A SIMPLIFIED METHOD TO PREDICT FRETTING-WEAR DAMAGE IN DOUBLE 90^{O} U-BEND TUBES

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Key Words : Fretting-wear damage, Random turbulence excitation, RMS vibration amplitude

Abstract

Fluid-elastic instability is believed to be a cause of the large-amplitude vibration and resulting rapid wear of heat exchanger tubes when the flow velocity exceeds a critical value. For sub-critical flow velocities, the random turbulence excitation is the main mechanism to be considered in predicting the long-term wear of steam generator tubes. Since flow-induced interactions of the tubes with tube supports in the sub-critical flow velocity can cause a localized tube wear, tube movement in the clearance between the tube and tube support as well as the normal contact force on the tubes by fluid should be maintained as low as possible.

A simplified method is used for predicting fretting-wear damage of the double 90º U-bend tubes. The approach employed is based on the straight single-span tube analytical model proposed by Connors, the linear structural dynamic theory of Appendix N-1300 to ASME Section III and the Archard's equation for adhesive wear.

Results from the presented method show a similar trend compared with the field data. This method can be utilized to predict the fretting-wear of the double 90º U-bend tubes in steam generators.

NOMENCLATURE

C_D ; Steady state drag flow coefficient
C_m ; Hydrodynamic mass coefficient
$C_{r1}(f_j)$, $C_{r2}(f_j)$;Turbulence excitation coefficients suggested for upstream and interior cylinders [$sec^{0.5}$]
D ; Tube outside diameter [m]
d_w; Tube wear depth [m^3]
f_j; Natural frequency of tube in jth mode shape [Hz]
FN_D ; Normal normal contact force on tube by steady drag flow [kg]
FN_d ; Dynamic normal contact force on tube [kg]
$G_f^{\,i}(f_j)$; Power spectral density of the forcing function for the ith span of a multi-span tube [$(kg/m)^2/Hz$]
g ; Gravitational acceleration [m/sec^2]
$J_{jj}^{\,i}$; Joint acceptance for the ith span
$\overline{K}^{i}$; Modal stiffness of simply supported in ith tube span [kg/m]
K_w ; Fretting-wear coefficient [Pa^{-1}]
L ; Supported tube span lengths [m]
$L_c^{\,i}$; Correlation length in ith tube span [m]
l_S ; Sliding distance per second [m/sec]
L_S ; Total sliding distance at the tube-to-tube support mating surface [m]
$m(x)$; Spanwise tube mass per unit length [kg_m/m]
m_t ; Average modal mass per unit tube length including hydrodynamic added mass [$kg\text{-}sec^2/m^2$]
M_j ; Modal masses of simply supported tube section in jth vibration mode shape [$kg\text{-}sec^2/m$]
P ; Tube array pitch [m]
V_w; Fretting-wear volume [m^3]
W ; Tube support width [m]
$U(x)$; Flow gap velocity [m/sec]
y^2 ; RMS vibration amplitude of a tube [m^2]
y_0 ; RMS vibration amplitude of out-of-plane at mid-span [m]
y_n ; RMS vibration amplitude for in-plane at tube-to-tube support mating surface [m]
α ; A half angle of tube wear volume [radian]
$\xi(f_j)$; Critical damping ratio(%) for jth natural frequency
ρ_i ; Flow density per unit length [kg_m/m^3]
$\phi_i(x)$, $\phi_j(x)$; Normalized displacement in ith and jth mode shapes

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1. Introduction

Fretting wear is one of the main degradation mechanisms of heat exchanger tubes. The heat exchanger tubes fret at support locations because of rubbing and impacting caused by flow-induced vibration. The traditional technique to predict fretting wear requires a time domain non-linear simulation of the tube dynamics in which the details of the dynamic interaction between the tube and its supports are modeled using the finite element technique. This approach requires a significant amount of effort and expertise because of the complicated nature of the problem and indefinite boundary conditions at the clearance supports [1].

As an alternative, some of simplified work-rate methods for a straight tube and for a half circle U-bend tube were proposed [2,3,4]. The concept of work-rate method is the modified form of the Archard's equation by replacing the force and sliding distance terms with an equivalent parameter as called work-rate [5]. These methods include the non-linearity effects due to clearances between the tube and its support.

In this study, however, based on the straight single-span tube analytical model proposed by Connors [6] and the well-developed linear structural dynamic analysis theory for arbitrary random loading of beams of Appendix N-1300 to ASME Section III [7] with the Archard's equation [8] for adhesive wear, a simplified method is used for predicting fretting wear damage by the random turbulence excitation for the double 90° U-bend tube.

2. Fretting-wear prediction

The assumption of a linear structure is justifiable for the small vibration associated with the turbulence excitation of weakly coupled fluid-structure systems, and the linear structural dynamic analysis theory for arbitrary random loading of beams is highly developed. In the case of lightly damped structures with well-separated shapes, mean square response is found by the integration of the power spectral density of the response over the frequency range interested.

The EPRI report [9] shows that the tube-to-tube-support clearance is the most important parameter affecting the tube wear. Excessive clearance increases impact and sliding at the supports. The work rate parameter determined through various methods shows a strong dependence on clearance size. Most methods predict a moderate wear increase with clearances up to 0.5mm and a drastic increase afterwards. The EPRI report also shows that squeeze film effects play an important role in the tube wear mechanism and the existence of a liquid film in the tube support plate reduces wear.

In case of the double 90° U-bend tubes, the nominal clearance of tube-to-tube support is 0.33mm (0.013in), with a maximum possible clearance of 0.66mm (0.026in). These require that tube wear analysis be needed.

In this study, the tubes are assumed simply supported at each support location. And also the vibration mode shapes, natural frequencies are assumed not to be altered by the small sliding motions of the tube by the steady forces that press the tube against the support.

2.1 Archard's Equation

The concept of Archard's equation [8] is that the tube wear volume is proportional to the normal forces that press the tube against the support and the sliding distance.

$$V_w = K_w \times FN \times L_S \qquad [\mathrm{m}^3] \qquad (1)$$

2.2 Fretting Wear Volume

It is conservatively assumed that the tube wear surface remains macroscopically flat and only the tube material is fret away. As the wear depth increases, based on the Figure 1, the tube wear depth and the removed tube wear volume are predicted by equations (2) & (3), respectively.

$$d_w = \frac{D}{2}(1-\cos\alpha) \qquad [\mathrm{m}] \qquad (2)$$

$$V_w = \frac{D^2 W}{8}(2\alpha - \sin 2\alpha) \qquad [\mathrm{m}^3] \qquad (3)$$

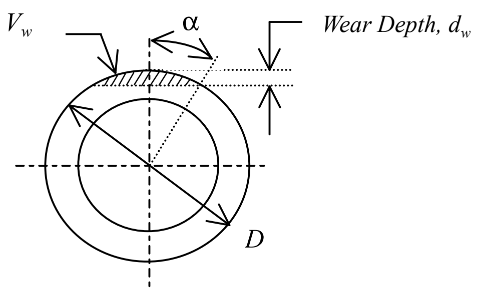


Figure 1. Tube Wear Depth and Wear Volume

2.3 Fretting-Wear Coefficient

In this study, based on the fretting-wear coefficient data proposed by Fisher et al. [10], the coefficient for design purpose is conservatively assumed as follows.

$$K_w = 50 \times 10^{-15} \qquad [\mathrm{Pa}^{-1}] \qquad (4)$$

3. Modal Analysis

The modal analysis technique for wear prediction is that the overall wear can be calculated by summing the individual contributions of vibration modes. Contribution of the vibration modes can be estimated using the frequency, modal damping ratio and RMS vibration amplitude of the mode of interest. In many U-bend tube cases, there may be more than one mode making a significant contribution to the overall vibration response. In modal analysis, individual modes can be treated as single degree of freedom systems with clearly defined frequency, modal damping and RMS displacement response [3].

In this study, four cases of the double 90° U-bend tubes are typically selected to predict wear damage according to the tube support conditions and tube lengths.

3.1 Hydrodynamic added mass

The hydrodynamic added mass is successfully used to characterize the fluid-structure coupling force created by the motion of a structure in a non-flowing fluid. This added mass increases the effective mass of a structure vibrating in a fluid.

In this study, the hydrodynamic mass coefficients of C_m=1.7 for the triangular tube array pitch of 1.00in. at the vertical tube section, and C_m=3.1 for the rotated square tube array pitch of 1.23in. at the upper U-bend tube section are used [11].

Based on the tube array patterns, the flow gap velocity factors inside tube bundle are P/(P-D)=1.0/(1.0-0.75)=4.0 for the triangular tube pitch array and P/(P-D)=1.23/(1.23-0.75)=2.6 for the rotated square tube pitch array, respectively.

3.2 Damping

The overall damping of the system is modeled using the Rayleigh proportional damping model in water environments as shown in Figure 2 [12].

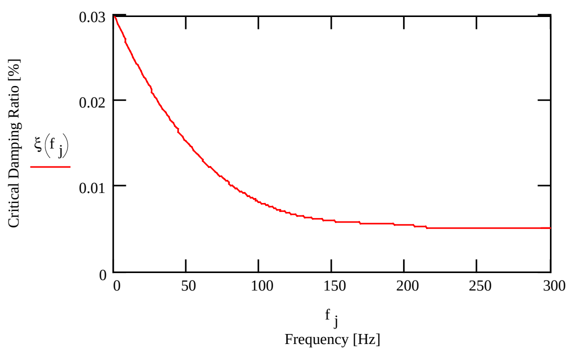


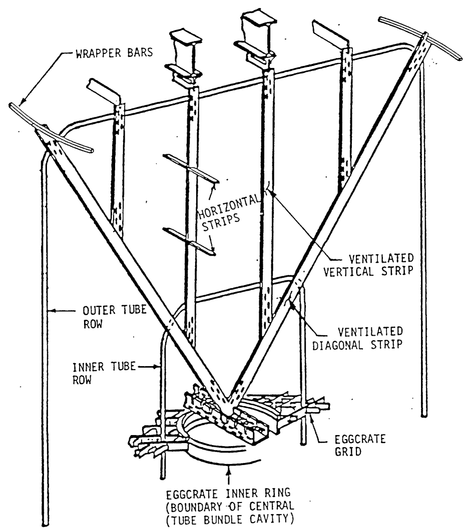
Figure 2. Critical Damping Ratio for Frequency

The range of frequencies covered is consistent with the dominant natural frequencies obtained from eigensolutions for idealized linear support conditions. A comparison of the damping ratio used in the simulations with those obtained from the proposed guidelines and measurements conducted on nuclear steam generator indicates that the damping values used herein provide a realistic envelope for operating conditions [13].

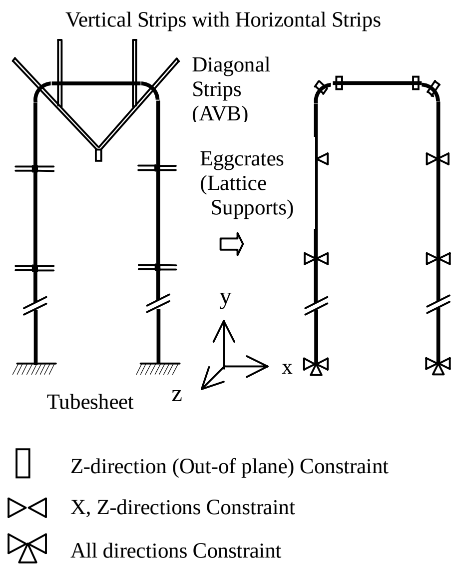
3.3 Modal Analysis Model

The tube modal analysis model for the double 90° U-bend tube is as shown in Figure 3. Tubes are supported by eggcrates that restrict both x- and z-direction movements in the vertical tube straight section, and by the diagonal strips(AVB) and vertical strips that restrict only z-direction.

The modal analysis is performed using the general-purpose ANSYS code and the thermal-hydraulic analysis of steam generator secondary side coolant is performed using the ATHOS3 code [14].



(a) Schematic View of Double 90° U-Bend Tube



(b) Simplified Analysis Model

Figure 3. Schematic View and Simplified Analysis Model of Double 90° U-Bend Tube

3.4 Modal Mass and Stiffness

In extending the straight tube formula to U-bend tubes, the selection of variables such as frequency, damping and mid-span RMS displacement may not be straightforward for U-bend tubes. In such cases, it may be more appropriate to use the dominant U-bend mode shape instead of the fundamental mode shape [3].

The double 90° U-bend tube is assumed as the three straight sections divided by two diagonal strips, and the horizontal tube section is also assumed to have the straight sections divided by two vertical strips. This enables to apply the straight modal mass equation to the double 90° U-bend tube section.

Modal mass of the straight tube section per unit length including the added mass of fluid outside the tube is as follows [6].

$$M_j = m_t \int_0^L \left(\sin\left(\frac{\pi \times z}{L} \right) \right)^2 dz \qquad [\text{kg-sec}^2/\text{m}] \qquad (5)$$

Modal stiffness of simply supported ith tube span in the significant jth mode shape is related to the tube modal mass [6].

$$\overline{K}^i = (2\pi \times f_j)^2 \times M_j \qquad [\text{kg/m}] \qquad (6)$$

4. Structural Response of Tubes

The linear structural dynamic theory recommended in ASME Section III is used to calculate the mean square response by random turbulence excitation.

4.1 RMS Vibration Amplitude

The mean square response (RMS vibration amplitude) of a tube is found by integration of the power spectral density of the response as follows [7]:

$$y(x)^2 = \sum_j \frac{L G_f^{\,i}(f_j) \phi_j^{\,2}(x)}{64\pi^3 M_j^{\,2} f_j^{\,3} \xi_j} \left(J^i{}_{jj}\right)^2 \qquad [\text{m}^2] \qquad (7)$$

These mean square responses are the mid-span displacement and assumed as the sliding distance at tube-to-tube support mating surface.

In the multiple span of non-uniform cross flow, the power spectral density is generated by the turbulent pressure field at the natural frequency of the jth vibration mode [7].

$$G_f^{\,i}(f_j) = \left(\frac{D}{2} \right)^2 C_r^{\,2}(f_j) \int_0^{L_i} [\rho_i U^2(x)]^2 \phi_j^{\,2}(x) dx$$
$$[(\text{kg/m})^2/\text{Hz}] \qquad (8)$$

The joint acceptance integral reflects the relative effectiveness of the turbulence forcing function to excite the jth vibration mode. It is defined as the ratio of correlation length and tube span [7].

$$\left(J_{jj}^i\right)^2 \approx L_c^{\,i} / L \qquad (9)$$

Pettigrew and Gorman defined the turbulence excitation coefficients as a function of natural frequency as shown in Figure 4 [7].

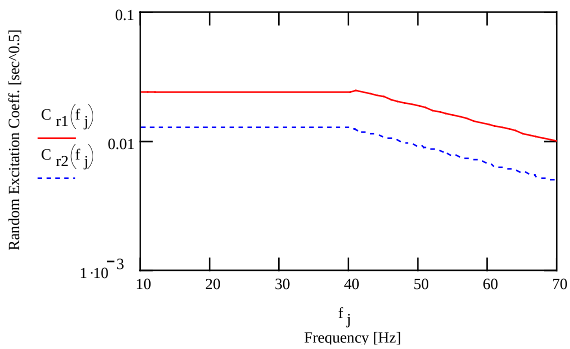


Figure 4. Random Excitation Coefficients for Arrays in the Cross Flow

Correlation length is a measure of the coherence range of the turbulent pressure field. The correlation length [15] in the ith span is

$$L_c^{\,i} = 6.8D \qquad [\text{m}] \qquad (10)$$

The mode shapes satisfy the orthogonality relation, and the total modal mass of the simply supported tube section in the jth vibration mode shape is

$$M_j = \int_0^L m_t(x) \phi_i(x) \phi_j(x) dx \quad [\text{kg-sec}^2/\text{m}] \qquad (11)$$

4.2 Total Sliding Distance

For random vibration by turbulence excitation, the total sliding distance should be expressed as a function of the RMS vibration amplitude and the sum of dominant in-plane sliding mode shapes.

The normalized displacements shown in Figure 5 are converted to RMS vibration amplitude using equation (7).

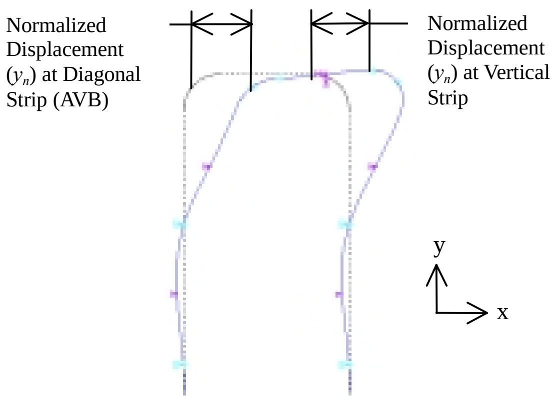


Figure 5. One of Tube Vibration Motion Shapes and Normalized Displacements

The total sliding distance along the support is calculated as follows:

$$l_S = \sum_i 4\times(f_i \times y_n) \quad \text{[m/sec]} \qquad (12)$$

$$L_S = l_S \times time \quad \text{[m]} \qquad (13)$$

5. Normal Contact Forces

The main causes of the normal contact force between the tube and the tube support are the distributed load by the cross flow on tube span and the dynamic tube vibration. It is conservatively assumed that the tubes always contact the support.

5.1 Steady Drag Force

It is believed that the tube is usually pressed against the support by the steady force, such as uniformly spanwise distributed load on the tube span as shown in Figure 6.

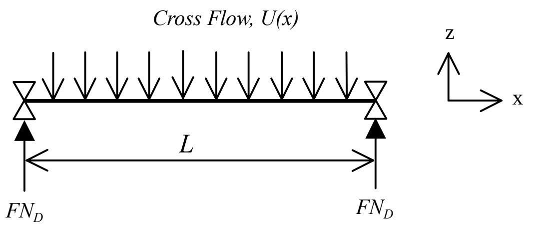


Figure 6. Steady Drag Force by Distributed Load

Assuming a steady state drag coefficient equals 0.20 [6], the steady drag force at the tube support is calculated by equation (14) as the reaction force by the distributed load along the beam.

$$FN_D = \frac{1}{2}\times C_D \times\left(\rho_i \times \frac{U^2(x)}{2g}\right)\times D \times L \quad \text{[kg]} \qquad (14)$$

5.2 Dynamic Contact Force

From the thermal-hydraulic analysis, the spanwise flow velocities normal to tube spans are applied to calculate the RMS vibration amplitude (y_0) obtained from equation (7).

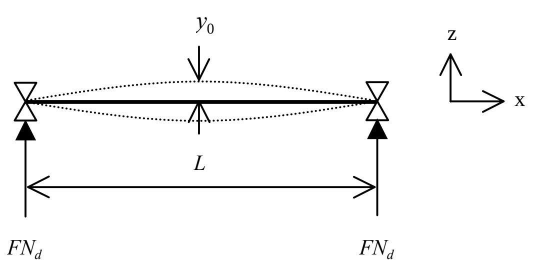


Figure 7. Dynamic Force by RMS Vibration Amplitude

The dynamic contact forces on the tube are calculated by equation (15) as the reaction force of the beam as shown in Figure 7 and are the sum of significant modes contributed to the overall vibration amplitude response.

$$FN_d = \frac{1}{2}\times\left(\bar{K}^i \times y_0\right) \quad \text{[kg]} \qquad (15)$$

6. Application to Steam Generator

According to the field inspection data, some excessive tube wear was observed around the center of the tube bundle within 2~3 years of reactor operation. These occurred at the relatively long unsupported tube span and in the higher cross flow velocity area.

In this study, based on the high fretting-wear inspection data, typical four cases of the tube rows (denoted A through D) in the center region of tube bundle are selected to predict the fretting-wear damage.

6.1 Expected Time of Fretting-Wear

Based on the Archard's equation (1), the removed wear volume of equation (3), the sliding distance of equation (13) and the normal contact forces of equations (14) and (15), the expected time of tube fretting-wear is predicted by equation (16) as follows and the analysis results of fretting-wear are shown in Figure 9.

$$time = \frac{\frac{D^2W}{8}(2\alpha - \sin 2\alpha)}{K_w \times (FN_D + FN_d)\times l_S} \quad \text{[year]} \qquad (16)$$

The presented method shows a similar trend compared with the field data shown in Figure 8 for the wear status of initial operation years.

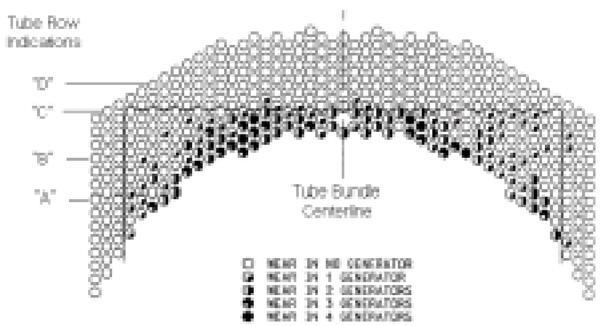


Figure 8. Field Inspection Wear Indication for Double 90° U-bend Tubes after 3 Cycle Operation

In the proposed model, however, it does not consider the effects of gap increment between the tube and tube support by tube wear. This effect might decrease the tube wall wear loss as the reactor operation years increase. Therefore, further refinements are required to consider the effects of gap increment according to the filed inspection data and material properties (i.e., tube material strain hardening and plasticity).

7. Discussion and Conclusion

A simplified method is used for predicting the fretting-wear damage of the double 90º U-bend tubes, which is based on the linear structural dynamic analysis and the Archard's equation for adhesive wear associated with turbulence excitations. It is found that

(1) This simplified method can provide a reasonable estimate at the design stage of steam generator for the fretting-wear damage without performing complicated non-linear analyses.

(2) The straight tube formula can be sucessfully applied to U-bend tubes with the double 90º U-bend tube. In this study, the double 90º U-bend tube is assumed as the three straight sections divided by two diagonal strips, and the horizontal tube section is assumed to have the straight sections divided by two vertical strips. This assumption enables to apply the straight modal mass and modal stiffness equations to the double 90º U-bend tube section.

(3) The magnitude of normal contact force at tube support by the steady drag force is more than 10 times that by the dynamic tube vibration. It is believed that the steady drag force on the tube is the main cause of fretting-wear.

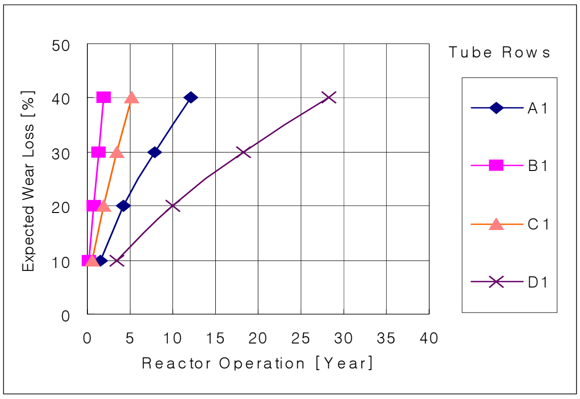


Figure 9. Expected Time of Fretting-Wear of % Tube Wall Thickness at Diagonal and Vertical Strip Surface

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