RELATIONSHIP BETWEEN VIBRATION BEHAVIOUR AND TWO-PHASE FLOW REGIME TRANSITIONS FOR PIPING WITH INTERNAL FLOW

C. Béguin & A. Ross & M. J. Pettigrew & N. W. Mureithi
BWC/AECL/NSERC Chair of Fluid-Structure Interaction, Department of Mechanical Engineering, École Polytechnique, P.O.Box 6079, succ. Centre-Ville, Montréal (Québec), Canada, H3C 3A7

ABSTRACT

Internal two-phase flow is common in piping components. Modeling such flow and in particular locating flow regime transition turns out to be a difficult task, due to the numerous parameters involved. Theoretical transition boundaries proposed by Taitel et al. (1980). for upward gas-liquid flow in vertical tubes seem to apply fairly well within a limited range of tube diameters. However, Anscutter et al. (2006) have shown recently that this model does not adequately reflect transitions for high flow velocity. They have suggested a simple empirical boundary to describe transitions from bubbly flow to churn or slug flow. This transition boundary is based on homogeneous velocity, surface tension, gas and liquid densities. The present paper is intended to verify the new suggested transition model. Experiments are carried out on various vertical clamped-clamped PVC tubes. Tubes of 9.0 to 23.8 mm inside diameter are subjected to air-water two-phase flow. Homogeneous flow velocities are varied between 1 m/s and 5 m/s. Volumetric qualities are varied in 5% *increments near transitions. Transitions are detected using both photographs and the "maximum two-phase dam-ping method" (Anscutter et al., 2006). Results shows that even if two-phase damping and flow regime are correlated, the "maximum two-phase damping method" is not a precise way to etablish transition. Finally this paper proposes a correction on two-phase damping model for bubbly flow.*

1. INTRODUCTION

In the nuclear and chemical process industries, 50% of piping elements operate with two-phase flows (Pettigrew and Taylor, 2004). Flow-induced vibration can lead to structural degradation, process malfunction, and component failure. Two-phase damping can significantly contribute to reducing vibration and thus, to prevent premature fatigue or wear. Therefore, it is desirable to identify some of the parameters that govern two-phase damping in pipes with internal two-phase flow. In the present paper, we propose to investigate some of those parameters.

The first damping experiments in two-phase flow were performed some 25 years ago by Carlucci (1980) on a series of tubes subjected to an axially confined air-water two-phase flow. His results showed that damping in two-phase flow strongly depends on void fraction; no significant relation was found with frequency or fluid mixture velocity. Many researchers have since contributed to the knowledge of two-phase damping. Recently, Gravelle et al. (2007) and Anscutter et al. (2006) shed new light on the parameters that govern two-phase damping in vertical tubes with internal air-water two-phase flows. The experiments showed that damping is affected by void fraction, flow velocity, and flow regime. The authors, as previously suggested by Hara (1988), showed that the interface surface area, which depends on flow regime, may be a dominant factor. This paper is intended to validate the model for two-phase flow damping in bubbly flow proposed by Anscutter et al. (2006) and, in particular, to verify the occurence of the transition when two-phase damping is maximum.

2. TWO-PHASE FLOW CONSIDERATIONS

2.1. Basic definitions

The proportion of gas in a two-phase gas-liquid mixture is characterized either by the void fraction ε or by the volumetric quality β (Collier and Thome, 1996). In one unit length ΔL of a two-phase mixture inside a tube, void fraction and volumetric quality are defined as:

$$\begin{aligned} \varepsilon &= \frac{V_g}{V_g+V_l} = \frac{A_g\Delta L}{A_g\Delta L+A_l\Delta L} = \frac{A_g}{A_g+A_l} \\ \beta &= \frac{Q_g}{Q_g+Q_l} = \frac{A_gU_g}{A_gU_g+A_lU_l} = \frac{\varepsilon}{\varepsilon+(1-\varepsilon)S_r} \end{aligned} \quad (1)$$

where V_g and V_l are the volumes of the gas and liquid phases in the mixture, A_g and A_l are the cross sectional areas of each phase in the tube section, Q_g and Q_l are the volume flow rates, U_g and U_l are the velocities of each phase. Void fraction and volumetric quality are related to each other through the slip ratio, $S_r = U_g/U_l$. The superficial velocities of each phase (U_{gs}, U_{ls}) and the homogeneous velocity ($U_{2\phi}$) are defined as :

$$\begin{array}{l} U_{gs} = \frac{Q_g}{A} = \frac{A_g U_g}{A} = \varepsilon U_g = \beta U_{2\phi} \\ U_{ls} = \frac{Q_l}{A} = (1-\varepsilon)U_l = (1-\beta)U_{2\phi} \\ U_{2\phi} = \frac{Q_g + Q_l}{A} = U_{gs} + U_{gl} \end{array} \qquad (2)$$

2.2. Damping in two-phase flow

Tube motion affects internal flow and allows energy transfer from tube to fluid and vice versa. If the fluid gains energy, the tube motion is damped; conversely if the fluid looses energy, the tube becomes unstable. Energy transfer is directly related to the initial energy in the fluid and, in particular, kinetic energy. Damping depends a priori on flow rates. Carlucci's experiments and theory show that energy transfer (damping) in two-phase flow is greater than in single-phase flows. Thus, the concept of two-phase damping was introduced to allow for this difference. The total damping in two-phase flows therefore includes the components of structural (ζ_s), viscous (ζ_v), flow dependent (ζ_f) and two-phase damping ($\zeta_{2\phi}$). Figure 1 shows the contribution of each component to the total damping ratio for confined annular air-water axial flow. Structural damping depends on the tube material and supports; it is not shown on this figure. Two-phase damping is preponderant and strongly depends on void fraction. Although Carlucci's tests were performed with an axially confined external flow, the various damping mechanisms in an internal flow are expected to be the same. The geometric configuration is different, but the motion of the tube and the dependence of two-phase damping on void fraction exhibit trends similar to internal flow. Carlucci et al. (1983) suggested that the total damping ratio ζ_t should be given by the sum of the various damping components:

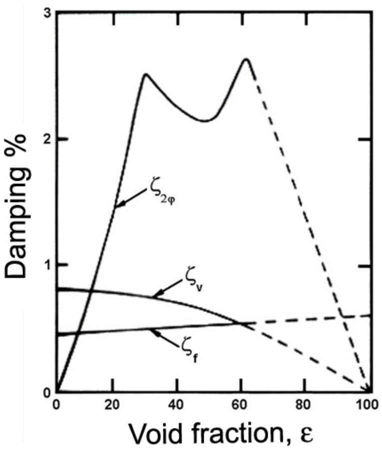


Figure 1: *Different components of damping*

$$\zeta_t = \zeta_s + \zeta_v + \zeta_f + \zeta_{2\phi} \qquad (3)$$

More recently Anscutter et al. (2006) carried out experiments for internal two-phase flow. They showed that the transition from bubbly flow to churn or slug flow occurs when damping is maximum for a given homogeneous velocity. The transition does not seem to depend on internal tube diameter, but only on homogeneous velocity. It was empirically correlated as :

$$\varepsilon_{trans} = 0.15 + 0.065 U_{2\phi} \qquad (4)$$

Difference between gas and liquid velocity is evaluated as the velocity of a single bubble in stagnant liquid (U_b is typically around 0.25 m/s) proposed by Harmatty (1960) :

$$U_b = U_g - U_l = 1.53 \left[\frac{g\sigma(\rho_l - \rho_g)}{\rho_l^2} \right]^{1/4} \qquad (5)$$

The greater is the homogeneous velocity, smaller is the difference between ε and β. Homogeneous model corresponds to $U_b = 0$ or $\varepsilon = \beta$. With equations (1,2 and 5), Anscutter et al. (2006) deduced a slip ratio model and a relation between volumetric quality and void fraction :

$$\begin{array}{lcl} \beta & = & \varepsilon + \varepsilon(1-\varepsilon)\frac{U_b}{U_{2\phi}} \\ \varepsilon & = & \frac{1}{2}\left[1 + \frac{U_{2\phi}}{U_b} - \sqrt{\left(\frac{U_{2\phi}}{U_b} + 1\right)^2 - 4\beta\frac{U_{2\phi}}{U_b}}\right] \end{array} \qquad (6)$$

As previously suggested by Hara (1988) and Gravelle et al. (2007), the two-phase damping ratio in bubbly flow is strongly correlated with interface surface area. Anscutter et al. (2006) have proposed a theoretical expression to evaluate interface surface area.

$$S = B_1 \varepsilon D_i^{38/25} L \left(\frac{\rho_l}{\sigma}\right)^{3/5} \nu_l^{2/25} U_{2\phi}^{18/25} \qquad (7)$$

They also found that two-phase damping is related as follows to interface surface area and homogeneous flow velocity :

$$\zeta_{2\phi} = B_2 \frac{S}{U_{2\phi}^{1.92}} \qquad (8)$$

In bubbly flow, the bubbles were considered as spheres and the interface surface area was written as:

$$S = \sum_{k=1}^{N} S_{b,k} = \sum_{k=1}^{N} \pi d_{b,k}^2 = N\pi < d_b^2 > \qquad (9)$$

where $S_{b,k}$ and $d_{b,k}$ are the surface area and diameter of the k^{th} bubble, $< d_b^2 >$ is the average squared diameter of the bubbles, and N is the number of bubbles. In addition:

$$N = \frac{V_g}{< V_b >} = \frac{\varepsilon \pi \left(D_i/2\right)^2 L}{\pi/6 < d_b^3 >} \qquad (10)$$

So

$$S = \frac{\varepsilon 3\pi D_i^2 L < d_b^2 >}{4 < d_b^3 >} = B_0 \frac{3\pi\varepsilon D_i^2 L}{4 d_{bmax}} \qquad (11)$$

where L and D_i are the length and inside diameter of the tube. Riverin (2005) suggested that bubble diameters follow a Rayleigh distribution ($\frac{<d_b^2>}{<d_b^3>} = \frac{B_0}{d_{bmax}}$ where B_0 is a constant). The maximum bubble diameter is suggested by Hinze (1955).

$$d_{bmax} = \kappa \left(\frac{\sigma}{\rho_l}\right)^{3/5} \tau^{-2/5} \qquad (12)$$

where κ is a constant and τ is the power dissipation per unit mass in turbulent pipe flow :

$$\begin{aligned} \tau &= \left|\frac{\partial P}{\partial z}\right| \frac{U_{2\phi}}{\rho_{2\phi}} = \left|\frac{2f}{D_i}\rho_{2\phi}U_{2\phi}\right| \frac{U_{2\phi}}{\rho_{2\phi}} \\ &= \left|0.092\left(\frac{\nu_l}{U_{2\phi}D_i}\right)^{1/5} \frac{\rho_{2\phi}U_{2\phi}}{D_i}\right| \frac{U_{2\phi}}{\rho_{2\phi}} \\ &= 0.092\nu_l^{1/5}\frac{U_{2\phi}^{2-1/5}}{D_i^{1+1/5}} = 0.092\nu_l^{1/5}\frac{U_{2\phi}^{9/5}}{D_i^{6/5}} \end{aligned} \qquad (13)$$

Finally Anscutter et al. (2006) came with the following relation for bubbly two-phase flow relation :

$$\zeta_{2\phi} = 1.02 \left[\varepsilon D_i^{1.52} U_{2\phi}^{-1.2} \left(\frac{\rho_l}{\sigma}\right)^{3/5} \nu_l^{2/5}\right] \qquad (14)$$

where D_i is expressed in inch. According to equations (11) to (13) it can been seen that the exponent of ν_l in equations (14) is incorrect. However, since all experiments were carried out in water, this error affects only the constant factor. Using international units and correcting the exponent of ν_l we have :

$$\zeta_{2\phi} = 3.3 \left[\varepsilon D_i^{1.52} U_{2\phi}^{-1.2} \left(\frac{\rho_l}{\sigma}\right)^{3/5} \nu_l^{2/25}\right], \varepsilon < \varepsilon_{trans} \qquad (15)$$

where void fraction (ε) is evaluated through the slip ratio model in equation (6).

3. TEST SECTION AND PROCEDURE

Experiments were carried on with water and air supplied at normal ambient pressure and temperature. Damping measurements were performed while the pre-measured mixture was injected at the bottom of the vertical tube, as shown in Fig. 2. Damping was measured using the logarithmic decrement technique with an initial transverse displacement on the tube. The tests have two purposes. One is to establish the behaviour of two-phase damping with respect to the internal diameter of the tube, and to confirm the effect of confinement on bubble coalescence. The second purpose is to confirm the validity of the method suggested above, by which flow regime transitions could be detected from damping measurements. For each test configuration, photographs were taken to determine the flow regime and possible bubble interactions. Table 1 shows the characteristics of the test performed by a) Gravelle et al. (2007) b) Anscutter et al. (2006) c) Beguin et al. (present data). Volumetric quality increment are shown in brackets [x%].

Test	Tube Material	D_i (mm)	β(%) [x%]	$U_{2\phi}$ (m/s)
a	Brass, PVC Polycarbonate	18, 19 and 21	0-100 [10%]	1.5-5
b	PVC	11, 15 and 21	0-100 [10%]	1.5-11
c	PVC	9, 13 18, 24	0-100 [5%]	1-5

Table 1: *Test characteristics*

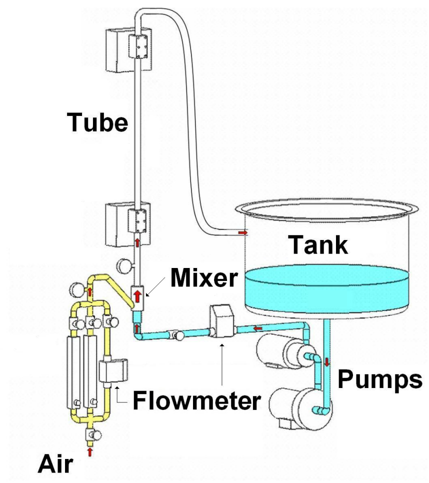


Figure 2: *Test section*

In the present experiments, we found that the viscous damping component ζ_v and the flow-dependent component ζ_f were small with respect to the two-phase component $\zeta_{2\phi}$ in the entire range of void fractions. The measured values of ζ_s range between 0.6% and 0.7%, which is much less than the two-phase damping ratios reported in the following sections. Total damping was measured at 0% and 100% void fractions (where $\zeta_{2\phi} = 0$), for various flow velocities. In air, the diffe-rence between total damping and structural damping ($\zeta_t - \zeta_s = \zeta_v + \zeta_f$) was less than the experimental error. In water, the difference is small (less than 5% of structural damping). Therefore, a very simple model was used to represent viscous and flow dependant damping. Figure 1 shows that both viscous and flow-dependent components are monotonic functions of void fraction. Thus, these two components (ζ_v and ζ_f) were considered as a linear function of volumetric quality from 0% to 100%.

$$(\zeta_v + \zeta_f)_{\beta,U_{2\phi}} = (\zeta_v + \zeta_f)_{\beta=0,U_{2\phi}=U_l}(1-\beta) \quad (16)$$

where $(\zeta_v + \zeta_f)_{\beta=0,U_{2\phi}=U_l}$ is measured for water flow with velocity U_l. Finally, in the two-phase flow experiments, $\zeta_{2\phi}$ was calculated as:

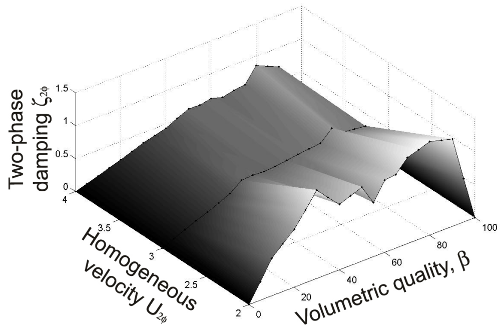
$$\zeta_{2\phi} = \zeta_t - \zeta_s - (\zeta_v + \zeta_f)_{\beta,U_{2\phi}} \quad (17)$$

4. RESULTS

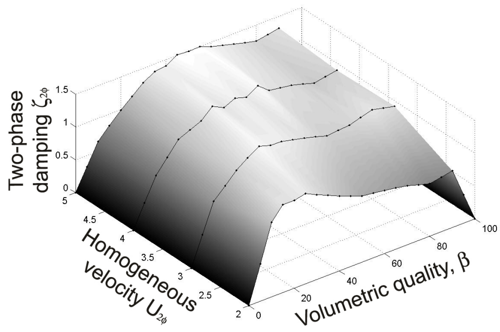
4.1. Transition from bubbly flow

Experimental results are presented for various tube diameter in Figure 3(a) to 3(d) as two-phase damping ratio with respect to volumetric quality and flow velocity. As previously noticed by Anscutter et al. (2006), for low volumetric quality corresponding to bubbly flow, the two-phase damping ratio increases linearly with volume-tric quality. In bubbly flow, higher velocity leads to lower two-phase damping ratio. Higher velocity also causes the transition of bubbly flow to churn or slug flow at higher void fraction. A larger tube leads to a higher two-phase damping ratio. Interface surface area drops at the transition, and so does the two-phase damping ratio. The maximum damping ratio occurs around the same value in the present tests as in previous tests. Figure 4 shows the void fraction at maximum damping with respect to homogeneous flow velocity in accordance with Equation (4)

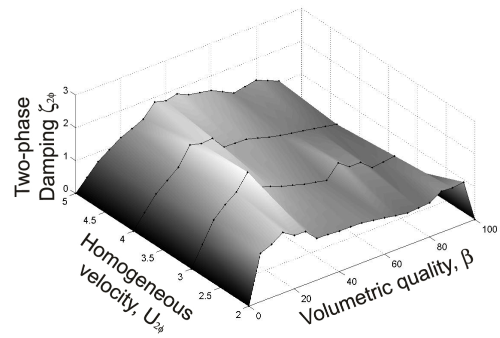
Figure (5) shows the two-phase damping ratio evaluated with equation (14) with slip ratio model (Equation (6)) and with the homogeneous



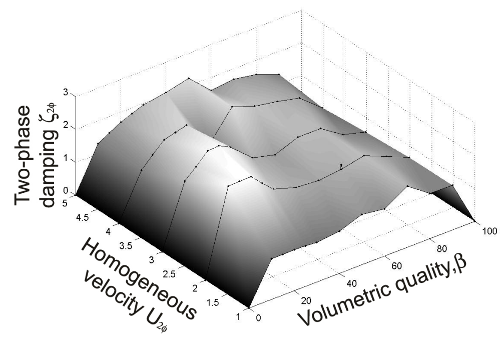
(a) *Tube ($D_i = 9$ mm)*



(b) *Tube ($D_i = 13.4$ mm)*



(c) *Tube ($D_i = 18.3$ mm)*



(d) *Tube ($D_i = 23.8$ mm)*

Figure 3: *Two-phase damping*

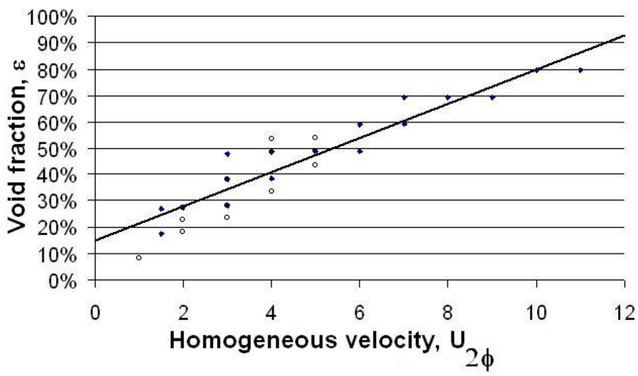


Figure 4: *Void fraction (ε) at maximum damping ratio with respect to homogeneous flow velocity. Diamond : Anscutter et al. (2006); circle : Present data*

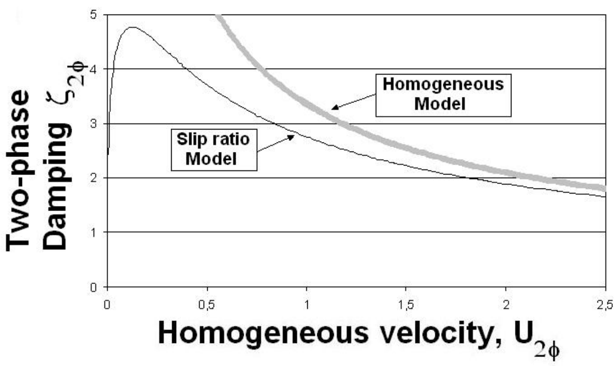


Figure 5: *Two phase damping ratio vs. homogeneous flow velocity for $\beta = 15\%$. Thick gray line : homogeneous model ; thin black line : slip ratio model*

model $\varepsilon = \beta$. The slip ratio model allows to have a zero limit of two-phase damping ratio when velocity goes to zero. Contrary to Anscutter et al. (2006), our photographs on Figure 6 show that the transition from bubbly flow to churn or slug flow does not always occur at the maximum damping. The transition seems to be more around +/- 10% of the maximum damping ratio. This observation leads one to believe that there exists an interaction between vibration and flow pattern transition. We have observed in some situations that the vibration can change the flow pattern. We choose to name "Transition Zone" the zone where flow pattern without vibration is not in accordance with the evolution of two-phase flow damping. This transition zone is larger, in terms of void fraction coverage, when vibration is present than it is without vibration. To be more conservative we will propose a new limit for the two-phase damping ratio model. The new limit

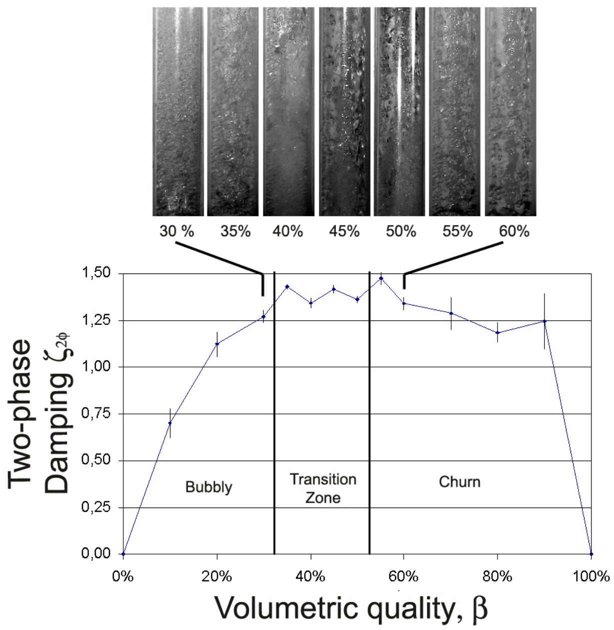


Figure 6: *Two phase damping vs. Volumetric quality for $U_{2\phi} = 4m/s$ and $D_i = 13.4mm$)*

is 10% lower than the previous limit :

$$\varepsilon_{lim} = 0.05 + 0.065 U_{2\phi} \tag{18}$$

The effect of vibration amplitude and tube diameter on the transition zone was not studied thoroughly enough to be conclusive. Photographs are not the ideal tool to study flow pattern.

4.2. Two-phase damping model

The correlation between $\zeta_{2\phi}/S$ and $U_{2\phi}$ leads to the following model :

$$\zeta_{2\phi} = 0.77 \left[\varepsilon D_i^{1.52} U_{2\phi}^{-0.68} \left(\frac{\rho_l}{\sigma} \right)^{3/5} \nu_l^{2/25} \right], \varepsilon < \varepsilon_{lim} \tag{19}$$

This model has been compared with the two-phase damping experiments of Anscutter et al. (2006) and our experimental data in Figure 7. We can conclude that this correlation can predict two-phase damping in bubbly flow within +/- 0.5%.

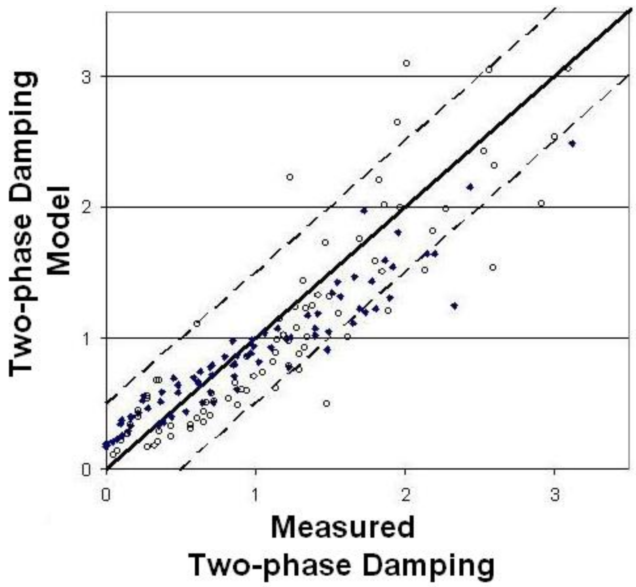


Figure 7: *Two-phase damping model vs measured two-phase damping. Diamond : Anscutter et al. (2006); circle : Present data*

5. CONCLUSION

The present and precedent studies have lead to the following conclusions :

- Two-phase damping in bubbly flow can be evaluated through equation (19).
- Vibration can modify the transition from bubbly flow to churn or slug flow
- Even with vibration, bubbly flow transition occurs in the range $0.05 + 0.065U_{2\phi} < \varepsilon < 0.25 + 0.065U_{2\phi}$, independently of the tube diameter.

In future work, the impact of vibration on bubbly flow transition should be studied more carefully.

6. ACKNOWLEDGMENT

The experiments reported were performed with the valuable and much appreciated assistance of M. Abdelali Raouhani.

7. REFERENCES

Anscutter, F., Béguin, C., Ross, A., Pettigrew, M.J. , Mureithi, N.W., 2006, Two-Phase Damping and Interface Surface Area in Tubes with Internal Flow *Proceedings, ASME Pressure Vessels and Piping Conference 2006* **9** : 537-547.

Carlucci, L.N., 1980, Damping and Hydrodynamic Mass of a Cylinder in Two-Phase Flow. *ASME Journal of Mechanical Design* **102**: 597-602.

Carlucci, L.N., 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 Design* **105**: 83-89.

Collier J.G. and Thome J.R., 1996, Convective boiling and condensation, *3rd ed., Clarendon Press, Oxford University Press, p6.*

Gravelle, A. , Ross, A., Pettigrew, M.J. and Mureithi, N.W., April 2007, Damping of tubes due to internal two-phase flow *Journal of Fluids and Structures* **23**(3): 447-462.

Hara, F., 1988, Two-Phase Fluid Damping in a Vibrating Circular Structure *ASME, Pressure Vessels and Piping Division (Publication) PVP* **133** : 1-8.

Harmatty, T.Z., 1960, Velocity of Large Drops and Bubbles in Media of Infinite or Restricted Extend, *AIChE Journal* **6**: 281.

Hinze, J.O., 1955, Fundamentals of the Hydrodynamic Mechanism of Splitting in Dispersion Processes, *AIChE Journal* **1**: 289.

Pettigew, M.J. and Taylor, C.E., November 2004, Damping of Heat Exchanger Tubes in Two-Phase Flow: Review and Design Guidelines. *ASME Journal of Pressure Vessel Technology* **126**: 523-533.

Riverin, J.L., 2005, Étude des forces flucuantes dans les élements de tuyauterie soumis à des écoulements diphasiques, *M.Sc.A Thesis, École Polytechnique, Montréal (Canada)* **145 p**.

Taitel, Y. Bornea, D.,Dukler A.E., May 1980, Modelling Flow Pattern Transitions for Steady Upward Gas-Liquid Flow in Vertical Tubes *AIChE Journal* **26**(3): 345-354.