Texas A&M University
Graduate Engineering Class Presentation

College Station, TX,
Feb 28, 2019

Yve L. Zhao

Yve Zhao is a Rotating Equipment Engineer working for BP's upstream engineering center. Having worked in the turbomachinery field for 25+ years, in her current position, Yve provides global technical support to the upstream business within BP with expertise on vibration diagnostics, rotor/thermodynamics, failure analysis, and health monitoring of rotating equipment and industrial standards development including API and JIP33.

Prior to joining BP, she worked as a a Staff Rotating Equipment Engineer for BHP's central engineering and midstream production unit. Prior roles includes Rotor Dynamicist/Analytical Engineer at CONMEC supporting centrifugal compressor, steam turbine, expander and axial compressor rerate projects and as a Principal Rotating Equipment Engineer at Air Products and Chemicals on Air Separation, HYCO and Chemical projects.

Yve Zhao received her BSME in1992 and MSME in1997. She has authored papers on turbulent flow seals, bearing dynamics, and case studies on machinery failure and root cause analysis. Currently, she holds positions in the Vibrations Institute - training committee.

WORLD-CLASS OUTSTANDING INTERNATIONAL

PROGRAM | EXHIBITION | NETWORKING

AXIAL VIBRATION FOR A SYNCHRONOUS MOTOR, GEARBOX, COMPRESSOR TRAIN

Lucy Zhao
TAM Presentation
Sept 2013

42nd Turbomachinery
29th Pump SYMPOSIA

GEORGE R. BROWN CONVENTION CENTER
9.30 – 10.3.2013

TEXAS A&M ENGINEERING TEES TURBOMACHINERY LABORATORY

A&M

Three Stories to Share:

The chalk - Axial Vibration for a Motor-Gearbox-Compressor Train

The bridge - Motor Pedestal Resonance

The rush - Vertical Turbine Pump Reliability Improvement

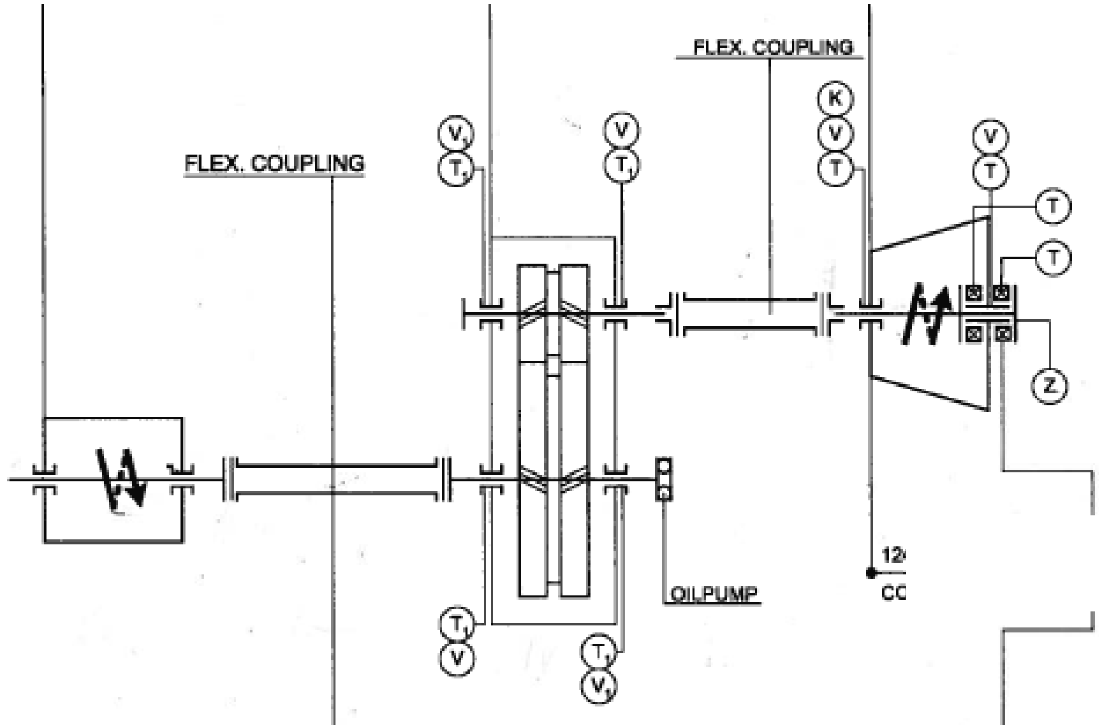
Content

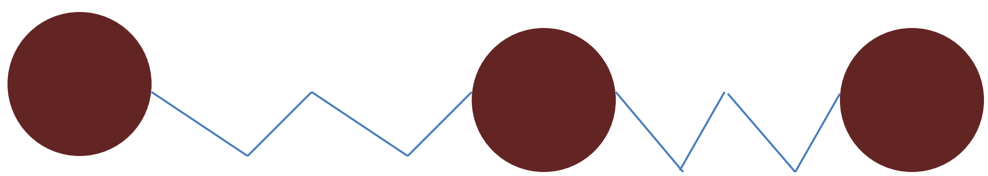
- Problem Statement
- Mathematical Model
- Calculation Results
- Discussion
- Conclusion and Recommendation
- Lessons Learned

Problem Statement

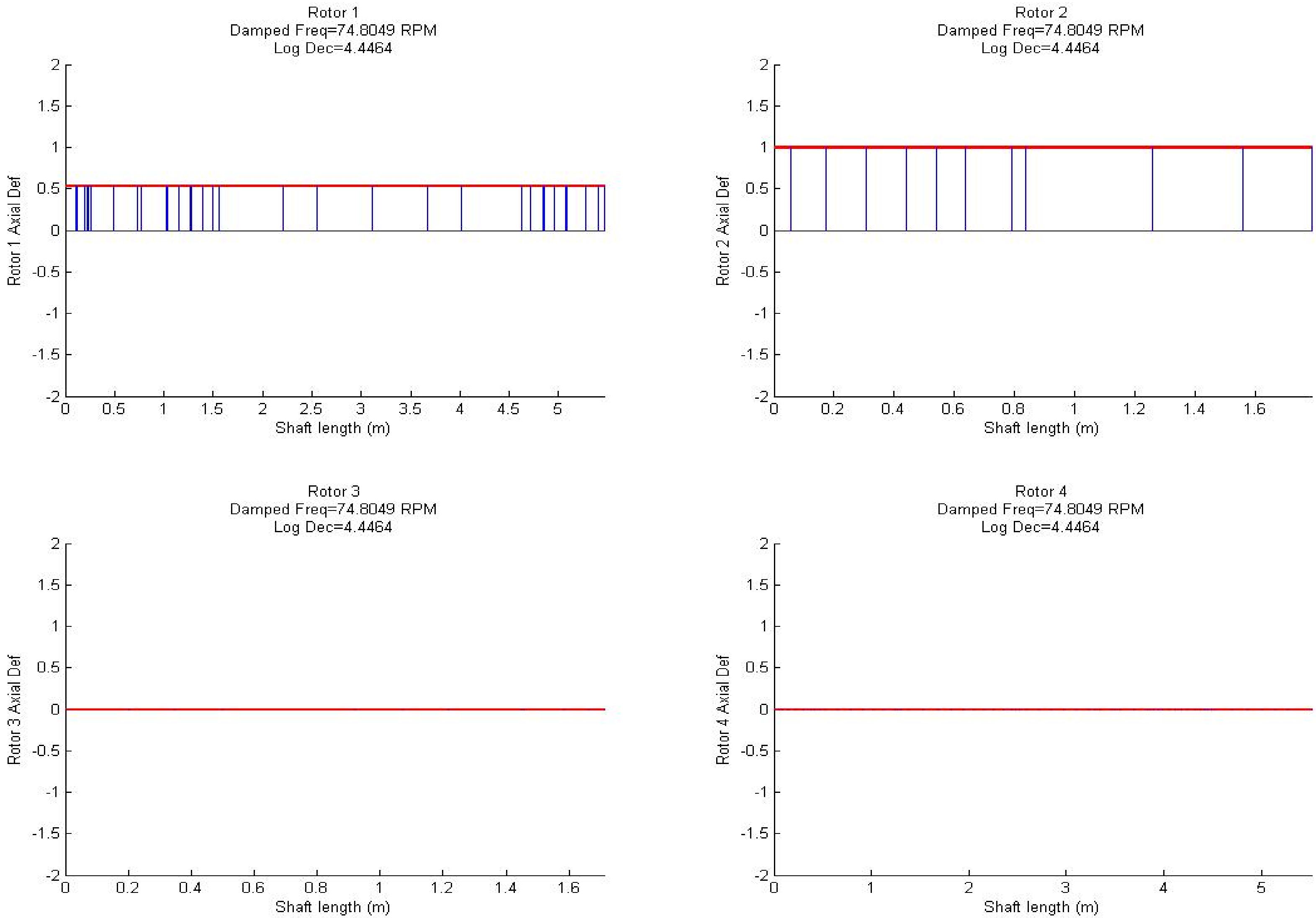
- Synchronous Motor (20KW), Gearbox, Inlet compressor
- Train installed 2007 with 2mm axial vibration mainly on the motor (low speed) side.
- Amplitude @ 2 mm with a frequency of 2.8 Hz(170/min). Gearbox running noisily.

Mathematical Model





MOTOR LS CPLG GB HS CPLG COMPR



Courtesy of Jason Kaplan, ROMAC/University. of Virginia

Assumptions

- Gear mesh is infinite rigid relative to other axial springs in the system and gear mesh damping effect is neglected
- The compressor rotor is axially stationary as the thrust bearing stiffness is much higher when compared to coupling axial stiffness
- The coupling stiffness non-linearity is ignored at low load
- The thrust Bearing is infinite rigid and its damping effect ignored

Calculation Results

- M-motor = 12000kg
- K-ls-cplg-axial = 1.7E6 N/M
- M-gb = 3500 kg
- K-hs-cplg-axial = 2.4E6 N/M

- Ncrit1-axial = 1.4 Hz (85 CPM)

- The axial oscillation frequency is 2.8 Hz.
- The rotor oscillates at 2X Ncrit1-axial and is visible to the observer.

Discussion

- API Standards do not discuss axial critical speeds
- The coupling axial gap is a variable dimension due to ambient temperature and rotor thermal condition changes .
- Axial resonances are rarely observed events. The Author is aware of only a few known cases (<5)
- There is little literature or research available on this subject.
- It is common belief if the axial alignment and thrust bearing clearances are set properly excitation forces will not be large enough to excite the axial natural frequency

Solution

- Motor magnetic center recheck with no change made
- Limited end float coupling installed. The vibration amplitude was reduced but not eliminated
- Extreme high axial alignment target was implemented with original disc pack coupling. Axial oscillation issue solved.

Conclusion & Recommendation

- The train experienced axial vibration at 2X of the train 1st axial natural frequency. Bull gear and pinion relative displacement was observed and calculated.
- Excitation force came from misalignment and motor magnetic centering force.
- The motor is not designed to run off magnetic center. The motor centering force is believed to be around 100-200 lbf/1000 HP.
- The motor centering force is believed to be non-linear to the axial displacement.
- The motor centering force coupled with misalignment was sufficient to trigger the observed axial oscillation.
- High axial vibration was resolved by better alignment.
- The study concluded axial vibration thresholds exist. They depend on axial excitation forces and axial mass-elastic property.
- More research in this directions is necessary to improve coupling design requirements and alignment criteria.

Lessons Learned

- Compressor trains with very low axial stiffness coupling tend to have low frequency axial oscillations.
- High alignment targets can reduce the excitation force. The process can be time consuming and misalignment change with ambient condition.
- Damping effect is low in the discussed motor compressor train which makes this simplified simulation valid and relates well with what was observed.
- Running at 2X axial natural frequency can cause axial vibration and LCF at coupling and gearbox. This can be a significant reliability issue.
- Precaution shall be given to coupling design to increase the lowest axial critical speed frequency.

43rd Turbomachinery
30th Pump SYMPOSIA
GEORGE R. BROWN CONVENTION CENTER
HOUSTON, TX | SEPT. 22 - 25, 2014

TWO POLE MOTOR VIBRATION DIAGNOSTICS, ANALYSIS AND SOLUTION

YU ZHAO – BHP BILLITON
NELSON BAXTER- ABM TECHNICAL SERVICES

TEXAS A&M UNIVERSITY
TEES TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

Content

- Problem Statement
- Field Measurements
- Finite Element Modeling
- Conclusion and Recommendation

Problem Statement

- High vibrations observed on motor casing and support frame
- Velocity reading was at 0.5 in/s in May, increased to 0.7 in/s in July.
- Cracks were seen on the grout and concrete foundation
- The support has a x-brace on the outboard end and a pate welded on the inboard end
- Question:
  - What caused the vibration and how to fix it.

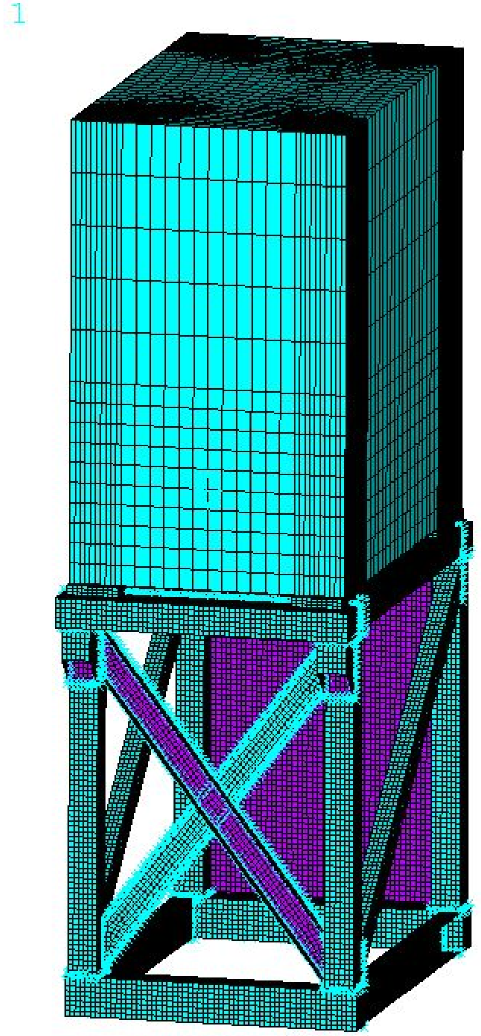
Field Measurement

- 1X dominant
- Found structural mode at 3000 rpm with cracked foundation
- Mode shape is mainly lateral vibration on the drive end, where the crack foundation is – at the anchor bolt location.
- Lose foot was foundation a year before and corrected, alignment done as well. Vibration amplitude unchanged.

Field Measurement

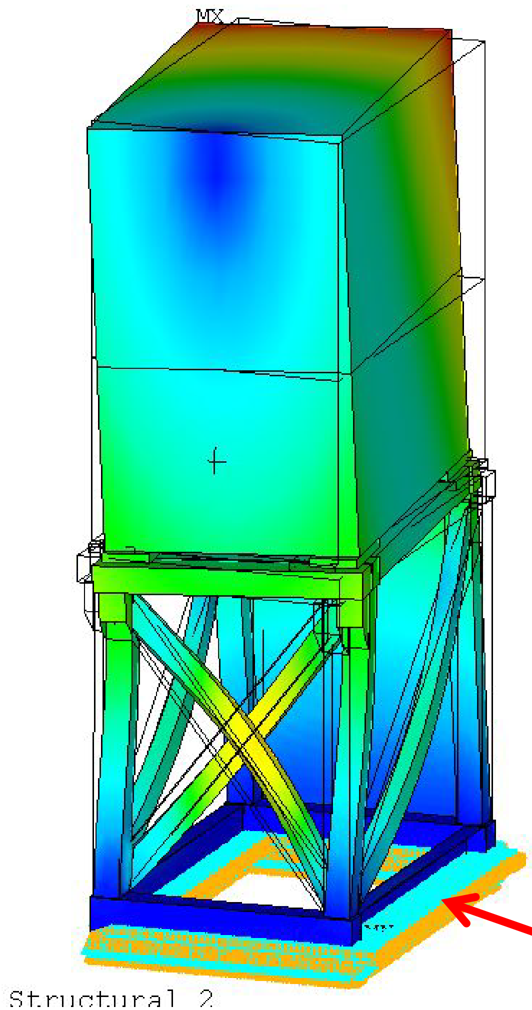
Continue on field measurements

Finite Element Model



Frame--Static Structural 2

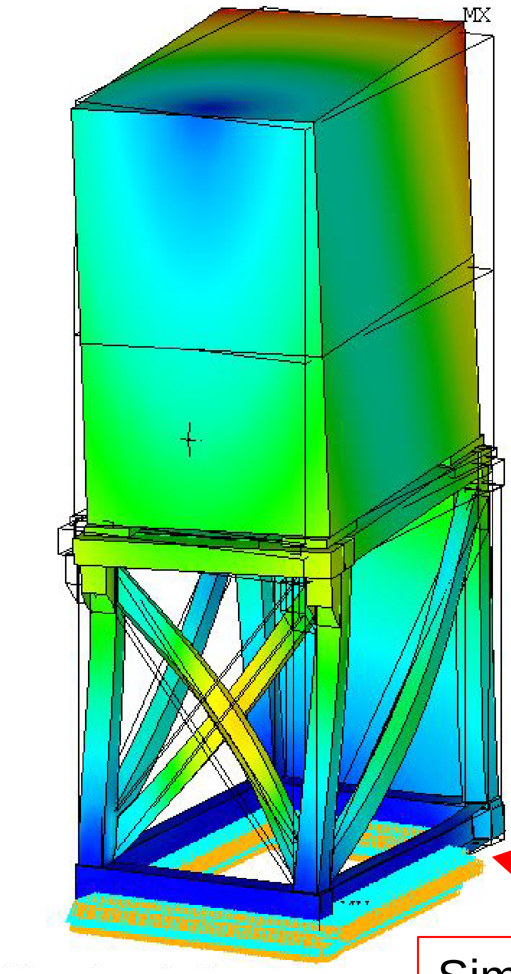
As-Design Condition



- The support has a natural frequency at 57.9 Hz, which is very close to the motor’s running speed of 60 Hz
  - This resonance was likely the original cause of the foundation crack
  - Simply fixing the loose foot due to the damaged foundation did not and will not solve the resonance problem
  - This mode must be pushed away from the motor’s synchronous excitation

All DOFs fixed on the bottom, no loose foot due to damaged foundation

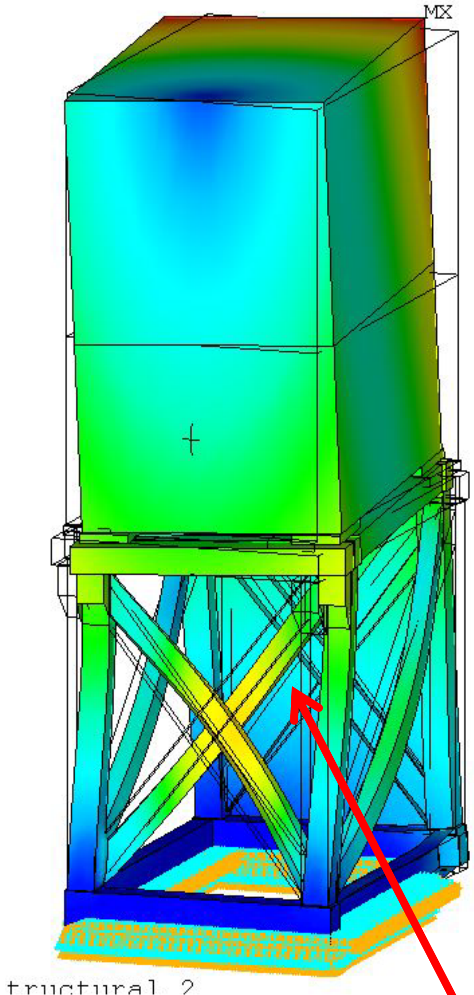
As-Found Condition



- The frequency of the problematic mode is reduce to 52.8 Hz
  - Both the frequency and the mode shape match the field measurements
  - This good correlation validates the FE model
  - Provides a good baseline to investigate solutions
  - The objective is to achieve 20% separation margin

Simulate the loose foot due to the damaged foundation by removing the constraints near the foot

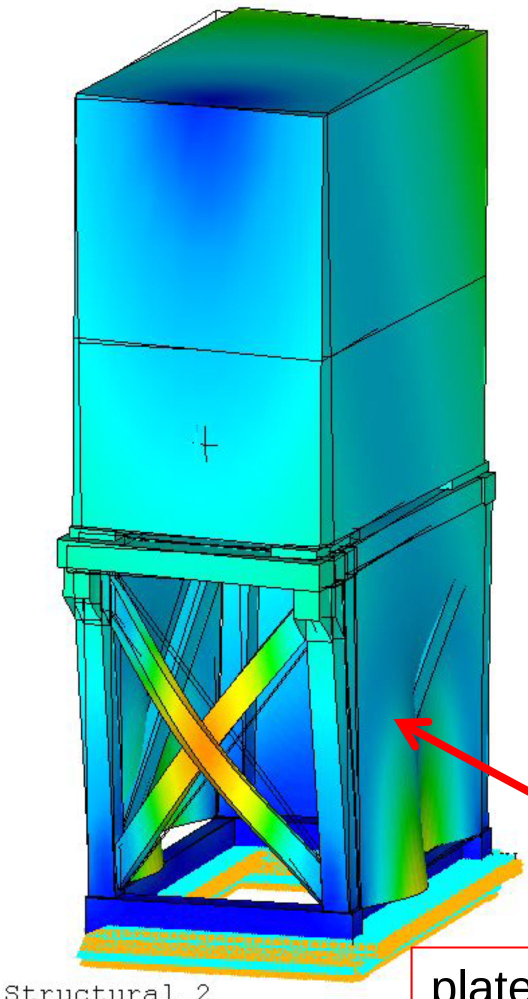
Initial Modification



- The frequency of the mode is increased to 53 Hz
  - Not a solution
  - Since the inboard end shows large displacement, it was intuitive to strengthen this end
  - However, this end is already very stiff due to the plate
    - Further stiffening is not effective
  - Strengthening device should be applied to the area showing the most twist, rather than the area showing the most displacement

X-brace added on the inboard end, loose foot

A Viable Modification



- Plates are added to the side surfaces
- The frequency of the problematic mode is increased to 72.8 Hz
  - 20% SM is achieved
- As a solution
  - Bottom of these plates should be welded as well
  - The plate should also be welded to the X-brace to reduce noise

Conclusion & Recommendation

- As designed, the support has a natural frequency at the motor’s running speed, causing resonance
- This resonance is likely the cause of the original foundation crack
- To increase the resonance frequency of the support outside the motor operating speed range, stiffening device should be added to areas with the most amount twist, instead of areas with the most displacement
- Based on the FE results, it is recommended that plates of ¾” to be welded to the East and West side of the support structure
- Turnaround team found out soft foot again that was fixed a year ago. Further prove the validity of the analysis and the softfoot was a result but not a solution to the structural resonance issue.

47TH TURBOMACHINERY & 34TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 17-20, 2018
GEORGE R. BROWN CONVENTION CENTER

Vertical Turbine Pump Reliability Improvement

Yve L. Zhao

Abstract

The subject pump was designed and installed when pipeline pressure was estimated high based on a higher production volume. The selected multistage vertical turbine-pumps are **generically** prone to vibration issues due to its flexible shaft design.

Due to the deviation between the pump design condition and its actual operating condition, flow turbulence and recirculation in the pump impellers produced enough vibration excitation forces that caused the mechanical seals to fail prematurely and to leak.

Pump restaging was not implemented due to the relatively high cost and uncertainty of a future line pressure. Since the pump is used for batch services, not its entire operating flow/pressure range is necessary to meet production needs.

Performance and Reliability Mapping (PRM) conducted instead thus ensuring a higher MTBF on the pump and seal components.

Note: This is a non-BP incidence

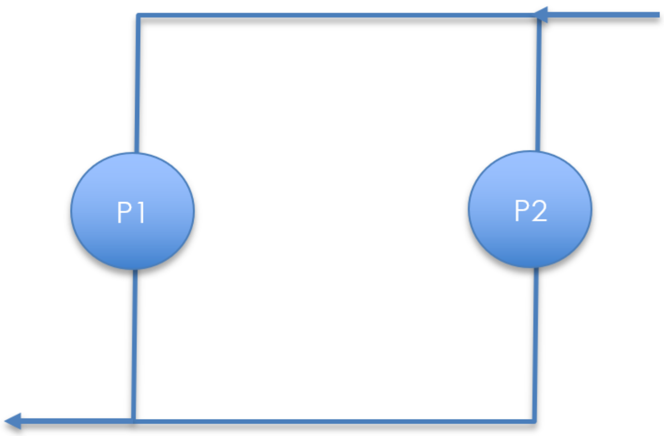
Sequence of Events

- Vertical Turbine/Pump used for pipeline transport service experienced high level of vibration that caused premature seal failures and condensate leakage.
- Radial Vibration produced severe damage of top two bushings (last two stages). 1st stage bushing showed edge loading wear pattern.
- Rotor dynamics report indicated rotor resonance at lower end of the operating speed range.
- During pump repair, tighter clearance bearing bushings were installed (3rd party repair shop). Rebuilt pump ran with loud metal scratching noise and leaking condensate from the mechanical seal over most of its operating speed range.

Sequence of Events – cont.

- Reviewed pump curve and compared with other transport pumps for similar service. The review indicated pump re-staging would bring pump to operate closer to its rated condition and enable an extended operating range. This recommendation was not implemented.
- Vibration signatures at various speed and back pressure taken to generate Performance & Reliability Map (PRM) to set a "**safe to operate range**".
- Transitional seal flushing orifice installed to reduce a turbulence flow effect on the rotor but results were marginal.
- In addition, a review of thrust loading showed absence of balance orifices in the impellers for VFD drive pump. Motor bearing selection needs to be reviewed to consider thrust load to prevent frequent bearing failure.

Pump System Configuration



Two (10 stage) pumps for on-stream backup.

Back pressure control valve.

VFD for speed control.

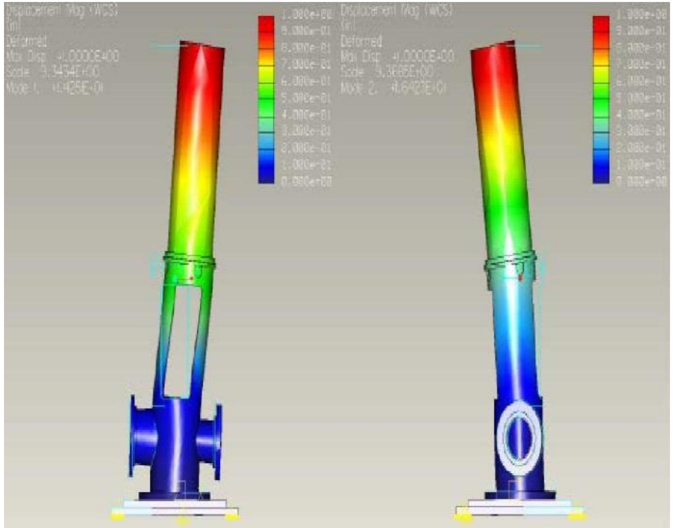
One pump with slightly higher system resistance – on the right.

Rotor Resonance

- Narrow operating speed range (885 – 985 RPM)
- Pump critical speed is highly reliant upon bushing clearances.
- Large clearance bushings were originally installed
  - Typical bushing clearance (per OEM):

    6 - 13 mils diametral (1.7” journal, Cd/D= 3.5~7.6)
  - Actual bushing clearance:

    11 - 21 mils diametral (1.7” journal, Cd/D= 6.5~12.4)

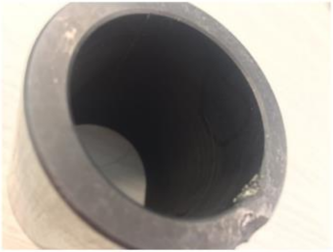


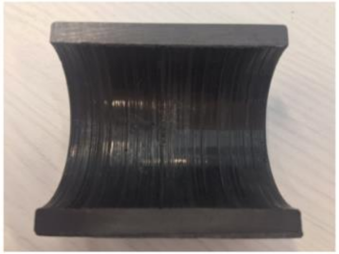
Operating Deflected Shapes at Resonance Speed

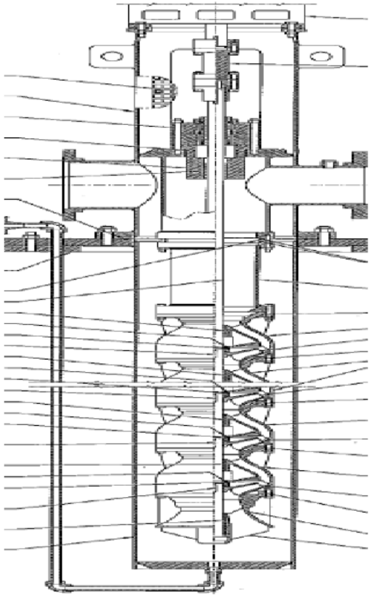
Wear Pattern – Shaft and Bushings

Flow recirculation on the top stage caused more damage to the top two bushings. Bottom 1st stage bushing was edge loaded.

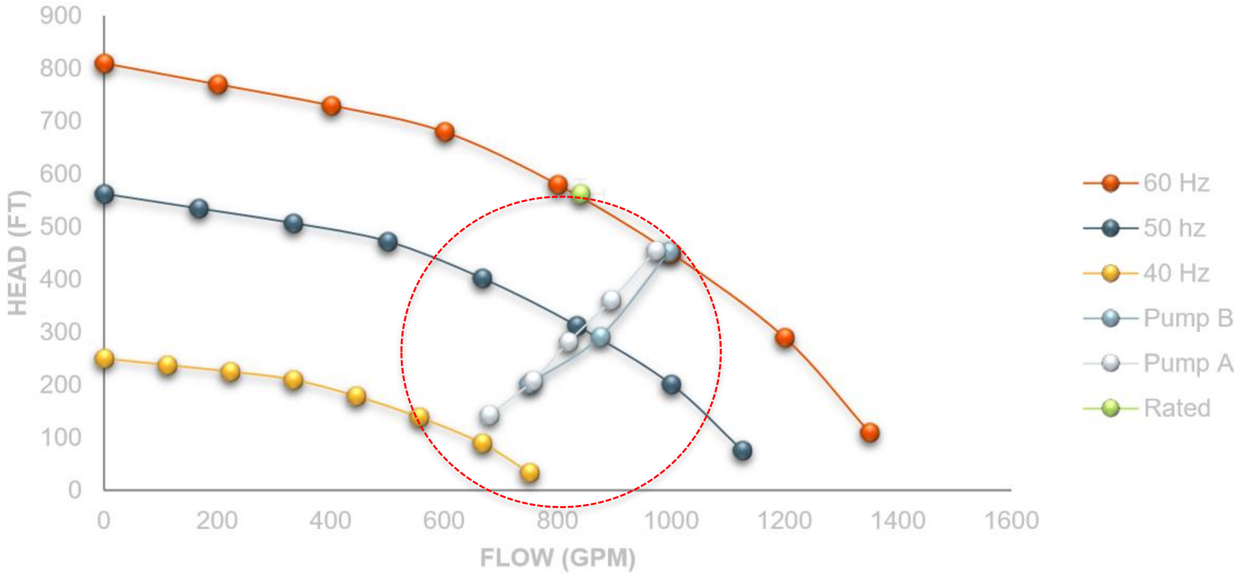
Pump has 10 stages. Catalog picture used to show configuration:



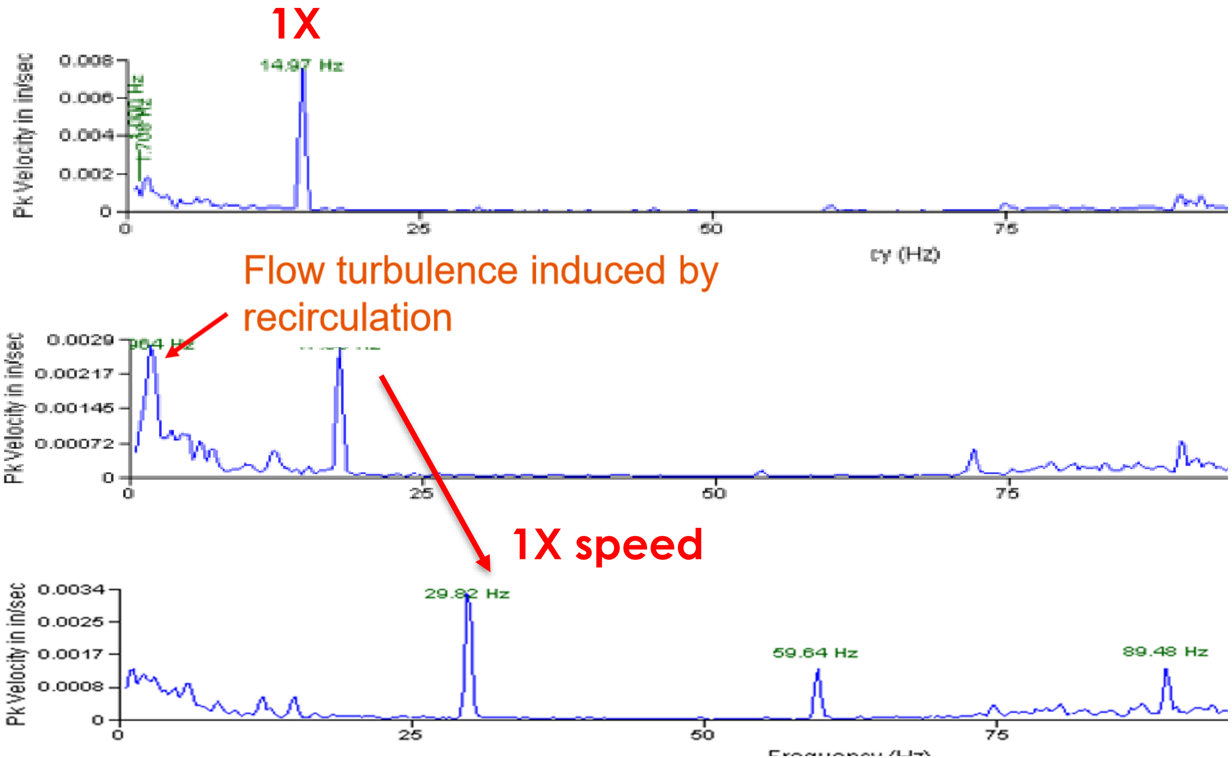




Pump Curve and Operating Points



Vibration Spectra At Various Speeds



Performance & Reliability Mapping (PRM)

speed

Frequency	Valve	Pump B
		Vibration
30 Hz	50%	good
30 Hz	100%	good
35 Hz	50%	good
35 Hz	100%	good
40 Hz	50%	good
40 Hz	100%	good
48 Hz	50%	light rub
48 Hz	100%	light rub
50 Hz	50%	-
50 Hz	100%	-
54 Hz	50%	rub
54 Hz	100%	rub
60 Hz	50%	rub
60 Hz	100%	-

Pump A

Frequency	Valve	Pump A
		Vibration
30 Hz	50%	-
30 Hz	100%	-
35 Hz	50%	light rub
35 Hz	100%	light rub
40 Hz	50%	good
40 Hz	100%	good
45 Hz	50%	good
45 Hz	100%	light rub
48 Hz	100%	light rub
50 Hz	50%	good
55 Hz	50%	good
55 Hz	100%	-
60 Hz	50%	good
60 Hz	100%	-

Pump B

Thrust Load

- Variable Speed Drive.
- Unbalanced impeller Thrust.
- Rigid Drive Shaft
- **Motor Bearing takes Thrust Load!!!**

Turbine/Pump Impellers – No Balance Orifice

Summary

Issue:

Vertical pump prone to vibration issues due to its flexible shaft design, highly turbulent flow and high pressure application with variable operating condition. Issue caused seal leak due to vapor created by heat generated when seal rubbed.

Solution:

Pump restaging was not implemented due to relatively high cost and uncertainty of future line pressure. As the pump is used for batch services, not the entire operating range is necessary, hence **performance mapping was performed instead of a reliability improvement**. Seal leaks stopped once implemented speed envelope.

Other Learning:

- Proper seal design is key to reliability of the vertical pump.
- VFD driven pumps need to be performance and reliability tested for their intended service→ to establish "good/safe to operate range."
- Thrust loading need careful reviewand results utilized for a proper motor bearing selection. Current system has a rigid drive coupling and lacks a thrust load control.

34h Pump Case Study:

Vertical Turbine Pump Reliability Improvement

Yve L. Zhao

Questions (?)