Study on Impact Vibration of Loosely Held Tube by Cross Flow (2nd Report)

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ABSTRACT

Loosely held tubes in heat exchangers are supported in a cross flow of coolant fluid. Impact vibration of these tubes is one of the major subjects to be solved. This paper presents an analytical treatment of the fluidelastic force and its extension to two-phase flow, and gives a comparison with actual test results.

1. INTRODUCTION

Tubes in heat exchangers have to be loosely held to relieve thermal stress and for ease of assembly. These tubes are forced to oscillate by the fluid flow and this sometimes causes impact vibration at the supports. There are two types of vibration induced by two-phase cross flow: fluidelastic instability and random vibration. This paper introduces a newly developed theory of fluidelastic force, that is an extension of our previous study (1), and the treatment of steam-water two-phase flow based on our physical interpretation of the phenomenon.

2. ANALYSIS

2.1 Fluidelastic force

A lot of papers on fluidelastic vibration have been published, however, the mechanism of the instability has not yet been clarified (2). In the first paper (1), fluidelastic force has been assumed to be a quasi-static one modified with a time lag term to introduce an unsteady effect, that is based on the theory of Price & Paidoussis (3). It requires some parameters to be estimated from test results, and there are insufficient data for liquid flow.

The following theory (6) has been introduced to estimate the fluidelastic force as physically as possible, while some assumptions had to be introduced into it. Here, the fluidelastic forces $F_p$ and $F_L$ that correspond to the in-flow direction and out-of-flow direction respectively, are derived from the following equation, where the pressure distribution function $C_p(\theta)$ around the tube plays an important role:
\[
\begin{aligned}
F_D &= \frac{1}{2} \rho U^2 \left[ \int_0^\pi (C_{P+}(\theta) + C_{P-}(\theta)) \cos \theta \cdot R d\theta \right] L
\end{aligned}
\]

\[
\begin{aligned}
F_L &= \frac{1}{2} \rho U^2 \left[ \int_0^\pi (C_{P+}(\theta) - C_{P-}(\theta)) \sin \theta \cdot R d\theta \right] L
\end{aligned}
\]

here, \( \rho \) = fluid density, \( U \) = flow velocity, \( R \) = tube radius, \( L \) = length

The plus "+" and minus "-" signs indicate the direction of the movement of the tube as shown in Fig.1. The pressure distribution function \( C_p(\theta) \) is assumed here to be as follows and is shown in Fig.2:

\[
\begin{aligned}
C_p(\theta) &= \begin{cases} 
1 - C_{p0} \cdot \sin^2 \left( \frac{\pi}{2 \theta_p} \right) & (0 \leq \theta \leq \theta_s) \\
1 - (1 - v)C_{p0} & (\theta \geq \theta_s)
\end{cases}
\end{aligned}
\]

Here, \( C_{p0} \) is the amplitude of the pressure coefficient that should include the narrow gap effect inside the tube array. The following formula is deduced from test results of a single tube in a narrow path:

\[
\begin{aligned}
C_{p0}(y) = 277.6 \left( \frac{y(t) + D}{T} - 0.25 \right)^4 + 3.1
\end{aligned}
\]

here, \( y \) = displacement of a tube in the lift direction, \( D \) = tube diameter, \( T \) = pitch

\( \theta_s \) means the separation point, that can be regarded as a function of tube displacement and velocity. By using the results of visual tests carried out on a tube array, the following empirical formula has been derived:

\[
\begin{aligned}
\theta_s(y, v) &= 102^\circ + 40^\circ \left( \frac{y(t) + D}{T} \right) + 150^\circ \frac{y(t - \Delta t)}{\Delta t}
\end{aligned}
\]

here, \( \omega \) = angular frequency

In this equation, \( \Delta t \) is a time lag term that introduces a destabilizing effect on the fluid force. Then \( F_D \) and \( F_L \) become the fluid-elastic forces.

2.2 Two-phase flow

Gas-liquid two-phase flows are classified according to flow patterns, such as bubbly flow, slug flow, mist flow and so on. Slug flow and froth flow, that might be important in many heat exchangers, are treated in this paper. These flow patterns can be regarded as an intermittent flow as shown in Fig.3, where liquid slugs and gas slugs flow alternately around the tube.

Superficial flow velocities \( j_g \) and \( j_l \), that are the gas and liquid flow velocities respectively, can be computed by thermal-hydraulic analysis for every position in the heat exchanger. By using these values, the intermittent flow velocities \( V_g \) and \( V_l \), that correspond to the intermittently rising speed of liquid and gas slugs respectively, can be estimated as follows:

\[
\begin{aligned}
V_g &= a(P) \cdot \frac{j_l}{1 - \alpha} \\
V_l &= \frac{j_s + j_l - V_g}{1 - P_g}
\end{aligned}
\]

Here, \( \alpha \) means the void fraction, that can be derived from various formulas. The following drift flux model is used in this paper.
\[ \alpha = \frac{J_s}{1.2(j_t + j_i) + 0.1} \tag{6} \]

The non-dimensional coefficient \(a(P)\) shows the relation between the actual intermittent liquid flow velocity and the theoretical liquid flow velocity, which depends on the pressure of the field. This has been deduced from our test results (7) to be as follows:

\[
a(P) = \begin{cases} 
1.6 & (P \approx 5.8\text{MPa}) \\
1.85 & (P = 0.5 - 3.0\text{MPa}) \\
2.5 & (P = 0.1\text{MPa}) 
\end{cases} \tag{7}
\]

The remaining parameter \(P_s\) in Eq.(5) means the existence ratio of the liquid slug. It could be estimated from test results, that show \(P_s\) might be a function of the pressure of the field, liquid flow volume and so on.

In addition to the above values, the period of the occurrence of the liquid slug \(\Delta T\) has to be estimated experimentally, while \(\Delta T_s\) in Fig.3 could be given as \(\Delta T_s = \bar{P} \Delta T\). However, these parameters have been given in our paper (7).

Then the response of a tube in a tube array caused by intermittent two-phase flow can be estimated by combining the estimation of two-phase flow with the fluidelastic force given in Eq.(1). If the sudden change of the separation point corresponding to the change of phase is ignored, the unsteady fluid force would be expressed by the following:

\[
\begin{align*}
F_s(t) &= \frac{1}{2} \left[ \rho_s \left( 1 - \frac{D}{T} \right) U_s(t) \right]^2 + \rho_g \left( 1 - \frac{D}{T} \right) U_g(t) \right]^2 \\
&\quad \times \int_0^\theta \left[ C_{r_s}(\theta) + C_{r_g}(\theta) \right] \cos \theta \cdot Rd\theta \\

F_g(t) &= \frac{1}{2} \left[ \rho_g \left( 1 - \frac{D}{T} \right) U_g(t) \right]^2 + \rho_s \left( 1 - \frac{D}{T} \right) U_s(t) \right]^2 \\
&\quad \times \int_0^\theta \left[ C_{r_g}(\theta) - C_{r_s}(\theta) \right] \sin \theta \cdot Rd\theta 
\end{align*} \tag{8}
\]

Here, \(\rho_s\) is the density of the liquid slug and \(\rho_g\) is the density of the gas slug. Furthermore, impact vibration at the tube support can be computed when the reaction force on the tube is estimated as one of the external loads in the following equation of motion:

\[
[M_s][\ddot{X}] + [C_s][\dot{X}] + [K_s][X] = \{F\} \tag{9}
\]

where vector \(\{F\}\) includes fluidelastic force and impact force.

3. TESTS

3.1 Single-phase flow

The test equipment (9) is shown in Fig.4. Although the figure shows only 5 rows there were actually 20 rows in the tube array. There were 5 columns of tubes with semicircular dummies attached to the walls. Only one tube in the center of the tube array is flexibly supported and the others are rigid. When the flexible tube becomes unstable, it causes an impact against the stoppers on both sides. The impact forces were measured with the strain gages at the ends of the stopper system.

Fig.5 shows the results of gas flow tests. These are compared with the calculated
data which show a good agreement with the test values. Fig.6 shows the same kind of test results as Fig.5 but for liquid flow and the same good agreement is observed.

3.2 Two-phase flow

Fig.7 shows the two-phase flow-induced impact vibration test equipment, that can be operated at up to 7.0MPa, 284°C. The number of tubes in the tube array is the same as in Fig.4, however, nine tubes in the middle of the tube array are flexibly supported. An adjacent tube is set with a 1mm gap clearance to measure the impact force when the flexible tubes become unstable. The tubes under test have a natural frequency of 19Hz in air and they become unstable in a liquid flow of 1.2m/s at 5.8MPa.

An example of the comparison between two-phase flow test results and the analytical data is shown in Fig.8. The maximum impact forces at various flow conditions are shown in Fig.9 to prove the possibility of the analytical method.

4. CONCLUSION

A quick overview of our study on the impact vibration of a tube in a tube array caused by two-phase fluidelastic forces has been given in this paper. There are plenty of remaining problems before this analytical treatment can be established for the design of heat exchangers. However, this paper may shed some light on the possibility of creating a fundamental framework for this subject.

REFERENCES

Fig. 6  Liquid Flow Test

Fig. 7  Two-phase Flow Test Equipment

(a) Measured

(b) Calculated

Fig. 8  Impact Vibration by Intermittent Flow
($p=5.8$MPa, $i=2.0$m/s, $j=0.4$m/s)

Fig. 9  Maximum Impact Force
vs. Liquid Flow Velocity