Chapter 8

Estimation of fretting wear limited life

8.1 Introduction

Flow-induced vibrations have led to premature failure of heat exchanger tubes. Even otherwise because of flow-induced vibration tube fretting-wear has been recognized as a significant contributor to the life of heat exchanger tube. To limit vibrations, supports (Baffles) are introduced at various locations along the tube. Tube-support assemblies are usually loose fitting due to manufacturing considerations, which require these clearances to facilitate tube bundle assembly. When the tube response due to flow-induced forces exceeds the clearance of the tube support, the tubes impact on their supports. In addition to impact, sliding forces also develop at the tube-support interface. Conditions are then favorable for tube fretting-wear, which leads to thinning of tube walls and eventual through wall wear and tube failure. Though there are many incidences of premature failures, technical details of only few of them have been reported in literature. Such cases from literature [62, 4, 71] are chosen to analyze heat exchanger tube failures due to flow-induced excitation.

Figure 8.1 shows schematic representation of fretting- wear damage assessment of heat exchanger tubes. The flow excitation mechanisms, vortex shedding and turbulent buffeting can cause vibration with significant amplitudes to heat exchanger tubes. Turbulence in the flow can affect the existence and strength of vortex shedding excitation mechanism. However, fluid elastic instability can be avoided by limiting the flow velocity [13]. Excitation forcing function is defined on a heat exchanger tube for the time simulation. The time domain response analysis is carried out to capture the tube to tube support dynamic interaction. The interaction force and
duration are important to estimate fretting wear-rate and life of heat exchanger tubes.

![Diagram](https://via.placeholder.com/150)

**Figure 8.1: Fretting wear damage assessment procedure**

An analytical tool is developed to carry out fretting wear damage assessment of heat exchanger tubes subject to vortex shedding and turbulent buffeting excitation mechanisms. The methodology involves definition of a structural dynamic model of the heat exchanger tube with intermittent baffle support having design clearance, definition of excitation forces, selection of the resulting dynamic equation of motion to obtain the response, computation of wear rate from the response data, extension to obtain the time for through-wall wear of the tube. The developed tool is validated solving two premature failure cases due to vortex shedding vibration and two premature failure cases due to turbulent buffeting vibration. The next section presents
different cases of premature failure due to both excitation mechanisms.

### 8.2 Premature failure cases

The life of heat exchanger tube have been verified studying various documented cases of premature failure from literature [62, 71, 64]. As a first case, floating-head-split-ring-pull-through bundle shell and tube heat exchanger is studied. The other cases are taken from DOE/ANL/HTRI/ heat exchanger tube vibration data bank [64]. HTRI prepares the data cases to be added to the tube vibration data Bank. Some cases have been forwarded to Argonee National Laboratory (ANL) to add into tube vibration data bank [64]. Each case has been analyzed separately as follows:

#### 8.2.1 Case 1: Thermo siphon reboiler tube arrangement

Paidoussis [4] reported this premature failure of shell and tube heat exchanger. This TEMA style heat exchanger failed within five to six days after start up. The problem faced by designer is 36 hours after start-up the tube bundle of shell and tube heat exchanger failed [4, 62]. The design information is shown in Table 5.1. As brought out in section 5.2.1.2, flow-induced vibration design check for this class of problem has been performed. It has been oberved that the cause of damage is vortex shedding excitation.

#### 8.2.1.1 Finite element simulation

FEA model of this class of problem is shown in Figure 5.2. The damping matrix is formed based on the Rayleigh proportional damping where the damping matrix is a linear combination of mass and stiffness matrices [4]. The tubes fixed in tube sheet, boundary conditions are considered as fixed-fixed. Support 2 experiences tube contact most frequently and with the greatest intensity since it is the restraint most effective in the lower modes. Hence the effect of higher modes becomes increasingly more important. The springs used to simulate the supports are very stiff. The higher stiffness could result in the dominance of contribution of higher modes. The time step is chosen as small fraction of the lowest natural period of the tube. It is necessary to use minimum value of support stiffness to capture tube dynamics accurately. As brought out in chapter 7 the minimum value of support stiffness is taken as 10000 times stiffness of the beam [72].
8.2.1.2 Time history response

Figure 8.2 shows time history response of node 13 (support 2) over 62.7 sec including first five modes, by reduced order method. Vortex shedding is harmonic in time, the forcing amplitude is a function of flow velocity, as per equation (5.1). Due to resonance, shedding frequency gets ‘locked in’ with the natural frequency of the tube. Resonance increases the amplitude till it goes and touches the baffles. After transients, the region exponentially reaches the steady state. The time history is computed over 10 seconds when response becomes asymptotic. The dynamics at impact is captured from 62 seconds to 72.1 seconds shown in Figure 8.3. The continuous hits at each cycle is observed. This duration of contact has been utilized while calculating work rate.

![Figure 8.2: Time history response (node 13)](image)

![Figure 8.3: Time history, when response becomes asymptotic for 10 sec.](image)
8.2.1.3 Fretting wear damage estimation and life prediction
Work rate is calculated to estimate volume of fretting wear. The work rate is computed utilizing interaction force and tube-support contact duration. The work rate is computed as 0.196 watt. For first 10 seconds the tube wall wear depth is calculated as 0.00012 mm. A time increment of $10^4$ seconds is used. The time is incremented until the tube wall wear depth reaches to the thickness of tube. Figure 8.4 shows that the tube wear wall depth increases with time. The wear calculation shows tube wall wear depth reaches to the thickness of tube in 34 Hrs.

![Figure 8.4: The predicted tube wall wear depth](image)

8.2.1.4 Discussion
Studying above case it is observed that, heat exchanger tube failed due to resonance, the cause of damage is vortex shedding mechanism. Time history response shown in Figure 8.2 shows that after 61.7 seconds steady state amplitudes are reached. The interaction force and time duration is utilized to compute work rate, which is used to estimate volume of fretting wear. For first 10 seconds tube wall wear depth is computed. A time increment is used to calculate the incremental wear rate. Proceeding further, it was found that at 34 Hrs fretting wear is equal to thickness. The actual life reported in literature [62] is 36 Hrs. It is seen that the current wear rate is in agreement with the reported life. One more premature failure case of TEMA style heat exchanger has been studied. The detail study of this failure is reported in case 2.
8.2.2 Case 2: Shell and tube TEMA style heat exchanger

Wambsganss et al. [71, 64] have reported a case of pre-mature failure. The heat exchanger tubes failed at the outlet span within six days after start up. The design is bundle of 1160 tubes 6.096 m long supported in baffles. The design information is shown in Table 8.1.

Table 8.1: Design information Shell and tube TEMA style heat exchanger

<table>
<thead>
<tr>
<th>Sr No</th>
<th>Parameters</th>
<th>Misc and Geom</th>
<th>Inlet Span</th>
<th>Mid Span</th>
<th>Outlet Span</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Tube span,(mm)</td>
<td>---</td>
<td>1325.63</td>
<td>3317.75</td>
<td>1452.63</td>
</tr>
<tr>
<td>2</td>
<td>Tube outside Dia.,(mm)</td>
<td>19.05</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>3</td>
<td>Tube inside Dia.,(mm)</td>
<td>17.8054</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>4</td>
<td>Velocity (m/sec)</td>
<td>---</td>
<td>0.4</td>
<td>0.4</td>
<td>0.4</td>
</tr>
<tr>
<td>5</td>
<td>Strouhal Number</td>
<td>---</td>
<td>0.157</td>
<td>0.157</td>
<td>0.157</td>
</tr>
<tr>
<td>6</td>
<td>Moment of inertia, (mm$^4$)</td>
<td>1530.2</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>7</td>
<td>Mass of tube (Kg)</td>
<td>2</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>8</td>
<td>Density of tube (Kg/m$^3$)</td>
<td>8027</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>9</td>
<td>Mod. of elasticity (N/mm$^2$)</td>
<td>194.9 x 10$^4$</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>10</td>
<td>Design press. Shell Side (N/mm$^2$)</td>
<td>0.517</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>11</td>
<td>Design press. Tube Side(N/mm$^2$)</td>
<td>0.75</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>12</td>
<td>Pitch,mm</td>
<td>25.4</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>13</td>
<td>Tube pattern</td>
<td>Rotated square 45$^0$</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>14</td>
<td>Fluid density (Kg/mm$^2$)</td>
<td>1000</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
</tbody>
</table>

8.2.2.1 Flow-induced vibration design check

The design check is performed at mid span, inlet span and outlet span. The lowest natural frequency for a beam shown in Figure 8.5 is calculated as 4.53 Hz. The vortex shedding frequency is calculated at each span. Resonance occurs at outlet span due to coincidence of vortex shedding frequency with natural frequency. The cause of damage is vortex shedding vibration.

8.2.2.2 Finite Element simulation

Figure 8.5 shows FEA model, representing a tube arrangement subjected to flow forces. Since the tubes are fixed in tube sheet, tube boundary conditions are considered as fixed-fixed. The distribution of flow forces is shown as uniformly distributed force over each span. The tube is
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![Diagram of FEA model](image)

- **Tube**: OD = 19.05 mm, L = 6.096 m
- **E**: $1.95 \times 10^{-5}$ N/mm²
- **ρ**: $8.027 \times 10^{-3}$ kg/m³
- **Gap**: 0.4 mm
- **Support**: Stiffness $K_{Gap} = 0.252 \times 10^6$ N/m
- **Material Const DAMP**: 0.25 N/m

**Tube damping**
- $C = \beta M + \gamma K$
- $\beta = 4.15$
- $\gamma = 0.323 \times 10^{-5}$

**Forcing function**
- $F_y = 0.83$ N/m
- $CL = 1.0$
- $f = 4.526$ Hz
- $\omega_s = 2\pi f_s = 28.42$

Figure 8.5: FEA model, representing a TEMA style heat exchanger tube configuration

excited by vortex shedding forces which are distributed along the entire length of the tube.

### 8.2.2.3 Time history response

Figure 8.6 shows time history response when steady state amplitudes are observed. The dynamics at impact is captured over 3.56 min. It is observed that heat exchanger tube hits the baffle continuously at each cycle. Tube support contact duration for first 10 seconds is used to calculate work rate.
8.2.2.4 Fretting wear damage estimation and life prediction

Normal work rate is computed as 29.826 watt. For first 10 seconds the tube wall wear depth is computed as $7.21 \times 10^{-6}$ mm. A time increment of $10^4$ seconds is used. The wear prediction shows tube wall wear reaches to the thickness of tube in 118 Hrs shown in Figure 8.7.

8.2.2.5 Discussion

Studying above case it is observed that tube failed due to coincidence of tube natural frequency and vortex shedding frequency. The time history response is checked over 3.56 min. For first 10 seconds tube wall wear depth is computed. The wear prediction shows tube wall wear reaches
to the thickness of tube in 118 Hrs. whereas the reported life in literature [64] is 144 Hrs.

8.2.3 Review of case 1 and case 2
The calculated wear for both the cases is in agreement with the reported wear. It should be noted that there are factors like leakages, number of hours of operation and hardening due to cold working, which contribute to wear and life of heat exchanger tubes. As wear progresses hardening due to cold working reduces wear rate.

8.2.4 Case 3: CEM TEMA style heat exchanger
FEA model of CEM TEMA style heat exchanger is shown in Figure 5.6. This heat exchanger was operated successfully for one and half years at the design conditions [64]. Minor cutting at the baffle is reported after one and half years. The design information of this class of problem is shown in Table 5.2.

8.2.4.1 Flow-induced vibration design check
For a beam of geometry shown in Figure 5.6 the lowest natural frequency is calculated as 4.41 Hz. The design criterion for the possibility of vortex shedding as an excitation source involves the parameter of reduced frequency and/or the determination of Strouhal number. The mechanism can be characterized by Strouhal number. Correlations of weaver et.al [16] give the Strouhal number \( S_u \) for 60° tube layout pattern. The Strouhal number is computed as 0.69. It is related to vortex shedding frequency. The vortex shedding frequency \( f_{vs} \) is calculated from flow velocity approaching the tube and tube diameter and is computed as 0.024 Hz. As there is difference between vortex shedding frequency and tube natural frequency no resonance takes place and “lock-in” is also avoided. Hence there is no possibility of vortex shedding as an excitation source. Even in the absence of vortex shedding the random vibration causes damage to the tube.

The design criterion to avoid fluid elastic instability as an excitation source is the ratio of actual to critical flow velocity should be less than one. For first eight vibration modes, ratio of actual to critical flow velocity is observed to be less than one. This means that fluidelastic instability is not possible for this heat exchanger tube.
For this class of problem there is no possibility of vortex shedding and fluid elastic instability vibration. The possibility of turbulent buffeting is studied further.

8.2.4.2 Time history response

FEA model of this heat exchanger is shown in Figure 5.6. Support 12 at the mid span experiences tube contact most frequently and with greatest intensity, hence middle support is checked. As brought out in Chapter 7, including adequate number of modes the dynamics can be captured accurately for this class of problems. Figure 8.8 shows, time history response of a heat exchanger tube over 15 minutes including first five modes, by reduced order method. Residue based variable time stepping Newmark integration solution scheme [11, 70] is used to capture dynamics. Time history shows intermittent hits due to random nature of vibration. Figure 8.9 shows time history over 2 seconds where intermittent hits are shown as circled. In turbulent buffeting vibration, due to intermittent contact there is need to analyze contact ratio over entire time period.

![Graph showing time history response over 15 minutes (Support 12)](image-url)

Figure 8.8: Time history response over 15 minutes (Support 12)
8.2.4.3 Response comparison

Figure 8.10 shows time history response, when response becomes asymptotic (refer section 8.2.2.3) due to vortex shedding vibration. This response is used to compare the hits observed over the response. One significant difference in the response is tube to support contact is intermittent shown in Figure 8.9 due to turbulent buffeting vibration whereas it is continuous at each cycle shown in Figure 8.10 due to vortex shedding vibration.

Figure 8.9: Time history over 2 seconds

Figure 8.10: Response, TEMA style heat exchanger tube configuration
8.2.4.4 Discussion on TEMA acceptance criteria

At this point it may be relevant to discuss the TEMA acceptance criteria for turbulent buffeting design check. A recommended TEMA [55] acceptance criterion is $y_{rms} \leq 0.02D$. Heat exchanger tubes respond in a random manner to turbulence in the flow field. Assuming tube vibrations represent steady-state random process, the mean square response is given by equation as follows [57]:

$$\overline{y_{max}^2} = \frac{[C_R f_0 \rho V^2 D]^2}{128 \pi^3 \zeta f_R^3 m^2} \quad (8.1)$$

$$y_{rms} = \sqrt{\overline{y_{max}^2}} \quad (8.2)$$

As brought out in section "5.3.3.1", this class of problem does not satisfy the TEMA acceptance criteria. The RMS value of mean square response is restricted to 0.02 $D$, which is small for the heat exchanger tube being considered, this could be equivalent to a small excitation. The time history response is computed taking this small value of equivalent forcing function. The time history response for this modified excitation is computed over 15 minutes as shown in Figure 8.11. It is observed that tube never hits the support over entire time duration considered.

![Figure 8.11: Time history using small factor of forcing function](image-url)
The response shown in Figure 8.8 is observed over 15 minutes. The contact of tube and baffle is intermittent. The fretting wear occurs at the contact of tube and baffle. Due to intermittent contact there is need to analyze contact ratio over entire time period. Larger time period is used to ensure that statistical parameters are adequately captured. Since response is random the convergence of contact ratio is studied as follows.

8.2.4.5 Contact Ratio

Percentage change in contact ratio is plotted by considering various duration’s of response as shown in Figure 8.12. The duration is increased till the convergence is obtained. It is observed that after 600 seconds there is negligible change in contact ratio value. The convergence study shows below 1% convergence is achieved at 600 seconds. Hence the time of 600 seconds is utilized to compute the work rate.

![Figure 8.12: Convergence of contact ratio](image)

8.2.4.6 Fretting wear damage estimation and life prediction

Work-rate is calculated to estimate volume of fretting wear. The work rate is computed utilizing interaction force and tube-support contact duration. The work-rate is computed as $1.92 \times 10^{-8}$ watt. The tube wall wear depth is calculated as $4.65 \times 10^{-6}$ mm. A time increment of $10^5$
seconds is used. The time is incremented until the tube wall wear depth reaches to the thickness of tube. Figure 8.13 shows that the tube wear wall depth increases with time. The wear calculation shows tube wall wear depth reaches to the thickness of tube in 23 months.

![Graph showing tube wall wear depth vs. tube life in months]

Figure 8.13: The predicted tube wall wear depth

8.2.4.7 Discussion

Time history shows random response. The interaction force and time duration is utilized to compute work rate, which is used to estimate volume of fretting wear rate. The time of 600 seconds is used to compute tube wall wear depth. A time increment is used to calculate the incremental wear rate. Proceeding further tube wall wear depth reaches to the thickness of tube in 23 months (24 hours of 30 days per month operation). In the actual case minor cutting of the tubes at the baffle is reported after 18 months. Considering various uncertainties like actual hours of operation, failure before through wall wear, wear rate changes with cold working etc. could contributes to wear rate and life of heat exchanger tubes. The cause of tube damage is turbulent buffeting excitation mechanism.

One more case of CEN type TEMA heat exchanger has been studied. This heat exchanger is used in refinery services. The detail study of this failure has been reported in case 4 as follows:
8.2.5 Case 4: CEN type TEMA heat exchanger

This 45-in. diameter by 40-ft long heat exchanger used in refinery services [64]. The shell side fluid is an olefin-isooctane mixture with hydrofluoric acid. It is reported that 14 tubes near the baffle tips at mid span damaged. The design information is shown in Table 8.2

Table 8.2: Design information CEN type TEMA heat exchanger

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>CEM TEMA Heat Exchanger</th>
<th>Parameters</th>
<th>Geometry</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Tube</td>
<td>Outside diameter (mm)</td>
<td>25.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Wall thickness (mm)</td>
<td>3.4036</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Material</td>
<td>carbon Steel</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Length (mm)</td>
<td>12021</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Tube Layout</td>
<td>Rotated Square 45°</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Tube Pitch (mm)</td>
<td>29.76</td>
</tr>
<tr>
<td>2</td>
<td>Shell</td>
<td>Orientation</td>
<td>Horizontal</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Inside diameter (mm)</td>
<td>1143</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Material</td>
<td>Carbon Steel</td>
</tr>
<tr>
<td>3</td>
<td>Baffle</td>
<td>Type</td>
<td>Double-segmental</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Thickness (mm)</td>
<td>15.875</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diametral Clearance (mm)</td>
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</tr>
<tr>
<td></td>
<td></td>
<td>Number of Baffles</td>
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</tr>
<tr>
<td>4</td>
<td>Shell side fluid</td>
<td>Shell Side Fluid Density (Kg/mm³)</td>
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<tr>
<td></td>
<td></td>
<td>Flow rate (Kg/hr)</td>
<td>1.49 x 10⁶</td>
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<tr>
<td></td>
<td></td>
<td>Flow Velocity (m/sec)</td>
<td>0.58</td>
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</table>

8.2.5.1 Flow-induced vibration design check

For a beam of geometry shown in Figure 8.14 the lowest natural frequency is calculated as 6.21 Hz. The strouhal number is computed as 0.42. The vortex shedding frequency \( f_{vs} \) is calculated from flow velocity approaching the tube, and tube diameter. It is computed as 9.59 Hz. Vortex shedding frequency and tube natural frequency does not match, no resonance takes place and “lock-in” is also avoided. Hence there is no possibility of vortex shedding as an excitation source.

The maximum mean square response is computed as 1.57 mm, which does not satisfy the TEMA acceptance criteria. Hence there is possibility of turbulent buffeting as an excitation source.
For first five vibration modes, ratio of actual to critical flow velocity is observed to be less than one. This means that fluid elastic instability is not possible for this class of problem.

### 8.2.5.2 Finite element simulation

Figure 8.14 shows FEA model, representing a heat exchanger tube arrangement subjected to flow forces. The model consists of 12.021 m long tube fixed at both ends in a tube sheet and loosely supported at the 12 baffle supports. The tube material is carbon steel having outside diameter 25.4 mm. The tube is fixed in a tube sheet, boundary conditions are considered as fixed-fixed. Tube failure is reported at baffles near the center of tube length. Support 6 at the mid span experiences tube contact most frequently and with greatest intensity. Hence time history response at support 6 is checked.

![FEA model diagram](image)

**Tube:**
- OD = 25.4 mm
- ID = 22 mm
- L = 12.021 m

**Support (Stiffness KGap):**
- K = 1.65 x 10^5 Nm
- Material Const. CDAMP = 0.25 N/m
- Number of supports = 12

**Tube damping (Fluid + Material):**
- C = βM + γK
- β = 4.15
- γ = 0.323 x 10^-5

**Material Properties:**
- E = 1.95 x 10^5 N/mm^2
- ρ = 8.027 x 10^3 kg/m^3

**Gap:**
- 0.39624 mm

Figure 8.14: FEA model, representing heat exchanger tube arrangement
8.2.5.3 Time history response
The time history response of a heat exchanger tube over 10 minutes including first five modes by reduced order method is shown in Figure 8.15. The intermittent hits are observed, which is shown as circled.

![Time history response over 10 minutes (Support 6)](image)

Figure 8.15: Time history response over 10 minutes (Support 6)

8.2.5.4 Contact ratio
Percentage change in contact ratio is plotted over the time duration shown in Figure 8.16. Convergence study shows below 2% convergence is achieved at 400 seconds. The time of 400 seconds is utilized to compute the work rate.

![Convergence of contact ratio](image)

Figure 8.16: Convergence of contact ratio
8.2.5.5 Fretting wear damage estimation and life prediction
Work-rate is calculated to estimate volume of fretting wear. The work rate is computed utilizing interaction force and tube-support contact duration. The work rate is computed as 0.01 watt. The tube wall wear depth is calculated as $1.82 \times 10^{-6}$ mm. A time increment of $10^5$ seconds is used. The time is incremented until the tube wall wear depth reaches to the thickness of tube. Figure 8.17 shows that the tube wear wall depth increases with time. The wear calculation shows tube wall wear depth reaches to the thickness of tube in 45.66 months.

![Graph showing tube wear depth vs. tube life in months.](image)

Figure 8.17: The predicted tube wall wear depth

8.2.5.6 Discussion
Studying the above case it is observed that the tube failed due to turbulent buffeting excitation. The time history response is checked over 10 min. For first 400 seconds tube wall wear depth is computed. The wear prediction shows tube wall wear reaches to the thickness of tube in 45.66 months (24 hours of 30 days per month operation) whereas this heat exchanger is operated successfully for 54 months.

8.2.6 Review of case 3 and case 4
In both the cases the calculated life is close to actually observed life. For the first case heat exchanger is successfully operated for 18 months whereas wear prediction shows tube wall
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wear depth reaches to the thickness of tube in 23 months. Considering various uncertainties predicted wear rate is in good agreement with the reported life of heat exchanger. However in the second case early tube failure is observed in 45.66 months against reported life of 54 months. As mentioned earlier this difference could be attributed to various factors such as the actual number of hours of operation, strain hardening effect, leakage etc.

8.3 Summary
This chapter presents estimation of fretting wear limited life of a shell and tube heat exchanger tubes. Analytical formulation, computer implementation, and qualification against actual reported failure were presented. Two documented premature failure cases due to vortex shedding mechanism and turbulent buffeting mechanism have been implemented in the program.

8.4 Conclusion
Studying various premature failure cases of heat exchanger due to vortex shedding and turbulent buffeting vibration, it was observed that the calculated life is in good agreement with the reported life. It should be noted that there are many factors which contribute to wear and life of heat exchanger tubes. As wear progresses hardening due to cold working reduces wear rate. However for some cases, early tube failure was observed against reported life of heat exchanger.