Importance of Added Mass and Damping in Flow-Induced Vibration Analysis of Tubes Bundle: An Overview

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ABSTRACT

Flow-induced vibration is of prime concern to the designers of heat exchangers subjected to high flows of gases or liquids. Excessive vibration may cause tube failure due to fatigue or fretting-wear. Tube failure results in, expensive plant upholding and suffers loss of production. Therefore, tube failure due to unwarranted vibration must be avoided in process heat exchangers and nuclear steam generators, preferably at design stage. Such vibration problems may be avoided through a comprehensive flow-induced vibration analysis before fabrication of heat exchangers. However, it requires an understanding of vibration mechanism and parameters related to flow-induced vibration. For an accurate vibration analysis, it is of prime importance to have good estimates of structural and flow related dynamic parameters. Thus dynamic parameters such as added mass and damping are of significant concern in a flow regime. The purpose of this paper is to provide an overview of our state of knowledge and role of dynamic parameters in flow-induced vibration on tube bundles due to current trend of larger heat exchangers. The present paper provides published data, analysis, evaluation, formulation, and experimental studies related to hydrodynamic mass and damping by a large number of researchers. Guidelines for experimental research and heat exchangers design related to added mass and damping mechanisms subjected to both single and two-phase flow are outlined in this paper.

Key Words: Tubes Bundle, Flow Induced Vibration, Damping Mechanism, Hydrodynamic Mass.

1. INTRODUCTION

Excessive vibration may lead to tube failure due to fatigue or fretting-wear. Tube failure results in, expensive plant upholding and suffers loss of production. Therefore, tube failure due to unwarranted vibration must be avoided in process heat exchangers and nuclear steam generators, preferably at design stage. Such vibration problems may be avoided through a comprehensive flow-induced vibration analysis before fabrication of shell-and-tube heat exchangers. However, it requires an understanding of vibration mechanism and parameters related to flow-induced vibration. For an accurate vibration analysis, it is of prime importance to have good estimates of structural and flow related dynamic parameters. Thus dynamic parameters such as added mass and damping are of significant concern in a flow regime.

Most of the work on vibration related problems was done in the late 60's and 70's. Unfortunately, very little attention was given to dynamic parameters except for Carlucci [1] in 1980 and Carlucci and Brown [2]. They conducted a

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symmetric study of dynamic parameters in two-phase flow simulated by air-water mixtures. Earlier a small number of researchers reported some damping measurements while investigating turbulence-induced vibration. Such as Gorman [3] reported much larger damping in two-phase axial flow than in liquid (single) phase axial flow. But this work is not exactly applicable to cross-flow. Vibration analysis experiments on small tubes bundle partially exposed to two-phase flow (air-water) cross-flow were conducted by Pettigrew and Gorman in 1973 [4], but they did not measure damping. Only Chen et al [5] carried out an analytical and experimental study of a cylindrical rod vibrating in viscous fluid, to obtain coefficient of added mass and damping. Helker and Vincent [6] also conducted some work in air-water cross-flow for a limited range of bundle geometries and flow conditions related to their steam generator design. The single span tube bundles used were exposed to flow over their entire length. They provided results on fluid-elastic instability and turbulence-induced excitation, unfortunately measured damping at critical flow velocity. Axisa et al [7] were the first to present work on vibration of tube bundles subjected to both steam-water and air-water cross-flow. Results reported by them on damping for tube bundles in a cross-flow are valuable due to the fact as the research has been carried out related to vibration of tube bundles subjected to steam-water cross-flow. A comparative study of square and triangular arrays has also been carried out by them, which was not done previously. The reported experimental results/data could be used as a reference purpose.

Fritz [8] studied the effects of liquids on the dynamic motions of immersed solids and suggested a theoretical relation for added mass. Chen, et. al. [9] for a circular cylindrical structure provided design guide for calculating hydrodynamic mass. However, experimental results reported by Morretti [10] are the milestone for hydrodynamic mass evaluation for a tube surrounded by rigid tubes. Pettigrew, et. al. [11] commenced an extensive program to study the vibration behavior of tube bundles subjected to two-phase cross-flow. They carried out analysis of hydrodynamic mass and damping characteristics of tube bundles and presented more reliable results. Reliability obtained through lower damping values results with single flexible tube surrounded by rigid tube than with all flexible tubes. They reported that hydrodynamic mass is practically independent of mass flux and the effect of bundle configuration for a given p/d ratio is not large. Thus, the early 1980's saw the research era of flow-induced vibration related to dynamic parameters hydrodynamic mass effect and damping in tube bundles in form of various international conferences. By the present time, importance and knowledge of dynamic parameters in the field of flow-induced vibration has evolved sufficiently. A profuse published data, monographs and design guidelines are available for the guidance of researchers and maintenance engineers in the industry.

The purpose of this paper is to provide an overview of our state of knowledge and role of dynamic parameters in flow-induced vibration on tube bundles due to current trend of larger heat exchangers. The enormous literature available in the field and the very modest space available here, the reference list and the papers incorporated are to be highly selective and not all published data, experimental work and studies can be covered. The present paper provides analysis, evaluation, formulation, published data and effects pertaining to dynamic parameters, experimental studies related to hydrodynamic mass and damping by a large number of researchers. Guidelines for experimental research and heat exchangers design related to dynamic parameters are provided.

2. DYNAMIC PARAMETERS

For analysis of flow-induced vibration and related effects of dynamic parameters on tubes bundle it is important to have the knowledge of damping and hydrodynamic mass.

2.1 Hydrodynamic Mass

As a result of flow-induced vibration, the tubes vibrate inside the shell, to accommodate the tubes, shell side fluid is displaced. The influence of the displaced fluid is taken care by augmenting the mass of the vibrating tubes by including the hydrodynamic mass or added mass. The same is incorporated while calculating the natural frequency of the tube. Simply hydrodynamic mass or added mass is defined as the equivalent external mass of the tube vibrating with tube [12]. The component of fluid force on the cylinder (any body or tube) is the mass of the fluid entrained by the cylinder (any body or tube). This force is called hydrodynamic mass or added mass and it also acts in the direction of fluid acceleration [13]. So, the added mass of fluid is equal to the mass of the fluid displaced by the tube times an added mass coefficient Cm. Added mass is a function of geometry, density of fluid and the size of the tube [10].

2.2 Damping

For the stability of the tubes array damping plays an important role. When the tube is excited by any one of the excitation mechanism damping limits the tube response. In heat exchanger tube bundles, a vibrating tube dissipates energy through a variety of means. In general term, damping is defined as the ability of a structure to dissipate vibratory energy or it is result of energy dissipation during vibration [8]. Damping in broad terms is classified into three categories as follows:

(a) Structural Damping

(b) Material/Internal Damping

(c) Fluid Damping

In case of a tube, damping is its ability to absorb energy. Damping is usually expressed in terms of a nondimensional damping ratio. A value much less than one is normal; the smaller the value, the less energy absorbed in each cycle [14]. According to Pettigrew, et. al., damping in multispan tube bundles or in a vibrating system is due to several possible dissipation mechanisms [15]. Here we will discuss three categories of damping.

2.2.1 Structural Damping

Structural damping is a strong function of number and thickness of the tube supports, the size of baffles hole clearance, the surface adhesion effects between tube and baffle materials. Lowery [16] reported that damping is roughly proportional to support thickness. Damping for a 5/8" plate is about twice that of a 1/8" plate. For very small clearance ~ 0.008" damping appears to be large [15, 17]. Pettigrew, et. al. [15] studied the effect of the number of spans and supports for a heat exchanger and they found a normalized damping ratio ζ_n as:

$$\zeta_n = \zeta \frac{N}{N-1} \tag{1}$$

where N is the number of spans.

2.2.2 Material/Internal Damping

This type of damping is due to yielding, heating and internal energy dissipation of materials [13]. The response of the structure is a function of excitation and damping. Material damping in simple words is termed as the internal stiffness of material to absorb energy. The flexibility of the material depends upon modulus of elasticity. Pettigrew, et. al. [15] concluded that material damping effect is negligible and its effect is found in tubes with no intermediate support, with welded ends. Where as Erskine, et. al. [18] highlighted the influence of material and young's modulus, due to variation in temperature as one of the cause of damage. Presently larger heat exchangers are being used in plants such as nuclear power plants which are exposed to higher temperatures. Pettigrew [15] therefore concluded based on facts the current trend to test tubes at higher temperatures (250-350°C) which affects the material composition of the test tubes and the material damping accordingly.

2.2.3 Fluid Damping

Fluid damping is the most ambiguous component. It varies with the flow viscosity of the shell side medium, the solidity of the tube array, and flow velocity of the shell side medium. Fluid damping is the result of the viscous shearing of the fluid at the surface of structure and flow separation [13]. Therefore the fluid damping is flow- dependent. Carlucci et al [2] reported that flow- dependent damping results from the action of the flow drag force on the cylinder.

3. MEASUREMENT TECHNIQUES FOR DYNAMIC PARAMETERS

Measurement of dynamic parameters includes the determination and measurement of hydrodynamic mass and damping. All measurement techniques are almost based on the same idea. Most common and reliable are discussed at present.

3.1 Hydrodynamic Mass Measurement Techniques

Most frequently used measurement techniques for hydrodynamic mass by various researchers are presented here. Hydrodynamic mass (m_h) is related to tube natural frequency f in two-phase mixture as discussed by Carlucci and Brown [2] and it can be calculated as given by:

$$m_h = m_t\left[\left(f_g/f\right)^2 - 1\right] \tag{2}$$

where m_t denotes the mass of the tube alone and f_g is the natural frequency in air. It is important to measure tube frequency at mass fluxes sufficiently below fluid elastic instability.

Blevins [13] suggested an analytical formula to determine added mass coefficient of a single flexible tube surrounded by an array of rigid tubes, as shown in Fig. 1(a). Approximation is also reasonable for more complex case of flexible cylinders. According to Pettigrew, et. al. [11], hydrodynamic mass depends on the ratio P/d (Pitch to Diameter) of the tube pitch P to diameter d.

Morreti, et. al. [10] presented experimental results for a single flexible tube surrounded by rigid tubes in a hexangular array and a square array as shown in Fig. 1(b). The experimental results were obtained for P/d ratio from 1.25-1.5. Hydrodynamic mass coefficient (C_m) evaluated from tests results for four different pitch to diameter ratios are given in Table 1 and Fig. 1(b). The Fig. 1(b) for determination of hydrodynamic mass coefficient C_m has been included by TEMA [21].

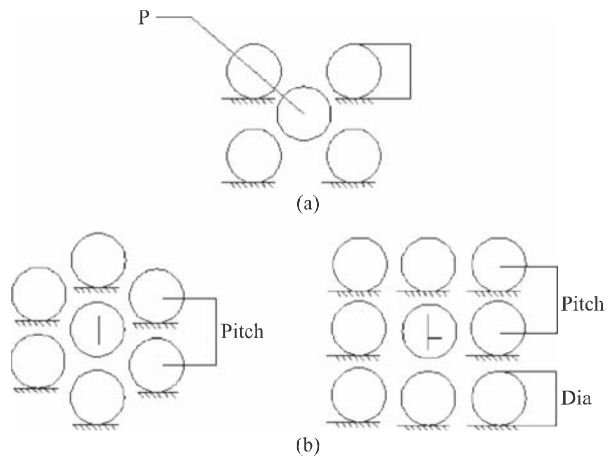

FIG. 1 (A). TUBE ARRANGEMENT FOR DETERMINING ADDED MASS COEFFICIENT,(a) BLEVINS MODEL [13] (b) MORRETTI MODEL [10, 20]

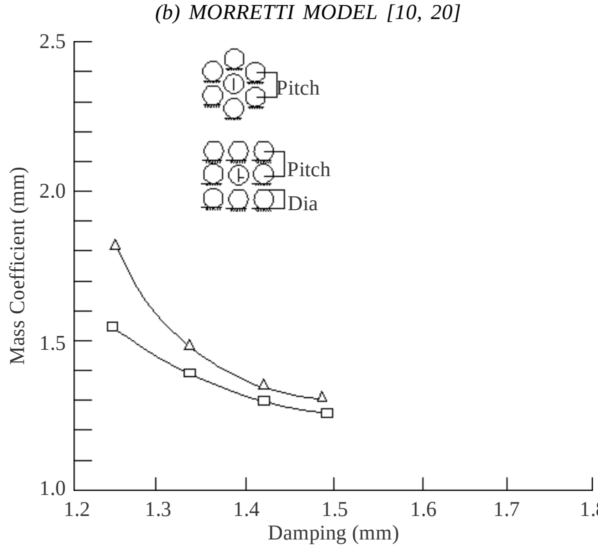

FIG. 1(B). ADDED MASS COEFFICIENT [10, 20]

3.2 Damping Measurements Techniques

All techniques for damping measurement are based on the same idea. Noteworthy are presented by Blevins[13] and Mitra [22], as mentioned below:

(a) The Free-Decay Method

(b) Half-Power Bandwidth Method

(c) Magnification Factor Method

(d) Random Decrement Technique

3.2.1 The Free Decay Method

In free decay technique damping is measured using a transducer, such as a strain gauge or accelerometer, an amplifier and a strip chart recorder [13,22]. The time history of strain gauge signal and the damping ratio is evaluated by determining the number of cycles (N) it takes for the initial amplitude to decay to half its value as evident from Fig. 2.

If N cycles are required for the amplitude to decay to one half of the initial amplitude, the damping factor is calculated as [13]:

TABLE 1. HYDRODYNAMIC MASS C_m [10, 20]

Pitch-to-Diameter Ratio	C_m	
	Triangular Pitch	Square Pitch
1.25	1.756	1.519
1.33	1.429	1.381
1.42	1.347	1.286
1.50	1.274	1.272

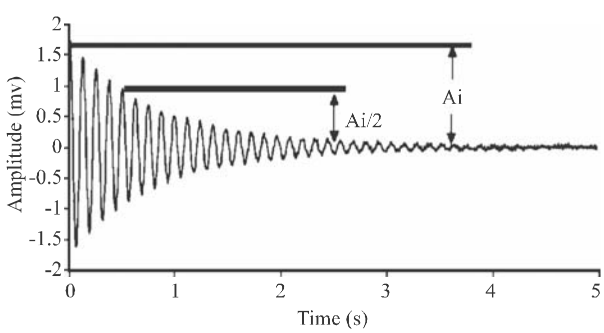

FIG. 2. TIME DECAY OF TUBE VIBRATION [22]

$$\zeta = \frac{\log_e 2}{2\pi N} = \frac{0.1103}{N} \quad (3)$$

3.2.2 Half Power Bandwidth Method

The bandwidth of the response $\Delta\omega$ is defined as the width of the frequency response at $1/\sqrt{2}$ times the peak amplitude. The excitation frequencies (ω_1 and ω_2) that produce these half power points response $1/\sqrt{2}$ times the resonant response as discussed in Blevins [13]. The peak resonance occurs at approximately the natural frequency a condition called resonance. The bandwidth is found to be proportional to damping factor, given by:

$$\zeta = \pi f/f_n \quad (4)$$

where $\Delta f = (\omega_2 - \omega_1)/2\pi$

To measure the damping from Equation (4), only the resonant frequency f_n and bandwidth must be obtained. So Fourier transforms of the strain gauge signal is obtained to determine the total damping in both single-phase and two-phase flow. The width of the peak of the amplitude spectrum is determined to calculate the damping ratio, as shown in Fig. 3. In the Fig. 3, f_n is the peak frequency and Δf is the frequency band corresponding to 70.7% of the peak value amplitude. The higher modes are not involved and log decrement levels can easily be controlled, which is the advantage of this method over the others.

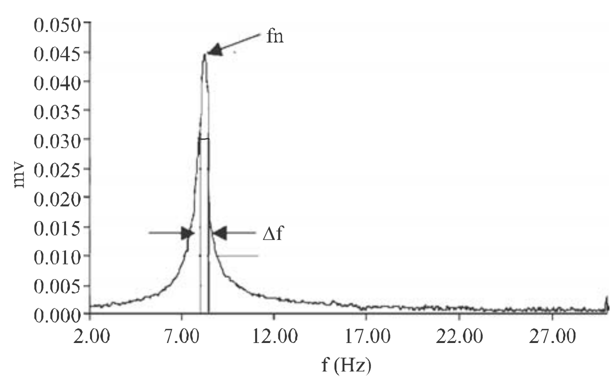

FIG. 3. DAMPING RATIO MEASUREMENTS FROM AMPLITUDE SPECTRUM [22]

3.2.3 Magnification Factor Method

The peak response of a lightly damped body at steady harmonic excitation is inversely proportional to damping [13]. The damping by magnification factor is evaluated as:

$$\zeta = \frac{F_o}{2kA\rho} = \frac{1}{2Q} \tag{5}$$

The ratio of the dynamic resonant response to the response to a steady load of same magnitude is given by [13]:

$$A_p/y_s = Q = 1/2\zeta \tag{6}$$

where Q is called the magnification factor. The Equation (6) may be used to determine the damping factor if the magnifications factor Q or the amplitude of resonant response can be measured. The result is dependent on the mode shape, mass distribution and details of the means used to excite the structure, unlike the free decay or bandwidth methods [13].

3.2.4 Random Decrement Technique

Random decrement technique may be used to determine the damping coefficient when a system is subjected to unknown random input and only measurable quantity is the system response. The technique is implemented by Yang, et. al. [23] and showed the applicability for random excitation utilizing only the response data. It utilizes assemble averaging technique so the results depend on the record length of the number of cycles of data.

4. DESIGN GUIDELINES

There is a strong need for establishing design procedures to avoid tube failures due to excessive flow-induced vibration prior to manufacture of heat exchanger preferably at design stage. Enormous published data and design parameters are available. Therefore, design guidelines are recommended to prevent tube failures due to unwarranted vibration and related to hydrodynamic mass and damping.

4.1 Hydrodynamic Mass Guidelines

As observed during the flow-induced vibration analysis, the vibrant tubes displace the shell side fluid. This hydrodynamic mass will have substantial effect on the natural frequency of the tubes. Carlucci and Brown [2] suggested following formulation for hydrodynamic mass:

$$m_h = m_t\left[\left(f_g/f\right)^2 - 1\right] \tag{7}$$

Assuming homogenous density for the two-phase flow hydrodynamic mass evaluation recommended by Rogers, et. al. [19] is given by:

$$m_h = \left(\frac{\rho\pi d}{4}\right)\left[\frac{(D_e/d)^2 + 1}{(D_e/d)^2 - 1}\right] \tag{8}$$

where D_e is equivalent diameter used to represent confinement due to surrounding tubes and is related to P/d ratio. De is given by:

For triangular tube bundle: $D_e/d = (0.96 + 0.5\ P/d)$

For square tube bundle: $D_e/d = (1.07 + 0.56\ P/d)$

In case of single flexible tube surrounded by an array of rigid tubes added mass coefficient suggested by Blevins [13] is given by:

$$C_m = \left[\frac{(D_e/d)^2 + 1}{(D_e/d)^2 - 1}\right] \tag{9}$$

The above approximation by Belvins [13] holds reasonable for all flexible tubes. The experimental results of Moretti et al [10] for a flexible single tube surrounded by rigid tubes have been included by TEMA [21] for determination of added mass coefficient. TEMA standards are being implemented for the design and manufacture of shell and tube type heat exchangers used for power plants and industrial applications.

4.2 Damping Design Guidelines

Determination of damping through experimental method is a bit complex phenomena. Empirical correlation is another way to determine damping. Some important design guidelines are presented here.

4.2.1 Damping in Gases

In heat exchangers with gas on shell side the dominant damping mechanism is friction between tubes and tube-supports, reported by Pettigrew, et. al. [15,24]. The design recommendation for friction damping ratio ζ_f, in percent is given by:

$$\zeta_f = 5\left(\frac{N-1}{N}\right)\left(\frac{L}{\ell_m}\right)^{1/2} \tag{10}$$

where L is support thickness, ι_m is span length and N the number of spans.

4.2.2 Damping in Liquids

Damping of multispan heat exchanger tubes with liquid on shell side is contributed by three energy dissipation mechanisms. The total damping is the sum of three. These are friction damping at support, viscous damping between tube and liquid, and squeeze-film damping in the clearance between tube and tube support. So

$$\zeta_T = \zeta_v + \zeta_F + \zeta_{SF} \tag{11}$$

Pettigrew and Taylor [24] recommended simplified formulation developed by Rogers, et. al. [19] which is also presented by Shahab [25] given by:

$$\zeta_v = \frac{100\pi}{\sqrt{8}}\left(\frac{\rho D^2}{m}\right)\left(\frac{2\nu}{\pi f D^2}\right)^{\frac{1}{2}}\left\{\frac{\left[1+\left(D/D_e\right)\right]^3}{\left[1-\left(D/D_e\right)^2\right]^2}\right\} \tag{12}$$

As the squeeze-film damping ζ_{SF} and friction damping ζ_F normally take place at the supports, semi-empirical expression are developed based on the available experimental data for total tube damping by Pettigrew, et. al. [26]. The same formulations are presented for design guidelines by Pettigrew and Taylor [24]. The squeeze-film damping is given by:

$$\zeta_{SF} = \left(\frac{N-1}{N}\right)\left(\frac{1460}{f}\right)\left(\frac{\rho D^2}{m}\right)\left(\frac{L}{\ell_m}\right)^{1/2} \tag{13}$$

and friction damping is given by:

$$\zeta_F = 0.5\left(\frac{N-1}{N}\right)\left(\frac{L}{\ell_m}\right)^{1/2} \tag{14}$$

where (N-1/N) is the ratio of the number of supports over the number of spans. So a total damping for tubes in liquid mentioned in Equation (15) is given by Pettigrew, et. al. [15]:

$$\zeta_T = \frac{100\pi}{\sqrt{8}}\left(\frac{\rho D^2}{m}\right)\left(\frac{2\nu}{\rho f D^2}\right)^{1/2}\left\{\frac{\left[1+\left(D/D_e\right)\right]^3}{\left[1-\left(D/D_e\right)\right]^2}\right\} + \left(\frac{N-1}{N}\right)\left[\left(\frac{1460}{f}\right)\left(\frac{\rho D^2}{m}\right)+0.5\right]\left(\frac{L}{\iota_m}\right)^{1/2} \tag{15}$$

Therefore, damping in multispan heat exchanger is an important parameter, which is evaluated as the sum of friction, viscous, and support damping. This total damping is incorporated for single phase or liquid flow.

4.2.3 Damping in Two-Phase Flow

Initial formulation for two-phase flow is given by Carlucci and Brown [2]. The total damping is evaluated as give by:

$$\xi_T = \zeta_s + \zeta_v + \zeta_F + \zeta_{TP} \tag{16}$$

Pettigrew and Taylor [27] reviewed the vibration behavior of shell and tube heat exchangers. They suggested design guidelines for total damping ratio of a multispan heat exchanger tubes in two-phase flow as:

$$\xi_T = \zeta_s + \zeta_v + \zeta_{TP} \tag{17}$$

For the conditions, liquid in between the tube and the support, Pettigrew, et. al. [26] suggested support damping as:

$$\xi_s = \zeta_{SF} + \zeta_F$$

$$\zeta_s = \left(\frac{N-1}{N}\right)\left[\left(\frac{1460}{f}\right)\left(\frac{\rho D^2}{m}\right) + 0.5\right]\left(\frac{L}{\ell_m}\right)^{1/2} \tag{18}$$

Considering a homogenous property of two-phase flow mixture and viscous damping in two-phase mixture analogous to viscous damping in single-phase [28] formulation is given by Equation (19):

$$\zeta_v = \frac{100\pi}{\sqrt{8}}\left(\frac{\rho_{TP} D^2}{m}\right)\left(\frac{2\nu_{TP}}{\pi f D^2}\right)^{1/2}\left\{\frac{\left[1+\left(D/D_e\right)\right]^3}{\left[1-\left(D/D_e\right)^2\right]^2}\right\} \tag{19}$$

For two-phase damping component in percent a semi-empirical expression from the available experimental data suggested by Pettigrew and Taylor [24] is as:

$$\zeta_v = 0.4\left(\frac{\rho_\ell D^2}{m}\right)\left[f\left(\varepsilon_g\right)\right]\left\{\frac{\left[1+\left(D/D_e\right)\right]^3}{\left[1-\left(D/D_e\right)^2\right]^2}\right\} \tag{20}$$

In case of two-phase damping, it is evaluated through friction, viscous, support and two-phase components of damping. Pettigrew, et. al. [29] carried experimentation for evaluating damping of tubes due to internal two-phase flow. Experimental data are reported, showing a strong dependence of two-phase damping on void fraction, flow velocity and flow regime. The two-phase damping ratio reaches a maximum value at the highest void fraction before the transition to a churn flow regime. They reported that there is a direct correlation between two-phase damping and the interface surface area, estimated assuming rigid spherical bubbles. It appears that viscous dissipation mechanisms govern the additional component of damping due to two-phase flow. Zhang, et. al. [30] carried out research to develop semi-analytical models for correlating vibration excitation forces to dynamic characteristics of two-phase flow in a rotated triangular tube bundle and for better understanding of the nature of vibration excitation forces. They reported that measurements of the dynamic characteristics of the two-phase flow indicate that quasi-periodic drag and lift forces are generated by different mechanisms that have not been observed previously. The relationships between the lift or drag forces and the dynamic characteristics of two-phase flow are established through fluid mechanics momentum equations.

5. CONCLUSIONS

The 20th century is the era of knowledge, analysis, and development of flow-induced vibration related problems. Especially much attention is provided to flow-induced vibration related parameters to the industrial heat exchangers and process steam generators. The current paper has reviewed the dynamic parameters related effects of flow induced vibration in tube bundles. Importance of hydrodynamic mass and damping is evident from the research work presented in the paper by various researchers. Following have been observed:

(i) It is observed during the flow-induced vibration analysis that hydrodynamic mass have substantial effect on the tubes in a bundle. It is difficult to measure damping in tube bundles and determination of damping through experimental method is a bit complex phenomenon.

(ii) Nevertheless hydrodynamic mass and damping are of major concern in the design and operation of shell and tube heat exchangers. Most of the vibration problems related to hydrodynamic mass and damping can be anticipated and considered appropriately at the design stage.

(iii) Failures due to flow-induced vibration must be avoided. It is not important to wait for the ultimate results, but industry requires utmost reliable information available. Thus, reliable design procedures and guidelines based on the currently available data related to hydrodynamic mass and damping are enumerated in this paper.

(iv) Present paper covers the hydrodynamic mass and damping parameters related to flow-induced vibration. It is recommended that other vibration causing factors such as fluid elastic instability, vortex shedding, and random turbulence excitations may be incorporated in the design of heat exchanger.

(v) Experimentation may be carried out on tubes bundle for two phase steam flow and thermal damping may be incorporated for experimental results.

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