Flow Induced Vibration of Shell & Tube Type Heat Exchanger*
(2nd Report, Confirmation of Countermeasures)

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Abstract
This paper describes the confirmation of countermeasure's effect for flow-induced vibration in a shell & tube type heat exchanger. This type of heat exchanger has often been used in the LPG gas carrier ship and has sometimes caused a fretting and finally comes to a fatigue failure. In flow-induced vibration, there are vortex-induced vibration, fluid elastic instability and buffeting. In this paper, the clip is adopted as the countermeasure and it is clarified that the clip can suppress the tube vibration drastically. In this experiment, another countermeasure, that is to say, the dummy tube array is also examined. However the fluid elastic instability occurs by use of dummy tube array. The reason is that the dummy tube was supported by baffle plates without clearance. Therefore the damping ability becomes very small and the natural frequency of the tube appears very clearly.

Key words: Flow Induced Vibration, Turbulence, Flow-meter, Fluid Elastic Instability, Heat Exchanger, Buffeting, Damping, Vibration Control Device

1. Introduction

In several types of heat exchangers, the shell & tube type heat exchanger has been used to evaporate a liquid propane gas (LPG) by sea water when the LPG is loaded on. This structure is shown in Fig.1. Many U bend tubes are supported by several baffle plates with the thickness of 12 mm and the LPG flows in the tubes. The sea water is introduced from the inlet of the shell and discharged from the outlet and the heat exchange is performed in this process. The diameter of holes of the baffle plates are 0.4 mm larger than that of the tubes to allow the elongation due to the heating. Then the tube vibration becomes non-linear system with the clearance and behaves with complexity. However, there is also merit that it is hard to occur the fluid elastic instability due to the large damping(1).

Many vibrations of this kind have been experienced and they are (1) the vortex induced vibration due to the Karmann vortex, (2) the fluid elastic instability, and (3) the random vibration due to the flow turbulence (buffeting) (2) ~ (5). There are many countermeasures of the vibration and the effects of them have been confirmed(6). However, little study was performed on the behaviors of tube vibration, and it is not clear what kind of vibration occurs easily.

In the previous paper, the vibration of the eight tubes selected among many tubes in the shell was measured(7). As a result, it was clarified that the vibration of the tubes located at the upstream was larger than that at the downstream. Furthermore the countermeasure of the two stages of tubes that were clamped by the clip is the best solution.
The purpose of this study is to clarify the causes of the tube vibration and confirmation of the countermeasure's effect on them.

2. Experimental procedure

2.1 Outline of experiment

The outline of the shell & tube type heat exchanger used in this experiment is shown in Fig.1. The number of tubes is two hundreds and the tube arrangement is triangular (P/d0=1.3) as shown in Fig.2 (b). The measured tubes are also shown in Fig.2. The acceleration sensors were attached at 30 mm apart from the fixed end and perpendicular to the flow. The tube diameter and its thickness were 25.4 mm and 0.8 mm, respectively. The material was Titan and the diametral clearance between the tube and the hole of baffle was 0.4 mm. The measured tubes were selected by the following points of view.

(1) The tubes ①, ②, and ⑦ were located at the most upstream and the left, the center and the right, respectively. They were measured to examine the symmetry of the vibration amplitudes.

(2) The tubes ③, ④, ⑤ and ⑥ were arranged from the upstream to the downstream and were measured to examine the difference of the vibration amplitudes.

(3) The tube ⑧ is the same as ①. It was measured to indicate the vibration in the flow direction. ① and ⑧ makes the Lissajous, namely locus.

(4) The vibration of tube ⑨ is measured to examine that whether the effect of countermeasure arises at just downstream of the two stages of tubes with clip or not.
2.2 Measurement items
The measurement items are as follows.
(1) Vibration characteristics of tube (Natural frequency and damping ratio)
(2) Vibration response of tube at each flow rate

The natural frequency of tube was obtained by hitting the outside of the shell by a impulse hammer (see Figure 1) and the damping ratio was obtained by a half power method. The range of the flow rate of water was Q=500~1100m³/h and it was measured by an ultrasound flow meter.

3. Experimental results and their considerations

3.1 Vibrations of each tube on operation before countermeasure

Figure 3 shows time histories of acceleration of each tube at the flow rate Q=950m³/h. From this Figure, it can be seen that the vibration of tubes ①, ②, ⑦ and ⑧ is larger than the other tubes. In addition, the amplitudes of the tubes ①, ② and ⑦ were not same. This means that the vibrations were not symmetrical. The vibration of tubes located at the end is larger than that at the center due to the existence of the impinging baffle.

3.2 Ideas of countermeasures
In the previous paper, it was shown that the occurrence of this phenomenon was not
caused by fluid elastic instability but the random vibration (Buffeting) due to the flow turbulence. As the countermeasures, the following ideas could be proposed.

1. Removing the impinging baffle
2. Adding the dummy tubes just on the tube array located at the most upstream
3. Making the large damping ability to connect the tubes by the belt
4. Clamping the 2 stages of tubes located at the upstream by a clip
5. Changing the baffle thickness from t=12 mm to t=25 mm
6. Making the clearance small

All of these countermeasures could not be tried to confirm because of using the actual thing in this experiment. Then the countermeasures (1) and (2) among them described above are confirmed here.

4. Confirmation of countermeasure’s effects

4.1 Countermeasure items

The vibrations of each tube were measured for three ideas of countermeasures described below. The measured tubes are the same as reported in the previous paper, but one more tube (NO.⑨) was added in the present study.

1. Clip
2. Dummy tube
After countermeasure (1), Clip

Fig. 6 Time histories after C.M. (1)

After countermeasure (2), Dummy tube

Fig. 7 Time histories after C.M. (2)

After countermeasure (3), Clip + Dummy tube

Fig. 8 Time histories after C.M. (3)
(3) Clip + Dummy tube
The clip and the dummy tube used here are shown in Fig.4 and 5 respectively.

4.2 Experimental results
The experiment was performed under six flow rate conditions such as Q=500, 600, 700, 800, 950 and 1100m$^3$/h. In this paper, only the result at Q=950m$^3$/h is shown to compare with the result before using of the countermeasure.

Figure 6, Figure 7 and Figure 8 show the time histories of vibration for each tube which are corresponding to the countermeasures of (1), (2) and (3) respectively. In Figure 8, the time history is broken off. This is due to the sensor breakdown. The axis of abscissa indicates time and full scale is 30 seconds and the vertical axis indicates the acceleration (m/s$^2$). However the vibrations at the countermeasures (2) and (3) became very large. So the vertical axes are obliged to be 2000m/s$^2$ of full scale.

The summaries of this experiment are as follows.

(1) The use of the clip (countermeasure(1)) reduced the vibration amplitude about 1/10 ~ 1/30 comparing with original idea (before countermeasure) in the flow range Q=500 ~ 1100m$^3$/h. It is shown clearly in Fig.6.
(2) The use of the dummy tube (countermeasure(2)) increased the vibration suddenly at the flow rate Q=900m$^3$/h and kept large with increasing the flow rate. On the other hand, the vibration stopped at Q=500m$^3$/h. The hysteresis phenomenon was observed. Therefore this countermeasure is not effective.
(3) The use of the clip and the dummy tube (countermeasure (3)) also produce the same results as the countermeasure (2). Therefore, this is also not effective.

From above results, it is possible to conclude that only use of the clip is the best countermeasure.

4.3 Consideration of experimental result
The use of the clip becomes effective due to the friction force generated at the contact plane when the tube vibrates and the vibration is suppressed. On the other hand, the tube vibrations become larger in case of the use of the dummy tube due to fluid elastic instability which occurs in case of small damping ability. The dummy tubes are supported without clearance. Therefore the structural damping is very small.

Next, the flow rate at which fluid elastic instability occurs will be examined. It is impossible to measure the vibration of dummy tube in water due to structural reasons, the natural frequency and the logarithmic decrement in air are first obtained by the impulsive test. The results are shown in Fig.9 and Fig.10 respectively. The values of the natural frequency and the logarithmic decrement are $f_n=128Hz$ and $\delta =0.0081$. It is reasonable to use the natural frequency and the logarithmic decrement in water when the critical flow velocity of fluid elastic instability is evaluated. These values in water are tried to presume by ones in air. The natural frequency of the dummy tube is modified by taking into account the added mass and the logarithmic decrement of that is modified by adding the difference to the logarithmic decrement in air already measured. The difference of $\delta$ is measured in the actual tube. As a result, the natural frequency and the logarithmic decrement of the dummy tube in water become 88Hz and 0.031 respectively. However the frequency at fluid elastic instability is 66Hz as shown in Fig.11 and Fig.12, and this value is different from natural frequencies of the dummy tube (88Hz) and the dummy rod (81.5Hz). So the reason is first considered. Let’s consider the vibration system consisted of the two vibration systems with water. Both systems have natural frequencies $f_1$ and $f_2$. According to the simple vibration analysis, the lower natural frequency of the connected system becomes $f=f_1 (=f_2)$ independent of the connecting rigidity in case of $f_1=f_2$. On the other hand, both natural frequencies increase with the connecting rigidity in case of $f_1>f_2$. Figure 13 shows
Fig. 9 Inertance of dummy tube in air

Fig. 10 Damping ratio of dummy tube in air

Fig. 11 Time history of acceleration for tube① (Q=950 m³/h)

Fig. 12 Spectrum of acceleration for tube① (Q=950 m³/h)
Fig. 13 Variation of natural frequency due to connecting stiffness of water

The relation between f and the connecting rigidity. Where \( f_1 = 78 \text{Hz} \) (the natural frequency of the dummy tube in water) and \( f_2 = 54 \text{Hz} \) (the natural frequency of the actual tube in water). From Fig. 13, the natural frequency of the connected system can be obtained and the value is 66Hz if the connecting rigidity is assumed \( 5 \times 10^4 \text{N/m} \).

The critical flow velocity of fluid elastic instability could be obtained by using these values as follows.

\[
U_{cr} = \frac{f_d \cdot K}{\rho \cdot d} = 66 \times 0.0254 \times 3.3 \times \frac{1.4545 \times 0.031}{1025 \times 0.025^2} = 1.44 \text{m/s}
\]

for dummy tube

This flow velocity means \( Q = 960 \text{m}^3/\text{h} \). This value coincides with one described above, namely about 950m\(^3\)/h.

Therefore the large vibration of the dummy tube over 1000m\(^3\)/h is due to the fluid elastic instability. The discussion described above is made by the values measured on tube \( \mathbb{1} \), not the dummy tube. But the dummy tube and the tubes located at the most upstream give the influence mutually through water. It is easily to predict that the dummy tube vibrates violently in the occurrence of instability.

4.4 State of tube vibration at instability occurrence

The tube vibration at the instability occurrence presents steady state where the amplitude is kept constant as shown in Fig. 7. In general, fluid elastic instability characterizes that the vibration has only the natural frequency and is sinusoidal. This vibration was also expected to be characterized as mentioned before. However, the impulsive wave with 66Hz was observed as shown in Fig. 11. The FFT result is shown in Fig. 12. The component of 133Hz in this figure is the second order harmonics. From these results, it is considered that the tubes located at the most upstream become the impulsive vibration due to the impinging of these tubes to the baffle. The exciting force arises due to
the large vibration of dummy tube through water.

Figure 14 shows the spectra of acceleration of tubes before and after countermeasure (1) in order to compare the countermeasure’s effects at the maximum flow rate \(Q=1100\text{m}^3/\text{h}\). In the figures, (a), (b) and (c) are correspondent to the cases of tube ①, tube ③ and tube ⑥ respectively. The tube ① is clamped by the clip and the effect is large. The tube ③ is not clamped by the clip. But the effect existed considerably. This fact means that the connecting force by the water contributes to the vibration considerably. On the other hand, the effect is little at all in the tube ⑥ which is located the lowest downstream. From these results, the effect of the vibration of the tube located at the most upstream on the vibration of the lower tubes becomes small with going to the downstream.

4. Conclusions

The previous paper showed that the vibration occurred in the shell & tube type heat exchanger is not fluid elastic instability but random vibration due to the flow turbulence\(^{(1)}\). In this paper, the following three ideas of countermeasures were proposed, that is to say, (1) the use of clip, (2) the use of dummy tube and (3) the use of clip and dummy tube. The effects of them were confirmed by the experiment. As a result, the following findings could be obtained.

(1) The use of clip is most effective( 1/10~1/30 reduced)

(2) The use of dummy tube is effective a little. But the large vibration occurs when the flow rate is over \(Q=1000\text{m}^3/\text{h}\) and this is inevitable. Furthermore it is able to occur at the lower flow rate.

(3) Fluid elastic instability occurred in the tube bundle with different natural frequencies \(f_1\) and \(f_2\) (\(f_1>f_2\)) vibrates with the frequency \(f\) which is different with \(f_1\) and \(f_2\) (\(f_2<f<f_1\))

(4) The wave form of fluid elastic instability in case of existing the clearance between the tube and the hole of the baffle has a constant amplitude. But it is not sinusoidal and vibrates with the regular impact.
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Appendix1: Natural frequency of dummy tube (in air)

The natural frequency of beam in both ends fixed is given as follows.

\[
f_n = \frac{\lambda_n^2}{2} \sqrt{\frac{EI}{\rho A}}
\]

where \( \lambda_1 = 4.73 \), \( \lambda_2 = 7.853 \)

(1) Dummy tube

From \( d_o = 25.4mm \), \( d_i = 22.2mm \), \( E = 1.156 \times 10^{11} Pa \), \( \rho = 4.5 \times 10^3 kg/m^3 \), \( L = 1m \)

\( f_1 = 152.3Hz \), \( f_2 = 420Hz \) can be obtained and these coincides with natural frequencies of the first and the second modes. Fig.A1 shows the coherence of Fig.9. the value 128Hz can not be regarded as the natural frequency.

(2) Dummy rod

\( d_o = 20mm \) and other values are the same. Then \( f_1 = 90.3Hz \)

![Fig.A1 Coherence of hammering test](image)

Appendix2: Derivation of natural frequency and logarithmic decrement for dummy tube and dummy rod in water

(1) Dummy tube

Dimensions of the dummy tube are as follows.
Outer diameter : \( d_o = 25.4\, \text{mm} \), Thickness : \( t = 1.6\, \text{mm} \), Inner diameter : \( d_i = 22.2\, \text{mm} \)

Mass of tube at unit length : \( m_i = 0.5383\, \text{kg/m} \),

Added mass at unit length : \( m_w = 0.5194\, \text{kg/m} \),

Density of sea water : \( \rho_w = 1025\, \text{kg/m}^3 \)

Where, \( m_w \) was calculated by \( \pi \rho_w (d_o / 2)^2 \).

Mass of sea water in tube at unit length : \( m_w = 0.3968\, \text{kg/m} \)

Natural frequency in air : \( f_a = 144.6\, \text{Hz} \) (Experimental value)

As \( f_a \) is regarded as the natural frequency in air, the natural frequency in sea water \( f \) becomes

\[
f = f_a \times \sqrt{\frac{m_i}{m_i + m_w + m_w}} = 144.6 \times \sqrt{\frac{0.5383}{0.5383 + 0.5194 + 0.3968}} = 88.0\, \text{Hz} \quad \text{due to mass ratio.}
\]

On the other hand, the logarithmic decrement was obtained in the manner that the logarithmic decrement in air 0.0047(Measured value) plus the difference 0.026(Measured value) between in air(0.094) and in water(0.12) of the real tube(THIS can be obtained in air and in water). Namely, \( \delta = 0.0047 + 0.026 = 0.031 \).

(2) Dummy rod

Outer diameter : \( d_o = 20.0\, \text{mm} \)

Mass of tube at unit length : \( m_i = 1.414\, \text{kg/m} \)

Added mass at unit length : \( m_w = 0.322\, \text{kg/m} \)

Natural frequency in air : \( f_a = 90.3\, \text{Hz} \) (Calculated value)

The natural frequency in sea water \( f \) can be obtained from mass ratio

\[
f = f_a \times \sqrt{\frac{m_i}{m_i + m_w}} = 90.3 \times \sqrt{\frac{1.414}{1.414 + 0.322}} = 81.5\, \text{Hz}
\]

The logarithmic decrement is not found because the measurement was not carried out.

Appendix 3: Derivation of natural frequency in case of connecting two tubes with different natural frequencies by water(connecting stiffness)

The analytical model is shown in Fig. A2. Two masses is connected by the stiffness due to the water. The equations of motion are given as follows.

\[
m_1\ddot{x}_1 + k_1x_1 + k(x_1 - x_2) = 0
\]

\[\text{(A2)}\]

\[
m_2\ddot{x}_2 + k_2x_2 + k(x_2 - x_1) = 0
\]

\[\text{(A3)}\]

Rewriting

\[
\ddot{x}_1 + \omega^2 x_1 + \alpha^2 x_1 - \alpha^2 x_2 = 0
\]

\[\text{(A4)}\]

\[
\ddot{x}_2 + \omega^2 x_2 + \alpha^2 x_2 - \alpha^2 x_1 = 0
\]

\[\text{(A5)}\]

The characteristic equation can be obtained as follows.
\[ \lambda^2 - (\omega_1^2 + \omega_2^2 + \alpha_1^2 + \alpha_2^2)\lambda + (\omega_1^2 + \alpha_1^2)(\omega_2^2 + \alpha_2^2) - \alpha_1^2\alpha_2^2 = 0 \]  
\hspace{1cm} \text{(A6)}

Where, \( \omega_1^2 = k_1/m_1, \ \omega_2^2 = k_2/m_2, \ \alpha_1^2 = k/m_1, \ \alpha_2^2 = k/m_2 \).  

Then, in case of two natural frequencies of tube1 and tube2 being the same, namely \( \omega_1^2 = \omega_2^2 \), \( \alpha_1^2 = \alpha_2^2 \), The characteristic equation becomes

\[ \lambda^2 - 2(\omega_1^2 + \alpha_1^2)\lambda + (\omega_1^2 + \alpha_1^2)^2 - \alpha_1^4 = 0 \]

Modifying this equation,

\[ (\lambda - \omega_1^2)(\lambda - \omega_1^2 - 2\alpha_1^2) = 0 \]  
\hspace{1cm} \text{(A7)}

The analytical results are shown in Fig.13.

**Appendix4: Connecting stiffness(By Tanaka&Takahara)**

In Ref.(7), the experiment in water was carried out by using lattice arrangement of tubes with P/D=1.33 in shown in Fig.A3. Where D=30mm and L=300mm. The cylinder(0) located in center was free and other tubes(U,L,D,R) were fixed. The range of Reynolds number based on the gap velocity is 6000 ~24000. The vibrational displacement was given and the force was measured by the strain gauge.

\[ F = (1/2)\rho V^2 CX_0 \sin(2\pi ft + \phi) \]  
\hspace{1cm} \text{(A8)}

Where, \( X_0 \) is the vibrational displacement and \( V \) is the gap velocity. Then the spring constant is given by the same phase component to \( X_0 \).

\[ K = (1/2)\rho V^2 C \cos\phi \]  
\hspace{1cm} \text{(A9)}

The results of Ref.(8) are shown in Fig.A4. The axis of abscissa shows the reduced velocity and vertical axis shows the fluid force coefficient. In 4.3, as the gap velocity is \( V=1.44 \sim 1.58 m/s \), the reduced velocities become \( V_R = V/\bar{d} = (1.44)/(66 \times 0.0254) = 0.86 \) for the dummy tube and \( V_R = V/\bar{d} = (1.58)/(66 \times 0.020) = 1.2 \) for the dummy rod.

From Fig.A4, \( C \cos \phi \) can be estimated 40 both directions of the horizontal(X) and the vertical(Y). So \( K = 0.5 \times 1025 \times 1.44^2 \times \cos(0) = 4.25 \times 10^4 (N/m)/m \) for the dummy tube and \( K = 0.5 \times 1025 \times 1.58^2 \times \cos(0) = 5.12 \times 10^4 (N/m)/m \) for the dummy rod. The meaning of the symbol :for example in Fig.A4, \( CYOX \) shows the fluid force coefficient of Y direction when the tube displaces in X direction.

![Fig.A3 Experimental model](image1)

![Fig.A4 Experimental result](image2)