



**43rd Turbomachinery
30th Pump SYMPOSIA**

GEORGE R. BROWN CONVENTION CENTER
HOUSTON, TX | SEPT. 22 - 25, 2014

**DRIVE SHAFT FAILURE ANALYSIS ON A MULTISTAGE VERTICAL TURBINE
PUMP IN RIVER WATER SUPPLY SERVICE IN A NICKEL AND COBALT MINE IN
MADAGASCAR - BASED ON ODS AND FEA**

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Problem Statement

- Pumps installed in 2011. One drive shaft suffered a catastrophic failure on 12/10/11 and was shortly followed by another on 1/26/12.
- Prior to any analysis by the authors, cladding and extra bracing were added to the engine support structure to reduce vibration on all three pumps. Additionally, plates were welded along the I-beams supporting the pump to the discharge on all three pumps. Lastly, grout was added to the internal cavity created by the cladding on one pump.
- The gearbox on one of the pumps suffered a crack on one of its centering feet after one drive shaft incident.
- The goal became to determine the root cause of the drive shaft failure, including identification of any resonance or other stress creators over the pump operating speed range. Based on this, a practical fix would be identified.

Photos Drive Shaft Failures



Failures located at the male end weld (same location each incident) at the engine-side of the assembly.

Outline Drawing

COS Pump Speed: **1760 rpm**

COS Engine Speed: **1800 rpm**

Condition of Service (COS)

Engine Cylinders: **12**

Gearbox Ratio: **1:1.0217**

EQUIPMENT DESCRIPTION

① GOULDS 189HC 7 STG PUMP - 700 M³/HR @ 275m RATED HEAD, PUMPING 100 S.G. RIVER WATER, 1780 RPM
 ② CAT 3512 INDUSTRIAL DIESEL ENGINE - CONTINUOUS RATING 1280 HP @ 1800 RPM
 ③ EXHAUST SILENCER-GRADE RESIDENTIAL COMMENTS INCLUDES S/S EXHAUST FLEX & SHIPPED LOOSE
 ④ AMARILLO VERTICAL GEAR - MODEL SH100, SHAFT HOLLOW, COOLING RIVER WATER RATIO 1:1, DOWTHRUST RATING 39,890 LBS.
 ⑤ SPICER GWS DRIVE SHAFT w/ GUARD-UNIVERSAL JOINT TYPE
 ⑥ STRUCTURAL STEEL BASE - NEW A-36, COMMENTS INCLUDES LIFTING PAD EYES, EARTHING LUGS, DRAIN PAN, w/ DRAIN CONNECTION.
 ⑦ METRON BATTERY CHARGER-230V 1PH 50HZ (SHIPPED LOOSE & MOUNTED OFF SKID)
 ⑧ BATTERIES - (2) 80-100 AMP 12 VOLT BATTERIES WIRED IN SERIES FOR 24 VOLT SYSTEM, TYPE LEAD ACID (ENCLOSED IN A CORROSION PROOF BATT. BDK.)
 ⑨ 12" FLANGED DISCHARGE HEAD BY GOULDS

NOTES

- Engine cooling water is to flow unobstructed & unrestricted to open atm. Do NOT "rainfall" into other piping system, as this could cause back pressure to the Pico.
- Center of Gravity determined by physical measurement after shop assembly.
- User is to insure skid is level and correctly centered over cushion before anchoring to the deck. Caisson must be "plumb" and perpendicular to the skid to insure column pipe is plumb and centered in the caisson.
- Dimensions shown are accurate to a tolerance of ±1/4". As-built will show actual dimensions.
- This pump unit is designed to be installed on a rigid foundation. The actual resonant frequencies observed on the unit are determined by the installation. Coordination with the system designer is essential to avoid operation at or near these frequencies. See Torsional Analysis Report.
- Impeller WT: 88lbs/33kg.

LIFT EYE DETAIL

NOTES:
 1. All lift eye openings & lift eye slots for lifting.
 2. Do not install pad eyes and vertical location of center of gravity in shop drawings.

GENERAL NOTES

- RADIUS TOP SIDE CORNERS-TYP. 4X
- ANCHOR BOLTS SHARE SAME CENTERLINE AS CROSSMEMBER BEAMS.
- ESTIMATED CENTER OF GRAVITY.

DIMENSIONING NOTES

- Dimensions stated in circles are listed from inside skin line.
- Dimensions in brackets are in millimeters, if others are in inches.

PUMP ASS'Y DETAIL

Not to scale

SECTION A-A

Scale: 3/4\"/>

WELDING PROCEDURES

- SEAL WELD.
- USE NEW A-36 STEEL P/ CONSTRUCTION.
- ROUND ALL SHARP CORNERS.
- CLEAN/REMOVE ALL SLAG, SPLATTER, & BUCKSHOT.
- GRIND & FILL ALL UNDERCUTS & CAVITIES.
- PAD EYE PLATE MATERIALS TO BE CUT TESTED PRIOR TO CUT OUT.
- "MAG PARTICLE TEST" PAD EYE TO SKID BEAMS REQUIRED.

DRY WEIGHTS

	LBS.	KG
BASE	7,515	3408.747
ENGINE	14,000	6350.293
GEAR	3,656	1658.334
DRIVE SHAFT/GUARD	360	163.293
HEAD/SUB BASE	1,896	855.475
BATTERIES	375	170.097
MISC. COMPONENTS	500	226.796
SUB-TOTAL	28,992	12933.04
MUFFLER & FLEX	750	340.194
PUMP ASSY.	5,508	2498.387
EST. TOTAL	34,550	15671.62

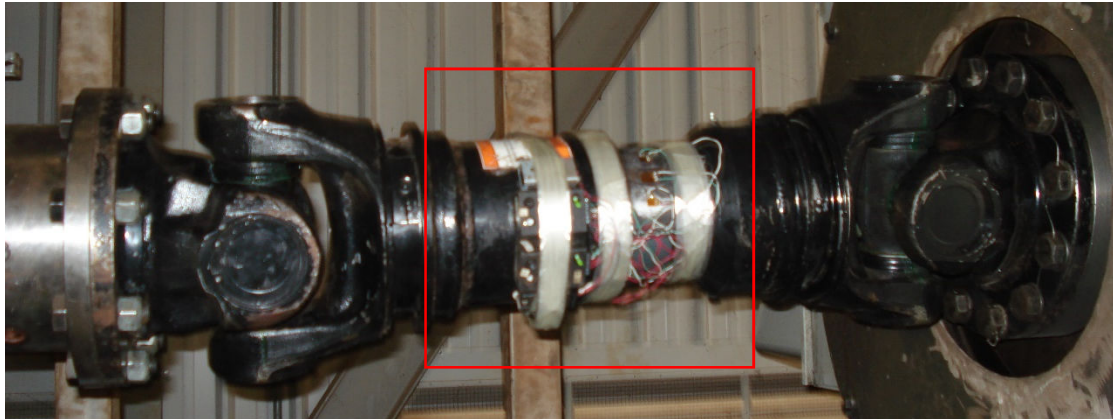
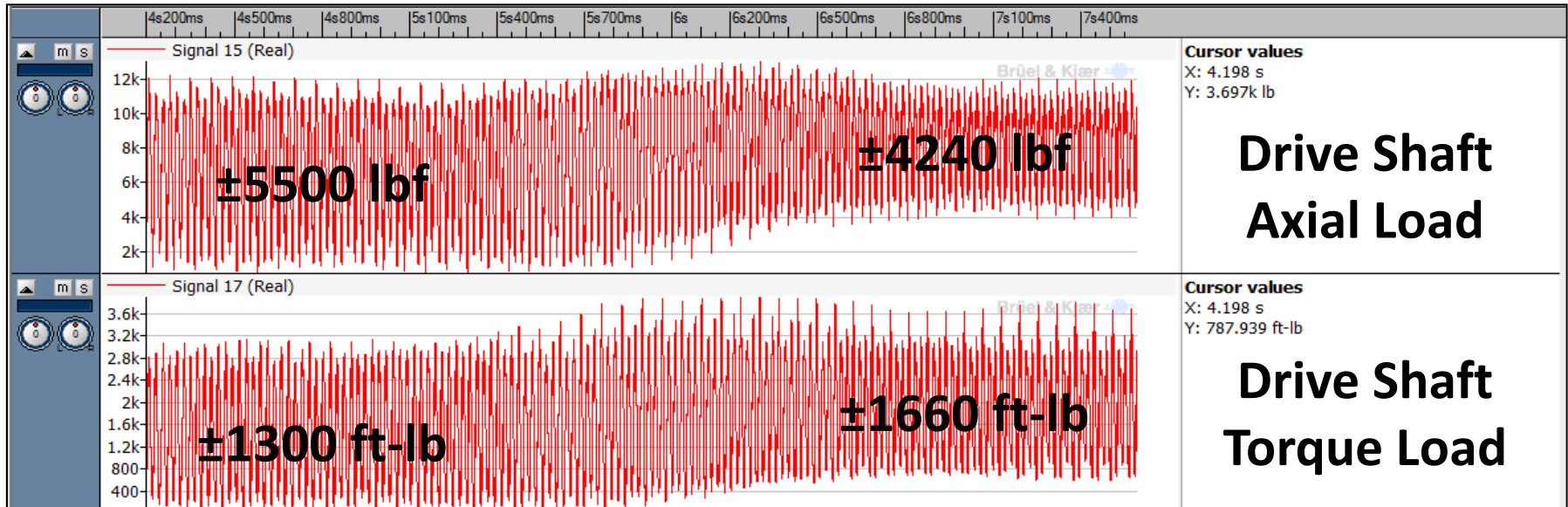
English Dimensional Outline Print General / Arrangement Drawing

CUSTOMER: ENOC Lavilla Inc
 Project Name: Amambatory Proj
 Project Number: 34-488
 P.O. #: 334488-05-01-00-0
 Murgoro River Water Supply Pumps - Diesel Drives Pumps
 TAG: 28-PU-05-A/B/C
 S.O. #: 31-032
 VDR Code: Q1, U4, M1, M6
 Client: Sherrill-Sherburne-Kerr
 314032-EDDM REV D

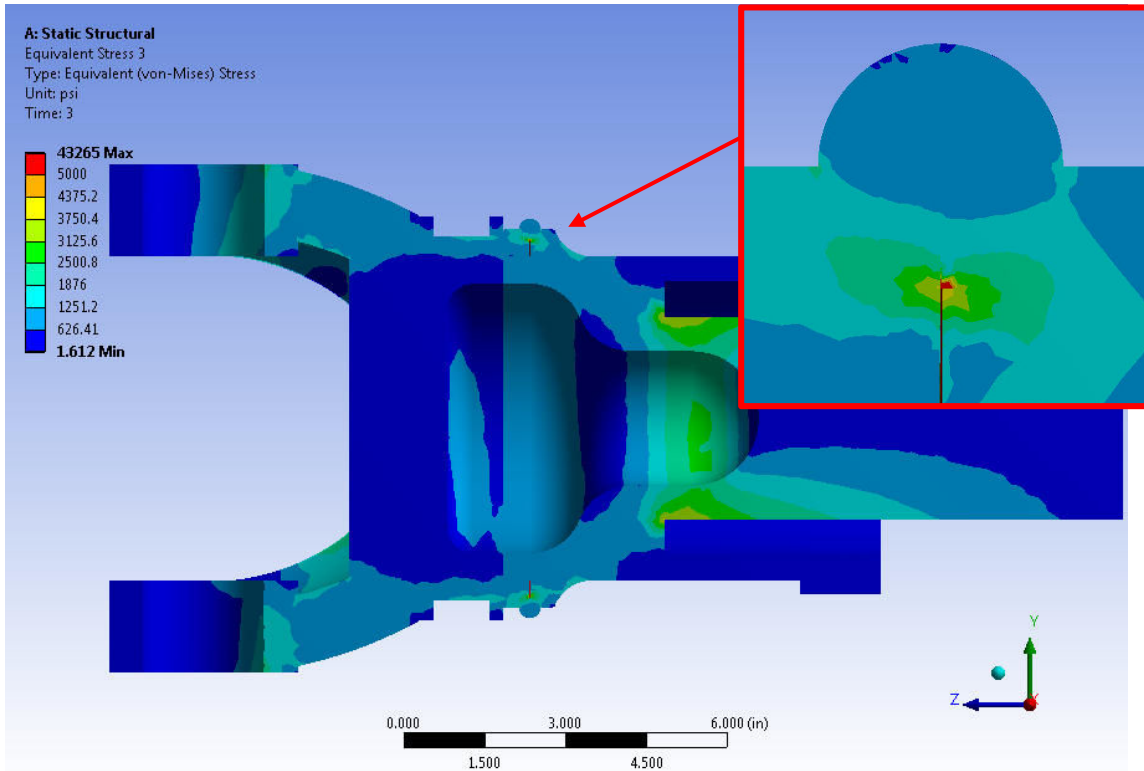
Analysis Method and Steps Taken

- Radio Frequency (RF) telemetry strain gauges measuring axial bending and torque were installed on the driveshaft of one pump.
- Time-transient vibration testing results on the pump, gearbox, drive shaft, and engine were collected using accelerometers and shaft sticks.
- An Operating Deflection Shape (ODS) test was performed to reveal dynamic behavior of the entire pump system.
- Experimental Modal Analysis (EMA) data was collected to find natural frequencies of the different system components.
- A test-calibrated Finite Element Analysis (FEA) based fracture mechanics analysis approach was used to predict the ability of detected stresses in the drive shaft to encourage initiation and propagation of the crack.

Time Domain Plots/ Drive Shaft 1,650 - 1,820 rpm



Finite Element Analysis



Stress Linearization Through Weld Bead

Results	
<input type="checkbox"/> Membrane	1407.2 psi
<input type="checkbox"/> Bending (Inside)	1739. psi
<input type="checkbox"/> Bending (Outside)	1739. psi
<input type="checkbox"/> Membrane+Bending (Inside)	3118.1 psi
<input type="checkbox"/> Membrane+Bending (Center)	1407.2 psi
<input type="checkbox"/> Membrane+Bending (Outside)	535. psi

If the weld on the driveshaft did not penetrate to the inner diameter of the welded components in the region near the weld, this would create a geometry that is basically a crack around the circumference of the shaft. This creates a high stress concentration at the inner edge of the weld. Such a circumstance can be quantitatively evaluated with Linear Elastic Fracture Mechanics (LEFM).

Fracture Mechanics Calculation

The peak membrane plus bending stress located at the region near the “crack” was calculated from the FEA model to be 3118 psi due to the observed alternating torque and axial loading.

Fatigue Analysis Due to Alternating Torque and Axial Load

Alternating Torque Load Stress $\sigma_a := 3118.1$ psi

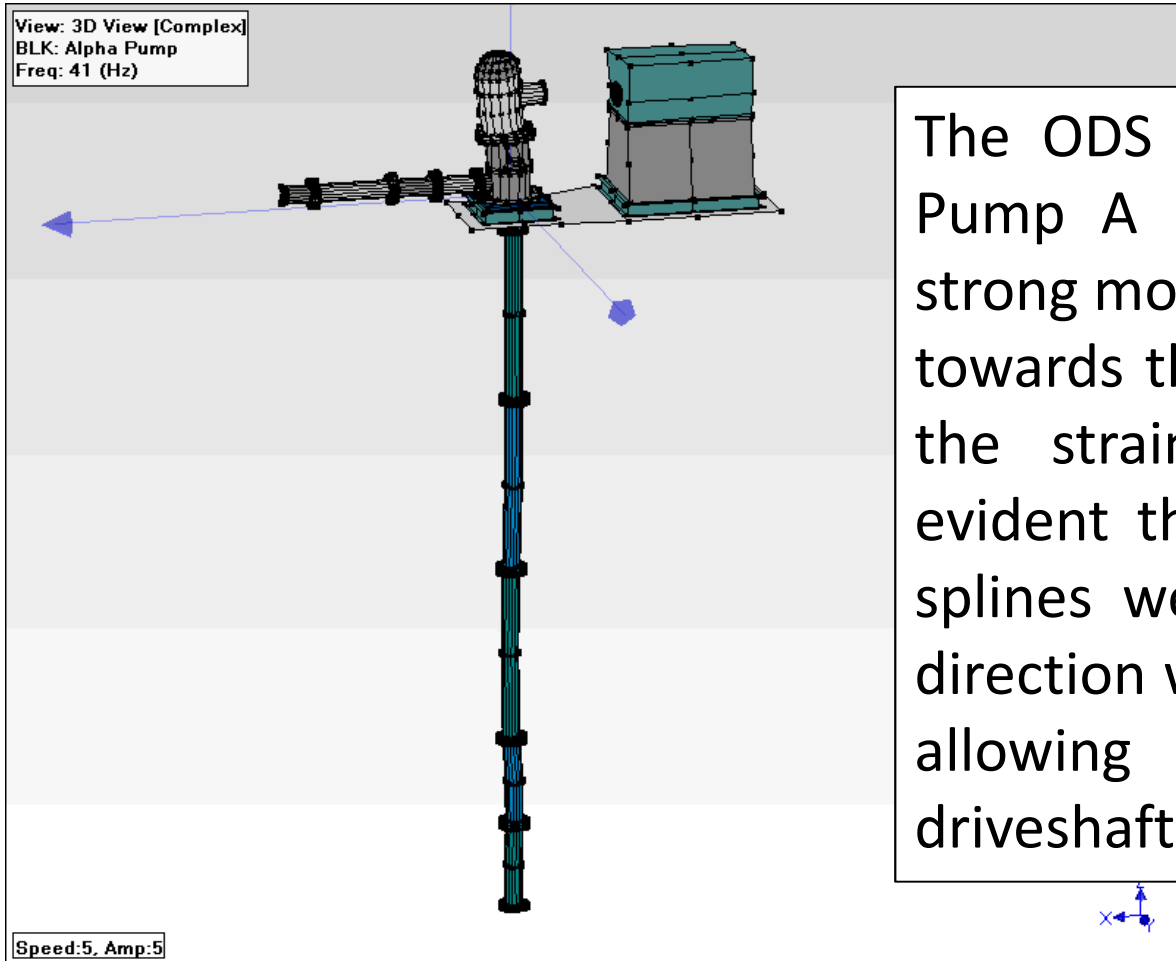
Crack Length $a := 3.3$ inches

Geometry Correction Factor $Y := 1.1$

Stress Intensity Factor $k_I := \sigma_a \cdot Y \cdot \sqrt{\pi \cdot a} = 1.104 \times 10^4$

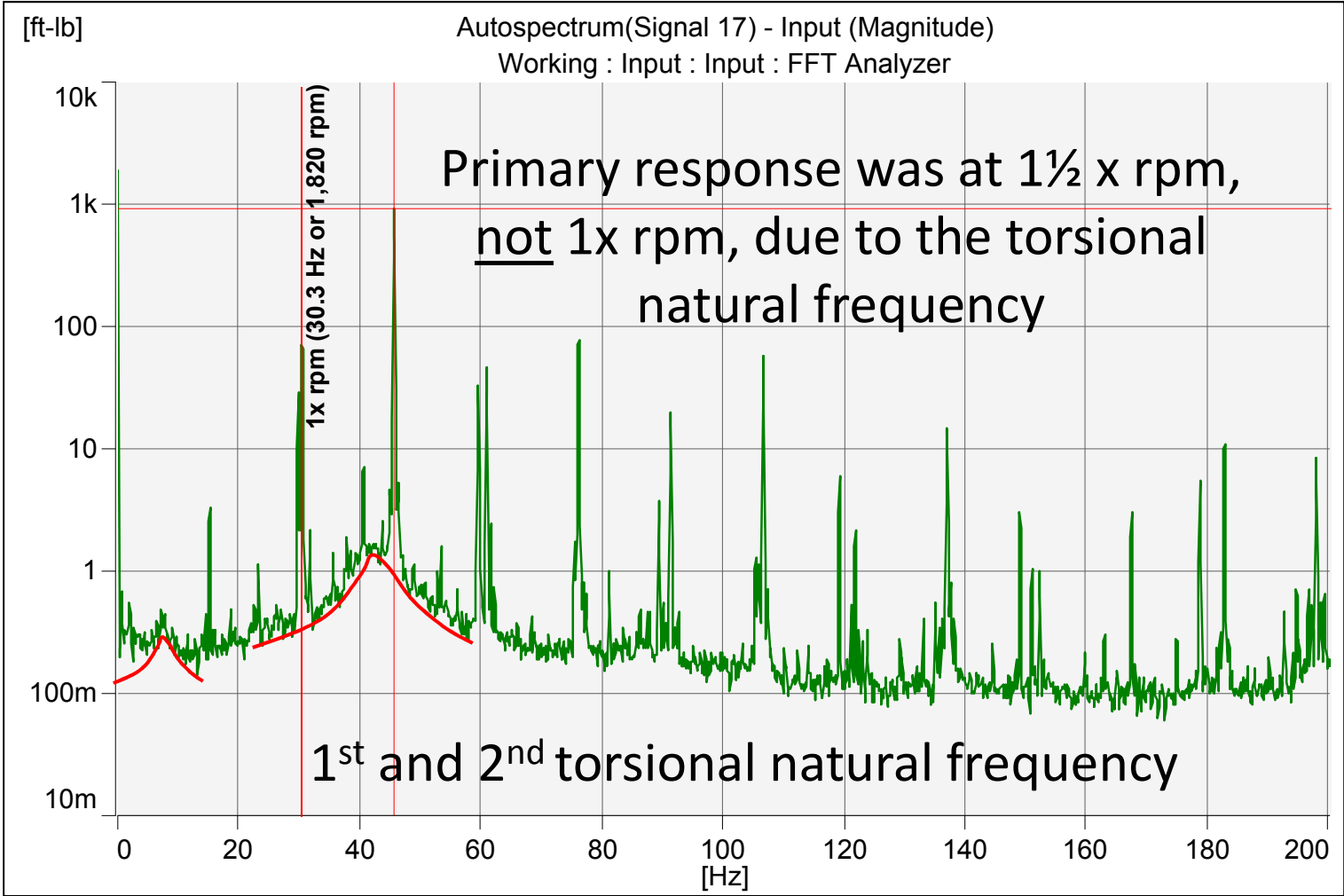
For carbon steels, exceeding a critical stress intensity factor of about $10,000 \text{ psi(in)}^{0.5}$ indicates that the crack or material flaw of radial length “a” has the ability to propagate under alternating load. This analysis calculated a possible k_I value of 11,040 which indicates significant but “borderline” probability of failure from fatigue loading of the weld bead.

Computer Model Using ME'Scope Program Plotting ODS Test Results



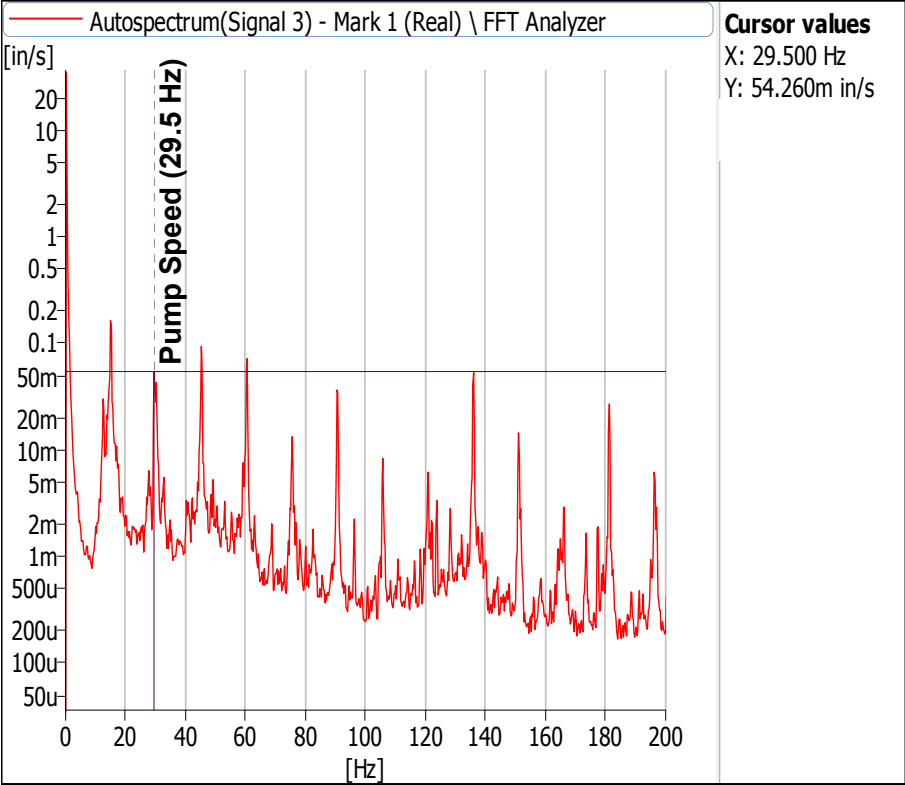
The ODS animation at 41 Hz on Pump A (1.5x RPM) indicated a strong motion of the gearbox head towards the engine. According to the strain gauge data, it was evident that the drive shaft axial splines were binding in the axial direction when under torque load, allowing this mode to enable driveshaft failure.

Strain Gage Test

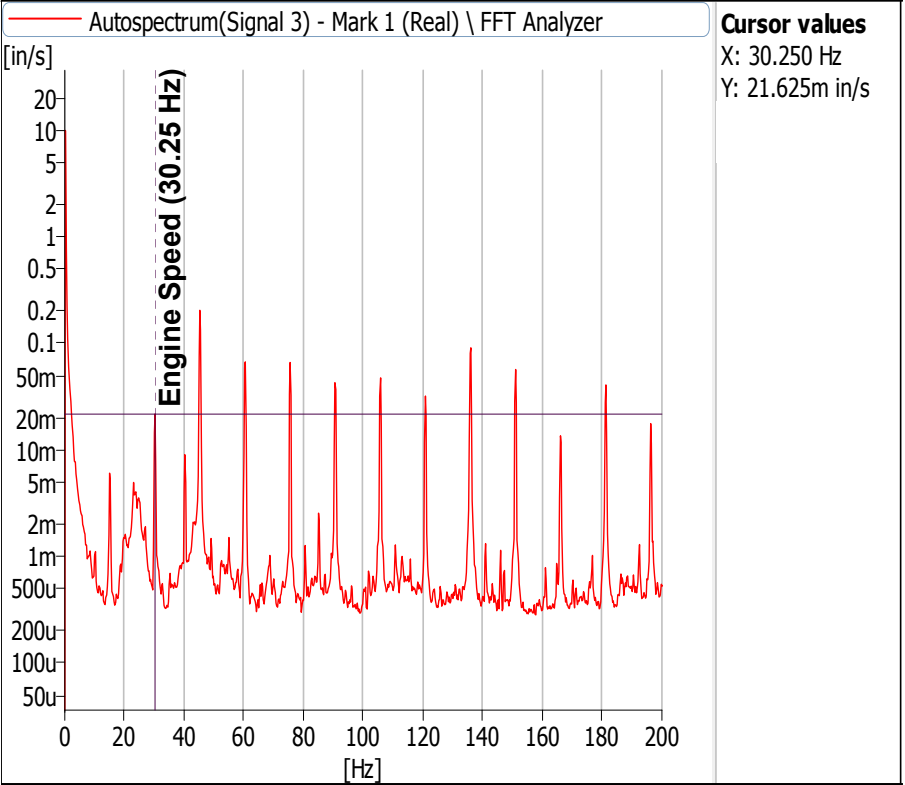


Drive Shaft Torque Spectrum and Torsional Natural Frequencies
Natural frequencies at 7.25 Hz and 42 Hz.

FFT Spectra of Vibration at Gearbox and Engine Top



Gearbox Top,
Input Shaft End



Engine Top,
Drive End

Notice that the $\frac{1}{2} \times \text{rpm}$ harmonics appear in both the engine and gearbox.

Mangoro Station Pump C Vibration vs. Speed

Signal #	Eng. Speed (rpm)	Pump Speed (rpm)	1x Pump (Hz)	Amp 1x in/s rms Pump	1.5x Eng. (Hz)	Amp 1.5x in/s rms Pump	Overall in/s	Mean Torque Load (ft-lb)	Power (HP)	Torque Oscillation (ft-lb) 0-pk	Oscillating Torque Load (%) 0-pk	Mean Axial Load (lbf)	Axial Oscillation (lbf) 0-pk	Oscillating Axial Load (%) 0-pk	Separation Margin (%) between 42 Hz and 1.5x
3	800	783	13.0	0.12	20.00	0.01	0.14	289	44	250	87%	2,414	1,500	62%	-52.94%
4				0.04		0.02	0.11								
3	1200	1174	19.6	0.14	30.00	0.01	0.20	600	137	550	92%	4,052	2,750	68%	-28.57%
4				0.07		0.02	0.18								
2	1650	1615	26.9	0.17	41.25	0.21	0.32	1,480	465	1,300	88%	6,827	5,500	81%	-1.79%
4				0.09		0.09	0.53								
2	1808.5	1770	29.5	0.09	45.21	0.11	0.35	1,490	513	1,400	94%	7,974	5,350	67%	7.65%
4				0.04		0.04	0.56								
2	1820	1781	29.7	0.07	45.50	0.15	0.24	1,900	658	1,660	87%	8,045	4,240	53%	8.33%
4				0.04		0.03	0.50								
2	1884	1844	30.7	0.08	47.10	0.16	0.48	1,910	685	1,625	85%	7,957	4,065	51%	12.14%
4				0.05		0.04	0.52								

Signal 2 – Top of the gearbox parallel to the discharge

Signal 3 – Top of the gearbox perpendicular to the discharge

Signal 4 – Top of the pump discharge head parallel to the discharge

*1,750 rpm column piping vibration was interpolated

Conclusions / Observations

1. The failure mechanism of the drive shaft was caused by the elevated axial and torsional oscillation loads in combination with the jammed driveshaft spline.
2. The situation became severe because an axial (horizontal parallel to the crankshaft) pump natural frequency and torsional shaft assembly natural frequency were simultaneously in resonance with an unexpectedly high 1.5x running speed harmonic, which appeared due to a poorly tuned engine (resulting in a 1/2x rpm fundamental and its harmonics).
3. The 2nd torsional natural frequency of the drive shaft was determined to be at 42.0 Hz (Pump C). The separation margin from 1.5x running speed is within 5% from both the pump and engine speed.
4. The torsional oscillation was observed to be as high as 94% zero-to-peak of the mean torque value.

Conclusions / Observations

5. The measured axial force oscillation imposed on the drive shaft peaked at 11,000 lbf pk-pk.
6. Since the weld on the driveshaft did not penetrate to the inner diameter of the material the region near the weld, this created effectively a crack around the circumference of the shaft. This resulted in a high stress concentration at the edge of the weld. The peak oscillating membrane plus bending stress amplitude located at the region near the “crack” was calculated from the FEA model to be 3118 psi due to the observed alternating torque and axial loading.
7. For carbon steels, exceeding a critical stress intensity factor of 8,000-10,000 indicates the effective initiated crack length “a” has the ability to propagate under alternating load. This analysis calculated a stress intensity factor value of 11,040 which explained the failure from fatigue loading on the weld bead.

Recommendations/ Results

1. The engine was re-tuned due to the observed mis-firing, since it was providing unusually strong torque impulses or "shocks" at the rate of $1/2x$ RPM, which caused strong frequency harmonics. This left no place to "park" the system natural frequencies to avoid resonance.
2. A torque shock absorbing coupling between the engine and the drive shaft was implemented. The entire drive shaft and coupling assembly was replaced including the u-joints at each end.
3. The highest vibration level dropped from 24 mm/s pk at the gearbox horizontal measurement location near the input shaft to 7.8 mm/s pk after change to a flexible coupling. Failures ceased.