Theoretical and Experimental Study on the Stochastic Characteristics of Fluid Elastic Instability of Condenser Tubes

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ABSTRACT

In this paper, the fluid elastic instability mechanism models of tube rows and tube arrays of condensers have been made. It is considered that each vibration mode of one tube is fluid dynamically coupled to all the modes of adjacent tubes. Mechanism modeling of single tube has been accomplished, and the vibration characteristics of single tube in all-support-effective state are calculated using analytic method. The Assumed Mode Method is used in calculating characteristics of other supporting states, the results can be substituted directly into the critical velocity computational formula to get the critical velocity of cross flow. And the stochastic characteristics results are obtained by using the manufacturing errors and theoretical gap between tubes and support baffles. All the analyses above are united to calculate the probabilities of fluid elastic instability happening to single tube of every possible supporting states so that the holistic instability can be evaluated. The tube bundle simulation experiment found the characteristics of single tube in different supporting states, and explored some rules of the FIV of the tube, pipe sound pulsation and acoustic resonance.

Keywords: Fluid Elastic Instability, Critical Cross Flow Velocity, Coupling Modes, Ineffective Support, Vibration Characteristics, Probability Analysis, Tube Bundle Simulation Experiment

1. INTRODUCTION

The fluid elastic instability of condenser tubes is caused by the external cross flow. When cross flow velocity exceeds a critical value, the work done on tubes by the oscillation surpasses the damping energy consumption, the transverse vibrations of tubes will diverge with amplitude magnified. No matter how slight is the shell-side flow rate increment relative to the critical value, it will lead to a sudden increase in the amplitude of the heat transfer tubes, so that collisions with adjacent tubes and shell would damage the tubes themselves. Therefore, the fluid elastic instability is one kind of flow-induced vibration phenomena causing maximum harm to the reliability of condensers and the life of heat tubes.

This critical value is always reckoned with an empirical expression which is positively correlated with natural frequency of the tube. The theoretical and numerical methods of calculating vibration characteristics of condenser tube have been used for decades. However, condenser working condition is complicated, so that the vibration characteristics of tube cannot be figured out precisely because of the properties of tubes, the thickness of support baffles and the manufacturing error of baffle holes.

This study constructs the dynamic model of single tube, then uses analytic method and assumed mode method to calculate the vibration characteristics of tube in different support conditions; estimates the stochastic characteristics of the tube with non-effective supports along it.

And a simulation experiment of tube bundle is designed to explore some rules about the FIV, pipe sound pulsation and acoustic resonance.

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2. FULID ELASTIC THEORY

2.1 Tube Array Dynamic Model

The motion of tube bundle disturbs the shell-side fluid, and the changes in flow field would generate feedback force on the tube bundle. The unsteady flow field generated by the interaction between the tube bundle structure and the cross flow, is an extremely complex fluid-solid coupling phenomenon, so that the theoretical model being used to study in fluid elastic instability mechanism of tube bundle cannot be very precise.

Cross sections of three adjacent tubes in cross flow is shown in Figure 1. Tubes are numbered $j-1$, $j$ and $j+1$ with uniform linear density $m$ (including added mass, the internal water mainly), stiffness $k$ and damping ratio $\zeta$. The displacement of tubes parallel or perpendicular to the mean flow $U$ are referred to as $x$ and $y$, and the distance along the tube as $z$.

![Figure 1 – Tube array dynamic model](image)

\[ m\ddot{x}_j + 2m\zeta \omega_x x_j + k \omega_x^2 x_j = F_{x,j} \]
\[ m\ddot{y}_j + 2m\zeta \omega_y y_j + k \omega_y^2 y_j = F_{y,j} \]  

(1)

The forcing model of $j$ tube in elastic fluid excitation is shown as Eqs. (2), which should be made based on the following assumptions:

1. Karman vortex shedding frequency is much greater than the natural frequency of the tube, so that vortex-induced vibration amplitude is not large;

2. The vibration displacement of one tube in a tube array is only related to the adjacent two tubes.

\[ F_{x,j} = \frac{\rho U^2}{4} [K_x(x_j - x_{j+1}) - K_x(x_{j+1} - x_j) + C_x(y_j - y_{j+1}) + C_y(y_{j+1} - y_j)] \]
\[ F_{y,j} = \frac{\rho U^2}{4} [K_y(x_j - x_{j+1}) + K_y(x_{j+1} - x_j) + C_y(y_j - y_{j+1}) - C_y(y_{j+1} - y_j)] \]

(2)

$C$ and $K$ are the the partials of fluid elastic force with respect to $y$ and $x$, $\rho$ is the shell-side fluid density and $U$ is the cross velocity in the narrowest gap between tubes.

All three tubes in the array model are non-rigid, and the wind tunnel test by Connors (1) have demonstrated that all movement and direction on $j-1$ and $j+1$ tube are inverted in phase, shown as Eqs. (3), and the force on $j$ tube in one direction($x$ or $y$) is only related to the amplitude of $j+1$ tube in the other direction, which means $K_x = C_y = 0$. So the forces on $j$ tube and $j+1$ tube in different directions are shown as Eqs. (4), it shows the coupling of adjacent tubes.

\[ x_{j+1} = -x_{j-1}, \quad y_{j+1} = -y_{j-1} \]
\[ \begin{cases} F_{x,j} = \frac{\rho U^2}{2} C_x y_{j+1} & F_{x,j+1} = -\frac{\rho U^2}{2} C_x y_j \\ F_{y,j} = \frac{\rho U^2}{2} K_x x_{j+1} & F_{y,j+1} = -\frac{\rho U^2}{2} K_x x_j \end{cases} \]

(3)

(4)

The solution to Eqs. (1) can be written in harmonic form as Eqs. (5), in which $\overline{X}_j$, $\overline{Y}_j$ and $\lambda$ are constant.
Substituting Eqs. (5) and two formulas in the first column of Eqs. (4) into Eqs. (1), respectively, gives the linear equation set of the vibration amplitudes of \( j \) and \( j+1 \) tube in both \( x \) and \( y \) direction.

\[
\begin{bmatrix}
\alpha_{x,j} & 0 & 0 & -C_x \\
0 & \alpha_{y,j} & \beta_i - K_y & 0 \\
0 & C_x & \alpha_{x,j+1} & 0 \\
\beta_i & 0 & 0 & \alpha_{y,j+1}
\end{bmatrix}
\begin{bmatrix}
\bar{x}_j \\
\bar{y}_j \\
\bar{x}_{j+1} \\
\bar{y}_{j+1}
\end{bmatrix} = 0
\]

where

\[
\begin{align*}
\alpha_{x,j} &= m \lambda^2 + 2 \zeta_m \omega_x \lambda + k_{ix} \\
\alpha_{y,j} &= m \lambda^2 + 2 \zeta_m \omega_y \lambda + k_{iy} \\
C_x &= \rho U^2 C_y / 2 \\
K_y &= \rho U^2 K_y / 2
\end{align*}
\]

\[
(\alpha_{x,j} \alpha_{x,j+1} + C_x K_y \alpha_{x,j+1} + C_y K_x) = 0
\]

\[
\Rightarrow \lambda^4 + a_1 \lambda^3 + a_2 \lambda^2 + a_3 \lambda + a_4 = 0
\]

As \( \lambda \) is the index number of natural logarithm \( e \), the amplitude of tube vibration cannot be amplified infinitely, so the real part of \( \lambda \) must be negative, so the coefficients of the polynomial (7) must follow the Hurwitz Criterion (2) shown in inequality set (8).

\[
\begin{align*}
a_1 &> 0; \\
a_2 &> 0; \\
a_3 a_2 &> a_4; \\
a_1(a_2 a_4 - a_3) &> a_4^2
\end{align*}
\]

The damping ratio and the coefficients \( C \) and \( K \) should be positive, so the minimum critical cross flow velocity of the start of instability can be derived as below, where \( D \) is the external diameter of the tube, subscripts \( j \) and \( j+1 \) indicate any two adjacent tubes.

\[
U_c = \left( \frac{2m}{\rho D^2} \right)^{\frac{1}{2}} \left( \frac{C_x}{C_y} \right)^{\frac{1}{4}} \left[ \left( \frac{\omega_{x,j} - \omega_{y,j}}{\omega_{x,j+1} - \omega_{y,j+1}} \right)^2 + 4 \left( \frac{\xi_{y,j+1} - \xi_{x,j}}{\omega_{x,j+1} - \omega_{y,j+1}} \right) \left( \frac{\xi_{x,j} \omega_{x,j} + \xi_{y,j} \omega_{y,j}}{\omega_{x,j+1} + \omega_{y,j+1}} \right) \right]^{\frac{1}{4}}
\]

Eq. (9) applies to all the tubes in array perpendicular to the cross flow even if the properties of tubes are variable, and it will be simplified into Eq. (10) when both of the adjacent tubes are the same in size, material and damping. Eq. (10) shows the critical velocity of this situation, \( f = \omega / 2\pi \) is the natural frequency of the tube, \( \rho \) is the density of shell-side steam.

\[
U_c = \left( \frac{2\sqrt{2\pi}}{\rho D^2} \right)^{\frac{1}{2}} \left( \frac{m(2\pi \zeta)}{\rho D^2} \right)^{\frac{1}{4}}
\]

The coefficient \( C_i K_i \) is related to the pitch diameter ratio (PDR) of the tube bundle and the layout arrangement of the tubes (in-line arrangement or staggered form). With the same PDR, the critical velocity across tube bundle is higher than that across tube array, and the critical velocity across staggered tube bundle exceeds that across the in-line form.

The actual design requires that the heat transfer tubes are stable at the operating steam velocity of the normal condition of condenser. For the tube bundle in uniform flow, the effective velocity of
cross flow is required not to exceed the corresponding critical velocity to any mode. However, for a multispan tube bundle in non-uniform flow, the frequencies corresponding to the respective modes are similar. Because of the coupling of adjacent tubes, the effective velocity of cross flow is higher and more conservative. The safest method of calculation is to assume that the cross flow is entirely uniform along the whole tube longitude, to calculate the maximum value of the effective velocity and and to analyze the stability span by span.

3. CALCULATION OF CHARACTERISTICS OF TUBE

For easy installation, the diameter of holes on the support baffles for the tubes to pass through are processed slightly larger than the outer diameter of tubes. Therefore, in condenser during operation, rubbing between tube and hole will occur at the intersection point. Every contacting scheme has corresponding natural frequencies and mode shapes, but the status of the contact is random, especially for multispan tube, the larger count of supporting baffles, the more randomness. For the non-equal-span tubes, there is no accurate analytic expression of natural frequencies and mode shapes. Different series of frequencies will lead to various critical cross flow velocity according to Eq. (10). So the probability function of support failure need to be carried out to prepare for the follow-up discussions about the stochastic characteristics of fluid elastic instability.

3.1 Simplified Physical Model and Calculation of Natural Frequencies

As the tube has a small slenderness, the vibration can be calculated without considering the transverse shear stress. The expanded and welding connection between the ends of the tube and tube sheets are simplified as clamped joints, the contacting points with support baffles and baffles are simply supported. So simplifying the heat transfer tube as multi-equal-span Euler-Bernoulli beam clamped at both ends and simply supported in the middle is shown in Figure 2.

Figure 2 – Simplified tube model

3.2 Natural Frequencies with All Supports Effective

The simplified calculating object which has natural frequencies (assuming all supports are effective) as follows shown in reference (3).

\[ f_i = \frac{\lambda_i^2}{2\pi^2} \sqrt{\frac{EI}{m}}, \text{ Hz} \]  (11)

In Eq. (11), \( l \) is span length; \( EI \) is the flexural rigidity of the beam; \( m \) is beam linear density with added mass (water in this paper). And \( \lambda_i \) is the solution of following transcendent equation, where \( N \) is the count of spans shown in reference (3).

\[
\cos \frac{n\pi}{N} = \frac{1}{\sinh \lambda} - \frac{1}{\tan \lambda}
\]

\[
n = 0,1,2,\ldots,N-1 \quad \text{modes in odd pass bands}
\]

\[
n = 1,2,\ldots,N \quad \text{modes in even pass bands}
\]

(12)

The solutions to Eq. (12) are distributed in several frequency pass bands shown in expression (13). In particular, there are \( N \) modes for an \( N \)-span beam with values of \( \lambda \) between \( m\pi \) and \((m+1/2)\pi\). The modes become more densely packed as the number of spans increase. Take a four-span straight beam for example, the first four modes are given by the values of \( \lambda \): 3.393, 3.927, 4.463, 4.730.

\[ m\pi \leq \lambda < (m+1/2)\pi \quad \text{in mth pass band} \]  (13)

Tube physical properties are as following table.
Table 1 – Physical parameters of tube

<table>
<thead>
<tr>
<th>Length/Span length, m</th>
<th>External/Internal Diameter, m</th>
<th>Modulus of elasticity, Pa</th>
<th>Density of tube material, kg/m³</th>
<th>Density of tube tube-side fluid, kg/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.2/0.3</td>
<td>0.016/0.0144</td>
<td>2.0e11</td>
<td>7930</td>
<td>1000</td>
</tr>
</tbody>
</table>

So the first four natural frequencies worked out by a simple program are shown in Table 2.

Table 2 – Natural frequencies of the tube with all supports effective

<table>
<thead>
<tr>
<th>Order</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>$f_1 = 443.6842$ Hz</td>
<td>$f_2 = 594.1284$ Hz</td>
<td>$f_3 = 767.6491$ Hz</td>
<td>$f_4 = 862.1369$ Hz</td>
</tr>
</tbody>
</table>

3.3 Natural Frequencies with Not All Supports Effective

Shell-tube condenser has baffles to support the tube bundle along its longitudinal direction at a certain distance. Out of the tube thermal expansion and installation considerations, the diameter of baffle holes is larger than the external diameter of tube, thus a gap between the tube and support is there. As a result, in the condenser working process, there might be non-effective support at the intersection point. Denoting effective support “1”, and non-effective “0” will name the all supports effective situation “111”.

And several other different supporting schemes have non-effective support “0” in them. Method of Eqs. (11) to (13) can still work in the situation of “010” and “000” statuses (results shown in Table 3 and 4), but cannot in situation of the non-equal-span “100”, “110” and “101” statuses, which can only be calculated in discrete approximate method.

Table 3 – Natural frequencies of the tube of “010” status

<table>
<thead>
<tr>
<th>Order</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>$f_1 = 148.5625$ Hz</td>
<td>$f_2 = 215.5784$ Hz</td>
<td>$f_3 = 380.3715$ Hz</td>
<td>$f_4 = 481.4928$ Hz</td>
</tr>
</tbody>
</table>

Table 4 – Natural frequencies of the tube of “000” status

<table>
<thead>
<tr>
<th>Order</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>$f_1 = 53.8946$ Hz</td>
<td>$f_2 = 139.3791$ Hz</td>
<td>$f_3 = 273.2724$ Hz</td>
<td>$f_4 = 451.6904$ Hz</td>
</tr>
</tbody>
</table>

In this paper, the beam structure has intensive restraint of five static indeterminations, maximum. So assumed mode method (AMM), one of the generalized coordinate methods, is used to calculate the natural frequencies of the non-equal-span situation.

AMM is to use the linear combination of finite number of assumed modes (admissible functions) to describe the vibration of elastic system. And then the vibration equation is established by using of Lagrange equation.

(1) For the status of “110” or “011”, the admissible functions are given as

$$
\phi_1(x) = x^2 \cdot (x - L)^2 \cdot (x - l) \cdot (x - 2l)
$$

$$
\phi_2(x) = \begin{cases} 
1 - \cos(2\pi x / l) & 0 \leq x \leq 2l \\
1 - \cos[\pi(x - 2l) / l] & 2l \leq x \leq L
\end{cases}
$$

(14)

(2) For the status of “101”, the admissible functions are given as

$$
\phi_1(x) = x^2 \cdot (x - L)^2 \cdot (x - l) \cdot (x - 3l)
$$

$$
\phi_2(x) = \begin{cases} 
1 - \cos(2\pi x / l) & 0 \leq x \leq l \\
1 - \cos[\pi(x - l) / l] & l \leq x \leq 3l \\
1 - \cos[2\pi(x - 3l) / l] & 3l \leq x \leq L
\end{cases}
$$

(15)

(3) For the status of “100”, the admissible functions are given as

$$
\phi_1(x) = x^2 \cdot (x - L)^2 \cdot (x - l)
$$

$$
\phi_2(x) = \begin{cases} 
1 - \cos(2\pi x / l) & 0 \leq x \leq l \\
1 - \cos[2\pi(x - l) / 3l] & l \leq x \leq L
\end{cases}
$$

(16)
The generalized stiffness and mass matrix of the system can be obtained based on the admissible functions (14)–(16). Thus the fundamental frequency of the system is worked out according to the eigen equation. So the fundamental frequencies of the three cases above of non-equal-span are obtained in AMM: 201.1758Hz, 162.8718Hz and 90.3929Hz.

The common point of the admissible functions of all the three status is that $\phi_2(x)$ has more enhanced restraint than the practical beam. So the frequency results obtained in AMM with admissible functions like this are larger than those of the practical tube or the results of comparison functions.

The calculation results indicate that when a failure of support occurs, the natural frequencies are much lower comparing with that of the original structure. The more supports go non-effective, the lower are the natural frequencies.

4. TUBE BUNDLE SIMULATION EXPERIMENT

4.1 Design of the Experiment

According to the structural characteristics of the condenser tube bundle, the test piece shown in Figure 3 is designed with removable upper and lower shell halves in order to move the bundle and install sensors. Air blows through the cylinder shell and directly act on the tube bundle, and water flows into the tubes from the water chamber at the tube sheet, which is shown in Figure 4. There are 19 tubes in total located in regular triangle. Five of them are replaceable (shown as concentric circles in Figure 4) with stuffing box support at the ends, other 14 are fixed. The physical parameters are exactly the same as the calculating model shown in Table 1.

![Figure 3 – Exploded view of the test piece](image1)

![Figure 4 – Diagram of tube sheet](image2)

The practical testing system is shown in Figure 5.

![Figure 5 – Practical testing system](image3)

4.2 Testing and Results

4.2.1 Modal Test on the Tube

Model tests are carried out on both replaceable and fixed tubes with and without water in, respectively. The natural frequencies of replaceable and fixed tubes are basically the same, which indicates that the stuffing box supports are as effective as the expansion joints and welding. Results are shown in Table 5.
Table 5 – Natural frequencies of the tube tested in the experiment

<table>
<thead>
<tr>
<th>Order</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without water, Hz</td>
<td>107.8</td>
<td>214.8</td>
<td>391.4</td>
<td>483.6</td>
</tr>
<tr>
<td>With water (0.88m/s), Hz</td>
<td>72.6</td>
<td>153.1</td>
<td>346.1</td>
<td>446.1</td>
</tr>
</tbody>
</table>

The lowest natural frequency of the tube with all supports effective with water inside is calculated before with the value of 443.7Hz, which is very close to the measured value of fourth order with water in Table 5. Because of the gap between tubes and supports, there will be random rubbing leading to failure of some supports. Then the value smaller than the theoretical solution appears in the test.

4.2.2 Test on OASPL and Vibration Frequency Peak

The vibration states are observed in the case of with and without flow inside, as well as the water pressure pulsation in the chamber induced by vibration.

1. Without water filled in the tube, the vibration amplitude variety of testing tube No.1 is shown in Figure 6. Stress value doesn’t stop increasing with the increase of wind speed until wind speed reaches 3.3m/s, then tends to be stable. The first order frequency ratio $f_1/f_s=0.94$, and the fourth order frequency ratio $f_4/f_s=0.97$ (data in Table 5). According to vortex shedding “lock zone” theory (5), the excitation frequency is very close to the natural frequency of the tube, resulting in a large amplitude, frequency in the “locked zone” is hardly changed. For an ideal rigid tube, frequency ratio should be close to 1.0. The tube is simply supported by the baffle hole, the fourth order fundamental frequency is equivalent to all the three supports effective. Not like the first order of no support effective, the weak coupling between the span displacement and vortex shedding causes relatively large frequency.

![Figure 6 – Transverse strain of testing tube No.1](image)

As shown in Figure 7, the overall sound pressure level (reference sound pressure of 20μPa) measured by the microphone increases with the increasing wind speed until reaching 3m/s, the overall sound pressure level (OASPL) reaches 110dB. Then the OASPL basically remains unchanged with the increase of wind speed. In addition, the frequency peak reaches 2553Hz with wind speed of 3m/s, then it increases rapidly to 2954.7Hz reach with wind speed of 3.3m/s. In 1987, Blevins’ experiments (6) of in-line tube bundle with PDR of 2 and 3 had similar curves with Figure 7. Acoustic resonance has been observed from the phenomena and data. And the range of the wind speed larger than 3.3m/s is the “locked area” for acoustic resonance where the acoustic resonance always occurs with the maximum SPL which will not grow with wind speed.

It is shown in Figure 8 that the transverse strain frequency of testing tube No.1 almost coincide with the frequency of the acoustic signal (blue curve is the strain data, and red the SPL). The tube bundle vibration grows severer with the increase of wind speed. The acoustic energy into the testing piece is greater than that radiated out, then resonance occurs with the continuous accumulation of acoustic energy. This acoustic resonance is caused by the flow-induced vibration, but by the impact of acoustic modes of tube bundle, 2545.313Hz should be close to the corresponding frequency.

![Figure 7 – OASPL and frequency peak](image)
2. With the internal flow rate of 0.88 m/s, tube vibration amplitude with the change of wind speed is shown in Figure 10. Transverse strain is greater than the longitudinal when the wind is slower than 2.8 m/s, it will be otherwise after that. It is well known that, in the event of a regular vortex shedding, the lift force on the tube is twice the drag force meaning this is beyond the explanation of vortex shedding. So fluid elastic instability is considered predominating in the vibration when the flow rate is larger than 2.8 m/s.

![Figure 8 – Signal of vibration and sound of empty tube with wind speed of 3.3 m/s](image)

![Figure 9 – Signal of vibration and sound of water-filled tube with wind speed of 3.8 m/s](image)

![Figure 10 – Strain RMS of testing tube No.1](image)

2.8 m/s.

Frequency peak and sound OASPL both are influenced by the presence or absence of water. The peak frequency of sound pressure has been reduced with water in the tube, the OASPL is lower than of the non-flow condition. This shows that the presence of flow within lowers the critical velocity of the vortex shedding, the fluid elastic instability and locking of acoustic resonance, on the other hand it reduces the OASPL and frequency peak of sound radiation. FIV mechanisms of situations of presence and absence of water are totally different.

Shown in Figure 9, when the shell-side cross wind speed is 3.8 m/s, the maximum frequency of vibration and sound pressure completely coincide. But the vibration peaks at low frequencies have no corresponding peaks in sound pressure spectrum. This indicates that although the acoustic resonance is caused by the vibration, its occurrence requires some certain conditions that vibration excitation and the acoustic mode are close or coincide. In summary, the OASPL and frequency peak of sound radiation are determined by specimen radiated by vortex shedding, the fluid elastic instability of bundle and acoustic modes together.

4.2.3 Test on the Sound Pressure Pulsation in Water Chambers

The curves of sound pressure pulsation of the cooling water inlet and outlet chamber are shown in Figure 11 and 13. As can be seen, changes in sound pressure level of 1250–3750 Hz are caused by the increasing-wind-speed-induced-vibration. Spectrum of sound pressure at the outlet with the wind speed of 0 (background spectrum) is more complex, meantime the sound pressure pulsation in chamber is mainly determined by the characteristics of entire pipeline the cooling water flowing through. The effect of FIV to the sound pressure pulsation in chambers grows greater than the characteristics of pipeline gradually with the wind speed increasing. Therefore, the sound pressure pulsation in chambers is in the FIV controlling zone with the current flow rate inside the tube.
OASPL in water chambers of every operating condition is shown in Table 6, OASPL increases with the increase of wind speed in general. The SPL in the inlet water chamber is higher than in the outlet water chamber in all the operating conditions. As the arrangement of the testing piece is completely symmetrical, and the internal water flow velocity is only 0.88 m/s. However, with the increase of wind speed, the sound pressure pulsation upstream caused by FIV grows larger, which indicates that the downstream acoustic damping is increased by FIV, leading towards the “asymmetry” of the the sound pressure pulsations.

<table>
<thead>
<tr>
<th>Wind speed</th>
<th>0 m/s</th>
<th>1.2 m/s</th>
<th>2.6 m/s</th>
<th>2.8 m/s</th>
<th>3.8 m/s</th>
<th>4.2 m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>OASPL</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inlet(dB)</td>
<td>159.24</td>
<td>162.23</td>
<td>163.78</td>
<td>165.75</td>
<td>163.28</td>
<td>165.15</td>
</tr>
<tr>
<td>Outlet(dB)</td>
<td>158.58</td>
<td>159.45</td>
<td>160.99</td>
<td>162.92</td>
<td>159.33</td>
<td>162.37</td>
</tr>
</tbody>
</table>

**CONCLUSIONS**

The critical velocity of the flow across the tube bundle has positive correlation with the natural frequency of the tube. The natural frequencies of the tubes with non-effective supports are far lower than the original structure, the more supports are non-effective, the lower are the natural frequencies. One vibration mode of the tube with non-effective support has no corresponding relation or necessary relation with the modes of other form of non-effective supports.

The test results of tube bundle simulation experiment indicate that the current design can meet the test requirements. Vortex shedding occurs to the fifth tube array. The acoustic resonance is caused by other FIV mechanisms, and reacts on the FIV changing the vibration amplitude. The sound pressure pulsation in tube-side flow caused by FIV is asymmetry, it is easier for sound pressure pulsation propagate upstream.

**REFERENCES**