Fretting Wear Analysis of PFBR Steam Generator

S. Jalaldeen, R. Srinivasan, P. Chellapandi and S. C. Chetal

Indira Gandhi Centre for Atomic Research, India

ABSTRACT: The fretting wear of steam generator tubes at the location of support has been estimated under various excitation mechanisms of flow induced vibration viz vortex shedding, fluid elastic instability and turbulence. The eccentricities of 0.0mm and 0.3mm are considered in the analysis. The vortex shedding is modelled by oscillating forces, instability by negative damping model and turbulence by DMT and ASME spectrum. The code TURB-GEN is used along with CASTEM-2000 for this analysis. The thinning of tube is 0.078mm at the end of life.

1.0 Introduction
Several mechanisms like turbulence, vortex shedding and fluid elastic instability can lead to flow induced vibration in the steam generator (SG) of PFBR. The material wear due to fretting at tube bundle supports have to be evaluated. Current ASME Appendix N does not address this problem. The fretting wear is a function of material, type of support, surface finish of the contacting surfaces, the diameter of the tube, the clearance between tube and support and the nature of vibration exciting mechanism.

In this note the material wear at the end of life is evaluated which is likely to arise from each of the possible mechanisms (vortex shedding, fluid-elastic instability, turbulence). Towards this, an in-house code 'TURB-GEN' has been developed based on state of art knowledge in this subject and used to estimate the wear rate for the SG. This code is a preprocessor to CASTEM 2000. Before doing the analysis, the literature survey has been made and the methodology of estimation of fretting wear of tubular structures has been formulated (Fig. 1).

2.0 Definition of excitation forces under various FIV mechanisms
The analysis is carried out under the three excitation mechanisms viz. vortex shedding, instability and turbulence. The excitation force-time history under each of these mechanisms are given below:

2.1 Exciting force due to vortex shedding
The oscillating lift and drag force produced on the tube are as follows:
\[ F_l = \frac{1}{2} \rho \nu^2 DL [\sin(2\Pi f_{s})] \]
\[ F_d = 0.1 \times \frac{1}{2} \rho \nu^3 DL [\sin(4\Pi f_{s})] \]
Where \( \rho \) - density of fluid, \( \nu \) - Velocity of flow, \( D \) - diameter of tube, \( L \) - Length of the span, \( f_s \) - Vortex shedding frequency.
2.2 **Excitation force due to fluid-elastic instability**

The critical velocity of the tube is calculated for each of the mode shape. The unstable mode shapes for which the actual velocity is exceeding the critical velocity are determined. The negative damping for these unstable modes are found out as follows:

\[ \xi = \xi_o \left[ 1 - \left( \frac{v}{V_o} \right)^2 \right] \]

where \( \xi_o \) - negative damping of unstable mode, \( \xi \) - damping of stable mode, \( V \) - Actual velocity of fluid, \( V_o \) - Critical velocity of unstable mode

2.3 **Excitation force due to turbulence**

Under turbulence the excitation forces are due to random forces whose power spectral densities are calculated from a given spectrum. DMT, CEA, Saclay, France is proposing one generalised spectrum called DMT envelope spectrum and ASME, Sec III, Appendix N is also proposing a spectrum which is very conservative compared to DMT spectrum.

**DMT envelope Spectrum:**

PSD of excitation force is calculated as follows:

\[ S_o = \frac{1}{2} \frac{p}{\nu^4 D} \frac{D}{\nu} \frac{L^2}{D_o} \frac{L_o}{L_o} \frac{D}{L_o} \frac{\phi_f(f)}{f^3} \]

Where \( D_o, L_o \) - Reference diameter and length of the tube.

\( \phi_f(f) = 4.0 \times 10^{-4} f^{-0.5} \quad 0.01 < f < 0.2 \)

\[ = 3.0 \times 10^{-6} f^{-3.5} \quad 0.2 < f < 3.0 \]

\( f = \frac{2 \pi D}{T} \)

\( f \) - natural frequency, \( a_o = 2/L_o \int \frac{\phi_f(s)}{\rho(s) u^2(s)} ds \)

**ASME Spectrum:**

Power spectral density of turbulent force per unit length of the tube:

\[ G(f) = [C_k(f)/2 \nu^3 D]^2 \]

where \( G(f) \) is PSD of turbulent force. \( C_k(f) \) is obtained from a given spectrum curve as shown in Fig. 2.

3.0 **Analysis for dynamic response**

3.1 **Input required for CASTEM 2000 Code**

3.1.1 **Geometry and flow details**

Tube diameter \( = 17.2 \) mm, Tube thickness \( = 2.3 \) mm

Tube length \( = 2100 \) mm (taken in the vicinity of inlet), Support clearance = 0.3 mm (radial)

Radial eccentricity = 0.0 to 0.3 mm (assumed)

Uniform cross flow velocity of 0.934 m/s is assumed throughout the span.

3.1.2 **Boundary conditions**

As shown in Fig 3, the bottom end of the tube is fixed and the top end is simply supported. Loose support is considered in between the two spans.

3.1.3 **Material data for modified 9 Cr 1 Mo**

Material properties of the tube at 798 K (525deg C) are:

Young’s modulus \( = 1.69 \times 10^9 \) MPa, Poison’s ratio = 0.3, Density = 7570 Kg/m³
3.1.4 Support details
The normal, tangential stiffnesses ($K_n$ and $K_t$) and tangential damping ($C_t$) are computed as shown below:

$$K_n = 1.9 \times E \varepsilon_0 \sqrt{\varepsilon / D}$$

where $E$ - Young's modulus (Pa), $\varepsilon$ - Tube thickness (m), $D$ - Tube diameter (m)

$$K_t = 10 \times K_n = 4.273 \times 10^8 \text{N/m}$$

$$C_t = K_t M = 2745 \text{N-s/m}$$

where $M$ - mass of the tube

Friction and adherence coefficients required for fretting wear calculations in CASTEM 2000 are given as 0.4. [4]

3.2 FEM modelling of tube
The tube is modelled using beam element of CASTEM-2000. The number of elements used is 40. Because of presence of sodium outside and water inside the tube, the added mass is calculated and the equivalent density for the tube is found to be 10900 kg/m$^3$.

3.3 Natural frequency analysis
The free vibration analysis is done using CASTEM-2000 to form a modal basis for the problem. The central support is not considered for the natural frequency analysis. Totally 18 mode shapes are considered for the analysis after confirming that with 22 natural frequencies and mode shapes, the results do not change significantly. The following table-I gives the frequencies of those modes. The first 9 natural frequencies (Hz) are 12.8, 41.5, 86.5, 147.9, 225.7, 319.7, 430.0, 556.6 and 699.4. The frequencies are the same in other mutually perpendicular directions.

3.4 Solution time step and Averaging time
The time step to be used in the non-linear dynamic response calculation is

$$dt = 1/(10 f_{max}) = 1.43 \times 10^{-4} \text{sec.}$$

where $f_{max}$ - maximum freq. in the modal basis.

Time averaging should be done well after the transient period of response. Averaging time of 5.0 s is assumed as recommended in the ref [3].

3.5 Results of analysis
3.5.1 For vortex shedding mechanism
The wear work rate ($w$), normal impact force ($F$) and rms displacement ($Y$) are calculated at the loose support using CASTEM-2000.

Radial eccentricities of 0.0 and 0.3 mm are considered for the analysis. The results are tabulated as follows:

<table>
<thead>
<tr>
<th>Eccentricity (mm)</th>
<th>F(N)</th>
<th>Y(m)</th>
<th>W(w)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>4.754</td>
<td>2.27x10^-4</td>
<td>3.15x10^-2</td>
</tr>
<tr>
<td>0.3</td>
<td>5.044</td>
<td>2.21x10^-4</td>
<td>3.97x10^-2</td>
</tr>
</tbody>
</table>

Fig. 4 gives locus of the tube under vortex shedding.
3.5.2 For instability

The critical velocity for the first mode of the tube is calculated as follows:

\[ V_c = f_n \sqrt{\frac{m_0 \beta}{\pi \rho}} = 0.52 \text{ m/s} \]

where, \( f_n \) - natural frequency, \( m_0 \) - mass per unit length, \( \beta \) - 4.5 for rotating triangular geometry, \( \rho \) - density of fluid, \( \delta \) - logarithmic decrement of damping

\( V_s = 0.934 \text{ m/s} \).

Since \( V_s > V_c \), the fundamental mode is unstable and the corresponding negative damping is calculated as follows:

\[ \xi = \xi_0 \left[ 1 - \left( \frac{V_s}{V_c} \right)^3 \right] = -2.22\% \]

Negative damping of -2.22% is given for first mode and +1% is given for other modes. The wear work rate, normal impact force and rms of displacement are calculated using CASTEM-2000. Excentricities of 0.0 and 0.3 mm are considered for the analysis. The results are tabulated as follows:

<table>
<thead>
<tr>
<th>Eccentricity (mm)</th>
<th>F(N)</th>
<th>Y(m)</th>
<th>W(w)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>0.669</td>
<td>2.08x10^4</td>
<td>1.67x10^3</td>
</tr>
<tr>
<td>0.3</td>
<td>0.188</td>
<td>2.31x10^5</td>
<td>2.07x10^4</td>
</tr>
</tbody>
</table>

Fig. 5 shows the locus of the tube under instability.

3.5.3 For turbulence

The power spectral densities for different modes by using the code 'TURB-GEN' code. The DMT envelope and ASME spectrums considered for the analysis are shown in fig 19 and fig 20 for drag and lift directions respectively. The spectrum for drag is one-fourth of that in the lift direction [4].

Table - 3: Wear work rate for turbulence (damping is 1 %)

<table>
<thead>
<tr>
<th>Spectrum</th>
<th>Eccentricity (mm)</th>
<th>F(N)</th>
<th>Y(m)</th>
<th>W(w)</th>
</tr>
</thead>
<tbody>
<tr>
<td>DMT</td>
<td>0.0</td>
<td>0.0</td>
<td>1.04x10^4</td>
<td>0.0</td>
</tr>
<tr>
<td>DMT</td>
<td>0.3</td>
<td>0.195)</td>
<td>2.27x10^5</td>
<td>2.00x10^4</td>
</tr>
<tr>
<td>ASME</td>
<td>0.0</td>
<td>0.65</td>
<td>1.57x10^4</td>
<td>6.54x10^3</td>
</tr>
<tr>
<td>ASME</td>
<td>0.3</td>
<td>1.28</td>
<td>1.33x10^4</td>
<td>9.66x10^3</td>
</tr>
</tbody>
</table>

Fig. 6 shows the locus of the tube under turbulence. Fig. 7 shows the tube displacement time history under turbulence.

XII-104
3.6 Estimation of thinning of the SG tube

Table-4: Summary of work rate (w) for different mechanisms:

<table>
<thead>
<tr>
<th>Mechanism</th>
<th>Eccentricities (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.0</td>
</tr>
<tr>
<td>Vortex-shedding</td>
<td>3.15x10^{-2}</td>
</tr>
<tr>
<td>Instability</td>
<td>1.67x10^{-3}</td>
</tr>
<tr>
<td>Turbulence-DMT</td>
<td>0.0</td>
</tr>
<tr>
<td>Turbulence-ASME</td>
<td>6.54x10^{-3}</td>
</tr>
</tbody>
</table>

From the above table it can be seen that out of all the mechanisms the vortex shedding results in highest value of wear work rate at the support location. The thinning of SG tube is determined based on the highest value of wear work rate (vortex shedding and 0.3 mm eccentricity).

i.e. Wear work rate = 3.97 x 10^{-2} W

3.6.1 Contact of tube with support:

The width of contact between tube and support depends on the following aspects:

(i) The nature of tube movement, i.e. inclined hitting or hitting with uniform contact.
(ii) The edge condition of the support.

Based on the above aspects, different values of width of contact are considered viz. 10, 15, 20 mm for the analysis. The effect of width of contact on the thinning of tube is given in Table-6 along with the specific wear coefficient.

3.6.2 Specific wear coefficient:

From the literature survey, the specific wear coefficient (k) to be used for PFBR has been obtained. Some values of K obtained from the literatures are tabulated below:

Table-5

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material (tube/support)</th>
<th>Type of fretting</th>
<th>K(525 C) m²/Nm</th>
<th>reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>9Cr 1Mo/9Cr 1Mo</td>
<td>Impact-slide</td>
<td>1.5x10^{-14}</td>
<td>[5]</td>
</tr>
<tr>
<td>2</td>
<td>9Cr 1Mo/Aluminised</td>
<td>&quot;</td>
<td>1.5x10^{-16}</td>
<td>[6]</td>
</tr>
<tr>
<td>3</td>
<td>&quot;</td>
<td>Rubbing</td>
<td>6x10^{-17}</td>
<td>[6]</td>
</tr>
</tbody>
</table>

Second and third values of K are considered for calculation of fretting wear, since case 1 value is not relevant to PFBR. However, it is included here to quantify the effects of aluminising.

For the combination of specific wear coefficients chosen above and various widths of contact (Para 3.6.1), the amount of thinning have been calculated at the end of life (2x10^3 h) and are tabulated as shown below:
Table 6: Amount of thinning (in mm) of tube at the end of life.

<table>
<thead>
<tr>
<th>Width of Contact [mm]</th>
<th>Specific wear coefficient [m³/Nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.5x10⁻¹⁶</td>
</tr>
<tr>
<td>10</td>
<td>0.078</td>
</tr>
<tr>
<td>15</td>
<td>0.052</td>
</tr>
<tr>
<td>20</td>
<td>0.039</td>
</tr>
</tbody>
</table>

4.0 Conclusion

The fretting wear of steam generator tubes at the location of support has been estimated under various excitation mechanisms of flow induced vibration viz vortex shedding, fluid elastic instability and turbulence. The eccentricities of 0.0 mm and 0.3 mm are considered in the analysis. In the case of vortex shedding, the oscillating forces are taken for the analysis. The fluid elastic instability is modelled using negative damping model. The turbulence is modelled using both DMT and ASME spectrum. The code TURB-GEN is used along with CASTEM-2000 for this analysis.

Out of all the mechanisms, the vortex shedding is found to be the most severe one. For the aluminised inconel 718 contact surface, the thinning estimated varies from 0.039mm to 0.078mm depending on the width of contact (complete 20 mm contact to 10 mm contact).

Considering the possible deviations in verticality due to manufacturing tolerances and possible bending of tube near the support location, the actual width of contact is likely to be lower than the support width of 20 mm. Corresponding to a width of contact of 10 mm, thinning of tube is 0.078mm at the end of 2x10⁵h (30 years of operation with 75% load factor).

References
2. ASME, SEC-III, Appendix N.
4. S. Jalaldeen 'Flow induced vibration of Steam Generator tubes in the presence of antivibratory bars under turbulence and fluid elastic instability' RAPPORF DMT/93 347.
Fig. 1 Flow sheet

Fig. 2 Geometry details of the tube

Fig. 3 ASME spectrum
Fig. 4 Locus of the tube under vortex shedding

Fig. 5 Locus of the tube under instability

Fig. 6 Locus of the tube under turbulence

Fig. 7 Displacement history under turbulence