Outline

- Background
- Measurements on gear casing
- Calculations
  - Static
  - Dynamic
- Comparing measurements
- Discussion
Background

- North sea platform production modification
- Two compressor trains were upgraded from 18 MW -> 21 MW
- Main modifications:

<table>
<thead>
<tr>
<th>Property</th>
<th>Modified</th>
<th>Original</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power [MW]</td>
<td>21.30</td>
<td>18.25</td>
</tr>
<tr>
<td>Speed in [rpm]</td>
<td>3600</td>
<td>3600</td>
</tr>
<tr>
<td>Speed out [rpm]</td>
<td>10718</td>
<td>10894</td>
</tr>
<tr>
<td>Module</td>
<td>5.6</td>
<td>6.4</td>
</tr>
<tr>
<td>Z1</td>
<td>44</td>
<td>38</td>
</tr>
<tr>
<td>Z2</td>
<td>131</td>
<td>115</td>
</tr>
<tr>
<td>Type</td>
<td>Single helical</td>
<td>Single helical</td>
</tr>
</tbody>
</table>
Problems encountered:
- Two consecutive gearbox failures within few weeks after commissioning
- ~ 500-1000 operating hours before failure
- Severe gearbox casing vibrations and excessive noise levels recorded
- Turbine side on bull gear experienced fractured teeth and cracks on the load surfaces

Actions taken:
- Full RCA initiated. This concluded poor final grinding as primary cause of failures
- Contributory causes had to be investigated as part of the RCA
- Torsional/Lateral vibration analysis was initiated by LRC as part of the RCA
Measurements

- Vertical accelerometer
- Horizontal accelerometer
Measurements

~1700 Hz  Gear mesh frequency (7000-8000 Hz)
Measurements

Requirements

<table>
<thead>
<tr>
<th>API613 5th Ed.</th>
<th>Velocity (RMS)</th>
<th>Acceleration (P-P)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency range</td>
<td>10 – 2500 Hz</td>
<td>2500 – 10 000 Hz</td>
</tr>
<tr>
<td>Overall</td>
<td>2.9 mm/s</td>
<td>8 g (~80 m/s²)</td>
</tr>
<tr>
<td>Discrete frequency</td>
<td>1.8 mm/s</td>
<td>-</td>
</tr>
</tbody>
</table>

10200 rpm = GMF @ 7480 Hz
Levels up to 1600 m/s² (~160 g) p-p
Measurements

Amplification factor $Q \approx 700$

44X filter applied on signal

$\sqrt{2}$
Pinion mode excitation

Bearing stiffness range
~1 – 2.5 E+9 N/m
(~6 - 14 E+7 lb/in)

88000 – 97000 cpm
(1470 – 1620 Hz)

1X = 10 718 rpm
8X = 85 744 rpm
9X = 96 462 rpm
10X = 10 7180 rpm
Pinion mode excitation

Damped eigenvalue analysis
- Mode no. 8: 1813.8 Hz, log. dec. = -6.149
- Speed: 10718 rpm

@ 0 rpm → 77 460 cpm (1291 Hz)
@ 10718 rpm → 78 840 cpm (1314 Hz)

1X = 10 718 rpm
7X = 75 026 rpm
8X = 85 744 rpm
9X = 96 462 rpm
10X = 10 7180 rpm
Torsional-Lateral Calculations

42 Elements
Torsional-Lateral Calculations

Displacement in gear contact

Tooth stiffness

\[ \delta_1 = -r_1 \theta_1 - x_1 \cos(\alpha) - y_1 \sin(\alpha) \]
\[ \delta_2 = r_2 \theta_2 - x_2 \cos(\alpha) - y_2 \sin(\alpha) \]

\[ P_i = \begin{cases} k(\delta_{1i} - \delta_{2i} - \delta_{ci}) & \delta_{1i} - \delta_{2i} > \delta_{ci} \\ 0 & \delta_{1i} - \delta_{2i} \leq \delta_{ci} \end{cases} \]

i = 1..43
Results – Dynamic calculations

Pinion bearing force [N]

Specific load [N/mm]

1780 Hz

7760 Hz
Conclusions

- The primary cause of failures was residual stress in tooth flanks from the manufacturing process. Contributory causes (such as torsional/lateral analysis) were investigated as part of the RCA.

- Spare gear set was sent onshore for a second “final grinding” to remove residual stress on the load flanks.

- The presented calculations were tuned with measurements from before and after modifications. The resulting loads were input to load flank fatigue calculations by the vendor.

- Fatigue calculations from before and after the machining could prove that the load flank fatigue life improved with the second grinding.

- Gear boxes are still in operation with no reported issues since commissioning March 2010 and June 2012 for the two trains, respectively.
Questions?