Fast and Ultimate Vibration Field Solution: From Problem Detection to Field Performance Validation

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26th International Pump Users Symposium
March 15 - 18, 2010
George R. Brown Convention Center
Houston, Texas
Summary

- The case
- Initial analysis
- Root cause analysis
- Solution implementation
- Results
- Conclusions
Description of the problem

During commissioning, customer reported unacceptable vibration levels on pumps tested with water.
The pump

The pump type/size is an API 12" discharge with top-top configuration, double suction impeller, double volute, antifriction bearings configuration, 360° mounted, center mounted (BB2)
First steps of the investigation

On site inspection to verify mechanical integrity of pumps

Campaign of vibration measurement on all installed pumps
Initial analysis

ISO 3945 limit = 7.1 mm/s (0.28 ips)

API 610 Limit 4.16 mm/s (0.16 ips)
Correction for RPM and Power

Pump A site (SG=1)

Pump B site (SG=1)

Normal Duty

Rated Duty

Vibration [mm/s RMS]

Q [m3/h]

1500 1600 1700 1800 1900 2000 2100

NDE H
NDE V
NDE A
DE H
DE V

Pump A Shop test 50Hz (SG=1) @ 1205 m3/h

Pump A Shop test 50Hz (SG=1)

Pump A Shop test 50Hz (SG=1)
Initial analysis

Spectra analysis and main outcomes

- Confirmation of the measure taken by customer
- Similar behaviour on the two pumps
- Frequency spectra with broadband showing peaks distributed for many frequencies up to 500 -700 Hz (low - medium range )
- Filtered vibrations at key characteristic frequencies ( 1x, VPF ) have amplitude around 1.0 – 1.5 mm/s (0.04 – 0.06 ips). But overall value is around 5-6 mm/s(0.2-0.24 ips), due to the high number of peaks
- Spectra instability, with high variations in different moments
- Phase not stable
- The higher vibration values were detected on pump casing, and not on the bearing
NDE vibration spectra

Initial analysis – Vibration spectra

Field data (November 2008)

- N=3580 rpm
- 1x=3580 rpm
- VPF= 7x=25060 rpm
- Q=1700 m$^3$/hr (7490 gpm)
- Close to Normal duty
- SG=1
- T=35°C
- NPSHA/NPSHR=2.14

Remarks:
- a) Low amplitude at VPF (< 1 mm/s = 0.04 ips)
- b) High activity mainly across a range up 500 Hz (30000 rpm)
Initial analysis – Vibration spectra

Field data (November 2008)

Remarks:

a) Max amplitude at VPF:
   1.4 mm/s = 0.06 ips

b) High activity distributed and dominant across a range up to 500 Hz
   (30000 rpm)
Initial analysis – Suction piping
Root Cause Analysis

Following the results and data collected in the first site campaign, a thorough Root Cause Analysis was conducted by pump designer.
## Potential Root Cause Analysis 1)

<table>
<thead>
<tr>
<th>POSSIBLE CAUSE</th>
<th>Why yes</th>
<th>Why not</th>
<th>Result</th>
</tr>
</thead>
</table>
| Mechanical behaviour of the pump | High level of vibration is due to the mechanic of the pump (misalignment, unbalance, etc) | 1. The spectra don’t show evidence of the mechanical problem  
2. Dismantling of pump A didn’t highlight any issue | EXCLUDED          |
| Major internal looseness, Broken parts | Extreme bearing wear, internal looseness or broken parts can justify a low noise level like background in the spectra | 1. Bearing when inspected, didn’t show any major damage  
2. Dismantling of pump A didn’t highlight any major looseness | EXCLUDED          |
| Resonance                    | Resonance can justify a unstable phase                                   | Resonance is centered on defined frequencies, and these frequencies are always the same. It’s not compatible with the spectra variations measured | EXCLUDED          |
### Potential Root Cause Analysis 2)

<table>
<thead>
<tr>
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<th>Why not</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid dynamics of the piping</td>
<td>Unsteady and random spectra with a broadband distribution of many</td>
<td>Piping designed according to customer best practice</td>
<td>PROBABLE CAUSE</td>
</tr>
<tr>
<td></td>
<td>peaks of low frequencies are indicative of intense turbulence.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Piping was not fully compliant with HI recommendations</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fluid dynamics of the pump</td>
<td>Unsteady and random spectra with a broadband distribution of many</td>
<td>Same type of pump running well in other applications</td>
<td>PROBABLE CAUSE</td>
</tr>
<tr>
<td></td>
<td>peaks of low frequencies are indicative of intense turbulence.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Pump operation at capacity below BEP is potential source of high</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>turbulence</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Implementation of 1st phase: Suction piping

- The customer modified the piping layout as to have it compliant to Hydraulic Institute recommendation.
- The results of the modification showed remarkable reduction in vibration level, though not within the required acceptance limits.
Implementation of 1st phase
Suction piping with flow straightener (February-March 2009)

The results of the modification showed remarkable reduction in vibration level, though not within the required acceptance limits.
The solution has been focused on the pump hydraulic, as the remaining cause pointed out in the Root Cause Analysis.

The hydraulic design of the pump was studied with respect to the vibration analysis.
Implementation of 2nd phase

The peculiarity of broadband frequency spectra with presence of many peaks up to 500–700 Hz could be associated with turbulent flow induced by flow separation inside the impeller either at inlet (suction recirculation) and/or at outlet (discharge recirculation). Vibration amplitude at VPF is in general a symptom more related with discharge recirculation which appears unlikely (low VPF level in all spectra).

Then the suction recirculation looks as the most probable mechanism of high turbulence and vibration source. Therefore the focus has to be directed to:

a) Pump operation: if and how much below BEP and/or

b) Impeller design: if suitable for the application (primarily inlet geometry)
**Implementation of 2nd phase**

*Recirculation*: For a trimmed impeller, the flow of onset suction recirculation may be closer to the normal point, even if this looks at first glance reasonable and complying with the API criteria.

*Incidence angle at blade tip*: An highly positive incidence far away from the shock-less condition may lead to flow separation (suction recirculation) with flow unsteadiness inducing vibrations. For pumps with high energy level at inlet associated with a peripheral velocity at the impeller eye diameter above 30 m/s (100 ft/s) the overall level of vibrations can be high even above acceptable limits for the bearing housings.
Implementation of 2nd phase
Hydraulic analysis

Design point:

- **N** = 3580 rpm
- **Q** = 2900 m³/h (12775 gpm)
- **H** = 418 m (1373 ft)
- **D₂** = 490 mm (19.3 inch)
- **NSPHR** = 28 m (92 ft)
- **Ns_{des}** = 1794
- **Nss_{des}** = 9630
- **Z** = 7 vanes, staggered
- **Dcw/D₂** = 1.08 (B-Gap)
- **D_{eye}** = 280 mm (11 inch)
- **U_{eye}** = 52.5 m/s (172.4 ft/s)
- **Qsl** = 3190 m³/h (14053 gpm)
- **Qsl/Q_{des}** = 1.1 (sl = shockless)
- **Qrs** = 2090 m³/h (9207 gpm)
  - (rs = suction recirculation)
- **Qrs/Q_{des}** = 0.72
- **Qrs/Qsl** = 0.65

Test Curves
Implementation of 2nd phase
Impeller trimming

Rated point:

\[
\begin{align*}
N &= 3580 \text{ rpm} \\
Q &= 2052 \text{ m}^3/\text{h} (9040 \text{ gpm}) \\
H &= 259 \text{ m} (850.6 \text{ ft}) \\
D_{2\text{duty}} &= 405 \text{ mm} (15.9 \text{ inch}) \\
NPSHR &= 19.4 \text{ m} (63.7 \text{ ft}) \\
NPSHA &= 39.6 \text{ m} (130 \text{ ft}) \\
NPSHA / NPSHR &= 2.04 \\
D_{2\text{duty}}/ D_{2\text{des}} &= 0.83 \\
Q_{\text{bepduty}} &= 2150 \text{ m}^3/\text{h} (9471\text{gpm}) \\
Q_{\text{rated}}/Q_{\text{bepduty}} &= 0.95 \ (\text{Seems good!}) \\
Q_{\text{rated}}/Q_{\text{design}} &= 0.74 \ (\text{Too low}) \\
Q_{\text{rated}}/Q_{\text{sl}} &= 0.64 \ (\text{Too low}) \\
Q_{\text{rated}}/Q_{\text{rs}} &= 0.98 \ (\text{Possibility of suction recirculation start}) \\
D_{\text{cw}}/D_{2\text{duty}} &= 1.30 \ (\text{B-Gap: very large i.e. low vibrations at VPF})
\end{align*}
\]
Implementation of 2nd phase
Impeller trimming

Normal point (specified):

N = 3580 rpm
Q = 1710 m³/h (7533 gpm)
H = 283 m (929.4 ft)
D₂duty = 405 mm (15.9 inch)
NPSHR = 18.5 m (60.7 ft)
NPSHA = 39.6 m (130 ft)
NPSHA / NPSHR = 2.14
Qnormal / Qrated = 0.83
Qnormal / Qbepduty = 0.8 (OK for API 610)
Qnormal / Qbepdes = 0.59 (Too low)
Qnormal / Qsl = 0.54 (Too low)
Qn = 59% of Qdes – RED FLAG
Qnormal / Qrs = 0.82

Suction recirculation is root cause of vibrations
Upgraded impeller design

New Impeller design point:

- \( N = 3580 \text{ rpm} \)
- \( Q = 2000 \text{ m}^3/\text{h} \) (8810 gpm)
- \( H = 300 \text{ m} \) (985 ft)
- \( D_2 = 445 \text{ mm} \) (17.5 inch)
- \( \text{NSPHR} = 23 \text{ m} \) (75.5 ft)
- \( N_{\text{des}} = 1911 \)
- \( N_{\text{s des}} = 9270 \)
- \( Z = 7 \text{ vanes, rake - no stagger} \)
- \( D_{\text{cw}}/D_2 = 1.18 \) (B-Gap, ample)
- \( D_{\text{eye}} = 255 \text{ mm} \) (10 inch)
- \( U_{\text{eye}} = 47.8 \text{ m/s} \) (157 ft/s)
- \( Q_{\text{sl}} = 2120 \text{ m}^3/\text{h} \) (9340 gpm)
- \( Q_{\text{sl}}/Q_{\text{des}} = 1.06 \) (sl = shockless)
- \( Q_{\text{rs}} = 1400 \text{ m}^3/\text{h} \) (6167 gpm)
- \( Q_{\text{rs}}/Q_{\text{des}} = 0.70 \)
- \( Q_{\text{rs}}/Q_{\text{sl}} = 0.66 \) (rs = suction recirculation)
Implementation of 2nd phase

- Incidence angle = \( \beta_{1\text{blade}} - \beta_{1\text{FLOW}} \)
Implementation of 2nd phase
Hydraulic analysis (March 2009)

- Incidence analysis *(existing impeller)*

<table>
<thead>
<tr>
<th>Point</th>
<th>Flow [m³/h]</th>
<th>β₁_blade (tip)</th>
<th>β₁flow</th>
<th>INCIDENCE</th>
</tr>
</thead>
<tbody>
<tr>
<td>DESIGN</td>
<td>2900</td>
<td>17°</td>
<td>15.5°</td>
<td>1.5°</td>
</tr>
<tr>
<td>RATED</td>
<td>2052</td>
<td>17°</td>
<td>10.2°</td>
<td>6.8°</td>
</tr>
<tr>
<td>NORMAL</td>
<td>1710</td>
<td>17°</td>
<td>8.4°</td>
<td>8.6°</td>
</tr>
</tbody>
</table>

Could lead to suction recirculation (flow separation) with high level of broadband vibration for high energy pumps.
New impeller design strategy

Constrains:

1) Upgrade impeller design with new pattern

2) Stringent expected delivery time from Contractor and End User

Impeller to be interchangeable with present pump configuration, i.e. double suction, double volute, existing bearing housing
Upgraded impeller design  
( April 2009 )

Incidence angle (design strategy for new customized impeller)

<table>
<thead>
<tr>
<th>Point</th>
<th>Flow [m³/h]</th>
<th>β1 blade (tip)</th>
<th>β1 flow (tip)</th>
<th>INCIDENCE</th>
</tr>
</thead>
<tbody>
<tr>
<td>DESIGN</td>
<td>2000</td>
<td>16°</td>
<td>15.2°</td>
<td>0.8°</td>
</tr>
<tr>
<td>RATED</td>
<td>2052</td>
<td>16°</td>
<td>10.2°</td>
<td>0.4°</td>
</tr>
<tr>
<td>NORMAL</td>
<td>1710</td>
<td>16°</td>
<td>8.4°</td>
<td>3.1°</td>
</tr>
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</table>

The incidence is near to the shockless condition for the rated capacity. Also it is far below the critical value (causing flow separation and suction recirculation) for the normal point.
Comparison upgraded vs original impeller

*Upgraded impeller (“Customized design”)*

N = 3580 rpm

\[ D_{2,\text{duty}} = 423 \text{ mm ( 16.7 inch )} \]

\[ D_{2,\text{duty}}/D_{2,\text{des}} = 0.95 \]

\[ D_{cw}/D_{2,\text{duty}} = 1.25 \]

\[ Q_{\text{bepduty}} = 1900 \text{ m}^3/\text{h ( 8370 gpm) } \]

*Rated point*

NPSHR = 23.2 M (76.2 ft)

NPSHA / NPSHR = 1.70

\[ Q_{\text{rated}}/Q_{\text{bepduty}} = 1.08 \text{ ( Good )} \]

\[ Q_{\text{rated}}/Q_{\text{design}} = 1.03 \text{ ( Good )} \]

\[ Q_{\text{rated}}/Q_{\text{sl}} = 0.97 \text{ ( Good )} \]

\[ Q_{\text{rated}}/Q_{\text{rs}} = 1.46 \text{ ( Well above suction recirculation onset )} \]

*Normal Point*

NPSHR = 21 m (69 ft)

NPSHA / NPSHR = 1.89

\[ Q_{\text{normal}}/Q_{\text{bepduty}} = 0.9 \text{ ( Good for efficiency )} \]

\[ Q_{\text{normal}}/Q_{\text{bepdes}} = 0.85 \text{ ( Reasonable )} \]

\[ Q_{\text{normal}}/Q_{\text{sl}} = 0.81 \text{ ( Acceptable )} \]

\[ Q_{\text{normal}}/Q_{\text{rs}} = 1.22 \]

*No suction recirculation*
Upgraded impeller design

Features: Blade rake – No stagger

3D Virtual solid model
Fast impeller production

The virtual solid model was post processed to obtain all the pattern components through Rapid Prototyping for fast production as required by Contractor and End User to complete the plant commissioning and release to production.
3D scanning for accurate casting inspection

Once the casting was obtained a 3D scanning of the impeller allowed the complete geometrical inspection to verify the compliance of the casting to the original design.

This step was needed because:

a) Incidence angle is very sensitive parameter.
   In relation to suction recirculation onset and cavitation behaviour only tight tolerance for incidence and inlet blade angle is allowed (+/-0.5°)

b) The new impeller could not be tested at the shop.
   The rotor had to be directly installed at site for quick plant restart, possibly avoiding any rework i.e. impeller outlet diameter to readjust head for any geometrical deviation (out of tolerance) at blade outlet (angle, span, thickness)
Shrouded Impeller blades are 3D scanned from casting (laser scan + point probe)

Blue = design model

Red = finished part
Machined impeller as shipped (June 09)

Remark: No shop test
Preliminary field results pump A
(July 2009)

Pump accepted: August 2009
Plant released to full production: September 2009

ISO 3945 limit = 7.1 mm/s (0.28 ips)
API 610 Limit 4.16 mm/s (0.16 ips)
Correction for RPM and Power
Field data comparison at normal capacity NDE (October 2009)

Initial field data with old impeller

<table>
<thead>
<tr>
<th></th>
<th>Old</th>
<th>New</th>
</tr>
</thead>
<tbody>
<tr>
<td>H mm/s(ips)</td>
<td>5.93 (0.23)</td>
<td>1.39 (0.05)</td>
</tr>
<tr>
<td>V mm/s(ips)</td>
<td>5.39 (0.21)</td>
<td>3.11 (0.12)</td>
</tr>
</tbody>
</table>

Final field data with new impeller
Field data comparison at normal capacity DE (October 2009)

Initial field data with old impeller

Final field data with new impeller

<table>
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<th>Old</th>
<th>New</th>
</tr>
</thead>
<tbody>
<tr>
<td>H mm/s(ips)</td>
<td>7.23 (0.28)</td>
<td>3.02 (0.12)</td>
</tr>
<tr>
<td>V mm/s(ips)</td>
<td>6.86 (0.27)</td>
<td>2.25 (0.09)</td>
</tr>
</tbody>
</table>
Conclusions

An analytical diagnostics approach has been applied along with experimental investigation for identifying the vibration root cause. The vibration source was identified as mainly an internal hydraulic excitation due to high vane inlet angle not suitable for the expected operating range.

A new impeller was designed with geometry fully optimized for the intended operating range, particularly the inlet geometry (customized design).

The new impellers were manufactured using a Rapid Prototyping process to meet customer impellent needs.

A 3D scanning protocol has been used to verify consistency of casting to the design and allow straight installation at site with minimal risk.

The new impellers have been installed in the pumps and field data show a drastic reduction of all vibration components below API acceptance level with full satisfaction of Contractor and End User for ultimate solution of pump vibrations with fast field implementation allowing the start of plant production according to schedule.