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A EVALUATION OF A TWO-PHASE DAMPING MODEL ON TUBE BUNDLES SUBJECTED TWO-PHASE CROSS FLOW

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ABSTRACT

The analytical model (Sim; 2007), to predict the two-phase damping ratio for upward cross-flow through horizontal tube bundles, has been evaluated. The damping model was formulated, based on Feenstra's model (2000) for void fraction and various models (homogeneous, Levy, Martinelli-Nelson and Marchaterre) for two-phase friction multiplier. The analytical results of drag coefficient on a cylinder and two-phase Euler number were compared with the experimental results by Sim-Mureithi (2010). The factor, a relation between frictional pressure drop and the hydraulic drag coefficients, could be determined by considering experimental results. The two-phase damping ratios, given by the analytical model, were compared with existing experimental results. It was found that the model, based on Marchaterre's model, is suitable for air-water mixture while the Martinelli-Nelson's model for steam-water and Freon mixtures. The two-phase damping ratio is independent on pitch mass flux for air-water mixture, but it is more or less influenced by the mass flux for steam-water/Freon(134) mixtures. The two-phase damping ratios, given by the present model, agree well with experimental results for a sufficiently wide range of pitch mass ratio, quality and *p/d* ratios.

INTRODUCTION

The integrity of steam-generator tubes is an important aspect of the long term reliable operation of nuclear power plants. Extensive research has been carried out in the region where the flow-induced vibration has been related to fluid-elastic instability in a tube array subjected two-phase cross-flow. Excessive vibration due to fluid-elastic instability often leads to tube failures or fretting wear damages in the exchangers. Such tube failures can be avoided by a comprehensive vibration analysis at the design stage. Damping is a result of energy dissipation during the vibration. Under operation conditions, damping of steam generator tube has various forms. As a result, some knowledge on tube damping mechanisms is required to avoid flow-induced vibration problems. The existing results of experiments on four-tube bundles configuration were presented (Pettigrew et al: 1989 a & b) and design guidelines were recently developed to prevent tube failures to excessive flow-induced vibration (Pettigrew et al; 2003).

Although the characterization of an energy-dissipation mechanism(s) can be reduced to a single parameter, damping ratio, the measurement of damping is not necessarily simple. To calculate damping ratios based on a measured vibration, various methods are used; amplitude logarithmic decrement, frequency-domain spectral response, exponential-decay curve fit and Nyquist Comparisons between these methods have been made for steam generator tubes by Janzen et al. (2005).

There is very little work done on damping in two-phase flow except Carlucci (1980) and Carlucci & Brown (1983) before 1985. They conducted a systematic study of damping of cylinder in axial two-phase flow simulated by air-water mixtures. They found that damping is highly dependent on void fraction. Some fundamental experiments were done with a single cantilevered tube immersed in two-phase mixtures generator by bubbling air through water. Hara and Kohgo (1982) studied the effect of void fraction, confinement and bubble size. Pettigrew and Knowles (1992) also investigated the effect of tube frequency and surface tension.

Little attention was given on vibration of tube bundles in two-phase cross flow before 1980. For a limited range of bundle geometries and flow condition pertaining to their steam generator design, Heilker and Vincent (1981) has done some work in air-water cross flow. Their single span tube bundles were exposed to flow over their entire length. Unfortunately, they measured the damping at the critical flow velocity for fluid-elastic instability, yielding unrealistically low damping values. Axisa et al. (1985) were the first to present results on vibration of the tube bundles subjected to air-water or steam-water cross flow. They tested three bundle configuration (i.e., normal-square, normal-triangle and rotated-triangular tube bundles), all of *p/d*=1.44. This yielded valuable results on damping. Later, Nakamura et al. (1986 a & b) also reported

data on fluid-elastic instability for a square bundle of cylinders of *p/d*=1.42 in both air-water and steam water cross flow. Unfortunately, no damping data are available. The semi-empirical approach by Pettigrew and Taylor (2003) was proposed by taking the lower envelope of the existing damping data.

Most of the previous researcher did not employ any means of measuring void fraction, and hence they relied on the homogeneous equilibrium model (HEM) to determine average fluid density and flow velocity of two-phase cross-flow. However, a proper determination of these quantities requires an appropriate, generally applicable two-phase void fraction model to account the velocity ratio of the phase. To predict the void fraction with the velocity ratio for upward cross-flow through horizontal tube bundles, a physically based model was developed by Feenstra etc. (2000). It agrees well experimental void fraction measurements in refrigerant 11 and air-water mixture for a sufficiently wide range of pitch mass flux, quality and *p/d* ratios.

An approximate analytical model, to predict the two-phase damping ratio for upward cross-flow through horizontal bundles, had been developed by Sim (2007). The model was formulated, based on Feenstra's model (2000) for void fraction and various models (homogeneous, Levy and Marchaterre) for two-phase friction multiplier. The important variables on the damping were identified. An empirical formulation of non dimensional pressure drop (Euler number) for single phase flow in tube bundles was proposed by Zukauskas et al. (1988). The model will allow researchers to provide analytical estimates of the damping ratios. However, it was requested to verify/improve the model by having comparison with experiments. Thus, two sets of experiments had been performed for various pitch mass fluxes of air-water mixture with changing void fraction. The analytical results of drag coefficient on a cylinder and two-phase Euler number were compared with experimental results by Sim-Mureithi (2010). The co-relation factor, a relation between frictional pressure drop and the hydraulic drag coefficients, was determined by considering experimental results. Taking into account the comparison and an additional model (Martinelli-Nelson; 1948) for two-phase friction multiplier, the approximate damping model has been modified /improved in the present work. The present results were evaluated with existing experimental results.

FEATURES OF TWO-PHASE CROSS-FLOW OVER TUBES WITHIN BUNDLE.

The pressure gradient, the fluid viscosity, and the Reynolds number control the flow over tubes. Flow over tubes within a bundle, shown Fig. 1(a), involves significant blockage of the flow passage; the pressure gradient at the tube surface is affected by the degree of flow constriction. The velocity distribution over the cross section of tube bundles may vary, depending on the net cross section area within bundle.

An important factor in determining the total drag on the tube bundles is the coefficient of pressure drag, which depends on the Reynolds number, the arrangement of tube within bundles, and the relative bundle pitches. The pressure drag coefficient and the longitudinal component of the pressure force, acting on a tube within bundle subjected to single-phase flow, are given by the following expressions, respectively;

$$c_w = \frac{2F_{px}}{\rho \hat{u}^2 A} \qquad F_{px} = \frac{L}{2}\int_0^{2\pi} Pd\cos\theta d\theta \tag{1}$$

where $A = \pi dL$ is the cross section area of the tube, perpendicular direction; L is the tube length. In the above equation, it is convenient for $\hat{u}$ to correlate data on the basis of the maximum velocity.

The dependence of the hydraulic drag on the bundles on the velocity will be clarified by analyzing the relationship between the static pressure drop in the bundle and the velocity. The pressure drop in a bundle exposed to a flow of fluid constant density can be defined in dimensionless form $Eu = \Delta P/(\rho \hat{u}^2) = f(\mathrm{Re}, p/d, z)$ where z is number of tubes.

The drag force on a staggered bundle increases with the reduction of the longitudinal pitch, since it is affected by the size of the space between tubes where eddies are generated. The drag on a tube bundle differs somewhat from that on a single row of tubes, since the drag is significantly affected by the flow turbulence generated by the upstream tube rows. A number of technologies employ compact tube bundles, in which the distance between the tubes is relatively small. Various experiments were performed with in-line tube bundles, $1.12 < p/d < 1.9$. Fortunately, the Euler number for single-phase flow is approximated by the expression (Zukauskas et al.);

$$Eu_{LO} = 0.307\,\mathrm{Re}_{LO}^{-0.1}(p/d-1)^{-0.36} \tag{2}$$

where subscript '*LO*' stands for liquid only. The Reynolds number is expressed as $\mathrm{Re}_{LO} = \rho_l \hat{u} d/\mu_l$. It is expected that the Euler number would be decreased with the pitch ratio and the Reynolds number.

In general, the hydraulic drag coefficient including viscous effect is expressed

$$C_D = \frac{2F_D}{\rho \hat{u}^2 A} = K \cdot Eu_{LO} \tag{3}$$

where the co-relation factor, *K*, is a relations between the Euler number and the hydraulic drag coefficients, which can be estimated empirically. In the equation, *A* is usually the projected area.

Pressure drop in two-phase flow is closely related to the flow pattern as defined by void fraction, α, and phase distribution. That information is therefore a prerequisite for evaluation of two-phase pressure drop – the subject matter of the preceding section. The two-phase friction pressure drop can be expressed in terms of the single-phase pressure drop of the liquid flow alone. The tube-wise overall value of two-phase friction multiplier, $\phi_{LO}^2 = (dp/dl)_{TP}/(dp/dl)_{LO}$, was proposed by Martinelli and Nelson (1948) versus exit quality x_e and pressure, assuming that the relationship between local quality increases linearly with flow direction and boiling starts at the inlet. Thus, the two-phase pressure drop can be obtained by

$$\left.\frac{dp}{dl}\right|_{TP} = \phi_{LO}^2 f_{LO}\frac{G^2}{2\rho_l d_e} + G^2\frac{1}{\rho_l}\left[\left(\frac{\rho_l}{\rho_g}-1\right)\frac{dx}{dl}\right] + \left[\alpha\rho_g + (1+\alpha)\rho_l\right]g \tag{4}$$

In the above equation, the friction coefficient of the liquid flow alone in a duct containing a tube bundle is independent of wall roughness and is given by Poiseuille's equation; $f_{LO} \approx 64/(\mathrm{Re}_{LO})$ for laminar flow, $f_{LO} \approx 0.316/(\mathrm{Re}_{LO}^{1/4})$ for turbulence flow.

Lockhart and Martinelli (1949) have developed a procedure for calculating the frictional pressure gradient of a adiabatic two-phase annular flow based on a correlation of the data obtained from horizontal flow of air and various liquids at atmospheric pressure. For prediction of the pressure drop during forced circulation boiling, Martinelli-Nelson (1948) assumed that the flow regime would always be 'turbulent(gas)-turbulent(liquid)'; .

$$\left.\frac{dp}{dl}\right|_{TP} = \left.\frac{dp}{dl}\right|_{LO} (1-x)^{1.75}\phi_{ltt}, \quad \phi_{ltt} = \left(1+\frac{C}{X}+\frac{1}{X^2}\right), \tag{5}$$

where ϕ_{ltt} denotes the ratio of the two-phase pressure drop to that which would exist if the liquid phase were to be following alone in the pipe. The subscript *tt* refers to turbulent gas flow and turbulent liquid flow, as an example, and value of *C* is tabulated in Table l. In the above equation, X is Martinelli parameter, expressed by

$$X = \left(\frac{\rho_g}{\rho_l}\right)^{1/(2-n)} \left(\frac{\mu_l}{\mu_g}\right)^{n/(2-n)} \left(\frac{1-x}{x}\right) \tag{6}$$

where n=0.2~0.25 are generally used for turbulent flow.

The friction multiplier for the homogeneous model is expressed as

$$\left.\frac{dp}{dl}\right|_{TP} = \left.\frac{dp}{dl}\right|_{LO} \left[1+x\frac{1/\rho_g - 1/\rho_l}{1/\rho_l}\right]\left[1+x\frac{\mu_l-\mu_g}{\mu_g}\right]^{-1/4} \tag{7}$$

A modification for non-homogeneous flow was suggested by Levy, based on a "momentum exchanger model".

$$\left.\frac{dp}{dl}\right|_{TP} = \left.\frac{dp}{dl}\right|_{LO} \frac{(1-x)^{1.75}}{(1-\alpha)^2} \tag{8}$$

This is quite similar to Martinelli-Nelson expression.

Marchaterre has suggested another modification that incorporates the mass flux, G and the equivalent diameter of flow channel, d_e;.

$$\left.\frac{dp}{dl}\right|_{TP} = \left.\frac{dp}{dl}\right|_{LO} \left[\frac{(1-x)^2}{1-\alpha} + \frac{g(\rho_l-\rho_g)\rho_l d_e}{2 f_{LO} G^2}\alpha\right] \tag{9}$$

In the present analysis for various geometry, gap mass flux, G_g, and diameter of tube, *d*, has been used instead of mass flux and the equivalent diameter. In the above equation, the friction coefficient of the liquid flow alone in a duct has been introduced; $f_{LO} \approx 64/(\mathrm{Re}_{LO})$ for laminar flow, $f_{LO} \approx 0.316/(\mathrm{Re}_{LO}^{1/4})$ for turbulence flow ,

Table 1. Value of *C* defined in ϕ_{lmm} -see eq. (5)

subscrip,*mm*	liquid	gas	*C*
tt	turbulent	laminar	20
vt	laminar	turbulent	12
tv	turbulent	laminar	10
vv	laminar	laminar	5

A VOID FRACTION MODEL FOR TWO-PHASE CROSS-FLOW IN HORIZONTAL TUBE BUNDLES.

To predict the void fraction for upward cross-flow through horizontal tube bundles, a void fraction model has been developed by Feenstra et al.(2000). The model development began with the relationship between void fraction, α, and flow quality, x ;

$$\alpha = \left[1+S\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)\right]^{-1} \tag{10}$$

where *S* is the ratio of gas velocity to liquid velocity and is the primary unknown in eq.(10) since quality, x, and gas and liquid phase densities, ρ_g & ρ_l, are usually easy to determine. The homogeneous void fraction is determined by substituting *S*=1 into the above equation. The problem was to identify the important variables that affected velocity ratio and to form dimensionless groups that were appropriate to the development of the model. After testing many correlations, the following form proved to fit the data in refrigerant 11 and air-water mixtures,

$$S = \frac{u_g}{u_l} = 1 + 25.7(Ri \times Cap)^{0.5}(p/d)^{-1} \tag{11}$$

where the Richardson number, *Ri*, and the capillary no, *Cap*, have the following forms,

$$Ri = (\rho_l - \rho_g)^2 g(p-d)/G_p^{\,2}, \quad Cap = \mu_l u_g/\sigma \tag{12}$$

In the above equation, G_p denotes the pitch mass flux. The capillary number requires knowledge of the surface tension, σ, and of absolute viscosity of the liquid phase, μ_l, both of which are readily determined from the fluid property tables. To obtain better agreement with the experimental data, the gas phase velocity is determined as flows;

$$u_g = \frac{xG_p}{\alpha\rho_g} \tag{13}$$

To solve the problem, initially it is required to know the value of void fraction that depends on the velocity ratio. Hence, calculating the capillary number is an iterative process whereby the velocity ratio is calculated starting from an assumed value and iterated until the assumed and calculated values agree within a desired degree of precision that in this case is about 0.1%.

It should be noted that the model was developed to correspond to the available void fraction data, most of which correspond to adiabatic flow. Experiments with boiling or simulated void fraction generation on tubes are scarce (Gidi et al., 1997; Schrage et al., 1988) and the results are difficult to interpret due to the constantly changing flow quality in the tube bundles. The calculated void fractions by the proposed void fraction model agree well with the available experimental data, mainly given for normal square array and normal triangle array. To apply the model for rotated array, it is question we can use gap mass flux instead of pitch mass flux. The relations between gap mass flux and the pitch mass flux are $G_g = G_p$ for normal triangle (*NT*) and normal square (*NS*) arrays, $G_g = 2G_p/\sqrt{3}$ for rotated triangle (*RT*) array and $G_g = G_p\sqrt{2}$ for rotated square (*RS*) array.

HYDRAULIC DRAG COEFFICIENTS ON A TUBE SUBJECTED TO TWO-PHASE FLOW.

Flow around a blunt object is usually treated empirically. To formulate the problem for the two-phase damping, we are interested primarily in the drag force on the body in the direction of the flow. The drag force acting on a tube within bundles subjected to two-phase flow can be expressed in terms of dimensionless drag coefficient,

$$\begin{aligned} F_D &= \frac{A}{2}\left[\rho_l(1-\beta)+\rho_g\beta)\right]u_p^{\,2}\hat{C}_{DTP} \\ &= \frac{A}{2}\left[\rho_l u_l^{\,2}(1-\alpha)+\rho_g u_g^{\,2}\alpha)\right]C_{DTP} \end{aligned} \tag{14}$$

where A is the projected area. In the above equation, drag coefficients, $\hat{C}_{DTP}\,\&\,C_{DTP}$, are defined for homogeneous and non-homogeneous two-phase parameters, respectively.

Considering the above relations and the information discussed in the previous sections, the hydraulic drag coefficients can be expressed in terms of Euler number, correction factor and the two-phase friction multiplier, as follows;

$$C_{DTP} = f\left[Eu_{LO}, K_{TP}, \frac{(dp/dl)_{TP}}{(dp/dl)_{LO}}\right] \tag{15}$$

where Marchaterre's expression to estimate the two-phase friction multiplier could be used for air-water mixture, while Levy's and Martinelli's expressions for for forced circulation boiling (steam-water and Freon).

It is shown that the value for air-water system depends on mass flux while those for the other systems do not depend on mass flux. Using the void fraction model proposed by Feenstra et. al., the properties of the two-phase flow can be calculated. As a result, the hydraulic drag coefficient can be approximated as

$$C_{DTP} \approx K_{TP}\cdot Eu_{LO}\cdot\phi_{LO}^{\,2}\cdot\frac{MF_{LO}}{MF_{TP}} = K_{TP}Eu_{TP} \tag{16}$$

$$\hat{C}_{DTP} \approx K_{TP}\cdot Eu_{LO}\cdot\phi_{LO}^{\,2}\cdot\frac{MF_{LO}}{MF_{TPH}} = K_{TP}\hat{E}u_{TP} \tag{17}$$

where MF denote the momentum flux.

$$\begin{aligned} MF_{TP} &= \left[\rho_l u_l^{\,2}(1-\alpha)+\rho_g u_g^{\,2}\alpha\right]d \\ MF_{TPH} &= \left[\rho_l(1-\beta)+\rho_g\beta\right]u_p^2 d \\ MF_{LO} &= \rho_l\hat{u}_{LO}^{\,2}d = G_g^{\,2}d/\rho_l \end{aligned} \tag{18}$$

LIFT-DIRECTION DAMPING RATIO IN TWO-PHASE FLOW

The study of damping in two-phase flow is difficult for several reasons. First, damping in two-phase flow depends on void fraction that is an additional parameter. Second, damping measurements are difficult to obtain; since, it is not possible to maintain a stagnant two-phase mixture. Third, damping in two-phase flow is dependent on flow regime. In spite of the above difficulties, it is essential to arrive at some design guidelines for damping in two-phase flow. The semi-empirical relationships were combined into a general damping formulation.

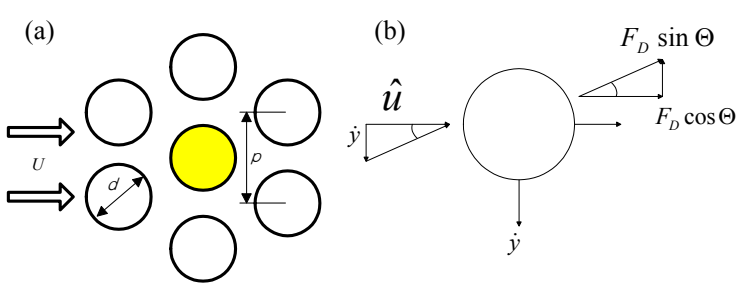

Figure 1. Section in a flow

Consider the elastically supported structure shown in Fig. 1(b), which is exposed to a high Reynolds number cross flow. As the structure vibrates, a relative component of flow velocity is induced. Using the linearizing approximations gives the following net vertical forces induced by the relative drag;

$$F_y = F_D\sin\Theta = \frac{C_{DTP}}{2}\left[\rho_l u_l^{\,2}(1-\alpha)+\rho_g u_g^{\,2}\alpha)\right]d\,\frac{\dot{y}}{u+\dot{x}} = c_y\dot{y} \tag{19}$$

In the above equation, the average velocity of two-phase flow, u is expressed as

$$u = \frac{\rho_l u_l(1-\alpha)+\rho_g u_g\alpha}{\rho} \tag{20}$$

where the mean density of the fluid is $\rho = \rho_l(1-\alpha)+\rho_g\alpha$.

Considering the equation of motion for vertical degree of freedom, the drag-induced damping ratio due to the cross flow is proportional to flow velocity and inversely proportional to the natural frequency in hertz, f_n,

$$\zeta_y = \frac{F_D}{um}\frac{1}{4\pi f_n} = \frac{C_{DTP}\cdot MF_{TP}}{um}\frac{1}{8\pi f_n} = K_{TP}\cdot Eu_{LO}\cdot\phi_{LO}^{\,2}\cdot\frac{MF_{LO}}{um}\frac{1}{8\pi f_n} \tag{21}$$

where $m(=m_t+m_h)$ is the total mass per unit length, including the hydrodynamic mass, m_h. Damping ratio is defined as the ratio of actual damping over critical damping. The hydrodynamic added mass is expressed as

$$m_h = \left[\rho_l(1-\alpha)+\rho_g\alpha\right]\bullet\frac{1}{4}\pi d^2\chi \tag{22}$$

in terms of added mass coefficient, $\chi = [(D_e/d)^2+1]/[(D_e/d)^2-1]$.

For a tube inside a triangular tube bundle, the equivalent diameter, D_e, is taken as $D_e/d = (0.96+0.5p/d)p/d$. Similarly, $D_e/d = (1.07+0.56p/d)p/d$ is proposed for a square tube bundle,

As discussed by Pettigrew et al. (1989 b), there are three important energy dissipation mechanisms that contribute to damping of multi-span heat exchanger tubes with liquids on the shell side. They are viscous damping, ζ_v, between tube and liquid, squeeze-film damping, ζ_{SF}, in the clearance between tube and tube-support and frictional damping, ζ_F, at the support. Thus, the total damping, ζ, is given by $\zeta = \zeta_{SF}+\zeta_v+\zeta_F$. The subject of heat exchanger tube damping in two-phase flow was reviewed recently by Pettigrew and Taylor (1997). The total damping ratio, ζ, of multi-span heat exchanger tube in two-phase flow is expressed as

$$\zeta_y = \zeta_s+\zeta_v+\zeta_{TP} \tag{23}$$

where ζ_v, ζ_S and ζ_{TP} are the viscous, structural(or support) and two-phase damping ratios, respectively. To obtain

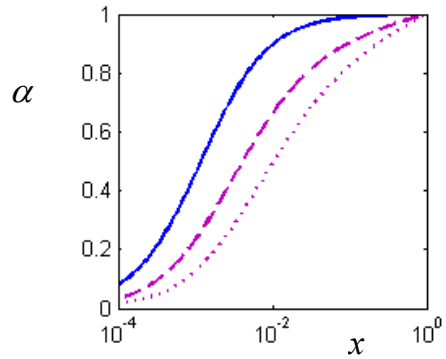

Figure 2. Void fraction v.s. mass quality given by homogeneous model: ___ and Feenstra model (G_p=800[kg/m^2s]; --- , G_p=200 [kg/m^2s]; ……)

the two-phase component of damping alone, structural and viscous components were subtracted, i.e.

$$\zeta_{TP} = \zeta_y - (\zeta_s + \zeta_v) \tag{24}$$

Roger et al. (1989) developed formulation for viscous damping, $\pi f D^2 / 2\nu$ >3300 and d / De <0.5 which covers most heat exchangers;

$$\zeta_v = \frac{\pi}{\sqrt{8}}\left(\frac{\rho d^2}{m}\right)\left(\frac{2\nu_{TP}}{\pi f_n d^2}\right)^{1/2}\left(\frac{[1+(d/D_e)^3]}{[1-(d/D_e)^2]^2}\right) \tag{25}$$

where ν_{TP} is the equivalent two-phase kinematic viscosity of fluid as per McAdams et al. (1942) for homogeneous two-phase flow;

$$\nu_{TP} = \frac{\nu_l}{1+\beta(\nu_l/\nu_g - 1)} \tag{26}$$

where β is the volumetric quality. Generally, squeeze-film and friction damping take place at the supports. However, in the present analysis, they are not considered.

COMPARISON OF DAMPING RATIO WITH EXPERIMENTS

In order to calculate damping ratio, the information of the flow pattern as well as two-phase pressure drop is prerequisite,

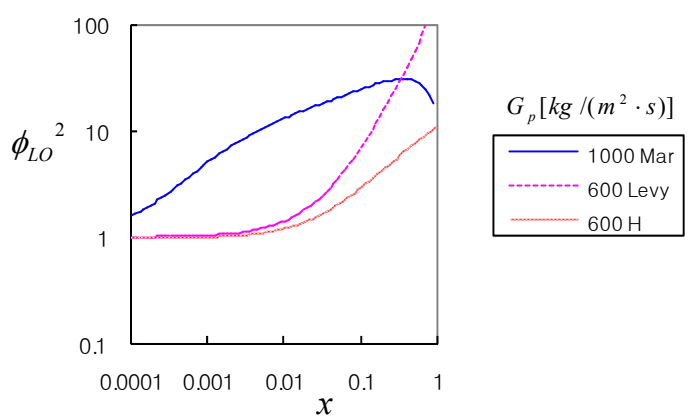

Figure 3. Two-phase friction multiplier given by each model $p/d = 1.47, d = 13mm, NS$

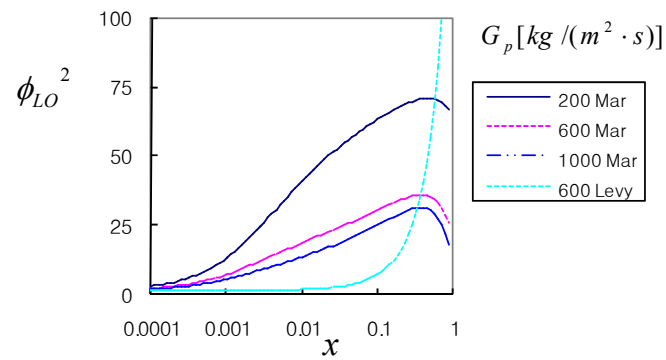

Figure 4. The effect of mass flux on two-phase friction multiplier for air-water mixture; $p/d = 1.47, d = 13mm, NS$

as discussed in the previous sections. In Fig. 2, the predictions of the void fraction model (Feenstra et al., 2000) and the homogeneous model are compared for the case of air-water cross-flow for $p/d = 1.5, d = 38mm, RS$. As shown in the figure, the void fractions given by the homogeneous model are over-estimated. The values given by Feenstra's model become closer to the homogeneous results with increasing the mass flux. For the similar cases ($p/d = 1.47, d = 13mm$, normal square array), the two-phase friction multipliers (ϕ_{LO}^2), given by eqs. (7,8&9) are shown in Fig. 3. In general, the multiplier for low-pressure case is much higher than that for high-pressure case. It is shown that the value given by homogeneous model is under-estimated. In order to show the effect of mass flux on the multiplier, Fig. 4 is presented for the same cases. With increasing mass flux, the multiplier, given by Marchatterre, decreases for air-water system of low-pressure. It is shown that the multiplier, given by Lecy's model, is much less than that by Marchaterre's model at relatively low mass quality while its trend is inversed at high mass quality. The typical results of two-phase friction multiplier for steam-water mixture (86 bar) are shown in Fig 5.

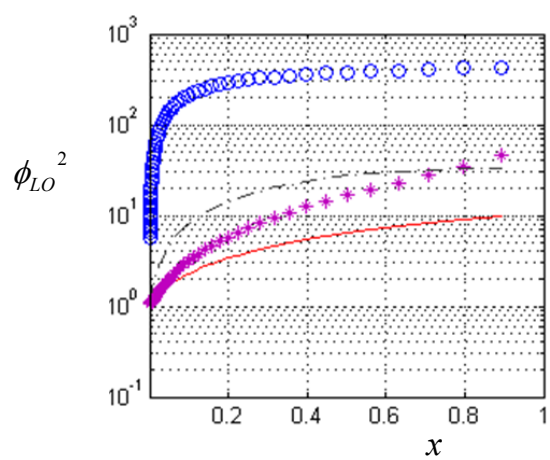

Figure 5. Two-phase friction multiplier given by each model for steam-water mixture $p/d = 1.5, d = 38mm, RS$;

Martinelli-Nelson;-----, Levy; ****, Marchaterre; ○ , homogeneous;____

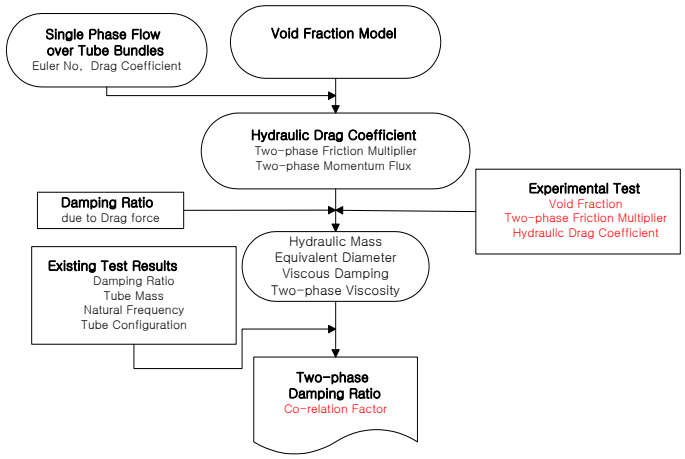

Figure 6. Developed algorithm for present model

The algorithm for present model is summarized in Fig. 6. As discussed in the previous sections, the properties of the two-phase flow can be calculated with the void fraction model proposed by Feenstra et al. Considering the momentum flux, the hydraulic drag coefficient can be approximated. And then, using the linearizing approximations gives the net vertical forces induced by the relative drag, from which the drag-induced damping ratio due to the cross flow can be calculated. However, still we need to decide the co-relation factor, K_{TP}. This value can be estimated by utilizing existing empirical results.

To verify the present analytical model, it is required to get more information about two-phase Euler number and drag coefficients keeping the pitch mass flux constant. For the purpose, experiments for air-water mixture had been performed. Typical experimental results of two-phase Euler number are presented in Fig. 7 and drag coefficient in Fig. 8. The results are compared with analytical results. To obtain the drag coefficients analytically, the co-relation factor, K_{TP}, was

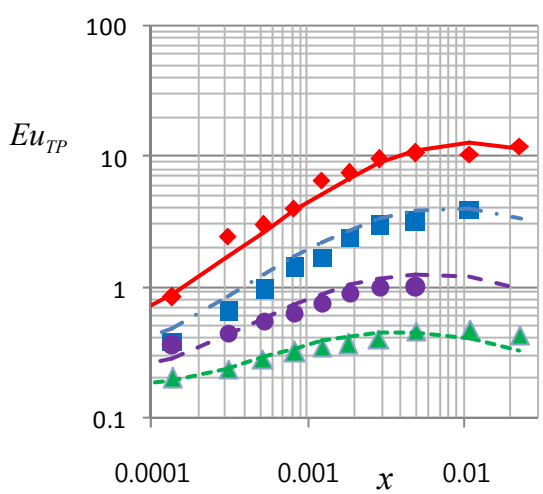

Figure 7. Two-phase Euler number for air-water mixture, obtained by present model (lines) and experiments (symbols) for $G_p\,[kg/(m^2 \cdot s)]$ = 100 (◆ ,__), 200 (■ , _ . _) , 400 (● ,__ __) and 800 (▲ ,---). $p/d = 1.5, d = 38mm, RS$

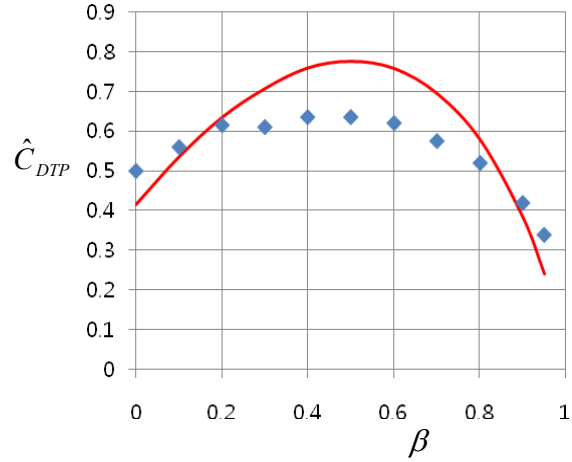

Figure 8. Two-phase drag coefficients obtained by present model (line) and experiments (◆) for G_p=800 $[kg/(m^2 \cdot s)]$

determined to be 3 by considering experimental results (Sim-Mureithi(2010). It is found the analytical results for air-water mixture, based on Marchaterre's model for two-phase friction multiplier, agree well with the experimental results.

As mentioned before, damping in two-phase flow is very complicated. Two-phase damping is highly dependent on void fraction and flow regime. As shown in eq. (21), total damping ratio is expressed in terms of the Euler No. (Eu_{LO}), the two-phase friction multipliers ($\phi_{LO}{}^2$) and the average velocity of two-phase flow (u). The Euler number decreases with Reynolds number given for liquid only-see eq. (2). The number is only function of mass flux for a given geometry. The friction multiplier of two-phase flow, given by Marchaterre's model (air-water), has a maximum value at a certain mass quality while the value, given by other models (Levy:steam-water), increases with the mass, as shown in Figs. 3 & 4. The average velocity increases with the quality. As a result, it is expected that the damping ratio has maximum value at certain mass quality. In Fig. 9, the effect of mass flux on the total damping ratio, based on Marchaterre's model, is presented for air-water mixture. The total damping ratio slightly increases with mass flux, since the two-phase friction multiplier decreases with mass flux while the momentum flux and average velocity

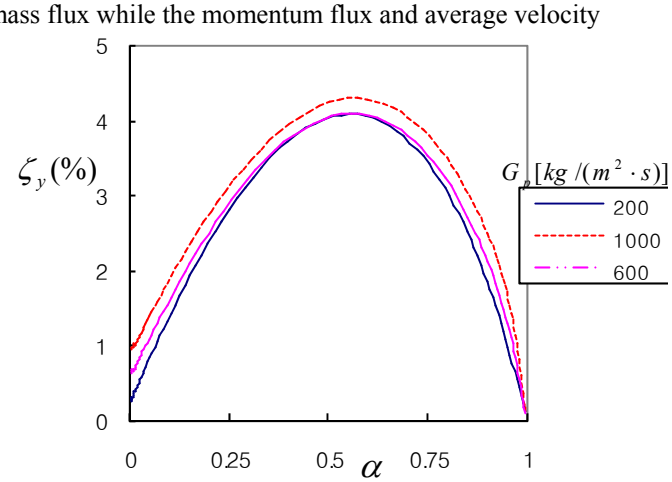

Figure 9. Total damping ratio for various mass fluxes, $p/d = 1.47, d = 13mm, m_t = 0.33kg/m, f_n = 29, NS$

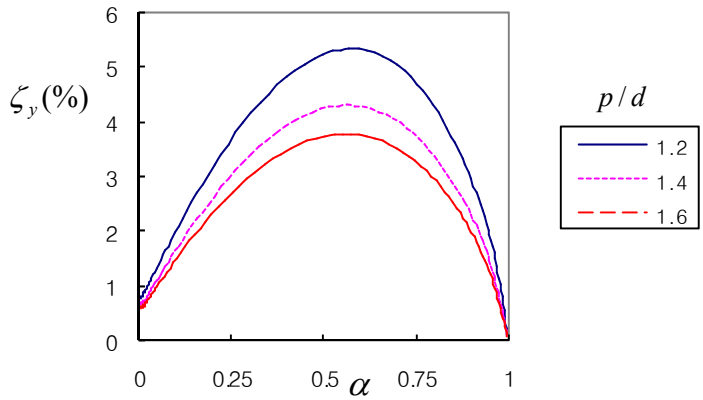

Figure 10. Total damping ratio for various pitch ratios, $G_p = 600kg/(m^2 \cdot s), d = 13mm, m_t = 0.33kg/m, f_n = 29, NS$

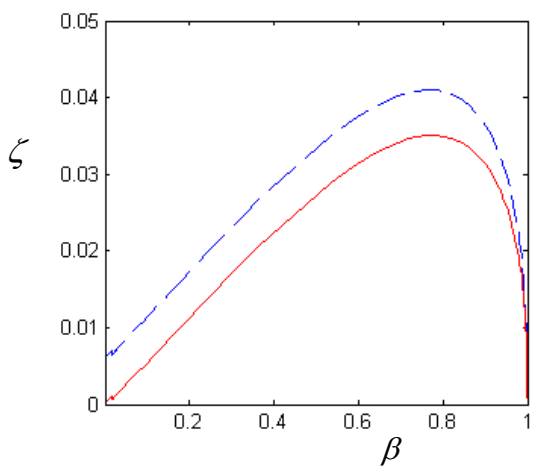

Figure 11. Typical results of total damping (---) and two-phase damping (___) ratios, based on Marchaterre's model; $G_p = 600kg/(m^2 \cdot s), p/d = 1.47,$ $d = 13mm, m_t = 0.33kg/m, f_n = 29, NS$

increase (see eq. (21) and Fig. 4). But, the damping ratio is strongly influenced by the pitch ratio, *p/d*, as shown in Fig. 10. The effect of the pitch ratio on the damping ratio could be mainly influenced by the Euler number. Typical results of two-phase damping and total damping ratios, based on Marchaterre's model for air-water mixture, are shown in Fig. 11. The difference between the ratios denotes the viscous damping ratio given by eq. (25). As expected, the damping ratio

Table 2. Summary of experimental conditions and tube array data

Author	Array Type	p/d	Tube dia. (mm)	Tube mass kg/m	Fluid	Pressure (Mpa)
Axisa et al.(1985)	*RT*	1.44	19.1	0.49	S-W	2.5
Nakamura et al.(2002)	*NS*	1.46	22.23	0.96	S-W	5.8
Janzen et al.(2001)	*RT*	1.5	12.7	0.27	R134	0.74
Pettigrew et al(1997)	*NT*	1.5	12.7	0.27	R22	1.2
Pettigrew et al (1989)	*RT*	1.47	13,	0.33	A-W	0.1

* * A-W;Air-water, S-W; Steam-water, R;Freon,

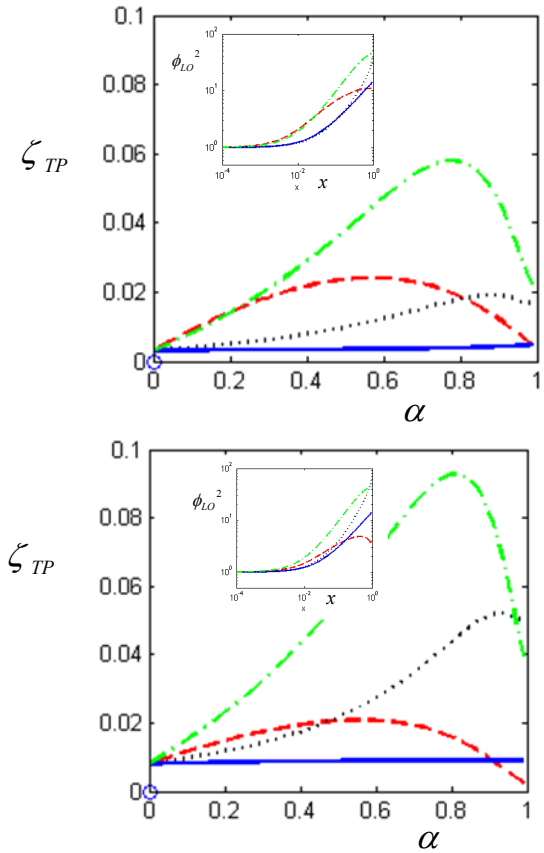

Figure 12. Two-phase damping ratios for Freon 134 mixture, based on Martinelli-Nelson; _.._., Levy;…..,, Marchaterre; ----, homogeneous; ___; (a) $G_p = 500kg/(m^2 \cdot s)$, (b) $G_p = 1000kg/(m^2 \cdot s)$ and $p/d = 1.47,\ d = 12.7mm,$ $m_t = 0.27kg/m, f_n = 38.8Hz, 0.74MPa, RT$

has a maximum value at a certain void fraction, $\alpha \approx 60\%$, for air-water system. For Freon 134, the two-phase damping ratios are shown in Fig. 12. It is found that the homogeneous results are under-estimated and the damping ratio is more or less influenced by the pitch mass flux.

To evaluate the present model, the present results are compared with the available experimental results. The experimental parameters extracted from the references are outlined in Table 2. For various tube bundle configurations and systems (air-water, steam-water and Freon-22 &134.), the present analytical results are compared to the existing experimental results in Figures 13, 14, 15, 16 & 17. The two-phase damping ratios, given by experiments, were measured at half of critical velocity due to fluid-elastic instability. For the present results, the co-relation factor, $K_{TP} = 3$, was used, based on the previous empirical evaluation for drag coefficients. In Fig. 13, two-phase damping ratios are shown for air-water mixture with flow condition. To estimate the analytical results, the pitch mass flux at the critical flow velocity, which is available data from reference, was used. In general, the two-phase damping is not influenced by the mass flux for air-water mixture. The results, based on Marchaterre's model, is abruptly increased at $\beta \approx 90\%$, since the flow is

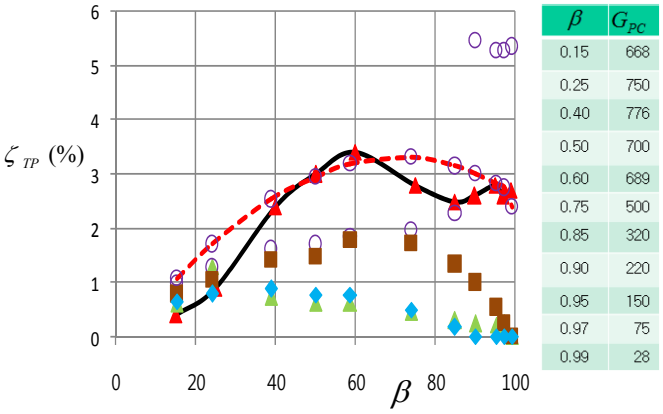

Figure 13. Comparison of the present results, based on Martinelli-Nelson; ▲ _._., Levy; ■ , Marchaterre; ○ , homogeneous; ◆ , with available experiments(Pettigrew et al, 1989);-▲ - and the previous result(Sim; 2007);-○ -for rotated triangular array.

changed from turbulent to laminar. It is shown that the previous results (Sim; 2007) have best agreement with the experimental results. But, the previous results were calculated using the $K_{TP} = 1.5$ and G_g. The gap pitch mass flux, G_g, was used for estimating void fraction with Feenstra's model. It is still question to apply them for the present model. At high void fraction, $\beta \geq 90\%$, the flow might be in intermittent flow regime or annular dispersed regime. Lian et al. suggest that damping is decreased as void increased in the intermittent flow regime. In a very similar test, Noghrehkar et al. and Pettigrew & Taylor (1997) found that intermittent flow resulted in maximum damping values. It is recommended that this flow regime be avoided in heat exchanger design.

For steam-water mixture with high void fraction, the present results are compared with existing experimental results (Axisa et al., 1985) and the previous results (Sim, 2007). The previous

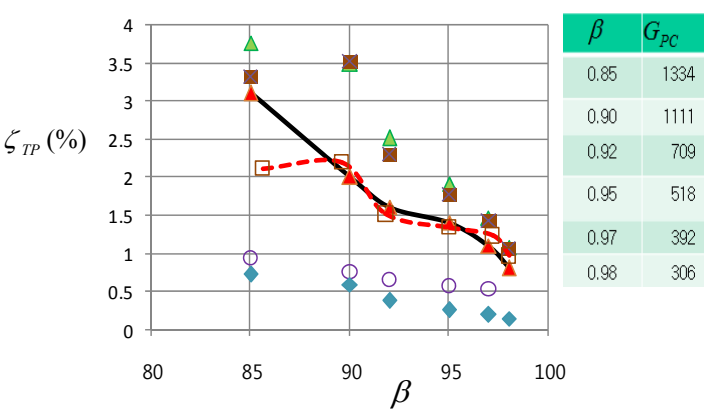

Figure 14. Comparison of the present results, based on Martinelli-Nelson; ▲ _._., Levy; ■ , Marchaterre; ○ , homogeneous; ◆ ;, with available experiments(Axisa et al, 1985);-▲ - and the previous result (Sim; 2007);-○ -for rotated triangular array.

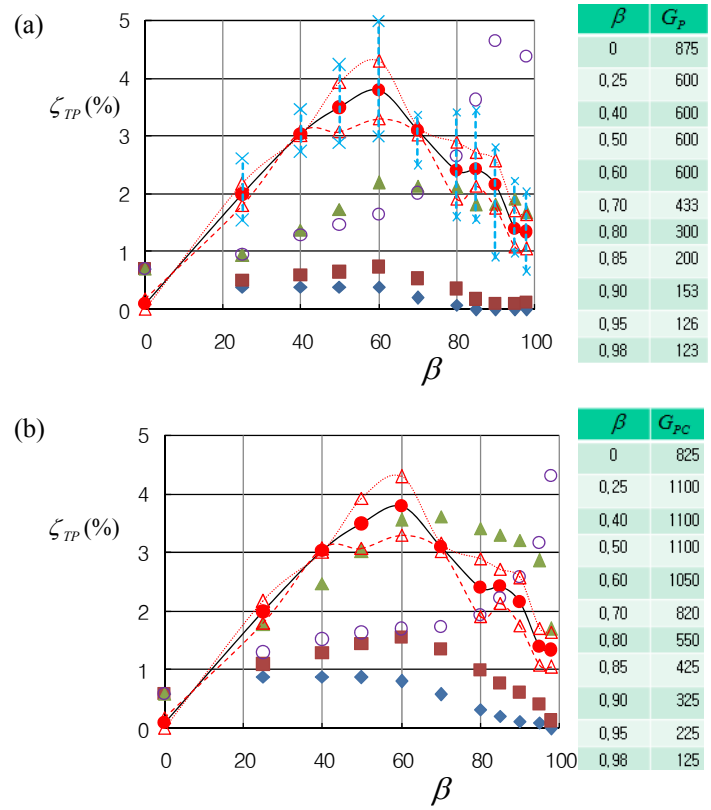

Figure 15. Comparison of the present results, based on Martinelli-Nelson; ▲ _._., Levy; ■ , Marchaterre; ○ , homogeneous; ◆ ;, with available experiments(Janzen et al, 2001);-● -(average) ..△ ..(drag) --△ --(lift), for rotated triangular array.

result was based on Levy's model with $K_{TP} = 1.5$. It is shown that for steam-water mixture, we can use the Levy's and Martinelli-Nelson's models for estimating the two-phase friction multiplier. For steam-water mixture, the values are under-estimated by the homogeneous and Marchaterre's models. For rotated triangular array, other experiments were performed by Janzen et al. (2001) in Freon 134 test loop. As shown in Fig 15, the experimental results are compared with the analytical results given by the present model. The analytical results are obtained using (a) the pitch mass fluxes where the damping ratios were measured and (b) the critical pitch mass fluxes where the system loses stability. In the figure, x--x denotes the range of the measured values. The measured values are scattered at relatively high void fraction. For Freon 134, the analytical results are influenced by pitch mass flux. The effect of mass flux is discussed in Fig. 12. It is shown that Martinelli-Nelson's model, to estimate two-phase friction multiplier, is better than others for Freon 134

Similar comparisons have been performed for normal square array and normal triangular array in Fig. 16 and Fig. 17 respectively. In Fig. 16, the results are given for steam-water mixture while in Fig. 17 for Freon 22. The previous result, in Fig. 16, was based on Levy's model with $K_{TP} = 2$. It is shown that Levy's and Martinelli-Nelson model could be applicable

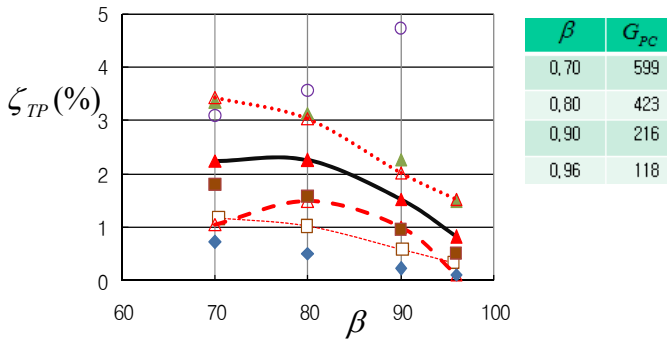

Figure 16. Comparison of the present results, based on Martinelli-Nelson; ▲ _._., Levy; ■ , Marchaterre; ○ , homogeneous; ◆ ;, with available experiments(Nakamura et al, 2002);-▲ -(average) ..△ ..(drag) --△ --(lift), and the previous result (Sim; 2007);..□ .. for normal square array.

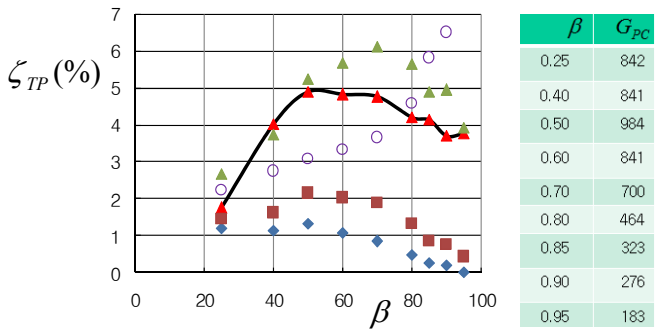

Figure 17. Comparison of the present results, based on Martinelli-Nelson; ▲ _._., Levy; ■ , Marchaterre; ○ , homogeneous; ◆ ;, with available experiments(Pettigrew et al, 1997);-▲ - for normal triangular array.

for steam-water mixture and Martinelli-Nelson's model for Freon 22 .

CONCLUSIONS

This paper outlines the development of a semi-analytical model to formulate damping of heat exchanger tube bundles in two-phase cross flow. Most of the available data on two-phase damping in tube bundles subjected to two-phase cross-flow have been reviewed. The formulation is based on information available in literature. The existing results of experiments on four tube bundles configurations were reviewed. The development of the present damping model stemmed from the two-phase multiplier of pressure loss and the momentum flux of the two-phase flow. The effects of several parameters such as flow velocity, void fraction, confinement, flow regime, fluid properties, two-phase multiplier of pressure loss and momentum flux of the two-phase flow are discussed. These parameters are taken into consideration in the formulation of a practical two-phase damping. For the present results of air water mixture, the co-relation factor, $K_{TP} = 3$, was used, based on the previous empirical evaluation for drag coefficients. But, still it is required to examine the value for steam-water mixture and Freon. The effect of the pitch ratio on damping ratio could be due to the Euler number. The two-phase damping ratio increases with decreasing the pitch ratio. It is also shown that the present approach, based on Levy's and Martinelli-Nelson model to estimate two-phase friction multiplier, could be applicable for steam-water mixture and Martinelli-Nelson's model for Freon 22 while the approach based on Marchaterre's model for air-water mixture. The analytical damping ratio is not influenced by pitch mass flux for air-water system but more or less influenced for steam-water mixture and Freon. The present results agree well with experimental damping ratios for a sufficiently wide range of pitch mass ratio, quality and *p/d* ratios using suitable co-relation ratio/factor, K_{TP} , between the Euler number and the hydraulic drag coefficients. It has shown this methodology to evaluate the two-phase damping ratio will be applicable for steam-water and Freons (22, 134). In order to improve the present model, it is required to get more information about two-phase friction multiplier, drag coefficient, co-relation factor and Euler number for steam-water mixture and Freon.

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REFERENCES

Axisa,F., Boheas, M.A. and Villard, b., 1985, "Vibration of Tube Bundles Subjected to Steam-Water Cross Flow: A Comparative Study of Square and Triangular Arrays", *Paper No. B1/2, 8th International Conference on Structural Mechanics in Reactor Technology*, Brussels

Carlucci, L.N., 1980, "Damping and Hydrodynamic Mass of a Cylinder in Simulated Two-phase Flow". *Journal of Mechanical Design*, **102**, pp597-602.

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

Feenstra, P.A., Judd, R.L. and Weaver, D.S., 1995, "Fluidelatic Instability in a Tube Array Subjected to Two-phase R-11 Cross Flow", *Journal of Fluids and Structure*,. **9** , pp. 747-771

Feenstra, P.A., Weaver, D.S. and Judd, R.L., 2000, "An Improved Void Fraction Model for Two-Phase Cross-Flow in Horizontal Tube Bundles", *International Journal of Multiphase Flow*, **26**, pp. 1851-1873

Gidi, A., Weaver, D.S. and Judd, R.L., 1997, "Two-phase Flow Induced Vibration of Tube Bundles with Tube Surface Boiling", *ASME Fluid-structure Interaction, Aeroelasticity, Flow-induced Vibration and Noise*, AD-Vol. **53-2**, pp.381-389

Hara, F. and Kohgo, O., 1986, "Numerical Approach to Added Mass and Damping of a Vibrating Circular Cylinder in a Two-phase Bubble Fluid", Proc. *ICCM(Tokyo)*, **2**, pp.255-260.

Heilker, W. J. and Vincent, R.Q., 1981, "Vibration in Nuclear Heat Exchanger Due to Liquid and Two-Phase Flow', *ASME Journal of Engineering for Power*, **103**, pp.358-365

Janzen, V.P., Pettigrew,C.E., Taylor,C.E. and Whan, T., 2001, "Vibration Behaviour of Rotated Triangular Tube Bundles in Two-phase Cross Flow", 2001 ASME Pressure Vessel and Piping Conference, Atlanta, USA

Janzen, V.P., Han, Y., Smith, B.A.W. and Fluit, S.M., 2005, "Vibration Damping of Stabilized Steam-generator Tubes",

2005 ASME Pressure Vessel and Piping Conference, Denver, USA, PVP2005-71666

Levy, S., 1960, "Steam Slip-Theoretical Prediction from Momentum Model", Trans. *ASME, series C, J. Heat Transfer*, **82**, pp. 113-124

Lockhart, R.W., Martinelli, R.C., 1949, 'Proposed Correlation of Data for Isothermal Two-phase two-component Flow in Pipes,' Chem. Eng. Prog. 45, pp.39-48.

Marchaterre, J. F., 1961, "Two-phase Frictional Pressure Drop Prediction from Levy's Momentum Model", *Trans. ASME, series C, J. Heat Transfe*r, **83(4)**

Martinelli, R. C. and Nelson, D. B., 1948, "Prediction of Pressure Drop during Forced Circulation Boiling of Water," *Transactions of ASME*, **70**, pp.695-702.

McAdams, W.H., Woods, W.K. and Herman, L.C., 1942, "Vaporization inside Horizontal Tubes-II-benzene-oil Mixtures, *Transactions of ASME*, **64**, pp.193-200.

Nakamura, T., Hirota, K., Watanabe, Y., Mureithi, N.W., Kusakabe, 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 Structure*, **16**, pp.123-136.

Nakamura, T., Yamaguchi, N. , Fujita, K. Sakata, K. and Saito, I., 1986a, "Study of Flow-Induced Vibration of a Tube Array by Two-Phase Flow (1st Report: Large Amplitude Vibration by Air-Water Flow)", *Trans. of Japanese Society of Mechanical Engineers*, **52(473 C)**

Nakamura, T., Yamaguchi, N. , Fujita, K. Sakata, K. and Saito, I., 1986b, "A Study of Flow-Induced Vibration of a Tube Array by Two-Phase Flow (2nd Report: Large Amplitude Vibration by Steam-Water Flow)", *Trans. of Japanese Society of Mechanical Engineers*, **52(473 C)**

Pettigrew, M.J. and Knowles, G.D,, 1992, "Some Aspects of Heat-Exchanger Tube Damping in Two-Phase Mixture", *Journal of Fluids and Structure*, **11**, pp.929-945

Pettigrew, M.J. and Taylor, C.E., 1997, "Damping of Heat exchanger tubes in Two-Phase", Proceedings of fourth International Symposium on FSI, AE & FIV+N, *ASME International Congress*, ASME Publication AD-Vol 53.2, Dallas, Texas, Nov. 16-21, pp. 407-418.

Pettigrew, M.J. and Taylor, C.E., 2003, "Vibration Analysis of Shell-and-tube Heat Exchangers; An Overview- Part 2: Vibration Response, Fretting-wear, Guidelines", *Journal of Fluids and Structure*, **18**, pp. 485-500.

Pettigrew, M.J., Taylor, C.E. and Kim, B.S., 1989a, "Vibration of Tube Bundles in Two Phase Cross Flow; Part 1 – Hydrodynamic Mass and Damping", *ASME Journal of Pressure Vessel Technology*, **111**, pp. 466-477.

Pettigrew, M.J., Tromp, J.H., Taylor, C.E. and Kim, B.S., 1989b, "Vibration of Tube Bundles in Two Phase Cross Flow; Part 2 – Fluid-Elastic Instability", *ASME Journal of Pressure Vessel Technology*, **111**, pp. 478-487

Rogers, R.J., Taylor, C. and Pettigrew, M.J., 1984, "Fluid Effects on Multispan Heat Exchanger Tube Vibration", *Proceeding of ASME Pressure Vessels and Piping Conference*, San Antonio, TX, June, ASME Publication Hoo316, Topics in Fluid Structure Interaction, pp.17-26.

Schrage, D.S., Hsu, J.T. and Jensen, M.K., 1988, "Two-Phase Pressure Drop in Vertical Cross Flow across a Horizontal Tube Bundle, *AIChE J*, **34**, pp.107-115

Sim, W.G., 2007, "An Approximate Damping Model for Two-Phase Cross-flow in Horizontal Tube Bundles", *2007 ASME Pressure Vessel and Piping Division Conference*, San Antonio, USA, PVP2007-26176.

Sim, W.G. and Mureithi, N.W., 2010, "Darg Coefficient and Two-phase Friction Multiplier on Tube Bundles Subjected to Two-phase Cross-flow", *2010 ASME Pressure Vessel and Piping Division Conference*, Bellevue, Washington, USA, PVP2010-25073.

Zukauskas, A., Ulinskas, R., Katinas, V. and Karni, J., 1988, " Fluid Dynamics and Flow-induced Vibrations of Tube Bank," *Hemisphere Publishing Corporation*, New York, pp. 97-114