Root Cause Analysis of a vibration problem in a propylene turbo compressor

Pieter van Beek, Jan Smeulers
Problem description

• A newly installed turbo compressor system for propylene showed vibrations in the piping system and rotor.
• After that supporting layout was significantly improved measurements showed that vibrations were within the allowable range.
• Still the rotor vibrations were not acceptable.
• A root cause analysis was carried that showed two likely causes.
The installation

- Large diameter piping (60 inch suction).
- Reducer to 48 inch just upstream of the inlet of the compressor.
- Large flows \(\sim 1500\) tonnes/h (25 m/s).
- Heavy gas \(\sim 44\) kg/kmol @ 6.6 barg.
- 2 phase flow after 2\(^{nd}\) stage condenser: liquid separation via large K.O. drum at the suction side (15 m height).
The compressor

• 2 stage radial turbo compressor ~ 2800 rpm & 23 MW (horizontally split).
• 1 inlet and 2 outlets underneath compressor.
• Discharge stages connected with suction via anti-surge valves (ASV 1 and 2).
The installation

- Essential part: separator, suction system, 1\textsuperscript{st} stage discharge (yellow), turbo-compressor.
1 - Pipe system design

- Original design did not account for pulsations and vibrations.
- Flexible / spring pipe supports with gaps.
- After first start-up large vibrations.
- Pipe supports have been reinforced significantly.
- Verification measurements showed acceptable vibration levels.
- Rotor vibrations still present.
- Root cause unknown.
Pulsation and vibration measurements

- Fixed measurement points: 5 puls. & 7 vibr. Locations.
- Also measurements with hand held equipment: 27 locations (tri-axial).

- Measurement program:
  - Varying ASV settings.
  - Varying RPM / load.
Measured vibrations

- Typical vibration spectra on V2 (close to compressor) show vibrations at low frequencies:
  - 0 – 15 Hz.
  - 20 – 40 Hz.
  - 45 Hz (compr. speed).
- Due to the improved pipe supports vibration levels are acceptable, both displacements and velocities.
Measured pulsations

- Typical pressure spectra show acoustic resonances at low frequencies → flow induced pulsations (FIPs).
- Pulsations reach the allowable pulsation levels of API 618 .... For reciprocating compressors!
- Pulsation levels up to approximately 16 kPa → vibration forces in the order of 10 kN on the piping @ 3.2 Hz!
Flow Induced Pulsations

- Pulsations are caused by vortex shedding in a T of a closed side branch.
- The vortex frequency depends linearly on the flow velocity and diameter of the side branch.
- Pulsations are amplified if the vortex frequency is equal to the resonance frequency of the side branch.

\[ f = Sr \cdot \frac{U_0}{D} \]

\[ Sr \approx 0.4 \]
Flow Induced Pulsations

- Examples for the present system:
- 1\textsuperscript{st} and 2\textsuperscript{nd} stage ASV lines, when valves (partially) closed.

<table>
<thead>
<tr>
<th>Percentage</th>
<th>Gas velocity</th>
<th>f vortex</th>
<th>1/4 lambda</th>
<th>3/4 lambda</th>
<th>5/4 lambda</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>10,0</td>
<td>3,1</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>45</td>
<td>11,3</td>
<td>3,5</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>50</td>
<td>12,5</td>
<td>3,9</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>55</td>
<td>13,8</td>
<td>4,3</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>60</td>
<td>15,0</td>
<td>4,7</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>65</td>
<td>16,3</td>
<td>5,1</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>70</td>
<td>17,5</td>
<td>5,5</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>75</td>
<td>18,8</td>
<td>5,9</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>80</td>
<td>20,0</td>
<td>6,3</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>85</td>
<td>21,3</td>
<td>6,7</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>90</td>
<td>22,5</td>
<td>7,1</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>95</td>
<td>23,8</td>
<td>7,5</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>100</td>
<td>25,0</td>
<td>7,9</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>105</td>
<td>26,3</td>
<td>8,3</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
<tr>
<td>110</td>
<td>27,5</td>
<td>8,6</td>
<td>7,5</td>
<td>22,5</td>
<td>37,5</td>
</tr>
</tbody>
</table>
Rotor vibrations

- Proximity probes on rotor show instability in orbits at > 90% load.

- Rotor vibrations have similar frequencies as both the vibrations and pulsations: < 10Hz, 20 – 40 Hz and 45 Hz.

[Graphs of rotor vibrations and displacement spectrum]

Typical rotor displacement spectrum
Rotor vibrations

- Although the API617 Level II stability criteria are met (log. dec. >0.1), still high rotor vibration occur.
- First critical of the rotor around 36 Hz (on site mech. run test) → first lateral resonance mode excited by a broadband source around this frequency.
- Vibrations in 20-40 Hz range increase with increasing compressor speed / flow.
- A Root Cause Analysis has been made.
Root Cause Analysis (RCA) rotor vibrations

First: overview suction side compressor, between K.O. drum and compressor inlet:
Suction side compressor

- Several **out of plane** sharp bends in the suction piping.
- Distance between sharp 90 degr. T joint and elbow <10D.
- Distance between elbow and inlet ~5D.
- Filter in T joint.
- Butterfly valve just upstream T joint.
- 60 → 48 inch reduced just before compressor inlet.
- Low point in between K.O. drum and inlet: 20 inch draining boot.
Root Cause Analysis (RCA) rotor vibrations

Schematic of possible mechanisms that can lead to rotor vibrations

- Acoustic excitation
  - Flow-induced pulsations
  - Turbulence
  - Compressor aerodynamic noise
  - ....

- Mechanical excitation
  - Pipe line vibrations
  - Compressor foundation vibrations
  - ....

- Flow excitation
  - Non-uniform inflow
  - Swirling flow
  - ....

- Multi-phase excitation
  - Liquid ingestion / slugs
  - ....
## RCA matrix

<table>
<thead>
<tr>
<th>Mechanism</th>
<th>Description</th>
<th>Mitigation measures</th>
<th>Judgment</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Acoustic – compressor aerodynamic noise</td>
<td>Tonal, high-frequent excitation, caused by rotor-stator interactions</td>
<td></td>
<td>Unlikely</td>
</tr>
<tr>
<td>2 Mechanical – piping vibrations</td>
<td>Connecting piping vibrations exciting the compressor and triggering the rotor instability</td>
<td></td>
<td>Unlikely</td>
</tr>
<tr>
<td>3 Mechanical – foundation vibrations</td>
<td>Concrete pedestal vibrations are mechanically exciting the compressor and rotor</td>
<td></td>
<td>Unlikely</td>
</tr>
<tr>
<td>4 Rotating stall in the compressor</td>
<td>Flow in impeller gets unstable at a certain load</td>
<td></td>
<td>Unlikely, RS occurs at reduced flow</td>
</tr>
<tr>
<td>5 Mechanical malfunction in compressor</td>
<td>Run out of clearance → rubbing</td>
<td></td>
<td>Unlikely</td>
</tr>
<tr>
<td>Mechanism</td>
<td>Description</td>
<td>Mitigation measures</td>
<td>Judgment</td>
</tr>
<tr>
<td>------------------------------------------------</td>
<td>------------------------------------------------------------------------------</td>
<td>--------------------------------------------------------------------------------------</td>
<td>-------------------------------</td>
</tr>
<tr>
<td>6     Acoustic – flow-induced pulsations</td>
<td>Resonance in closed side branch; vortex shedding</td>
<td>Relocation of valve; reduce flow speed in main piping; apply restriction in branch</td>
<td>Likely to occur, but not the critical effect for rotor vibrations</td>
</tr>
<tr>
<td>7     Acoustic – pressure fluctuations caused by turbulence in flow</td>
<td>Broad-band, low-frequent excitation of impeller and rotor</td>
<td>Reduce flow speed</td>
<td>Likely</td>
</tr>
<tr>
<td>8     Multi-phase excitation – liquid ingestion</td>
<td>Accumulated liquid is entrained into the compressor; varying liquid → unsteady load on the rotor</td>
<td>Improve separator, avoid liquid accumulation in upstream piping; thermal insulation piping</td>
<td>Likely</td>
</tr>
<tr>
<td>9     Flow excitation – non uniform inflow</td>
<td>Short radius elbows → varying load on compressor</td>
<td>Apply large radius elbows, flow straightener</td>
<td>Likely</td>
</tr>
<tr>
<td>10    Flow excitation – swirling inflow</td>
<td>Double out-of-plane elbows induce swirling flow that may not be re-developed before impacting on the compressor</td>
<td>Increase distance between elbows and compressor, flow straightener</td>
<td>Likely</td>
</tr>
</tbody>
</table>
RCA rotor vibrations

1. Liquid in suction flow.

The internals of the K.O. drum have been modified:
- Liquid carry over to compressor inlet mitigated.
- Compressor now runs stable up to 106% compressor speed!
- Rotor vibration amplitudes still high (50 μm pk-pk). However, this is acceptable according to compressor manufacturer.
RCA rotor vibrations

2. FIPs in combination with flow distortion.

The high vibration amplitudes can be caused by FIPs and flow distortion:

- Sharp bends in the suction piping can induce unsteady flow distortion.
- Double out-of-plane bend will cause (unsteady) swirl in flow.
- Reducer close to compressor inlet can increase flow distortion.

→ CFD analysis compressor inlet section performed.

Note: especially combination of rather undamped rotor and flow FIP / distortion can lead to high rotor vibration amplitudes.
2. FIPs in combination with flow distortion – CFD analysis.

- High Reynolds number and large geometry dimension require super-fine boundary-layer mesh.
- Also very fine mesh needed at butterfly valve and filter section.
- Code-to-code comparison carried out; separation behaviour checked.
2. CFD analysis - results.

- Filter dominant obstacle:
  - Imposing the main pressure drop.
  - Redirection of flow at large scale vortical structures and small scale turbulence.
- K.O. drum inflow turbulence no significant impact on flow topology.
- Generally no flow separation.
RCA rotor vibrations

2. CFD analysis - Compressor inlet conditions; velocity, turbulence kinetic energy and vorticity (z-direction).

Only weak, counter rotating vortices at the inlet.
Conclusions

- Design philosophy did not consider *Flow Induced Pulsations (FIPs)*.
- Improved pipe supports reduced vibrations but do not eliminate the source.
- Rotor instability mainly caused by liquid carry over K.O. drum to compressor inlet.
- No fluctuating swirling flow into the compressor, mainly due to pressure drop over filter and high Reynolds number flow.
- Rotor most likely too susceptible for disturbances. Not critical anymore, but additional improvements planned.
Lessons learned

- FIPs can cause serious vibration problems at low frequencies.
- A pulsation and vibration analysis for this large diameter pipe systems should be part of the design.
- 3D pipe bend configuration in the suction piping can lead to flow distortion. To avoid this long radius bends should be applied or guide vanes could be installed in the bends.
- Take actual inlet flow into account in rotor damping (seals) and stiffness (bearing clearance) design.
Thank you for your attention!

Pieter van Beek
TNO
Heat Transfer & Fluid Dynamics
tel. +31 (0)88 8666366
Pieter.vanbeek@tno.nl