Root Cause Analysis on a Multistage Centrifugal Pump in a Power Plant due to Shaft Crack Based on ODS and FEA

27th International Pump Users Symposium
George R. Brown Convention Center
Houston, TX
September 12 - 15, 2011

By:

Maki Onari
Mechanical Solutions, Inc.
Whippany, NJ 07981 USA
Tel: (973) 326-9920
www.mechsol.com
Introduction

• Two multistage barrel type pumps for each 1200 MW Unit were installed in 1986 in a large power plant.

• The 11 stage centrifugal pumps were designated as Pump(s) 1A/B and Pump(s) 2A/B. All impellers were designed with 6 vanes and 8 diffuser vanes.

• The pumps are driven by 600 HP four-pole induction motors (1,770 rpm or 29.5 Hz) through a gear increaser (output speed of 4,860 rpm or 81 Hz).

• Rated capacity: 350 GPM and 2,516 ft of TDH

• Normal capacity: 150 GPM (43% BEP).
Introduction

Barrel Type - 11 stage Pump
Outline Drawing
Introduction

Barrel Type - 11 stage Pump
Three pump shaft failures (cracks) have been detected since Pump 1B installation (MTBF=7.7 years).

In 1989, the first failure took place, but the failure analysis was not properly documented and the failed shaft was not saved.

In 1999, a crack was found at the rear end of the 9th stage hub.

In December 2007, the last failure was discovered under the 11th stage impeller hub with evidence of fretting. A circumferential crack was found at the keyway with 132 degrees arc. The vibration level of the pump had been always considered low and adequate.
Background History

2007 Failure after 8 years of operation. Crack detected under 11th stage impeller hub.
Background History

2007 Failure after 8 years of operation.
Crack detected under 11th stage impeller hub with the origin away from the keyway.
Specialized Testing

- Experimental Modal Analysis (EMA) test to determine the natural frequencies of the impeller, pump structure, and the rotor system with their mode shapes.

- Monitoring test during transient and steady operation to monitor the vibration amplitude and natural frequencies.

- Dynamic pressure transducer data from suction, discharge, and balance line.

- Operating Deflection Shape (ODS) testing during steady operation.
11th Stage Impeller
EMA Testing
Vibration EMA Test

11th Stage Impeller Modal Impact Test
Vibration EMA Test

11th Stage Impeller Modal Impact Test

Impeller Model

FRF Wet
Vibration EMA Test

11th Stage Impeller Modal Impact Test

2 Nodal Diameter / 0 Nodal Circle at 2620 Hz
Vibration EMA Test

11th Stage Impeller Modal Impact Test (Wet)

Interference Diagram
11th Stage of 2 1/2 RLJ Pacific Pump (Wet Impeller)
6 Rotating Vanes and 8 Diffuser Vanes

Interference Diagram

Frequency (Hz)

Nodal Diameter
Specialized Field Testing
Vibration Monitoring

Pump 1B Typical Spectrum OBB in the Vertical Direction

**Autospectrum(Signal 5) - Mark 1 (Magnitude)**
*Working: ODS2: Input: FFT Analyzer*

- 0.15 in/s peak @ 1x rpm
- 0.16 in/s peak @ 4x motor rpm

Pump Operating at 145 GPM

---

Pump 2B Typical Spectrum OBB in the Vertical Direction

**Autospectrum(Signal 1) - Mark 1 (Magnitude)**
*Working: 2B Linear Average: Input: FFT Analyzer*

- 0.06 in/s peak @ 1x rpm
- 0.06 in/s peak @ 4x motor rpm

Pump Operating at 160 GPM
Vibration Monitoring

Pump 1B Axial Proximity Probe FFT at OBB

Autospectrum(Signal 1 2) - Mark 1 (Magnitude)
Working : 1B Linear Ave at 100 GPM : Input : FFT Analyzer

0.46 mils pk-pk @ ~30 Hz

Pump 1B Axial Proximity Probe Time Signature at OBB

Time(Signal 1 2) - Mark 1
Working : 1B Linear Ave at 100 GPM : Input : FFT Analyzer

29.5 Hz
Vibration Monitoring

Pump 2B Axial Proximity Probe FFT at OBB

- 0.4 mils pk-pk @ 81 Hz
- 0.04 mils pk-pk @ ~30 Hz

Pump 2B Axial Proximity Probe Time Signature at OBB

- 81 Hz
Pump Rotor Modal Impact Testing During Operation
Experimental Modal Analysis
“Bump” Test

Pump 1B Axial Proximity Probe FRF

- Frequency Response (Signal 12, Signal 17) - Mark 1 (Magnitude)
- Working: Axial Hits Channel 12 Axial Prox Probe 100 GPM: Input: Enhanced

- 29.5 Hz (1x motor speed)
- 121.2 Hz
- OBB natural frequency

Pump operating at 160 GPM

Pump 2B Axial Proximity Probe FRF

- Frequency Response (Signal 12, Signal 17) - Input (Magnitude)
- Working: Input: Input: Enhanced

- 81 Hz (1x pump speed)
- 118.2 Hz (4x motor speed)
- OBB natural frequency
Operating Deflection Shape
(ODS)
Operating Deflection Shape

- Natural excitation signature of the pump train structure.
- Over 700 vibration measurements.
- Data base of amplitude vs. frequency and phase angle.
- 3-D CAD model assigning motion to each individual vibration data point.
- Create animations of the pump
Operating Deflection Shape

Pump 1B ODS @ 29.9 Hz
Operating Deflection Shape

Pump 1B ODS @ 81 Hz
Operating Deflection Shape

Pump 2B ODS @ 29.9 Hz
Operating Deflection Shape

Pump 2B ODS @ 81 Hz
Finite Element Analysis (FEA)
Axi-symmetric Model and FEA Breakup
Axi-symmetric Model
Impeller Loading

Axial displacement in displacement (mils pk-pk) and acceleration (g’s)
Axi-symmetric Model
Impeller Loading

**Load Step 1:**
- Initiate and complete the interference fit between the impeller and the shaft (2 mils diametral interference)
- Constrain the outboard end of shaft at the thrust bearing

**Load Step 2:**
- Ramp up an axially applied pressure of 75 psi load to the inboard end over 0.0001sec (30 g’s peak acceleration).

**Load Step 3:**
- Ramp pressure down to a 0 psi over an additional 0.0001sec.

**Load Step 4:**
- Run for an additional 0.001 sec to monitor the traveling of the acoustic waves and the impeller interface conditions.
Axi-symmetric Model
FEA Results

- Transient dynamic analysis revealed that acoustic wave propagation past the impellers can result in micro-motion at the press fit interfaces.

- An idealized axi-symmetric model of the pump shaft and the last stage impeller predicted an impeller to shaft sliding condition.

- This sliding condition led to fretting damage and the crack initiation.
Axi-symmetric Model
FEA Results
FEA-Based Fracture Mechanics Approach

• Assume that relative sliding velocity between the impeller and shaft is equal to the peak axial shaft velocity measured on the end of shaft near the thrust bearing.

• Assume that fretting condition and asperities on shaft surface cause the impeller to axially lock-up against the shaft.

• Model the lock-up behavior as an axial impact between the shaft and impeller in the region of the assumed 5 mil crack.

• Using transient dynamic FEA predict the stress distribution near the crack and calculate the stress intensity factor.

• Check whether crack propagation can be expected.
FEA-Based Fracture Mechanics Approach

0.005 inch crack explicitly modeled

Impact Velocity 31 in/s pk (15 g’s pk at 29.9Hz)
FEA-Based Fracture Mechanics Approach

Double Peak due to Impeller Flexing and Spring-back

von Mises Stress (psi) at Root of Crack During Impact Event
FEA-Based Fracture Mechanics Approach
von Mises Stress Distribution (psi) Macro View
FEA-Based Fracture Mechanics Approach
von Mises Stress Distribution (psi) In Vicinity of Crack

Impact Location
Crack Root

\[ \Delta K = \Delta \sigma \pi \sqrt{a} (1.1) \]

ANSYS Calculated Stress Intensity Factor \( \Delta K \) = 16.3 ksi√in
FEA-Based Fracture Mechanics Approach

ANSYS Ki Output

Calculate mixed-mode stress intensity factors
• Assume plane strain conditions.
• Assume a full-crack model (use 5 nodes)
• Extrapolation path is defined by nodes: 2044520, 2044672, 2044683, 2047304, 2047315
• With node 2044520 as the crack-tip node.
• Use material properties for material number 1 with Ex = 0.29e+08 μxy = 0.30 at temp = 0.00.

Ki = 16315. , Kii = 7826.4 , kiii = 0.0000
Implemented Fixes

• A new gear set was installed in the last outage in Fall 2009.
• Both gear couplings were replaced.
• New pump element installed.
• Monitoring of pump shaft axial vibration and gear drive vibration adjusted to address imposed duty on pump shaft due to less than smooth mesh of gears.
• Gear set removed mapped.
Conclusions

- The root cause of the repetitive cracking of the pump shaft is due to a fatigue process while the pump shaft is oscillating axially driven by the gear tooth circumferential run-out.

- The axial mismatch of the helical gear set (apex) was the cause of the axial displacement of the pump shaft at the driver’s operating speed, generating an impulsive displacement load due to a geometric abnormality of the gear teeth of the input shaft gear (axial run-out).

- The ODS test performed on Pump 1B at the running speed of the motor (30 Hz) indicated high axial motion of the pump shaft driving the OBB with a vertical rocking motion. The gearbox moved axially with some phase lag with respect to the pump shaft and the motor casing.
Conclusions

• The vibration test performed on Pump 2B did not indicate any abnormal axial motion of the pump shaft at the running speed of the motor (0.04 mils pk-pk versus 0.5 mils pk-pk measured on Pump 1B).

• Traditional troubleshooting approaches probably would not have indicated a gear/pump inter-related problem, and would not have provided such clear visual evidence for decision makers.

• ODS / EMA coupled with appropriate analysis is a powerful troubleshooting tool to facilitate and visually understand the most difficult vibration problems in turbomachinery and pumping systems.
Thank you

Any Questions...?