Flow Induced Vibration of Shell & Tube Type Heat Exchanger*
(1st Report, Understanding of Phenomena)

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Abstract
This paper describes the phenomenon of flow-induced vibration in a shell & tube type heat exchanger. This type 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 vibrations, fluid elastic instability and buffeting. Experiments are carried out by use of an actual heat exchanger in order to clarify the cause of tube vibration. The vibration of a lot of tubes is measured for various flow rates and the natural frequency and its damping ratio are also measured. As a result, the cause of the vibration of tubes is not the fluid elastic instability and the vortex-induced vibration but the buffeting due to the turbulence of upstream flow. The main reason is that these tubes are supported by many baffle plates with clearance and then the vibration system becomes the nonlinear.

Key words: Flow Induced Vibration, Turbulence, Flow Meter, Fluid Elastic Instability, Heat Exchanger, Buffeting, Vortex Induced Vibration

1. Introduction
In several types of heat exchangers, a shell & tube type heat exchanger has been used to evaporate a liquid propane gas (LPG) by sea water when it is loaded on. This structure is shown in Fig.1. A lot of U bend tubes are supported by several baffle plates and the LPG flows in the tubes. The sea water is introduced from the inlet of the shell and suctioned from the exit and the heat exchange is performed in this process. The diameters of holes made on baffle plates are 0.4 mm larger than those of tubes to allow the elongation due to heating. Then the tube vibration becomes a non-linear system with the clearance and behaves with complexity.

A lot of investigations of this kind have been carried out for many years. Fricker studied the unstable and the impulsive behaviors of tubes analytically and experimentally from the point of view that the clearance between the tube and the support could generate fluid elastic instability easily(1). Gordon and Lebret studied the effect of clearance on fluid elastic instability experimentally(2), Cai et.al proposed a mathematical model on fluid elastic instability in non-uniform flow(3). Weaver and Parrondo evaluated the unstable behavior of fluid elastic instability under the conditions of many supports and non-uniform flow(4). However, the natural frequency of the tube with the clearance cannot be generally obtained theoretically and it is considered to be hard to occur fluid elastic instability due to the large damping.
Generally, tube vibrations are classified into three kinds such as (1) the vortex induced vibration due to the Karmann vortex, (2) fluid elastic instability, and (3) random vibration due to flow turbulence called as buffeting. And there are many countermeasures of these vibrations and effects of them were confirmed. However little study has been performed on the detail of the tube vibration and it is not clear what kind of the vibration occurred easily.

The purpose of this study is to clarify the causes of tube vibrations and to confirm the countermeasures of them. Whereas, vibrations of eight tubes selected among a lot of tubes in the shell are measured to deepen our understanding of this phenomenon.

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 measured tubes are also shown in Fig.2. The number of tubes were two hundreds and the tube arrangement is triangle (P/d₀=1.3) as shown in Fig.2 (b). Acceleration sensors are attached at the position 30 mm apart from the fixed end and perpendicular direction to the flow. The tube diameter and its thickness were 25.4 mm and 0.8 mm, respectively. The tube material wasTitan and the diametral clearance between the tube and the hole of baffle was 0.4 mm. The measured tubes were selected by the following considerations,

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

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

The tube ⑧ is the same as ① and indicates the vibration to the flow direction.
2.2 Measurement items

Measurement items are as follows.

(1) Vibration characteristics of tube (Natural frequency and damping ratio)
(2) Vibration response of tube at each flow rate

In this experiment, the natural frequency of the tube was obtained by hitting the outside of the shell (A) by an impulse hammer as shown in Fig.1. And the damping ratio was obtained by a half power method. The range of flow rate of water is \(Q=520 \sim 1100 \text{ m}^3/\text{h}\), whereas it was measured by an ultrasound flow meter.

3. Experimental results and their considerations

3.1 Natural frequency

Figure 3 shows an example of the tube natural frequency (Tube ① in water). Natural frequencies below 100Hz are 6.6Hz, 16.1Hz and 53.5Hz.

3.2 Vibration modes

It is difficult to obtain the natural frequency of the tube supported with clearance theoretically. Here, Noticing that most of the failure occurs at first plate from the end plate “A”, the natural frequency of the tube which length is 1000mm and has the boundary condition “Fix-Simply Supported” can be obtained as 63.9Hz by calculation of Eq.(1). This value is relatively close to the measured value 53.5Hz. Therefore the vibration mode can be assumed to be one as obtained under the boundary condition “Fix-Simply Supported”.

\[
f_o = \frac{n^2 \beta^2}{L^2} \sqrt{\frac{EI}{m}}
\]

Where values used here are as follows.

\(d_o = 25.4\text{mm} \), \(d_i = 23.8\text{mm} \), \(L = 1000\text{mm} \), \(\rho_i = 4.5\times10^3 \text{ kg/m}^3 \), \(\rho_o = 1025 \text{ kg/m}^3 \)
3.3 Logarithmic decrement

The damping ratio of this mode is calculated by the half power method and becomes 0.12 as shown in Fig. 4.

3.4 Vibrations of each tube in flow and their causes

Figure 5 shows time histories of vibration (acceleration) of each tube at Q=950m³/h. From Fig. 5, it is revealed that the vibration of tubes ①, ②, and ⑦ are larger than the other tubes and these amplitudes are not symmetrical. This means that the vibrations of the tube bank do not vibrate uniformly, whereas the tubes at the upstream vibrate larger than those at the downstream.

In addition, the spectrum of vibration of tube ① at high flow rate (Q=950m³/h) is shown in Fig. 6. From these results, it is presumed that this vibration is not caused by fluid elastic instability but from random vibration due to flow turbulence. As the vibration is random, we attempt to calculate the overall vibration of each tube as shown in Fig. 7. This figure shows that the vibration of tubes ①, ②, ⑦ and ⑧ suddenly increases as the flow rate is over Q=800m³/h. On the other hand, the vibration of the tubes ④, ⑤ and ⑥ increase with the 1.5~2.5 power of the flow rate(Q1.5~Q2.5). The former reminds us the occurrence of fluid elastic instability. However it is denied due to the random characteristics and the result that the vibration amplitude (displacement) of dominant frequency components gently increases with the flow rate as shown in Fig. 8.

The critical flow velocity can be evaluated by Connor’s expression (2) as follows.

\[ \frac{U}{fd_0} = K \sqrt{\frac{m\delta}{\rho d_0^2}} = K\sqrt{\delta_s} \]

where, \( m \): Mass of tube per unit length containing inner fluid and added mass (0.798 kg/m), \( U \): Gap velocity, \( f \): Natural frequency of tube in water (53.5 Hz), \( d_0 \): Diameter of tube (0.0254 m), \( \delta \): Logarithmic decrement of tube in water (0.12), \( \rho \): Fluid density (1025 kg/m³). Then, the reduced velocity becomes,

\[ \frac{U}{fd_0} = K \sqrt{\frac{m\delta}{\rho_d d_0^2}} = 3.3 \times \sqrt{\frac{0.7976 \times 0.12}{1025 \times 0.0254^2}} = 1.255 \]
Therefore,

\[ U_{cv} = 1.255 \times f d_w = 1.255 \times 55 \times 0.0254 = 1.75 \text{ m/s} \]
If the flow rate is constant, the flow velocities at each position are different due to the difference of the flow area as shown in Fig.2. Therefore the flow rates vary in the range of $Q=1170\text{m}^3/\text{h} \sim 1490\text{m}^3/\text{h}$ at each position when the flow velocity is $1.75\text{m/s}$. The result shows that fluid elastic instability does not occur in the flow rate range $0 \sim 1100\text{m}^3/\text{h}$. Because the critical flow velocity is larger than the examined maximum flow rate in this experiment, it is concluded that this phenomenon does not occur from fluid elastic instability.

Next, the cause of vibration can be considered from the random vibration due to turbulence generated by the impinging baffle plate as shown in Fig.1. Figure 9 shows the spectrum of the vibration of tube $\text{(1)}$ at $Q=520\text{m}^3/\text{h}$ where the tube vibration is small. The difference can be seen in the range of over 30Hz in comparing with Fig.6 and Fig.9. This means that the vibration becomes large and the tube impinges the baffle when the flow rate increases. This results in vibration with many higher modes.

3.5 Ground of random vibration and its consideration

It is presumed that the cause of vibrations of tube may be from the random vibration due to turbulence generated by the impinging baffle plate located at the upstream of the tube bundle. This assumption is considered based on the experimental facts as follows.

(1) Vibrations of tubes are large at the most upstream
(2) Vibration spectra show the broad band characteristics

Blevins [6] proposed the expression to predict the vibration response due to turbulence
of flow. It is defined as follows;

$$\frac{Y_{rms}}{d} = \frac{1}{16\pi^{3/2}} \frac{\rho d^2}{m} \left( \frac{U}{f_d} \right)^{1.5} J_f \Phi(f_d/U)^{3/2} \phi(x)$$

(3)

where

$$\Phi(\tilde{f}) = 4 \times 10^{-4} \tilde{f}^{-0.5} \quad 0.01 \leq \tilde{f} \leq 0.2$$

(4)

$$\Phi(\tilde{f}) = 3 \times 10^{-6} \tilde{f}^{-3.5} \quad 0.2 \leq \tilde{f} \leq 3$$

(5)

Where

- $m$: Modal mass
- $\phi(x)$: Vibration mode
- $d$: Outer diameter of tube
- $\rho$: Fluid density
- $f$: Natural frequency of tube
- $\zeta$: Damping ratio

**Fig. 9 Spectrum of acceleration in case of Q=520m$^3$/h**

**Fig. 10 Non-dimensional force power spectrum**
\[ U \]: Gap velocity

\[ \Phi(\tilde{f}) \]: Non dimensional spectrum of fluid force

\[ \tilde{f} = \frac{fd}{U} \]: Reduced velocity

\[ J \]: Joint acceptance (Maximum=1.0)

In this prediction, the outer diameter of the tube is used as the characteristic dimension in Eq. (3) and Eq.(5). However in this phenomenon, it is reasonable to use the turbulence scale \( L_s \) as the characteristic dimension, because the fluid exciting force is resulted from the impinging baffle plate set at upstream.

Having tried to evaluate the response by use of Eq.(3) and \( \Phi=0.02 \) (Recommended value by reference[6]), the maximum displacement became \( 0.3 \) mm at \( Q=1100m^3/h \). This value may not be able to produce the failure. Because it is within the diametral clearance \( 0.4mm \).

Ishihara and Aoki studied the random vibration of the thermo-well in the past[7]. Where the fluctuating velocity just at downstream of a valve set in a pipe was measured to examine the characteristic of flow turbulence. Figure10 shows the normalized power spectrum density of the fluctuating velocity of the flow at just downstream of the valve. The scale of this piping system(\( \Phi=340 \)) is almost the same as one(\( \Phi=330 \)) used in the present experiment as shown in Fig.1a. Figure10 shows the cases of the valve opening ratio being \( \theta=20^\circ \) and \( \theta=40^\circ \). The same spectra were also obtained in other opening ratios. In Fig.11, the abscissa indicates the reduced frequency based on \( L_s \) and the normalized power spectrum density is kept constant by the reduced frequency \( \frac{fL_s}{U} \) being \( 0.2 \) and reduces \(-2\) power of the reduced frequency when \( fL_s/U \) is over 0.2. This reduced frequency(0.2) is called the critical reduced frequency. Therefore the critical reduced frequency based on the characteristic dimension \( d \) is necessary to rewrite as follows.

From \( fL_s/U = 0.2 \)

\[ \frac{fd}{U} = (\frac{fL_s}{U}) \cdot \frac{(d}{L_s}) = 0.2(\frac{d}{L_s}) \quad \quad \quad (5) \]

According to the reference [7], the turbulence scale \( L_s \) is \( 40mm \). Then
The gap velocity $U$ at the most upstream becomes 1.65 m/s when $Q=1100\, \text{m}^3/\text{h}$, $fd/U$ is 0.85 and $fLs/U$ becomes 1.34. The value of $\Phi$ is evaluated about 0.5 from Fig.10. Therefore the random vibration is evaluated by using Eq.(3), whereas $\Phi=0.5$. The maximum displacement became 1.5 mm. This value may be possible to be able to produce the failure. Because it is larger than the diametral clearance 0.4 mm.

In the present phenomenon, it can be seen that the O.A value of the acceleration suddenly increases at high flow rate ($Q=950\, \text{m}^3/\text{h}$) comparing with at a little flow rate ($Q=520\, \text{m}^3/\text{h}$). From Fig.7, the O.A. value of the tube is 3.5 m/s$^2$ at $Q=520\, \text{m}^3/\text{h}$ and 17.5 m/s$^2$ at $Q=950\, \text{m}^3/\text{h}$. Now provided that the fluid force is proportional to $Q^2$, then the acceleration O.A. value becomes 11.7 m/s$^2$ at $Q=950\, \text{m}^3/\text{h}$. This fact suggests that the acceleration value is not due to only the flow rate but due to various parameters such as the vibration mode, the natural frequency, the configuration of the impinging baffle plate and its located position.

By comparing Fig.6 and Fig.9, the difference can be seen in the frequency region over 30 Hz. As the spectrum shifts the left with the flow rate, the fluid force coefficient and the dynamic pressure become large. It makes the large fluid force.

The random vibration is the phenomenon that only the natural frequency component is excited due to the random force. In the present vibration system, the numerous natural frequencies exist. Then all of the natural frequency components arise and the spectra become broad band shape at the large flow rate.

3.6 Lissajous of tube
Figure 11 shows the vibration locus of the tube. It is shown that the shape of the Lissajous is ellipse. This means that the tube impinges and changes the direction, and the vibration in flow direction is larger than that in perpendicular direction of the flow.

4. Conclusions
The vibrations of a lot of tubes were measured by using the actual shell & tube type heat exchanger in order to grasp the cause of vibrations. The results are summarized as follows.

(1) In case of existing the clearance between the tube and the hole of the baffle plate, the tube impinges on the baffle plate over some extent of flow rate and it is observed that the acceleration increases discontinuity at a glance. The acceleration increasing occurs due to numerous natural frequency components excited by the strong impinging.

(2) The location of the tube array with large vibrations is the most upstream and the vibrations do not occur in tubes located downstream. The cause of the large vibrations is the random vibration due to turbulence.

(3) Fluid elastic instability, which is often experienced in the tube bundle, is hard to occur in the case of with clearance comparing the case of without clearance. This is due to not existing the clear natural frequency and due to a large damping generated at the clearance.

References
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Appendix

Table 1a. Comparison of Re bet. Ref(7) and present

<table>
<thead>
<tr>
<th>Va (m/s)</th>
<th>Re (Ref.7)</th>
<th>Q (m³/h)</th>
<th>Vw (m/s)</th>
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Fig. 1a Comparison between both test equipments