MECHANICAL INTEGRITY DESIGN FOR FLARE SYSTEM
ON FLOW INDUCED VIBRATION

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ABSTRACT

In recent baseload LNG plants of larger capacity, the risk of vibration for large diameter pipes increases, especially for a flare system with relatively high velocity. One such kind of piping vibration is FIV (Flow Induced Vibration) caused by turbulence. It is known that FIV occurs at locations where high turbulence is present such as downstream of a combining tee. However, no paper has yet been published that focuses in detail on the characteristics of this FIV at the tee. In this paper, a study is presented on FIV, downstream of a 45 degree combining tee. Experiments and a CFD / FEA approach were used to investigate the characteristics of FIV. The results of the experiments show that the shell mode vibration of the pipe is larger than the beam mode vibration. This finding contradicts the previous understanding that FIV generally initiates beam mode vibration. The CFD results show that large turbulence energy occurs a little downstream from the combining point. Random vibration analysis was executed with FEA based on the experimental and CFD results, which demonstrated that the CFD/FEA approach is practical method to evaluate the overall vibration characteristics of FIV. In consideration of the basic characteristics of FIV, an index is proposed, called “Vibration Velocity Number”, which expresses the magnitude of the vibration velocity. The experimental results show that this index is effective in evaluating the vibration risk of FIV for wide range of branch area ratios to the main.

INTRODUCTION

In recent baseload LNG plants of larger capacity, the risk of vibration for large diameter pipes increases, especially for a large diameter flare system with relatively high velocity. One such kind of piping vibration is FIV (Flow Induced Vibration) caused by turbulence. It is known that the FIV occurs at locations where high turbulence is present such as downstream of a combining tee [1]-[3] or downstream of a bend [4]. For FIV downstream of a bend, Hirota et al. investigated the non-dimensional relationship between pressure fluctuation and fluid momentum energy with experiments and vibration analysis [4]. However, there is no publication which expresses in detail the characteristics of the vibration phenomena of FIV downstream of a combining tee. The author has recently investigated this FIV by analytical and experimental methods [2], [3], and found that the shell mode vibration of the pipe is larger than the beam mode vibration with large diameter and relatively thinner wall pipes. This finding contradicts the previous understanding that FIV generally initiates beam mode vibration [1]. The author has also demonstrated that CFD (Computational Fluid Dynamics) / FEA (Finite Element Analysis) analysis is effective in investigating the characteristics of FIV and introduced an index called “Vibration Velocity Number” which expresses the magnitude of the vibration velocity.
This paper presents the study results of FIV downstream of a 45 degree combining tee using experiments and CFD / FEA analysis. It also demonstrates the concepts of the “Vibration Velocity Number” index which is derived from the basic random vibration theory and expresses the magnitude of the vibration severity. The experimental results show that this index is effective in evaluating the vibration stress of FIV for a wide range of process conditions and the branch flow area ratios to the main pipe.

**METHOD OF EXPERIMENT**

Figures 1 and 2 show the experimental facility which has an air chamber at the upstream end. The piping system is connected to the air chamber through a ball valve and a RO (Restriction Orifice) and there is a 45 degrees tee downstream of the orifice. After opening the ball valve, air will flow from the pressurized air chamber into the tee section. Two types of sensor, strain gage type (TP1-TP4) and piezoelectric type (TPA1-TPA5), are used to measure the absolute pressure and pressure fluctuation, respectively. Twenty one strain gages for the circumferential direction are attached downstream of the combining point, as shown in Figure 3, in order to measure the shell mode vibration in detail. Table 1 shows the five experimental cases that were executed to investigate the effect of the branch flow area ratio to the main pipe.

**Figure 1. Experimental Facility (4"x1.5")**

**Figure 2. Experimental Facility around Combining Tee (4"x1.5")**

**Figure 3. Measuring Points for Strain**
Table 1. Experimental Cases

<table>
<thead>
<tr>
<th>Case Definition</th>
<th>Main Pipe</th>
<th>Branch Pipe</th>
<th>Branch Area Ratio to Main</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>O.D.</td>
<td>Wall. Thick.</td>
<td>D/t</td>
</tr>
<tr>
<td>4&quot; x 4&quot;</td>
<td>114.3 mm</td>
<td>2.1 mm</td>
<td>54.4</td>
</tr>
<tr>
<td>4&quot; x 3&quot;</td>
<td>114.3 mm</td>
<td>2.1 mm</td>
<td>54.4</td>
</tr>
<tr>
<td>4&quot; x 1.5&quot;</td>
<td>114.3 mm</td>
<td>2.1 mm</td>
<td>54.4</td>
</tr>
<tr>
<td>4&quot; x 3/4&quot;</td>
<td>114.3 mm</td>
<td>2.1 mm</td>
<td>54.4</td>
</tr>
<tr>
<td>4&quot; x 1/4&quot;</td>
<td>114.3 mm</td>
<td>2.1 mm</td>
<td>54.4</td>
</tr>
</tbody>
</table>

**METHOD OF NUMERICAL ANALYSIS**

Figure 4 shows the analytical model for the 4" x 1.5" case. First, CFD analysis is executed using the commercial code STAR-CCM+ V5.2 to obtain an area of high turbulence where high pressure fluctuations occur. The CFD results in Figure 5 show that the area of high turbulence occurs a little downstream of the combining point and this indicates that the high turbulence is caused by the jet flow from the branch impinging on the main pipe wall. Corresponding to this area of high turbulence, the exciting force area is determined in the FEA model. Figure 6 shows the PSD (power spectrum density) of the pressure fluctuation at TPA4 (refer to Figure 2) for the 4"x1.5" case with 26 bara at the RO upstream. This pressure fluctuation is treated as the exciting force acting on the area of high turbulence as explained above and the random vibration analysis is executed by using the commercial FEA code, ABAQUS V6.10.
RANDOM VIBRATION THEORY

Statistical processing is required [5] in order to evaluate the random vibration in which the overall RMS (Root Mean Square) of the vibration is equivalent to the standard deviation. For simplicity a simple vibration model is considered with a random excitation force, \( F \), expressed by equation (1).

\[
m \ddot{x} + c \dot{x} + kx = F(t) \tag{1}
\]

Here, \( m \), \( c \) and \( k \) and \( F \) are the mass, damping constant, spring constant and displacement, respectively. PSD (Power Spectrum Density) of the vibration displacement, \( S_x \) can be written by equation (2).

\[
S_x(\omega) = |H(\omega)|^2 \cdot S_w(\omega) \tag{2}
\]

Here, \( \omega \), \( H \) and \( S_w \) are the angular frequency, transfer function and PSD of exciting force, respectively. Transfer function, \( H \) can be written by equation (3).

\[
H(\omega) = \frac{1}{\sqrt{(k - m\omega^2)^2 + (c\omega)^2}} \tag{3}
\]

Integrating of equation (2) derives equation (4) [5].

\[
x_{rms}^2 = \int_{-\infty}^{\infty} S_x(\omega) d\omega = \int_{-\infty}^{\infty} |H(\omega)|^2 \cdot S_w(\omega) d\omega = \frac{W_0}{4kc} \tag{4}
\]

Here, \( x_{rms} \) is the root mean square of the displacement and \( W_0 \) is a coefficient which shows the PSD of the exciting force at a certain frequency and is assumed to be constant. \( k \) and \( c \) can be written as follows;

\[
k = (2\pi f_n)^2 m, \quad c = 2\zeta \sqrt{mk} = 2\zeta m\omega_n = 4\pi\zeta m f_n \tag{5}
\]

Here, \( f_n \), \( \omega_n \) and \( \zeta \) are the natural frequency, natural angular frequency and damping ratio, respectively. From equations (4) and (5), equation (6) is derived.

\[
x_{rms} \propto \sqrt{\frac{W_0 f_n}{m f_n^2}} \propto \sqrt{\frac{W_0 f_n}{m f_n^2}} \tag{6}
\]

In equation (6), \( \zeta \) is assumed to be constant and \( f_g \) which expresses the frequency range of the excitation force is used instead of \( f_n \). Generally From the vibration velocity, \( V_{rms} \) is equal to \( 2\pi f_n x_{rms} \). And the vibration stress is basically proportional to the vibration velocity. Therefore, the following equation can be derived;

\[
\sigma_{rms} \propto V_{rms} = 2\pi f_n x_{rms} \propto \sqrt{\frac{W_0 f_g}{m f_n}} \tag{7}
\]
Here, $\sigma_{rms}$ is the root mean square of the vibration stress. The excitation force is generally proportional to the product of the pressure fluctuation and excitation area which would be proportional to the total fluid momentum $\rho_gAV^2$ at the end of the branch just before the combining point, here, $\rho_g$, $A$ and $V$ are the fluid density, flow area of branch and velocity at the branch, respectively. Therefore, $\rho_gAV^2$ can be used instead of $\sqrt{W_{r_0}}$ in equation (7). The mass of the pipe, $m$ can be expressed as $\pi \rho_p D^2 t$, here, $\rho_p$, $D$ and $t$ are the density of the pipe material, the outside diameter of the main pipe and the wall thickness of the main pipe, respectively. Thus the equation (8) can be derived.

$$\sigma_{rms} \propto V_n = \frac{\rho_g AV^2}{\pi \rho_p D^2 t f_n}$$  \hspace{0.5cm} (8)

In equation (8), an index, $V_n$ expresses the magnitude of the vibration velocity which would show the vibration stress, i.e. the vibration severity. From this point of view, the index $V_n$ is named as “Vibration Velocity Number”.

Where critical flow occurs with sonic speed at the end of the branch pipe just before entering the main pipe, there would be a pressure discontinuity at the combining point with a rapid expansion of air. This rapid expansion of air would have the effect of increasing the turbulence energy at the combining point. The turbulence energy, including this rapid expansion effect, could be assumed to be proportional to the total pressure change $(1/2)\rho V^2 + \Delta p$ at the combining point; here, $\Delta p$ is the pressure discontinuity at the combining tee for the critical flow condition. From this assumption, equation (8) can be modified to equation (9) in case of the critical flow occurrence at the end of the branch.

$$\sigma_{rms} \propto V_n = \frac{\rho_g AV^2 + 2 \Delta p A}{\pi \rho_p D^2 t f_n}$$  \hspace{0.5cm} (9)

RESULTS AND DISCUSSION

Figures 7 and 8 show typical examples of the measured time histories of pressure fluctuation and strain respectively. As shown in these figures the time history data seem to have random vibration characteristics with no periodic vibration characteristics. Figure 9 shows the probability distribution of the measured strain. Figure 9 shows that the measured probability distribution is almost close to the normal probability distribution. This is one of the typical characteristics of random vibration [5].

![Figure 7. Time History of Measured Pressure Fluctuation](image1)

**Figure 7. Time History of Measured Pressure Fluctuation**

(HPA4: Upstream Pressure of RO is 28 bara, 4"x1.5")

![Figure 8. Time History of Measured Strain](image2)

**Figure 8. Time History of Measured Strain**

(ST2: Upstream Pressure of RO is 28 bara, 4"x1.5")
Figure 9. Probability Distribution of Measured Strain
(ST2: Upstream Pressure of RO (TP1) is 28 bara, 4”x1.5”)

Figure 10 shows typical examples of PSD of the pressure fluctuations upstream (TPA1) and downstream (TPA4) of the combining point. As shown in this figure, the pressure fluctuation downstream of the tee (TPA4) is considerably larger than that upstream of the tee (TPA1). This means the most of the large pressure fluctuation would be generated by the turbulence just after the combining point.

Figure 10. Power Spectrum Density of Pressure Fluctuations at Upstream and Downstream of Tee
(Upstream Pressure of RO (TP1) is 26 bara, 4”x1.5”)

Figures 11 and 12 show the shapes of the vibration modes obtained by measurement and analysis, respectively. In Figure 11 the vibration strains at the points from 22 to 40 are assumed to be equal to the measured vibration strains at the points from 1 to 21. These figures show that the analysis agrees well with the measured data, not only for the vibration mode shapes but also vibration frequencies. Figure 13 shows the overall vibration stresses obtained by the measurement and analysis. From this figure the analytical vibration stress has good agreement with the measured one though the analytical vibration stress is a little larger than that of the measured one.
Figure 11. Vibration Modes (Measurement, 4”x1.5”)
(Upstream Pressure of RO (TP1) is 26 bara)

Figure 12. Vibration Modes (Analysis, 4”x1.5”)

Figure 13. Comparison of Vibration Stresses between Analysis and Experiment (4”x1.5”)

Figure 14 shows the relationship between the measured vibration stress and vibration velocity number, $V_n$. In this figure $Ab/Am$ is the branch flow area ratio to the main pipe, and the maximum measured strain among the 21 measuring points is depicted. The proposed vibration index $V_n$, "Vibration Velocity Number", is quite effective to evaluate the order of the vibration stress for wide range of process conditions including the occurrence of critical flow and a branch area ratio from 0.011 to 1.0.
CONCLUSION

The FIV (Flow Induced Vibration caused by turbulence) at 45 degree combining tee was investigated by measurement and analysis. The results drew the following conclusions:

(1) The turbulence generated at the combining point creates the large pressure fluctuation and results in shell mode vibration in the case of relatively thinner wall pipes.
(2) The vibration has random characteristics and CFD/FEA approach with random vibration analysis is effective to evaluate the vibration phenomena.
(3) The proposed vibration index called “Vibration Velocity Number”, Vn is quite effective to evaluate the order of the vibration stress for a wide range of process conditions including the occurrence of critical flow and a branch area ratio from 0.011 to 1.0.

REFERENCES