Proceedings of the ASME 2010 3rd Joint US-European Fluids Engineering Summer Meeting and 8th International Conference on Nanochannels, Microchannels, and Minichannels
FEDSM-ICNMM2010
August 1-5, 2010, Montreal, Canada

FEDSM-ICNMM2010-30143

FLUIDELASTIC INSTABILITY AND PERIODIC FLUID FORCES IN A NORMAL TRIANGULAR TUBE BUNDLE SUBJECTED TO AIR-WATER FLOW

G. Ricciardi*
M. J. Pettigrew , **N. W. Mureithi**
BWC/AECL/NSERC Chair of Fluid-Structure Interaction,
Department of Mechanical Engineering,
Ecole Polytechnique,
Montréal, QC, Canada, H3C 3A7
guillaume.ricciardi@gmail.com

ABSTRACT

Two-phase flow in power plant steam generators can induce tube vibrations, which may cause fretting-wear and even fatigue cracks. It is therefore important to understand the relevant two-phase flow-induced vibration mechanisms. Fluidelastic instabilities in cross-flow are known to cause the most severe vibration response in the U-bend region of steam generators. This paper presents test results of the vibration of a normal triangular tube bundle subjected to air-water cross-flow. The test section presents 31 flexible tubes. The pitch-to-diameter ratio of the bundle is 1.5, and the tube diameter is 38 mm. Tubes were flexible in the lift direction. Seven tubes were instrumented with strain gauges to measure their displacements. A broad range of void fractions (from 10% to 90%) and fluid velocities (up to 13 m/s) were tested. Fluidelastic instabilities were observed for void fractions between 10% and 60%. Periodic fluid forces were also observed. The results are compared with those obtained with the rotated triangular tube bundle, showing that the normal triangular configuration is more stable than the rotated triangular configuration.

*Address all correspondence to this author.

INTRODUCTION

Steam generators are major components of a nuclear power plant. They work as heat exchangers between the primary loop and the secondary loop. For obvious safety reasons, fluid exchange between the two loops has to be avoided. Two-phase flow in steam generators can induce tube vibrations that may cause fretting-wear and even fatigue cracks.

Vibration due to fluidelastic instabilities in the U-bend region are known to cause the most severe vibration response [1]. The two-phase flow fluidelastic instability is a complex mechanism that depends on the array geometry (tube diameter, pitch, triangular or square configuration), and on the flow conditions (void fraction, velocity, density and flow direction). Data are available for steam-water flow [2], freon vapor-liquid [3] and air-water [5] two-phase flow. Most of the data is for a pitch-to-diameter ratio (P/D) around 1.5, and a tube diameter around 20 mm, because this is what exists in most nuclear power plant steam generators.

Steam-water experiments are very expensive, therefore it is easier to perform tests in freon vapor-liquid and even cheaper in air-water. Tests performed on a normal triangular tube bundle configuration subjected to freon vapor-liquid flow showed [3] a well defined fluidelastic instability threshold at each void fraction tested, from 25 % to 95 %. Price and Zhang [4] obtained fluidelastic instabilities in air-flow, in drag and lift directions, they also shown that an acoustic resonance could cause a flexible array to go unstable. No periodic fluid forces were observed in [3] and [4]. Steam-water flow is closer to freon vapor-liquid flow than to air-water flow, since first two are both single-component fluids, whereas the latter is a two-component fluid. However, considering homogeneous flows, air-water is suitable to simulate fluidelastic instabilities in steam-water flows. In the case of intermittent flows, the use of air-water flow is less relevant, but still less expensive. There are very few data available on air-water flow for a normal triangular tube bundle configuration, thus the present study is a contribution to these data.

We present flow-induced vibration results for a normal triangular tube bundle in air-water flow, with a pitch-to-diameter ratio P/D=1.5, and a tube diameter of D=38 mm, which is larger than what is usually tested. We used the same geometry parameters (D and P/D) as those of previous work with a rotated triangular configuration, which makes it possible to compare these two configurations. All tubes are preferentially flexible in the lift direction. Seven tubes are instrumented for vibration response measurement. Several void fractions were tested varying from 0 % to 90 %.

EXPERIMENTAL APPARATUS

Test loop

The air-water loop (Fig. 1) comprised a 2500 L tank, a 30 L/s variable speed pump, and an air supply system (500 scfm). The air and water were homogenized with an undulated plate designed mixer. A magnetic flow meter was used to measure the water flow rate Q_w, and an air flow meter gave standard air flow rate Q_a at atmospheric pressure P_a. Since the pressure in the test section was not necessarily equal to one atmosphere, the flow rate was corrected based on the pressure in the test section. Two pressure measurements P_u and P_d were taken down stream and upstream of the tube bundle. The pressure used to calculate the correct air flow rate Q_{ac} was the average of the two measured pressures,

$$Q_{ac} = \frac{2P_a}{P_u + P_d} Q_a. \tag{1}$$

The homogeneous void fraction ε was calculated from the water flow rate and the correct air flow rate :

$$\varepsilon = \frac{Q_{ac}}{Q_{ac} + Q_w}, \tag{2}$$

and the pitch velocity U_p is given by :

$$U_p = \frac{Q_{ac} + Q_w}{S} \frac{P}{P - D}, \tag{3}$$

were S is the free stream cross-sectional area.

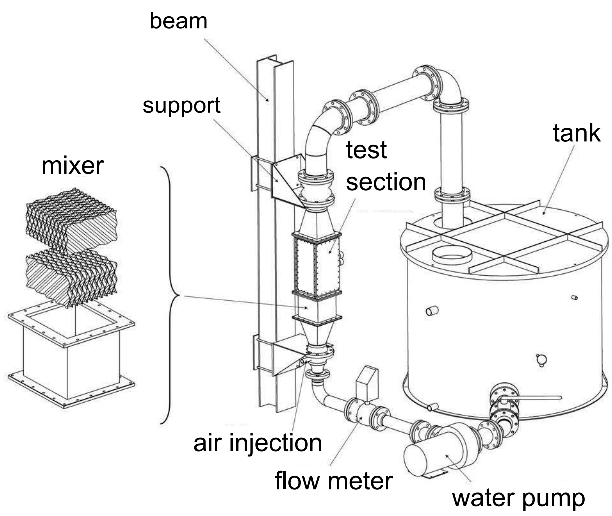

FIGURE 1. TEST LOOP.

Test section

The test section which has a rectangular cross section (229×191 mm), consisted of 31 flexible tubes of 38 mm diameter with a pitch-to-diameter ration of 1.5 (Fig. 2). The tubes were in a normal triangular configuration, with the flow direction perpendicular to the base of the triangle.

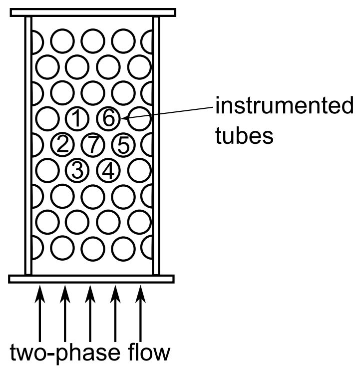

FIGURE 2. TEST SECTION.

The tubes were limited to move in one direction since they were supported with a flexible cantilever beam (Fig. 3) which was thin (4 mm) in one direction and thick (25 mm) in the other. The tube natural frequency in air was around 14.4 Hz in the thin

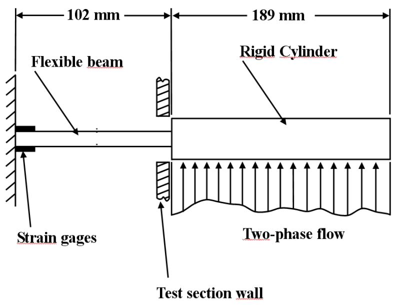

FIGURE 3. FLEXIBLE TUBE.

direction and 81 Hz in the thick direction. Seven tubes were instrumented with strain gauges, located on the flexible beams to measure tube displacements. The signals were numerically recorded with an OROS 32 dynamic data acquisition system. Fig. 2 shows the location of instrumented tubes.

Tests were performed with all tubes flexible in the lift direction, over a homogeneous void fraction range from 0 % to 90 %.

RESULTS

Fig. 4 presents the total response vibration amplitude of instrumented tubes for 10 % void fraction. A clearly defined instability occurs at 2.2 m/s with an abrupt increase in vibration amplitude beyond this flow velocity.

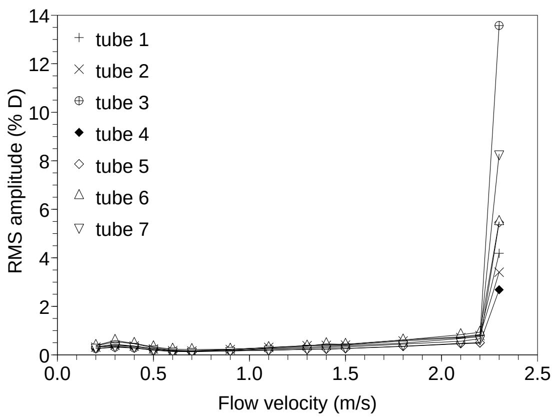

FIGURE 4. RMS VIBRATION AMPLITUDE VERSUS FLOW VELOCITY, FOR 10 % VOID FRACTION.

For 60 % void fraction (Fig. 5), instability occurs at 3.2 m/s, but the increase in amplitude is not as abrupt as it is for 10 % void fraction. We also observe a local maximum in amplitude below 3.2 m/s that may be due to periodic fluid forces as observed by Zhang et al. [6]. In this case the amplitude reaches a value of 5 % of the tube diameter at 2.4 m/s. This suggests that periodic fluid forces may have non negligible consequences in terms of fretting-wear damage.

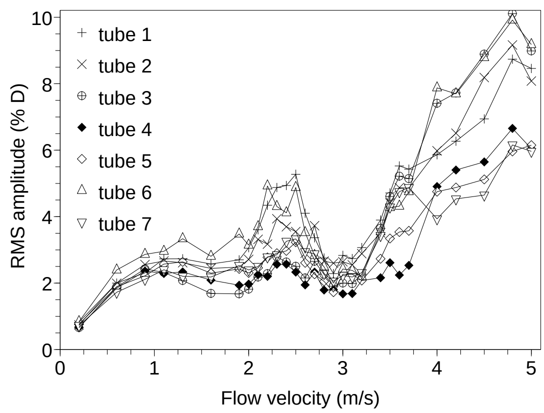

FIGURE 5. RMS VIBRATION AMPLITUDE VERSUS FLOW VELOCITY, FOR 60 % VOID FRACTION.

A waterfall plot of the response spectra of Tube 1 for 60 % void fraction is shown in Fig. 6. The spectra contain sharply defined vibration peaks. Normally such peaks would be expected at instability. However, in this case they appear at velocities well below the critical velocity at 3.2 m/s. This raises the question of whether what we interpreted as periodic fluid forces may not be a particular mechanism of fluidelastic instability.

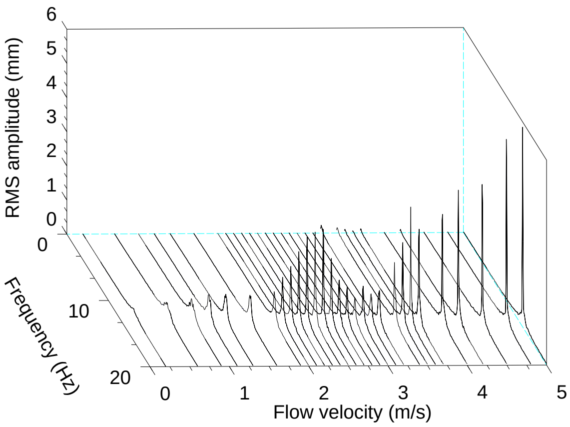

FIGURE 6. VIBRATION RESPONSE SPECTRA OF TUBE 1 VERSUS FREQUENCY AND FLOW VELOCITY, FOR 60 % VOID FRACTION.

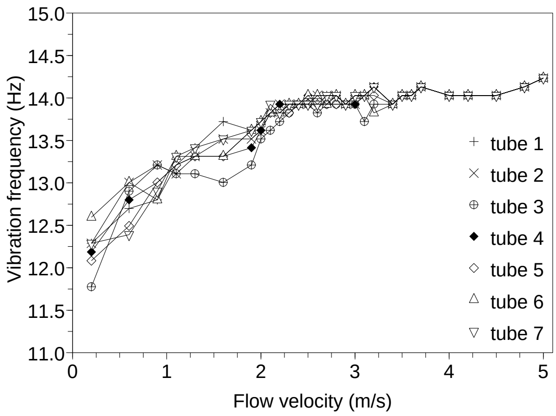

FIGURE 7. VIBRATION FREQUENCY OF THE TUBES VERSUS FLOW VELOCITY, FOR 60 % VOID FRACTION.

To answer the previous question let us consider the response frequency (Fig. 7) and the coherence between the tubes (Fig. 8) for 60 % void fraction. We observe three zones. In the first, between 0 and 1.9 m/s, the coherence is very low and the response frequencies of the tubes are very different. In the second zone, between 1.9 and 3.2 m/s, the frequencies are close but not strictly identical, whereas the coherences are still far from one. And finally in the third zone above 3.2 m/s the coherences are uniformly close to one, and the seven instrumented tubes oscillate at exactly the same frequency. The third zone obviously corresponds to the fluidelastic instability regime where all the tubes oscillate at the same frequency with a constant phase. This confirms our first critical velocity identification at 3.2 m/s. The second zone corresponds to the resonance of tubes, where each tube oscillates at almost the same frequency but independently from each other, as the coherence values testify. The resonance could be explained by a periodic fluid force of which the frequency would coincide with the natural frequency of the tube. On the other hand the comparison of tube displacements between the second and the third zones (Figs. 9 and 10) shows that the displacement is much more irregular in the second zone than in the third zone. The irregularity of the second zone may be due to a strongly pulsating intermittent flow that would strike the tubes with Dirac impulses causing the tubes to oscillate at their natural frequencies between successive "impacts". Nevertheless, these last assumptions need to be demonstrated by flow measurements to determine whether the principal oscillation frequencies correspond to fluid forces or not.

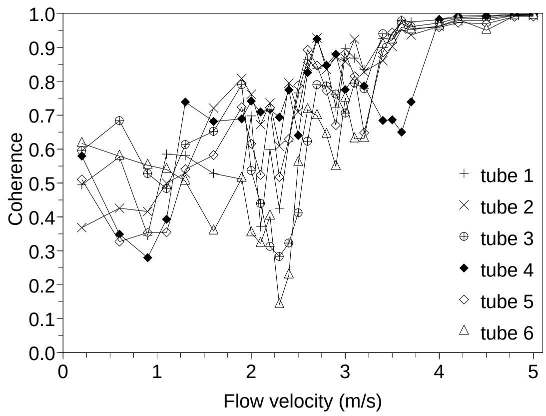

FIGURE 8. COHERENCE BETWEEN TUBE 7 DISPLACEMENT AND THE OTHER TUBES DISPLACEMENTS AT THE OSCILLATION FREQUENCY VERSUS FLOW VELOCITY, FOR 60 % VOID FRACTION.

A possible scenario may be the occurrence of intermittent fluidelastic instability, as observed for instance by Nakamura et al. [5]. In this case the average intertube coherence may remain low due to constant interruption of instability (Fig. 11). Note that according to the map Fig. 16, the intermittent flow regime is expected for $\varepsilon > 70\%$ particularly for high gas velocities. This is perhaps the simplest explanation.

The frequencies (Fig. 14) and the coherence (Fig. 15) for 90 % void fraction, show two zones. The first zone, from 0 to 5 m/s, is characterized by close vibration frequencies and low coherence values.

In the second zone, from 5 to 13 m/s, the frequencies di-

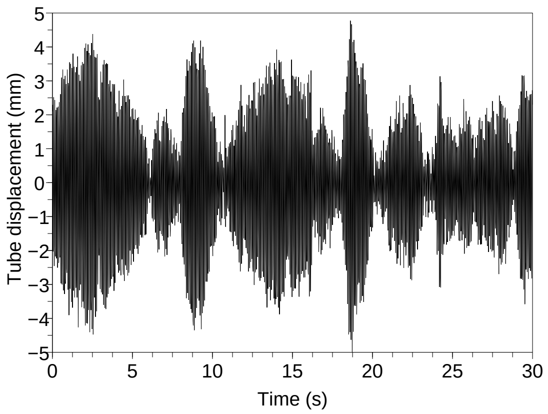

FIGURE 9. TUBE 1 DISPLACEMENT VERSUS TIME AT 2.4 M/S, FOR 60 % VOID FRACTION.

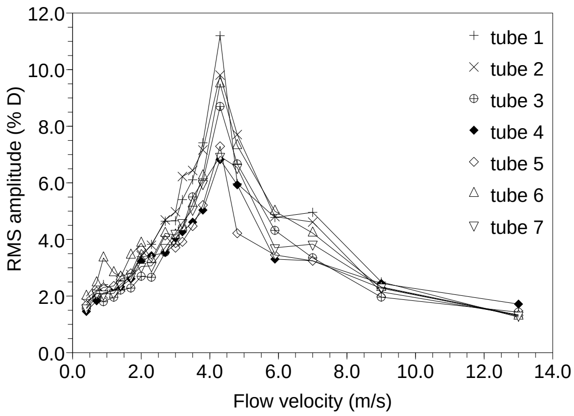

FIGURE 11. RMS VIBRATION AMPLITUDE VERSUS FLOW VELOCITY, FOR 90 % VOID FRACTION.

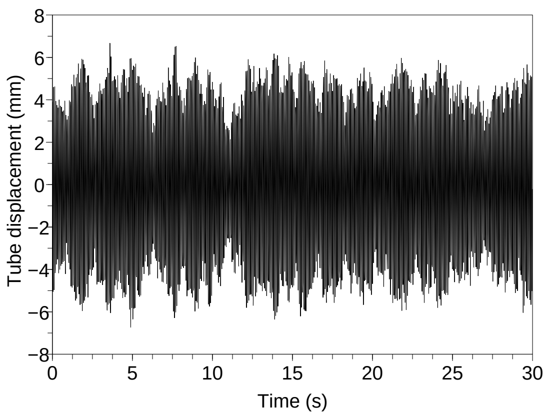

FIGURE 10. TUBE 1 DISPLACEMENT VERSUS TIME AT 5.0 M/S, FOR 60 % VOID FRACTION.

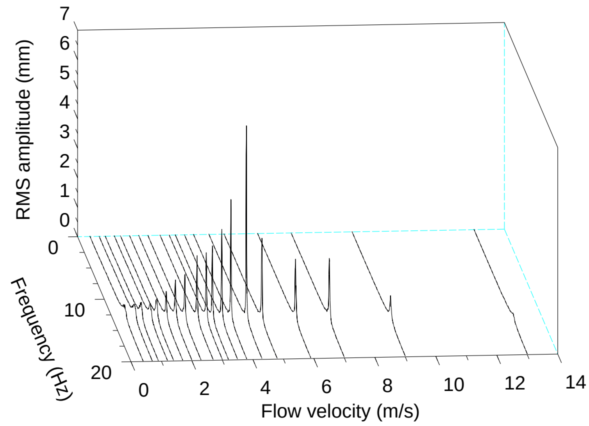

FIGURE 12. VIBRATION RESPONSE SPECTRA OF TUBE 1 VERSUS FREQUENCY AND FLOW VELOCITY, FOR 90 % VOID FRACTION.

verge and the coherences, interestingly, increase to reach values between 0.85 and 0.95. The first zone corresponds to high amplitude vibrations, and the second zone to low amplitude vibrations (Fig. 12). Fig. 16 presents a flow regime map, boundary separating spray, bubbly and intermittent flows were drawn based on experimental observations, but the flow regime does not change suddenly, therefore, points located close to the boundary, as shown in the figure, should be considered as intermittent flow. The high coherence value in the second zone is due to intermittent flow effects, because the seven instrumented tubes bundle scale is small compared with the flow structure, especially at high flow velocity. The observation of frequencies and coherences, allows us to conclude that there is no stiffness controlled instability.

The low coherence value leads us to conclude that the large amplitude responses observed for 60 % void fraction below 3.2 m/s and for 90 % void fraction are not due to a typical coupled stiffness controlled instability, but it could be due to a damping controlled instability, in which each tube is independent. In the latter case, the inter-tube coherence need to not be high since the 1 d.o.f instability is independent of inter-tube coupling. The decrease of amplitude could be due to a change of flow pattern. To confirm these explanations, the value of fluidelastic damping and the flow pattern need to be known. Nevertheless, the negative fluid damping is difficult to measure since one must measure fluid forces acting on a moving tube. This will be done in future work.

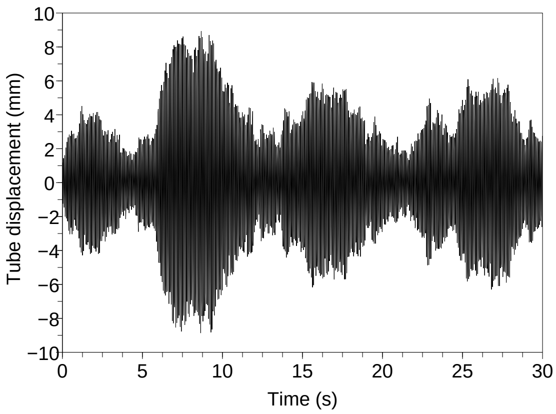

FIGURE 13. TUBE 1 DISPLACEMENT VERSUS TIME AT 3.8 M/S, FOR 90 % VOID FRACTION.

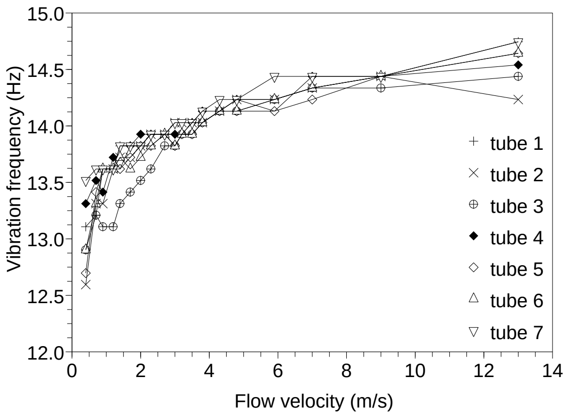

FIGURE 14. VIBRATION FREQUENCY OF THE TUBES VERSUS FLOW VELOCITY, FOR 90 % VOID FRACTION.

DISCUSSION

A comparison (Table 1) with the rotated triangular configuration results taken from previous work [7], shows that the normal triangular configuration is significantly more stable than the rotated one. This is an important result for engineers designing steam generators.

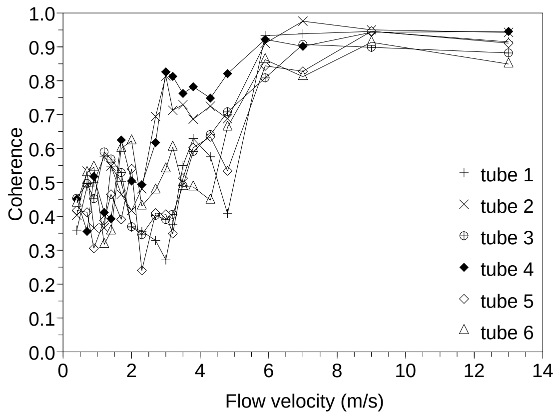

FIGURE 15. COHERENCE BETWEEN TUBE 7 DISPLACEMENT AND THE OTHER TUBES DISPLACEMENTS AT THE OSCILLATION FREQUENCY VERSUS FLOW VELOCITY, FOR 90 % VOID FRACTION.

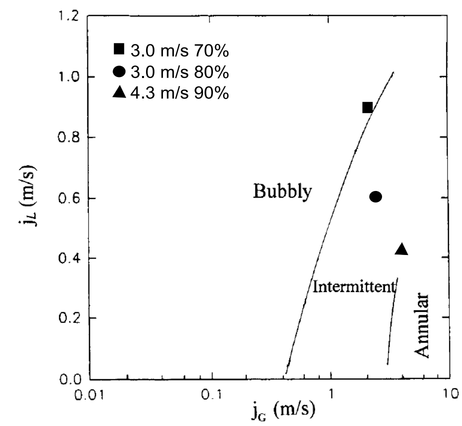

FIGURE 16. FLOW REGIME MAP FOR ROTATED TRIANGULAR TUBE BUNDLE IN TWO-PHASE CROSS FLOW, [10].

When designing a steam generator, engineers use the Connors relation to ensure that fluidelastic instability will not occur [8, 9] :

$$\frac{U_p}{fD} = K\sqrt{\frac{2\pi\zeta m}{\rho D^2}}, \tag{4}$$

where U_p is the pitch velocity, f is the tube natural frequency

TABLE 1. COMPARISON OF FLUIDELASTIC INSTABILITY CRITICAL VELOCITY IN THE NORMAL TRIANGULAR CONFIGURATION WITH THE ROTATED TRIANGULAR CONFIGURATION.

Void fraction (%)	U_{pc} normal (m/s)	U_{pc} rotated (m/s)
60	3.1	1.1
70	2.0*	1.2
80	1.8*	1.2
90	2.7*	1.3

$*$ inconclusive

in the two-phase mixture, D is the diameter of the tube, m is the mass per unit length including hydrodynamic mass, ρ is the homogeneous density of the two phase mixture, ζ is the total damping ratio and K is the fluidelastic instability factor. It is recommended [9] to take a value of $K = 3.0$ to avoid fluidelastic instability (Fig. 17) for all tube bundle with pitch-to-diameter ratio $P/D > 1.47$. We can see in Table 2 that this guideline is verified by the present results, except for the 90 % void fraction case which is inconclusive for now.

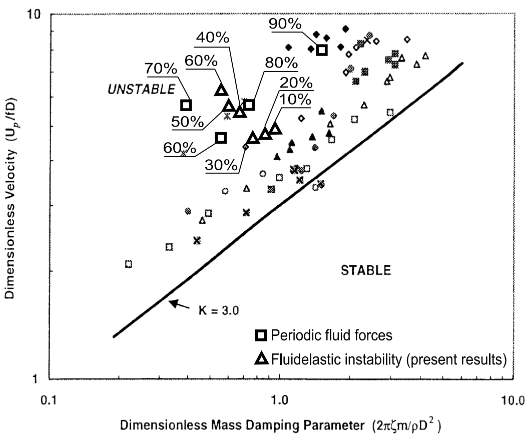

FIGURE 17. FLUIDELASTIC INSTABILITY DATA IN TWO-PHASE FLOW COMPARED TO DATA FROM PETTIGREW AND TAYLOR [9].

Periodic fluid forces have already been found in the rotated triangular configuration [6]. In the normal triangular case the vibration amplitude can reach up to 8 % of the tube diameter at relatively low fluid velocities. Such amplitudes could cause significant fretting-wear damage. Periodic fluid forces have not been observed on tube bundles with smaller diameters typically

TABLE 2. CONNORS EQUATION PARAMETERS.

ε (%)	U_{pc} (m/s)	f (Hz)	m (kg/m)	ζ (%)	ρ (kg/m^3)	$\frac{U_{pc}}{(fD)}$	$\frac{2\pi\zeta m}{\rho D^2}$	K
0	> 2.3	10.6	4.20	2.5	1000	$>$ 5.7	0.46	> 8.5
10	2.2	11.8	4.08	4.7	900	4.9	0.93	5.1
20	2.3	12.7	3.97	3.9	800	4.8	0.84	5.2
30	2.4	13.3	3.85	3.2	700	4.7	0.77	5.4
40	2.7	13.6	3.74	2.5	600	5.2	0.68	6.3
50	2.9	13.8	3.62	1.9	501	5.5	0.60	7.1
60	3.1	13.9	3.52	1.4	401	6.1	0.53	8.3
70	2.0*	14.1	3.40	0.8	301	3.7	0.39	5.9
80	1.8*	14.3	3.29	1.0	201	3.3	0.71	3.9
90	2.7*	14.4	3.18	1.0	101	4.9	1.37	4.2

$*$ inconclusive

used in steam generators [11–13]. Thus, the presence of these forces may be related to the larger spacing between the tubes. This needs to be confirmed by measuring the fluid forces on the present geometry and on smaller diameter tube bundles. It is important to clearly understand this phenomenon, and to state if we have to account for it in steam generator designs.

One may also wonder if these increases and decreases of amplitude were already observed in previous studies and incorrectly interpreted as fluidelastic instability. In many studies, once a certain amplitude is reached at a certain velocity, authors usually do not perform tests beyond this velocity, for many reasons such as avoiding damage to the experimental apparatus. Thus, perhaps they miss the decrease of amplitude we observed at high void fractions. Assuming this, we can wonder if there exists an instability beyond 13 m/s for 90 % void fraction (Fig. 11). But when designing a steam generator, beyond 13 m/s flow rate some other problems due to important drag forces may occur.

A decrease of the amplitude just before the instability as for 60 % void fraction (Fig. 5) has already been observed by Joly et al. [7] with a rotated tube bundle for 70 % void fraction. From [3], which deals with freon vapor-liquid, there are two regions of instability, in the first at low void fraction the instability is very well marked with an abrupt increase of amplitude, in the second one instability is not that well marked with a lower slope of the instability curve, which make it difficult to extract a precise value of critical velocity. This observation is confirmed by the present work and by Violette et al. [14] on a rotated triangular tube bundle in air-water flow. Nakamura et al. [5] proposed tests on a square tube bundle in air-water and steam-water flow, they observed a low slope of the instability curve as well but only for the air-water case. They also observed a decrease of vibration amplitude after the instability in the air-water case. But the curves proposed are not at constant void fraction, thus the comparison with present results is not obvious. But still, this leads us to make the assumption that the phenomenon observed in the present study will not occur in steam-water flow.

CONCLUSION

Experimental results on the vibration of a normal triangular tube bundle subjected to a air-water flow and constrained to vibrate only in the lift direction have been presented. Fluidelastic instabilities were observed. Comparison with previous studies showed that the normal triangular configuration is more stable than the rotated triangular configuration.

Significant vibration amplitudes, that increase with void fraction, were observed. This phenomenon is not well understood and might be due to the diameter of the tubes, which is larger here than which exists in real steam generators. We know only little about the origins of the phenomenon, and weather they are due to periodic fluid forces or damping controlled instability. Thus if we want to clearly understand this phenomenon we need to determine the prevailing two-phase flow regime by locally measuring the dynamic characteristics of the two-phase flow (i.e. velocity, void fraction, bubble size, flow direction).

Acknowledgment

This work was sponsored by the BCW/AECL/NSERC Industrial Research Chair and the French Commissariat à l'Energie Atomique (CEA). Their support and permission to publish this paper is gratefully acknowledged.

REFERENCES

[1] Pettigrew, M.J., and Taylor, C.E., 1994, ”Two-Phase Flow-Induced Vibration : An Overview”, ASME Journal of Pressure Vessel Technolgy, Vol. 116 p. 233-253.

[2] Axisa, F., Boheas, M.A., Villard, B., and Gibert, R.J., 1985, ”Vibration of Tube Bundles Subjected to Steam-Water Cross Flow: a Comparative Study of Square and Triangular Pitch Arrays”, 8th Internationnal Conference on Structural Mechanics in Reactor Technology, Paper No. B1/2, Bruxelles, Belgium.

[3] Pettigrew, M.J., Taylor, C.E., 2009, ”Vibration of a Normal Triangular Tube Bundle Subjected to Two-Phase Freon Cross Flow”, ASME Journal of Pressure Vessel Technolgy, Vol. 131 p. 051302-1-7.

[4] Price, S.J., Zahn, M.L., 1991, ”Fluidelastic behavior of a normal triangular array subjected to cross-flow”, Journal of Fluids and Structures Vol. 5, 1991, p. 259-278.

[5] Nakamura, T., Fujita, K., Kawanishi, K., Yamaguchi, N., and Tsuge, A., 1995, ”Study on the Vibrational Characteristics of a Tube Array Caused by Two-Phase Flow. Part II: Fluidelastic Vibration”, Journal of Fluids and Structures Vol. 9, 1995, p. 539-562.

[6] Zhang, C., Pettigrew, M.J., and Mureithi, N.W., 2007, ”Vibration Excitation Force Measurements in a Rotated Triangular Tube Bundle Subjected to Two-Phase Cross Flow”, ASME Journal of Pressure Vessel Technolgy, Vol. 129 p. 21-27.

[7] Joly, T., Mureithi, N.W., and Pettigrew, M.J., 2009, ”Effect of Angle of Attack on the Fluidelastic Instability of a Tube Bundle subjected to Cross-Flow”, ASME Pressure Vessels & Piping Conference, Paper No. 77052, Prague, Czech Republic.

[8] Pettigrew, M.J., Taylor, C.E., Jong, J.H., and Currie, I.G., 1995, ”Vibration of a Tube Bundle in Two-Phase Freon Cross-Flow”, ASME Journal of Pressure and Vessel Technology, Vol. 117, p. 321-329.

[9] Pettigrew, M.J., and Taylor, C.E., 2003, ”Vibration analysis of shell-and-tube heat exchangers: an overview-Part 1: flow, damping, fluidelastic instability”, Journal of Fluids and Structures Vol. 18, 2003, p. 469-483.

[10] Noghrehkara, G.R., Kawaji, M., Chan, A.M.C., 1999, ”Investigation of two-phase flow regimes in tube bundlesunder cross-flow conditions”, International Journal of Multiphase Flow Vol. 25, 1999, p. 857-874.

[11] Sasakawa, T., Serizawa, A., and Kawara, Z., 2005, ”Fluidelastic vibration in two-phase cross flow”, Experimental Thermal and Fluid Science Vol. 29, 2005, p. 403413.

[12] Hirota, K., Nakamura, T., J. Kasahara, J., Mureithi, N.W., Kusakabe, T., and Takamatsu, H., 2002, ”Dynamics of an in-line tube array subjected to steam-water cross-flow. Part III : fluidelastic instability tests and comparison with theory”, Journal of Fluids and Structures Vol. 16, 2002, p. 153-173.

[13] Feenstra, P.A., Weaver, D.S., and Nakamura, T., 1995, ”Vortex shedding and fluidelastic instability in a normal square tube array excited by two-phase cross-flow”, Journal of Fluids and Structures Vol. 17, 2003, p. 793-811.

[14] Violette, R., Pettigrew, M.J., and Mureithi, N.W., 2005, ”Fluidelastic Instability of an Array of Tubes Preferentially Flexible in the Flow Direction Subjected to Two-Phase Cross-Flow”, ASME Pressure Vessels & Piping Conference, Paper No. 1133, Denver, Colorado.