ratio between the actual added mass and the added mass of a tube surrounded by liquid. The continuous line represents the theoretical hydrodynamic mass ratio, which is the lineal mass of fluid displaced by the tube and corrected for confinement of the nearest tubes. It can be calculated as:

$$m_{R,theor.} = \frac{[(D_e/D)^2+1]}{[(D_e/D)^2-1]}\left(\frac{\pi\rho D^2}{4}\right), \qquad (9)$$

In Figure 7, it can be seen that the HEM over-predicts the void fraction, producing larger values of hydrodynamic mass ratio than the theoretical prediction. On the other hand, the RAD results show that the real values of added mass are lower than expected, and tend to level off at void fractions larger than about 75%. The authors believe that at this point, a vapour film is formed around the tube and the effective fluid added mass becomes insensitive to further increases in void fraction. The

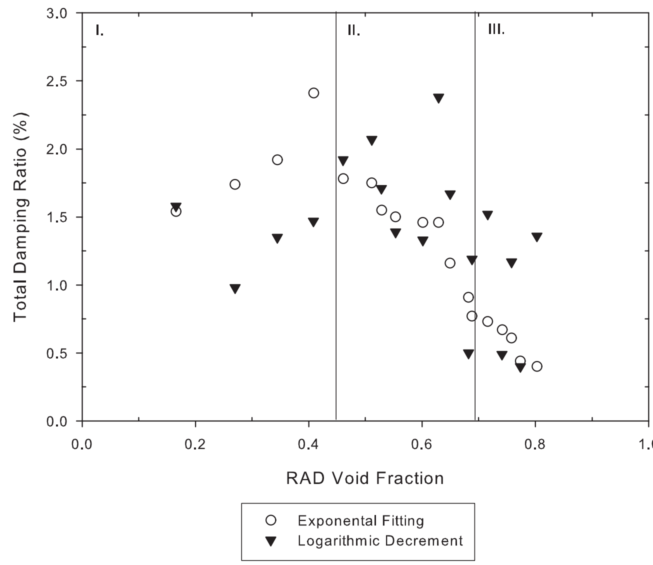

Figure 8. Comparison between the exponential fitting and the logarithmic decrement methods, for a mass flux of 250 kg/m^2 s. The numbers I., II. and III. indicate the regions where bubbly, intermittent and annular flow (respectively) were observed.

present experimental rig does not permit RAD void fractions above 85% (HEM void fraction about 99%).

In terms of the frequency shifting, the presence of this phenomenon will be much weaker when the added mass remains constant. For lower void fractions, where the hydrodynamic mass ratio is strongly dependent on void fraction, the shifting is much more pronounced.

Damping Measurements

Figure 8 shows a comparison between the exponential fitting and the logarithmic decrement methods, for a pitch mass flux equal to 250 kg/m^2 s. It can be observed that the logarithmic decrement method values have a significant scatter. On the other hand, the exponential fitting method is more consistent, not only over the flow velocity range, but also between the three measurements performed for each value of flow velocity. The flow regimes were determined based on the flow regime map proposed by Ulbrich and Mewes [17] and corroborated by visual observation.

It is clear that the logarithmic decrement approach is very sensitive to the vibration amplitude peak-values used in the calculations, even when averaging is used. The shifting frequencies caused by variation in fluid added mass and irregular response due to turbulent two-phase flow excitation, make this method rather unreliable for this application. On the other hand, the smoothening of these effects by fitting the decay curve provides more consistent results. For this reason, only the exponential

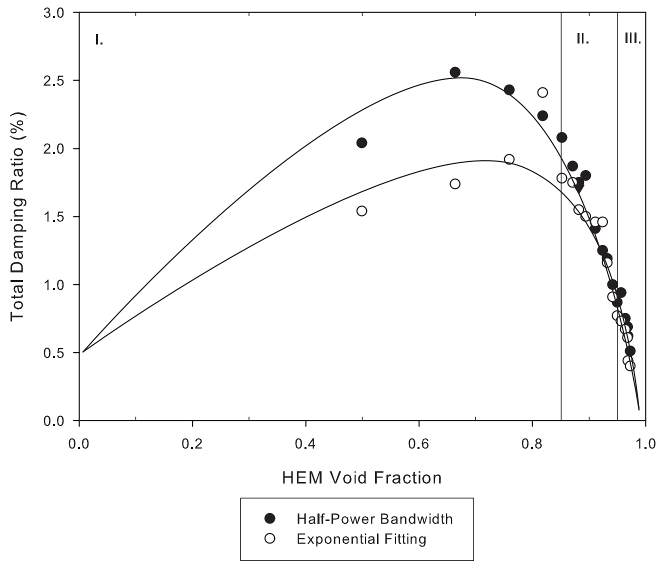

Figure 9. Comparison between total damping ratio obtained by the half-power bandwidth method and the exponential fitting for a mass flux of 250 kg/m^2 s. The numbers I., II. and III. indicate the regions where bubbly, intermittent and annular flow (respectively) were observed.

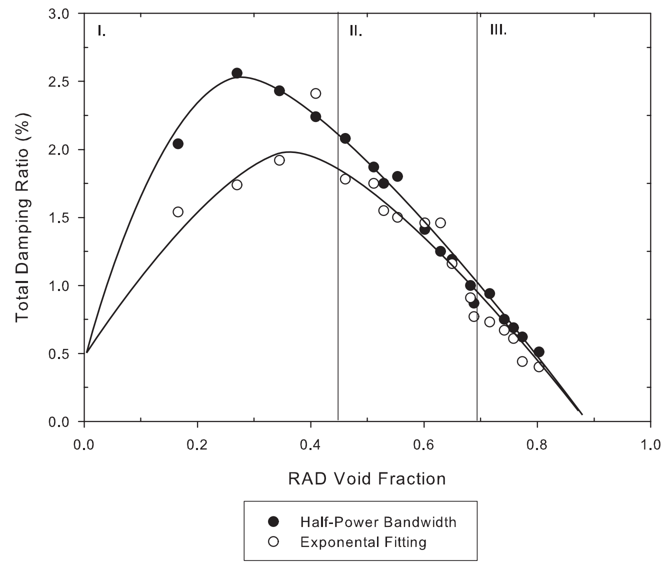

Figure 10. Total damping ratio obtained by the half-power bandwidth method and exponential fitting versus RAD void fraction, for a mass flux of 250 kg/m^2 s. The numbers I., II. and III. indicate the regions of bubbly, intermittent and annular flow, respectively.

curve fitted damping results will be used for comparison with the half-power bandwidth method in the discussion below.

Damping measurements obtained using the half-power and exponential decay methods are plotted against HEM void fraction in Figure 9. It can be observed that the results based on the half-power bandwidth method are generally higher than for the exponential decay method. This agrees with the trend reported by Janzen et al. [12] and Chandler [13]. It is important to note that, even though those researchers did not carry out experiments in two-phase flow, they showed that there can differences in the damping values obtained depending on the measurement method. The trends observed in Figure 9 were seen in all of the present experiments.

If we plot the results shown in Figure 9 as a function of the RAD void fraction, the trends identified above are maintained. However, as we can see in Figure 10, the flow regime zones are more spread out, indicating a more progressive transition between flow patterns.

The differences in damping values obtained between the two damping measurement methods is the greatest in the bubbly flow regime and reduces as the flow moves into the intermittent and annular flow regimes. These transitions between flow regimes at a given mass flux are accompanied by increasing void fraction. The effects of varying void distribution on fluid added mass, and therefore on tube natural frequency, reduce as the void fraction increases. Thus, it seems that the half-power bandwidth method is more sensitive to the effects of frequency shifting at lower void fractions than the exponential decay curve fitting approach. This explains why the former method gives much larger values of damping at lower void fraction than the latter method, and why the difference in results between the two methods reduces with increasing void fraction. Interestingly, the effects of intermittent flow on damping measurements appear to be less important than those of variations in fluid added mass.

Damping and Flow Regime

In order to improve our understanding of two-phase flow damping, we have plotted isocontours of the total damping ratios obtained by the exponential fitting method on a grid of RAD void fraction and pitch mass flux. Figure 11 shows these contours, the numbers indicating the total damping ratio.

It is interesting to note that the damping is not very sensitive to mass flux at low and high values of RAD void fraction, but has a substantial peak at intermediate RAD void fraction. The height of this peak is strongly dependent on pitch mass flux and occurs about in the middle of the mass flux range used in the present study. This peak is in the bubbly flow regime, and is therefore not the result of intermittency in the flow. This is not an expected outcome and needs further study.

CONCLUSIONS

An experimental study in two-phase flow-induced vibrations was conducted to compare three different damping measurement methods. The model tube bundle was subjected to two-phase flow of R-11, and the void fraction of the mixture was measured

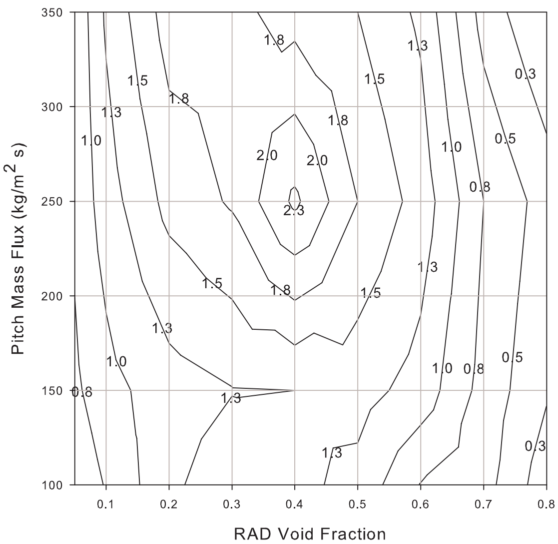

Figure 11. Isocontours of total damping ratio on a grid of RAD void fraction and pitch mass flux.

by means of a gamma densitometer. An electromagnetic device was used to essentially pluck the tube from the outside of the test section, allowing for the capture of the amplitude decay response and the implementation of the logarithmic decrement and exponential fitting methods. On the other hand, the half-power bandwidth method was based on the tube turbulence frequency response spectra. The results obtained by implementing the latter agree with those reported in the past. A summary of findings of this research are as follows:

1. The present data for two-phase flow damping was normalized using the Pettigrew et al. [7] relationship. The results agree with those presented by previous research. The use of HEM does substantially over-estimate the void fraction, especially at lower flow velocities.
2. After comparing the three damping measurement methods, it can be concluded that the half-power bandwidth produces higher values of damping than the logarithmic decrement and the exponential fitting methods. The latter seems to generate more consistent values, and should be used as a reference for other damping studies. The logarithmic decrement method used in this study produces very scattered values.
3. The trends on the damping values obtained by the three methods were the same for all the mass fluxes studied. The difference between the results of the half-power bandwidth method and the exponential decay curve fitting method are largest at lower void fractions, and reduce at high void fractions. This may be due to the frequency shifting effects of fluid added mass, which increases with increasing void.
4. Two-phase flow damping is highest at intermediate RAD void fractions, and is influenced significatively by pitch mass flux in this region. At low and high RAD void fractions, the damping is not very sensitive to pitch mass flux.

REFERENCES

[1] Pettigrew, M. J., and Taylor, C. E., 1994. "Two-phase flow-induced vibration: An overview". *Journal of Pressure Vessel Technology,* **116**(3), pp. 233–253.

[2] Carlucci, L. N., 1980. "Damping and hydrodynamic mass of a cylinder in simulated two-phase flow". *Journal of Mechanical Design,* **102**, pp. 597–602.

[3] Carlucci, L. N., and Brown, J. D., 1983. "Experimental studies of damping and hydrodynamic mass of a cylinder in confined two-phase flow". *Journal of Vibration, Acoustics, Stress and Reliability in Design,* **105**, pp. 83–89.

[4] Hara, F., 1987. "Vibration of circular cylindrical structures subjected to two-phase flows". *JSME International Journal,* **30**(263), pp. 711–722.

[5] Axisa, F., Wullschleger, M., and Villard, B., 1988. "Two-phase cross-flow damping in tube arrays". In ASME PVP Conference, Vol. 133.

[6] Pettigrew, M. J., and Knowles, G. D., 1997. "Some aspects of heat exchanger tube damping in two-phase mixtures". *Journal of Fluids and Structures,* **11**, pp. 929–945.

[7] Pettigrew, M. J., Taylor, C. E., and Kim, B. S., 1989. "Vibration of tube bundles subjected to two-phase cross-flow: Part I - hydrodynamic mass and damping". *Journal of Pressure Vessel Technology,* **111**, pp. 466–477.

[8] Rogers, R. G., Taylor, C. E., and Pettigrew, M. J., 1984. "Fluid effects on multispan heat exchanger tube vibration". In ASME PVP Conference.

[9] Pettigrew, M. J., and Taylor, C. E., 2004. "Damping of heat exchanger tubes in two-phase flow: Review and design guidelines". *Journal of Pressure Vessel Technology,* **126**, pp. 523–533.

[10] Baj, F., and de Langre, E., 2003. "Scaling of damping induced by bubbly flow across tubes". *Journal of Fluids and Structures,* **17**, pp. 351–364.

[11] Nakamura, T., Hirota, K., Wanatabe, Y., Mureithi, N. W., Kusanabe, T., and Takamatsu, H., 2002. "Dynamics of an in-line tube array subjected to Steam-Water Cross-Flow. Part I: Two-phase damping and added mass". *Journal of Fluids and Structures,* **16**(2), pp. 123–136.

[12] Janzen, V. P., Han, Y., Smith, B. A., and Fluit, S. M., 2005. "Vibration damping of stabilized steam generator tubes". In ASME PVP Conference, Paper No. 71666.

[13] Chandler, C., 2005. "Methods of determining a damping ratio from a single impact test". In ASME PVP Conference, Paper No. 71041.

[14] Moran, J. E., Weaver, D. S., and Feenstra, P. A., 2005. "A new method for measuring damping in two-phase flows".

In 20th Canadian Congress of Applied Mechanics (CANCAM).

[15] Feenstra, P. A., Weaver, D. S., and Judd, R. L., 2002. "Modelling two-phase excited damping and fluidelastic instability in tube arrays". *Journal of Fluids and Structures,* **16**(6), pp. 811–840.

[16] Feenstra, P. A., Weaver, D. S., and Nakamura, T., 2003. "Vortex shedding and fluidelastic instability in a normal square tube array by two-phase cross-flow". *Journal of Fluids and Structures,* **17**, pp. 793–811.

[17] Ulbrich, R., and Mewes, D., 1994. "Vertical, upward gas-liquid two-phase flow across a tube bundle". *International Journal of Multiphase Flow,* **20**(2), pp. 249–272.