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Damping estimation of a loosely supported single U-bend tube used in a steam generator

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Abstract. An evaluation method for estimating the damping of loosely supported single U-bend tube used in a steam generator colliding with a support plate is proposed. First, we performed experimental modal analysis and obtained natural frequencies, modes and damping without collision by Impulse Modal Test and then analysed natural frequencies, modes and damping with collision employing FEM analysis taking account of the collision force. In modelling the characteristics of the collision force, we applied Bijlaard’s model for the spring constant and assumed hysteresis. After that, we performed experiments for measuring the damping ratio by changing the gap size and the support plate position. Comparison between calculated and experimental results is made which shows good agreement. Experimentally observed fact showing damping coefficient increases with the initial amplitude is well explained by theoretical model.

1. Introduction

It is well known that heat exchanger tubes used in a steam generator start to vibrate beyond critical speed. This phenomenon is called fluid-elastic vibration and the critical speed is usually estimated by Connors’ equation¹,

\[
\frac{U_c}{fD} = K \left[ \frac{m\delta}{\rho D^2} \right]^{1/2}
\]

where \(U_c, f, D, m, \delta, \rho\) denote critical velocity, natural frequency, outer diameter, mass per unit length, damping ratio of single tube, density of the fluid and \(K\) denotes coefficient called critical factor. In order to estimate critical velocity accurately, evaluation of damping ratio under collision is a key issue²,³.

To evaluate damping ratio under collision, restitution factor is often introduced and model target selected is a cantilever pipe⁴. The idea of restitution coefficient is convenient however it lacks energy dissipation process during collision. In this study, to propose more accurate method, we propose a method using energy loss coefficient proposed by Bijlaard⁵,⁶ and single U bend tube as a target.

First, we performed experimental modal analysis and obtained natural frequencies, modes and damping without collision by Impulse Modal Test and then analysed natural frequencies, modes and damping with collision employing FEM analysis taking account of the collision force. Finally, analytical results are compared with experiments.
2. Time history analysis of single U bend tube
As shown in Fig.1, we first performed experimental modal analysis and obtained natural frequencies, modes and damping without collision by Impulse Modal Test. Information on target object is tabulated in Table 1.

![Fig. 1 Single U bend tube and Impulse Modal Test](image)

| Material                          | Stainless Steel |
|----------------------------------|-----------------|
| Height                           | 1510mm          |
| Outside Diameter of the tube ($D_T$) | 22.23mm        |
| Inside Diameter of the plate ($D_B$) | 24.23mm        |
| Thickness of the tube            | 1.27mm          |
| Radius of curvature of bending beam | 381.1mm        |
| Clearance between tube and hole  | 1.0mm           |

Then, time history under collision is analysed by Finite Element Method taking collision effect into consideration. In our model, collision effect is represented by a linear spring shown in Fig.2.

![Fig.2 Collision model](image)
Introducing virtual displacement as shown in Fig. 3 and assuming hysteresis characteristics of collision force as shown in Fig. 4, then radial force acting on the surface can be described by Eq. (2).

\[
F_N = \begin{cases} 
  k \times \delta & (\dot{\delta} \geq 0) \\
  \max[k\delta_{max} - k(\delta_{max} - \delta)/(1 - \alpha), 0] & (\dot{\delta} < 0) 
\end{cases}
\]  

(2)

In the above equation, \( k \) denotes spring constant determined based on Bijlaard\textsuperscript{(5),(6)}, \( \delta \) denote virtual displacement defined by

\[
\delta = \gamma - \left( D_b - D_T \right)/2
\]

(3)

where \( \gamma \), \( D_b \) and \( D_T \) denote radial displacement of a tube, inside diameter of the plate and outside diameter of the tube, respectively.

In addition, \( \delta_{max} \) in Fig. 4 is defined as corresponding virtual displacement when time derivative of \( \delta \) equal to zero and \( \alpha \) in Eq. (2) is defined as energy loss factor representing the ratio between strain energy and hysteresis damping energy.

Then, equation of motion for a single U bend tube taking collision effect into account can be written as
\[ [M]\{\ddot{U}\} + [C]\{\dot{U}\} + [K]\{U\} = \{F\} \]  \hspace{1cm} (4)

where \([M],[K]\) denote mass and stiffness matrices and \(\{U\},\{F\}\) denote nodal displacement and nodal force obtained by FEM analysis. When we calculate time history, we inserted the data obtained by experimental modal analysis without collision to damping matrix \([C]\).

In Fig.5, one example of calculated transient virtual displacement is shown.

3. Experiment with single U bend tube under collision

In Fig.6, experimental setup and coordinate system is shown. Single U bend tube is fixed by two blocks and support plate is equipped in between the top of U bend tube and blocks. Gap between support plate and U bend tube is set 1mm.

Fig.5 Transient virtual displacement

Fig.6 Experimental setup
Fig. 7 shows transient wave form in the direction of y measured on the top of U bend tube when applied initial amplitude is 2mm. One can find that the envelope of decay during collision corresponding to the first three wave forms in Fig. 7 is almost linear but not exponential. In addition, higher mode is included as shown in enlarged wave form of Fig. 7.

4. Analytical results
In Fig. 8, calculated results are shown in two cases with energy loss factor $\alpha$ is 0.25 and 0.5. In the calculation, the value of $\alpha$ was determined by logarithmic decay of time history. Comparing Fig. 7 and Fig. 8, similar trend on the envelope can be found. In case of low energy loss factor which means duration time of collision is longer than the case of high energy loss factor, fundamental period of vibration becomes shorter than the case of high energy loss factor. Therefore, we can conclude the trend that the fundamental natural frequency becomes higher is exerted by the increase of strain energy due to the local deformation when a tube collides with a support plate.

Finally, in Fig. 9, calculated and experimentally obtained damping ratio are shown against initial displacement. Both results show good agreement.
Let us explain more on the detail of vibration. In the case that initial displacement is 1.5mm and 1.7mm, U bend tube collides with a wall with one end and in the case of 2.0mm and 2.2mm, U bend tube collides with a wall with both ends. Therefore, higher damping ratio is realised in the case of larger initial amplitude.

5. Conclusion

In this paper, the method for evaluating damping ratio of single U bend tube vibration taking account of collision against support plate is proposed. Comparing calculated results with experimental results, this method is found to be effective for the purpose of accurate damping evaluation.

Reference

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