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AN EXPERIMENTAL STUDY ON THE VIBRATION INDUCED BY A TWO-PHASE AIR - WATER CROSSFLOW IN A NORMAL TRIANGULAR TUBE BUNDLE

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Abstract. *Flow-Induced Vibration (FIV) is probably the most critical dynamic issue in the design of shell and tubes heat exchangers. This fluid - structure phenomenon may generate high amplitude vibration of tubes or structural parts, which leads to fretting wear between the tubes and supports, noise or even fatigue failure of internal components. Many test benches have been constructed to study this phenomenon, however, some vibration mechanisms, mostly those related to multiphase flow, are not yet fully understood. Therefore, in this work, an experimental study on the vibration induced by a two-phase air - water vertical upward crossflow in a tube bundle is presented. For this purpose, a 19mm OD stainless steel tube was flexibly mounted as a cantilever-beam in a test section. The dynamic response of the tube was measured by using piezoelectric microaccelerometers installed at its free-end. The surrounding rigid tubes were installed in order to complete a normal triangular configuration with pitch-to-diameter ratio of 1.26. This paper presents tests in air and water single-phase flows aiming at addressing the resonance frequencies of the dynamic structure. Also, damping was calculated by using the half-power bandwidth method. The influence of flow velocity on vibration amplitudes and frequencies was analyzed. Finally, tests for two-phase air - water flow were carried out; the influence of void fraction on vibration amplitude, resonance frequencies and damping was checked; the results were compared with the data available in the open literature.*

Keywords: *cantilever-beam, flow - induced vibration, multiphase flow, void fraction, resonance frequency*

1. INTRODUCTION

Shell-and-tube heat exchangers are still one of the most important heat exchanger type used in industrial processes (Kanizawa *et al.*, 2012). These devices are also used as steam generators in PWR nuclear power plants, which correspond to a critical operational condition since a mechanical failure could lead to economic losses due to unexpected shut downs, as well as critical radioactive accidents. Efficient designs of shell-and-tube heat exchangers often require high fluid velocities on the shell side while an optimum design would necessitate as much reduction in the structural support as possible (Noghrehkar *et al.*, 1999), thus favoring flow-induced vibration (FIV) in tube bundles of the heat exchanger, being the most important issue at the design stage.

Pettigrew and Taylor (2003) indicate that, for these heat exchangers, forces generated by flow in the axial direction are negligible in comparison with cross flow condition. Based on the revision presented by Taylor and Pettigrew (2001) as well as in the literature in general, it is accepted that the most important FIV mechanisms during single-phase flow are: (i) fluidelastic instability, (ii) vortex shedding, (iii) random vibration and (iv) acoustic resonance. According to Noghrehkar *et al.* (1999), almost half of shell-and-tube heat exchangers in the industry operate under two-phase flow condition in the shell side, which also lead to FIV. Nonetheless, the number of research and publication about FIV during two-phase flows is reduced (Pettigrew *et al.*, 2001), which is not surprising since single-phase FIV, a simpler topic, is not yet fully understood.

Vibration during two-phase flow depends on flow patterns, i.e. gas and liquid phase distributions, which in turn depend on void fraction and mass flux, even though the available investigations in the literature point out that flow patterns is a complex issue to take into account. Actually, as mentioned in Taylor and Pettigrew (2001), the available thermalhydraulic models fail to explain some deviations of dynamic parameters measurements that must be related to flow pattern transitions. As pointed out in Khushnood *et al.* (2012), it is speculated that the same excitation mechanisms of single-phase flow are present during two-phase flow conditions. However, recently another mechanism seemed particularly significant: quasi-periodic forces (Zhang *et al.*, 2007; Perrot *et al.*, 2011). It can be seen that more experimental studies are required in order to understand FIV during two-phase flow.

In order to study those missing details on FIV characteristics, a test bench was designed and constructed. As a starting point, in this work measurements of dynamic parameters such as: resonance frequency, hydrodynamic mass and damping ratio are presented. These results are compared with experimental data available in the open literature, to confirm that the designed experimental facility is capable of performing reliable experiments for FIV investigation during two-phase flows. The experimental apparatus description, experimental results and conclusions are detailed in the foregoing sections.

2. EXPERIMENTAL APPARATUS

The experimental facility used for FIV and dynamic parameters evaluation consists of a tube bundle counting with injection and conditioning sections, a water loop and an air compression and conditioning system. The water loop comprises a reservoir, a 15 CV centrifugal pump, electromagnetic volumetric flow transducers, a heat exchanger to control water temperature, and the water injector that consists of a perforated tube. Air is compressed by a 40 HP rotary screw compressor, and passes sequentially by a heat exchanger, reservoir, regulating pressure valve, turbine volumetric flow meters, needle and globe valves and is injected by 7 membrane injectors. The water is injected in the bottom region of the test section, followed by the injection of air. The two-phase mixture passes by a static mixer just upstream the test section. Downstream the test section, the flow is directed to a cyclone type separator, from which the water is directed to the reservoir and the air to laboratory outside.

The test section consists of a triangular tube bundle, counting with 19 mm (3/4") OD, 381 mm long stainless steel tubes, with transversal pitch-to-diameter (P/d) ratio equal to 1.26. The tubes are distributed in 20 rows, where even rows have 4 tubes and odd have 3 tubes plus two half tubes positioned in the section wall to avoid bypass flow. Thus, the experiments are performed for upward crossflow condition. All tubes are rigidly installed, except the instrumented one which is 378 mm long. This tube is cantilevered mounted on a wall of the test section and it is located at the center of the 13th row, where it is expected that the two-phase flow is fully developed.

The dynamic response of the tube is measured by using piezoelectric uniaxial microaccelerometers model 352A24, from PCB Piezotronics. They are installed inside the tube and close to the free end. Data is acquired with a LMS SCADAS Mobile system running LMS Test.Lab. This set-up is presented in Fig.1.

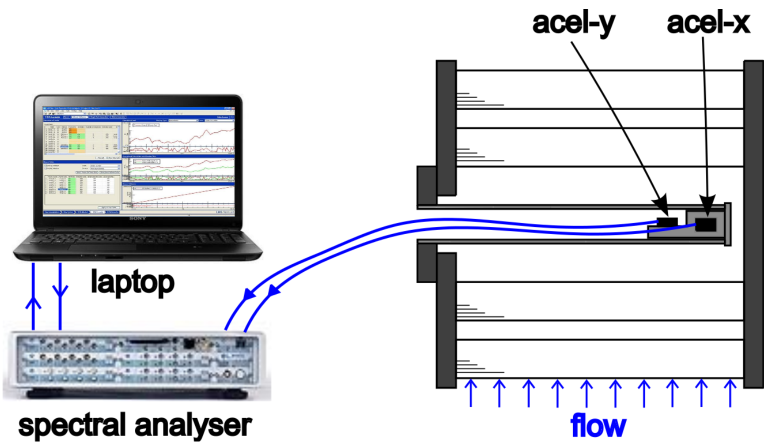


Figure 1: Set-up scheme for measuring tube response.

3. EXPERIMENTAL RESULTS

The dynamic response of a vibrating tube in a heat exchanger (as well as for every mechanical system) depends on its inertia, stiffness and energy dissipaters (damping). System's inertia and damping during fluid flow are different from that measured in air since hydrodynamic mass and damping are known to be dependent on fluid properties as well as functions of component geometry and adjacent boundaries, whether rigid or elastic (Khushnood *et al.*, 2012). In addition, hydrodynamic mass and damping depend on void fraction when the tube is submerged in two-phase flow.

Tests were performed in order to verify the effect of flow velocity and void fraction on the vibration level of the tube and dynamic parameters such as resonance frequency, hydrodynamic mass and total damping ratio. Figure 2 depicts the experimental conditions desired and also the Ulbrich and Mewes (1994) flow pattern map. According to this figure, it is expected to perform experiments during bubbly and intermittent flow patterns. The desired experimental conditions were defined for fixed void fractions values, which are given by the homogeneous model, and for the range of feasible superficial velocities. As pointed in Taylor and Pettigrew (2001), none of the slip models allowed a better collapse of these data. The velocities were limited by the measuring range of the volumetric flow transducers and by the pump and compressor capacities. In the next subsections, each dynamic aspect is described in detail. Sections 3.1 to 3.3 deal directly with dynamic parameters as resonance frequency, hydrodynamic mass and damping ratio. Finally, the tube response during two-phase flow is analyzed in section 3.4.

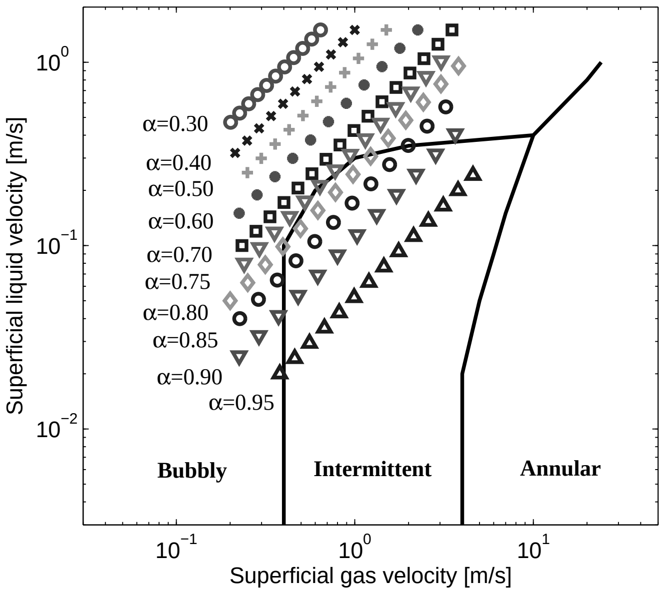


Figure 2: Test conditions during two-phase flow.

3.1 Resonance frequency

Tests have been performed with different flow velocities of water and air as single-phase flow, and void fractions from 30% to 95% during two-phase flow. These experiments were carried out at room temperature ($25°C$) and pressure close to 95 kPa. The dynamic response was measured using a sampling frequency of 1024 Hz with 8192 spectral lines and results are obtained by linear averaging of 40 windows with zero overlap. Power Spectral Densities (PSD) of acceleration responses were calculated, thus the resonance frequency of the tube for every experimental condition can be identified. As an example, the PSD measured at 30% void fraction is presented in Fig.3. In general, for every tested condition, there is just one resonance frequency in the range of interest (0 - 200 Hz) bandwidth, which matches with the behavior predicted by a previously performed finite-element dynamic model of a cantilever beam.

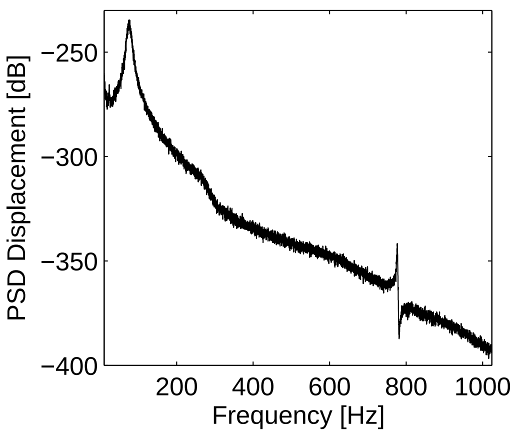
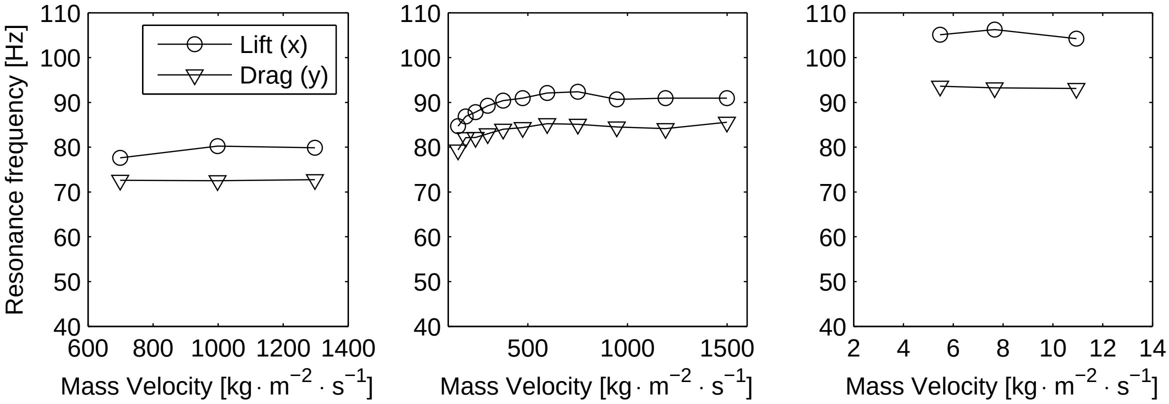


Figure 3: PSD measured during two-phase flow $\alpha = 30\%$.



(a) (b) (c)

Figure 4: Tube's resonance frequency a) $\alpha = 0\%$ b) $\alpha = 60\%$ b) $\alpha = 100\%$.

As expected, it was found that the resonance frequency of the tube depends on the kind of flow in which it is immersed. As it can be seen from figs.4a and 4c, the resonance frequency of the tube measured during water flow is lower than that measured in air flow, which is explained by the hydrodynamic mass effect (see section 3.2). Further, resonance frequency does not seem to be significantly affected by mass flux under these conditions. On the other hand, the resonance frequency during two-phase flow varies within the limits set by those measured in air and water single-phase flow. It increases with void fraction due to hydrodynamic mass effect. Moreover, as it can be noticed from Fig.4b, the resonance frequency in two-phase flow seems to vary slightly during low mass fluxes, however, it stabilizes for some ranges in which it can be assumed as constant.

3.2 Hydrodynamic mass

Hydrodynamic mass (also called as added mass) is defined as the equivalent external mass of fluid vibrating with the tube (Pettigrew *et al.*, 1989), thus modifying its dynamic behavior. Carlucci and Brown (1983) measured hydrodynamic mass, m_h, in a bundle of cylinders subjected to axial two-phase flow, they found that it is related to the resonance frequency of the tube, f, in two-phase mixture, and is given by:

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

where m_t is the mass of tube alone per unit length, f_g is the resonance frequency of the tube in air. Additionally, Khushnood *et al.* (2012) noted that the tube frequency must be measured at mass fluxes sufficiently below fluidelastic instability, otherwise significant shifts in tube's dynamic response are observed. In this paper, hydrodynamic mass is measured at velocities roughly half the flow velocity for fluidelastic instability assuming a Connors' constant of 2.8 and the exponent of the mass-damping parameter equal to 0.5.

Experimental measurements can be compared against the theoretical model presented in Rogers *et al.* (1984) for liquid flow. This theory is based on an equivalent diameter, D_e, to model the confinement due to surrounding tubes. Thus, hydrodynamic mass may be formulated for two-phase flow by:

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

where d is the tube diameter and ρ is the fluid density, which, according to the references, can be calculated by assuming the homogeneous model. For a tube inside a triangular tube bundle, D_e is taken as:

$$\frac{D_e}{d} = \left(0.96 + 0.5 \frac{P}{d} \right) \frac{P}{d} \tag{3}$$

Experimental hydrodynamic mass can be calculated by replacing the measured resonance frequencies in Eq.1. Results are compared against the theoretical model as shown in Fig.5. As it can be seen, experimental hydrodynamic mass follows the tendency predicted by Rogers *et al.* (1984), which indicates reduction of added mass with void fraction. However, hydrodynamic mass is higher than expected for $\alpha > 80\%$. According to Pettigrew *et al.* (1989), for most tube bundles, the measured hydrodynamic mass is close to that calculated using the homogeneous density of two-phase mixture, except for void fraction values higher than 80%. This may be attributed to the fact that a great parcel of the experiments for these conditions ($\alpha > 80\%$) correspond to intermittent flow pattern. Alternatively, this result can be due to the higher slip between the phases at this condition, which would lead to higher liquid hold up and consequently to a higher added mass. It must be highlighted that the homogeneous model was adopted for the evaluation of void fraction, which corresponds to the maximum α possible. Additionally, hydrodynamic mass in drag and lift directions present a similar tendency, which agrees with results presented in Taylor and Pettigrew (2001) for Freon 22, air-water, steam-water and Freon 11.

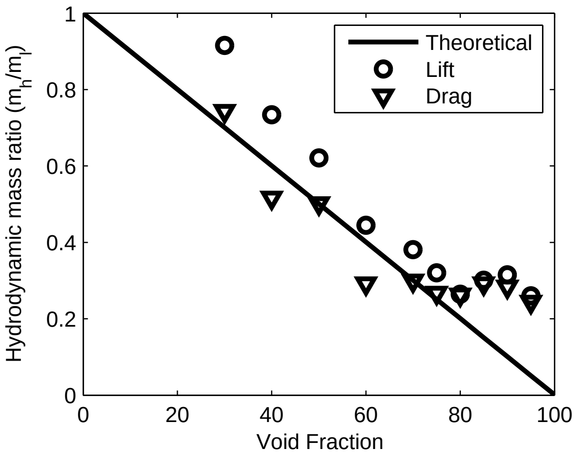


Figure 5: Hydrodynamic mass.

3.3 Damping ratio

In general, real systems dissipate energy, while vibrating, by several different mechanisms. The dissipative process is therefore the simultaneous result of all those mechanisms, and is difficult to identify and to model accurately (Maia and Silva, 1998). There are several forms of damping in tube bundles subjected to two-phase flows. Generally, total damping is equal to the sum of structural damping, ζ_S, viscous damping, ζ_V, and a two-phase damping component ζ_{TP}. Therefore, if ζ_S and ζ_V are known, then ζ_{TP} can be estimated for each void fraction.

Structural damping includes damping generated by material losses, support frictional losses and sealing losses. This kind of damping is measured in air and it is expected to remain invariant during all void fractions. In the present study, ζ_S was found to be less than 4.7 percent, which is higher than values reported in the open literature. This must be related to the characteristics of the support components, principally with those used to avoid leakage from the test section. On the other hand, ζ_V in two-phase mixtures is taken to be analogous to viscous damping in single-phase fluids, as introduced in Pettigrew *et al.* (1986):

$$\zeta_V = \frac{100\pi}{\sqrt{8}} \left(\frac{\rho_{TP} d^2}{m}\right) \left(\frac{2\nu_{TP}}{\pi f d^2}\right)^{0.5} \left\{\frac{\left[1+(d/D_e)^3\right]}{\left[1-(d/D_e)^2\right]^2}\right\}, \tag{4}$$

where ζ_V is the viscous damping ratio in percent, ρ_{TP} is the homogeneous density of the two-phase viscosity, m is the tube mass per unit length including the hydrodynamic mass, f is the tube resonance frequency in two-phase mixture and ν_{TP} is the kinematic two-phase viscosity evaluated according the McAdams model (Collier and Thome, 1994).

In this work, the total damping ratio, ζ, was measured at the same mass velocities as for hydrodynamic mass analysis. It was used the random vibration method in order to estimate damping values during two-phase crossflow. It was considered that two-phase flow delivers a random dynamic force to the tube as a white noise. Thus, the measured PSDs will appear similar to an FRF, and PSD amplitudes at resonance and half-power frequencies have the same ratio as in a FRF. For recognizing half-power points it was necessary to filter the PSDs by using a moving average with seven elements (Álvarez *et al.*, 2015). Once the total damping ratio is measured, and its viscous and structural components are calculated, then ζ_{TP} can be estimated. The arithmetic mean value of the lift and drag damping components is calculated, and the variation of this parameter with void fraction is depicted in Fig.6a. Furthermore, the resulting two-phase damping component can be normalized to take into account the effect of confinement and tube mass by using the confinement factor, $C(d/D_e)$, and the mass ratio R, respectively.

$$\zeta_{TPn} = \zeta_{TP} \left(\frac{\left[1+(d/D_e)^3\right]}{\left[1-(d/D_e)^2\right]^2}\right)^{-1} \left(\frac{\rho_l d^2}{m}\right)^{-1} \tag{5}$$

According to Fig.6a, two-phase damping ratio varies significantly with void fraction, similar to the results found in the open literature. Despite damping was not measured at low void fractions, it is possible to note a maximum at $\alpha = 50\%$. Then, damping ratio decrease until $\alpha = 85\%$, thenceforth it is almost constant. A similar damping variation was reported in Pettigrew *et al.* (1989) and Pettigrew *et al.* (2001) for normal triangular tube bundles of $P/d = 1.47$ and 1.22, respectively, during air-water mixtures.

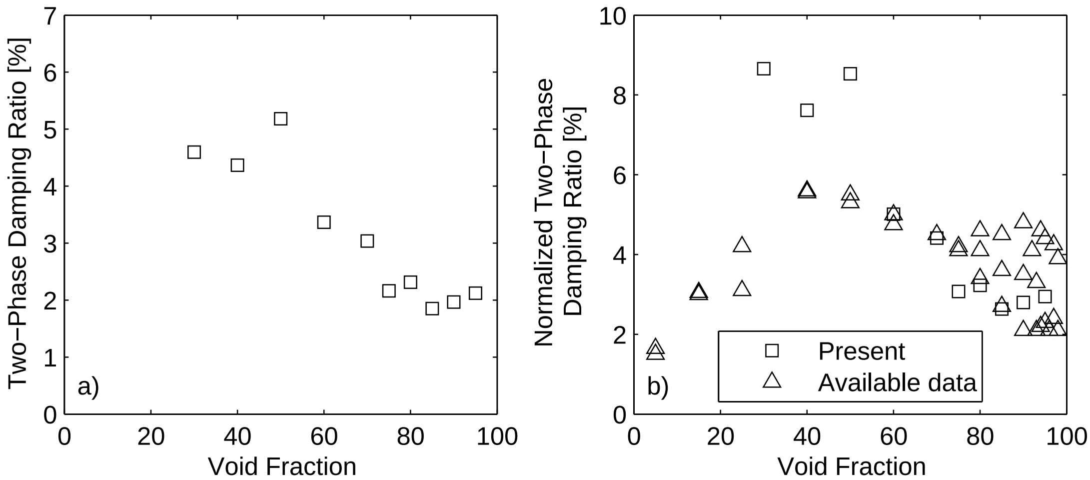


Figure 6: a)Two-Phase Damping Ratio, b) Normalized Two-Phase Damping Ratio in comparison against data available in open literature for normal triangular tube bundles during air-water mixtures (Pettigrew and Taylor, 2004).

The particular pattern of data plotted in Fig.6a suggests the damping ratio dependence on void fraction. However, according to Lian *et al.* (1997) and Noghrehkar *et al.* (1999), it would be another mechanism that is affecting the damping behavior. Lian *et al.* (1997) showed that measured void fraction fluctuations have the same void fraction dependence as damping ratio, that is, the RMS amplitude of local void fraction fluctuations increases with increasing void fraction up to a void fraction of about 40% (local time average void fraction) in bubbly flow pattern, reaches a peak at $\alpha = 50\%$ and then decreases with void fraction thereafter in the intermittent and annular-dispersed droplet flow patterns. Furthermore, they suggest that in bubbly flow at low void fractions or in annular-dispersed droplet flow at high void fractions, the temporal changes in fluid momentum are small and thus, the fluid force and energy transfer associated with these flow patterns would be relatively small. Differently, at intermediate void fractions ($40\% < \alpha < 60\%$), intermittent flow pattern appears, thus fluctuations in the fluid momentum would be large as indicated by the larger values of RMS amplitude of void fraction, and as a result, the fluid force and energy transfer or damping ratio would be at maximum.

In this work, the damping ratio peaks at $\alpha = 50\%$ during dispersed bubble pattern. This flow pattern was predicted by an extensive experimental study performed by Kanizawa (2014) in the same experimental facility. On the other hand, according to Ulbrich and Mewes' map (see Fig.2), this flow pattern is predicted as bubbly flow. As it can be noticed, the present maximum two-phase damping appears in the expected void fraction range ($40\% < \alpha < 60\%$), but differently from pointed by Lian *et al.* (1997), it did not occur during intermittent flow pattern. This fact suggests that, for this test section, somehow void fraction fluctuation during dispersed bubble pattern may be similar to that measured during intermittent flow pattern in Lian *et al.* (1997) and Noghrehkar *et al.* (1999). Anyway, it can be seen that more experimental studies are required in order to understand the influence of flow pattern on two-phase damping. Moreover, as it can be noticed in Fig.6b, ζ_{TPn} becomes larger especially at low void fractions. Actually, normalized two-phase damping at $\alpha = 30\%$ is even larger than that at $\alpha = 50\%$, which is related to the large hydrodynamic mass measured at that void fraction. Despite of this, normalized two-phase damping until $\alpha = 50\%$ is noticeably higher than those at $\alpha \geq 60\%$.

3.4 Tube response during two-phase flow

In order of evaluating the influence of void fraction on the vibration level, the RMS values of tube displacement for each α values was computed in the 10 - 1024 Hz bandwidth for each flow velocity, and are presented in Fig.7. The components within the 0 - 10 Hz bandwidth were not considered since in this band appears a resonant peak that corresponds to a vibration mode of the whole structure used for supporting the test bench. The Connors' model was used for estimating the instability threshold in the present tube bundle. All the tests were performed below fluidelastic instability, thus it is expected that a random vibration mechanism governs the oscillation of the tube.

According to Fig.7, FIVs during two-phase flow correspond to higher amplitudes in comparison with water single-phase flow for the same interval of mass velocities which confirms the dynamic structure sensibility to multiphase flow excitation. Furthermore, an important remark is that, for all void fractions, the displacement can be assumed roughly as

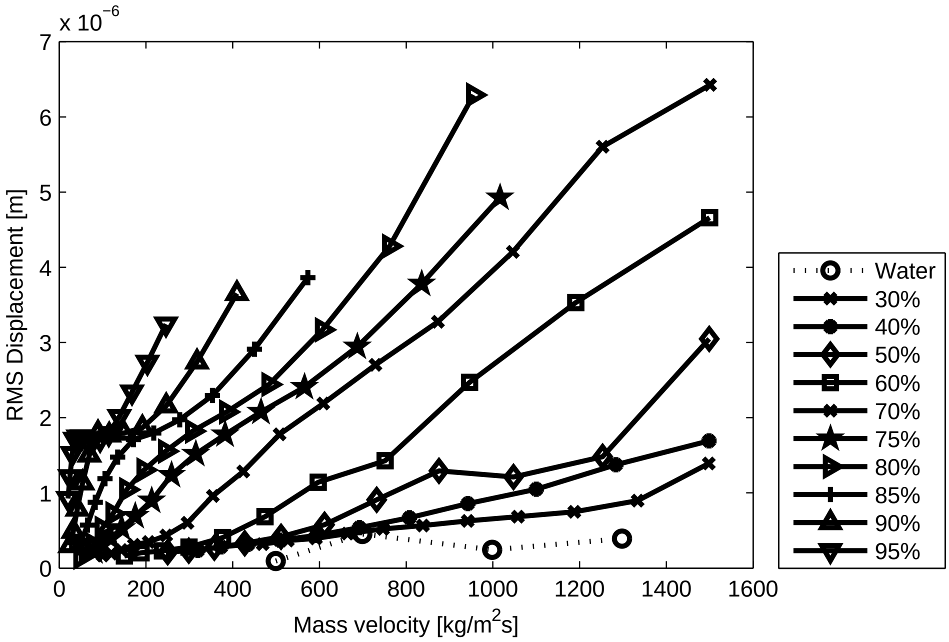


Figure 7: Two-phase flow damping component.

a linear function of mass velocity. According to Taylor *et al.* (2001), this is a typical dynamic response of a tube excited by two-phase flow during random vibration mechanism. Also, it can be noticed from Fig.7 that the slope of the curve increases with void fraction. This variation is more pronounced from 60 to 75% void fraction, passing through 70% as a transition zone. After this void fraction, slopes are considerably higher if compared with those at low void fractions. This effect must be related with a change in flow pattern. Actually, the Ulbrich and Mewes' map predicts a transition from bubbly to intermittent flow in this range of void fractions, which may explain the rising slope variation.

Also, the flow pattern identification performed in Kanizawa (2014) predicts that most of tested conditions up to 60% void fraction correspond to bubbly flow and higher void fractions may generate bubbly, churn and intermittent flow pattern. This agrees with predicted by Ulbrich and Mewes and suggests that churn and intermittent flow patterns generate large excitation forces. This criterion was also drawn in Taylor and Pettigrew (2001), thus the authors recommended the avoidance of intermittent flow patterns for heat exchangers operation.

4. CONCLUSIONS

Flow-induced vibration during two-phase crossflow in a normal triangular tube bundle was studied in the present work. It was used air-water mixtures to simulate water-steam systems, which are commonly present in heat exchanger equipment. Regarding to dynamic parameters, the tube's resonance frequency does not vary significantly with mass velocity. Furthermore, resonance frequency clearly depends on void fraction, which was expected due to hydrodynamic mass effect. Hydrodynamic mass measurements agree with those predicted by the theoretical model. However, during high void fractions ($\alpha > 80\%$), the measured values are higher than theoretical. These results show that, as indicated in the open literature, the homogenous void fraction model is not appropriate for condition of high void fractions values, and portions of liquid can be present around the tubes. Two-phase damping ratio agrees with those presented by references. However, more experimental studies are required to understand the influence of flow pattern on two-phase damping. The vibration induced by two-phase flow is higher than that induced by single-phase flow; additionally, it was found that the relationship between the displacement RMS and mass velocity is approximately linear, in which the slope increases with void fraction. It is speculated that the variation of the slope is due to flow patterns transition, since higher slopes are verified for conditions characteristic of intermittent and churn flow patterns. In summary, measurements of dynamic parameters and dynamic response agree with data presented by references. Therefore, the present test section with an instrumented tube mounted in cantilever can be used for future experimental studies of flow-induced vibration problems.

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