

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

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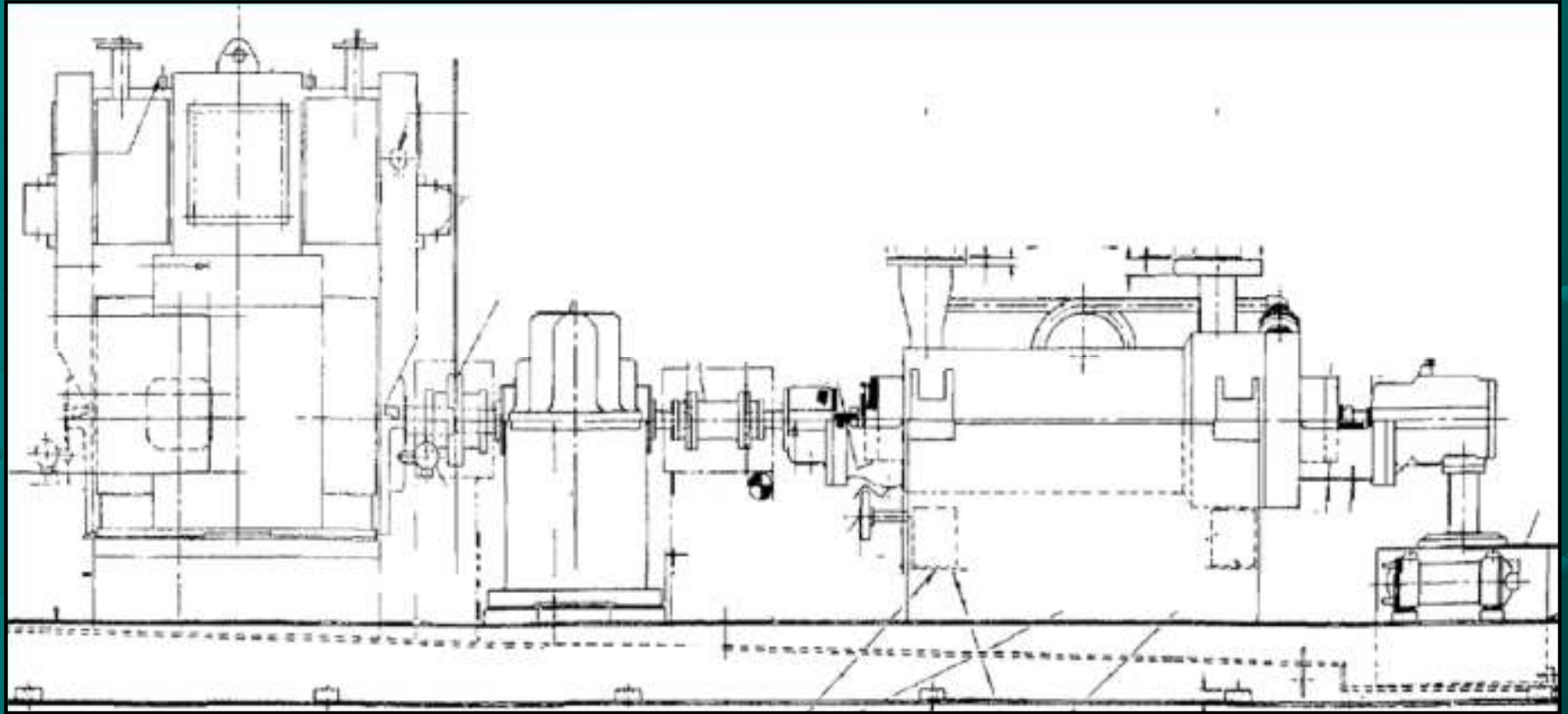
*By:*

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## Introduction

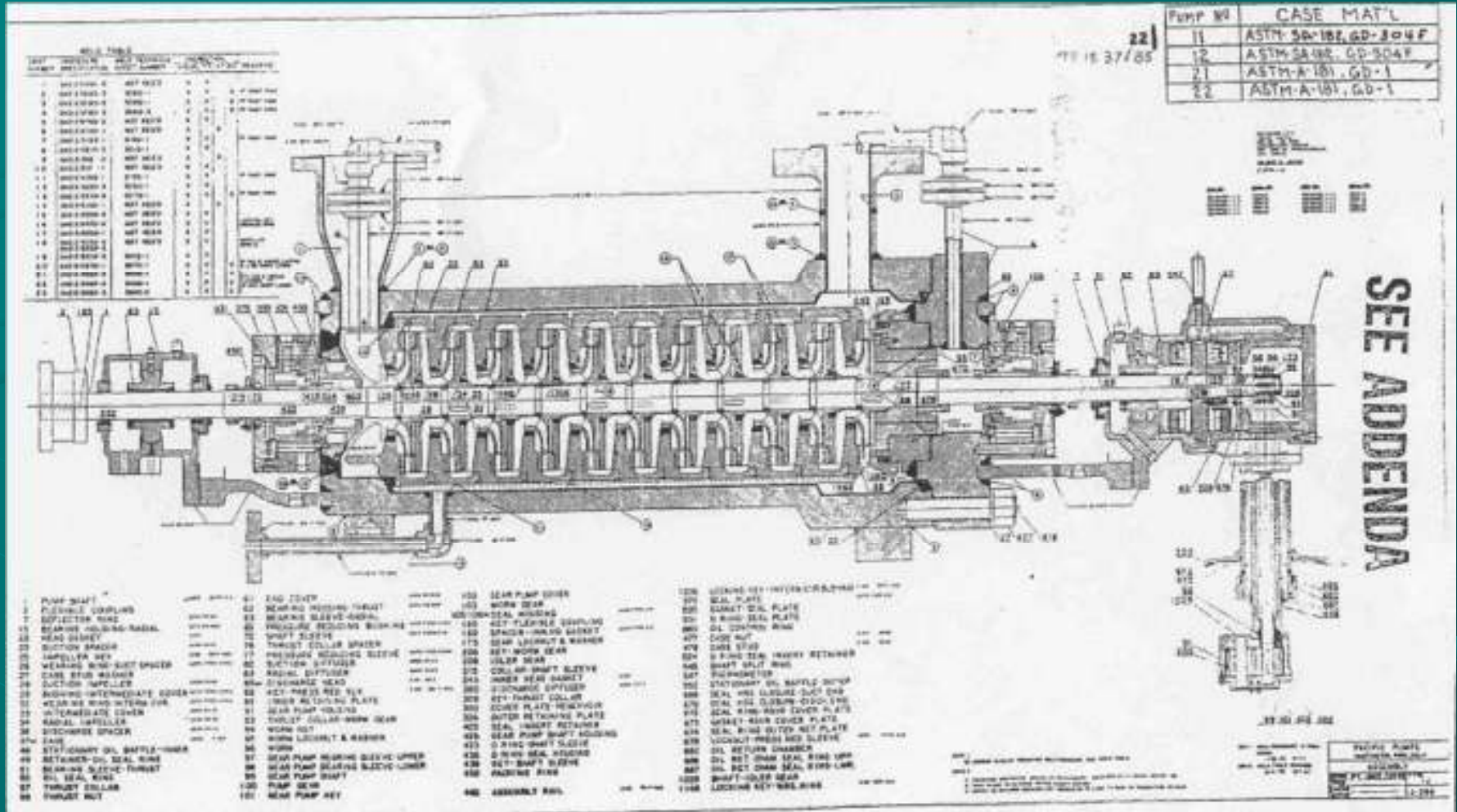
- Two multistage barrel type pumps for each 1200 MW Unit were installed in 1986 in a large power plant.
- The 11 stage centrifugal pumps were designated as Pump(s) 1A/B and Pump(s) 2A/B. All impellers were designed with 6 vanes and 8 diffuser vanes.
- The pumps are driven by 600 HP four-pole induction 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



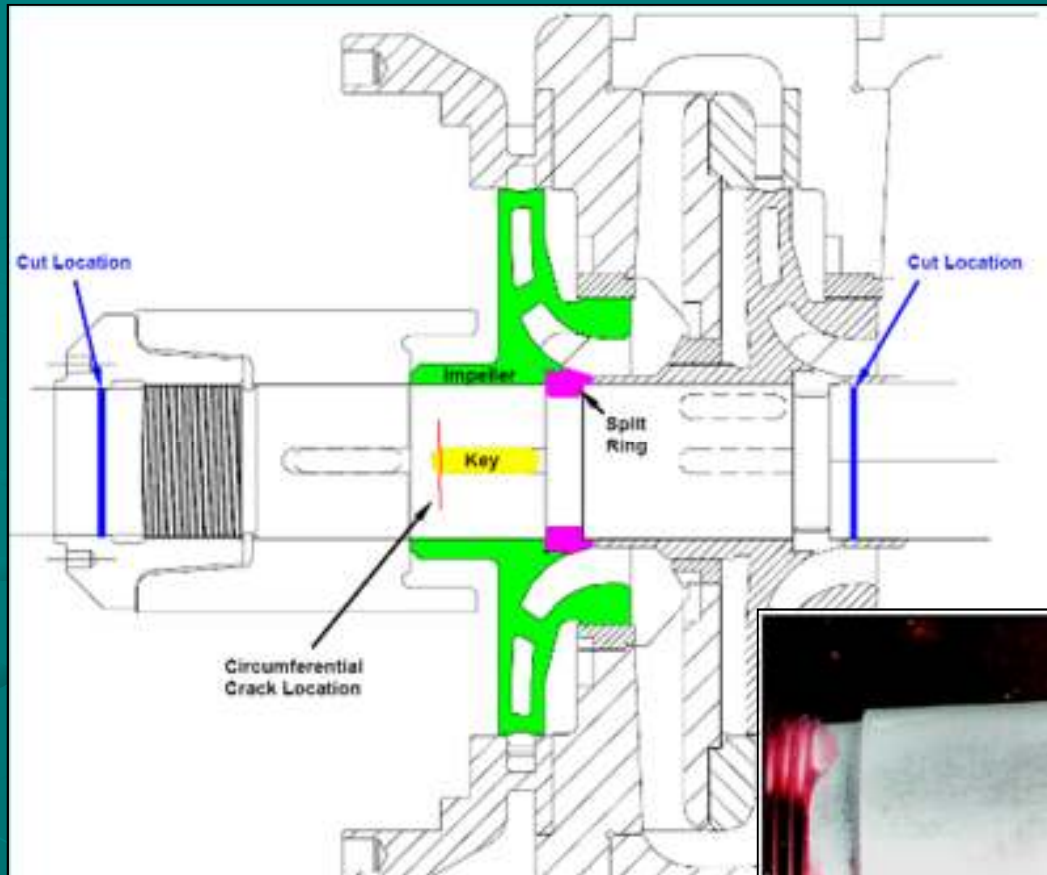
**Barrel Type - 11 stage Pump**

## Background History

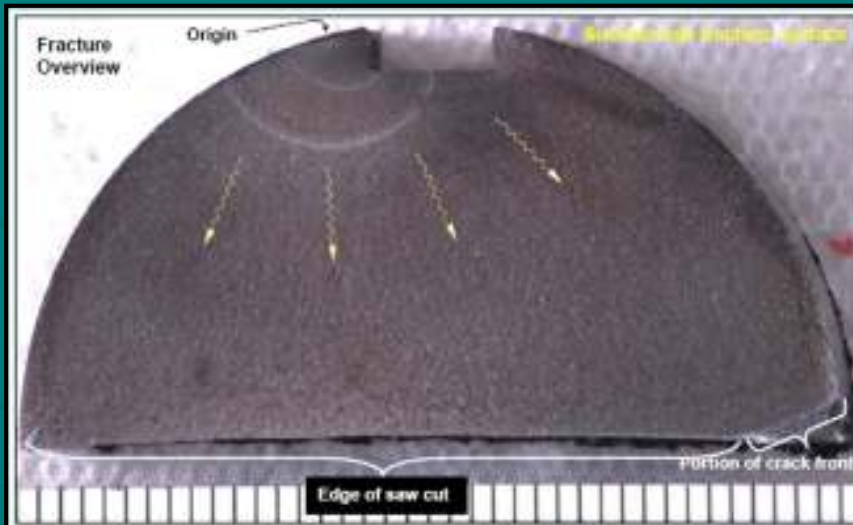
- Three pump shaft failures (cracks) have been detected since Pump 1B installation (MTBF=7.7 years).
- In 1989, the first failure took place, but the failure analysis was not properly documented and the failed shaft was not saved.
- In 1999, a crack was found at the rear end of the 9<sup>th</sup> stage hub.
- In December 2007, the last failure was discovered under the 11<sup>th</sup> 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 11<sup>th</sup> stage impeller hub.



# Background History



2007 Failure after 8 years of operation.  
Crack detected under 11<sup>th</sup> stage impeller hub with the origin away from the keyway.



# Specialized Testing

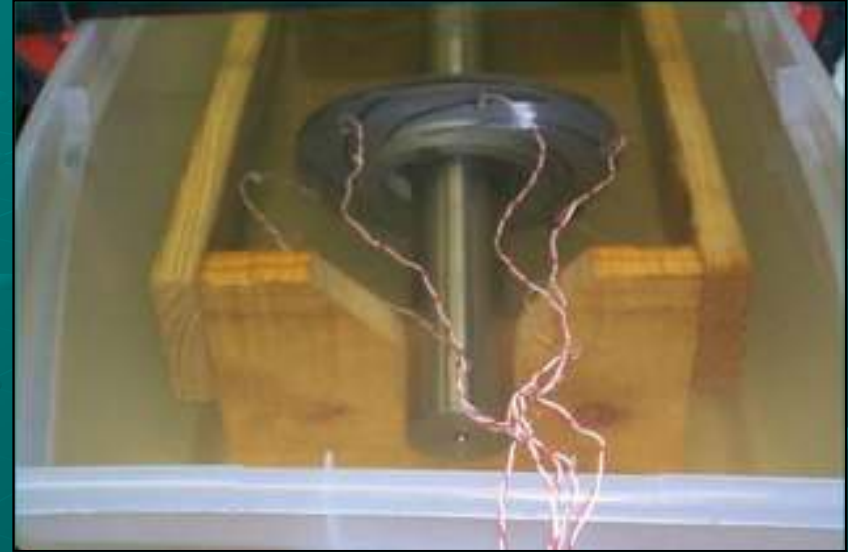
- ❖ **Experimental Modal Analysis (EMA) test to determine the natural frequencies of the impeller, pump structure, and the rotor system with their mode shapes.**
- ❖ **Monitoring test during transient and steady operation to monitor the vibration amplitude and natural frequencies.**
- ❖ **Dynamic pressure transducer data from suction, discharge, and balance line.**
- ❖ **Operating Deflection Shape (ODS) testing during steady operation.**



# **11<sup>th</sup> Stage Impeller EMA Testing**

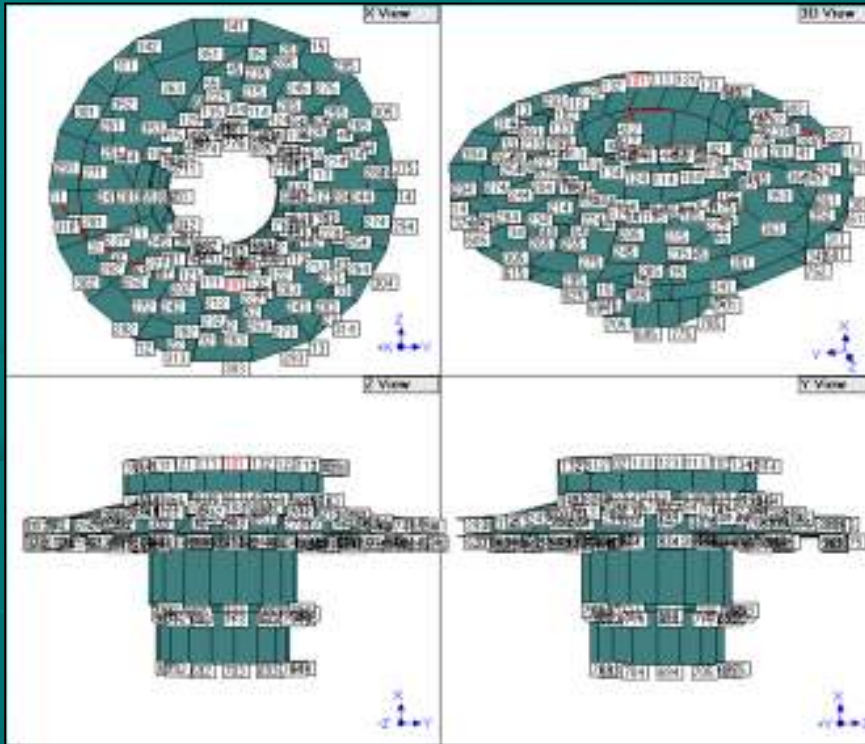
# Vibration EMA Test

## 11<sup>th</sup> Stage Impeller Modal Impact Test

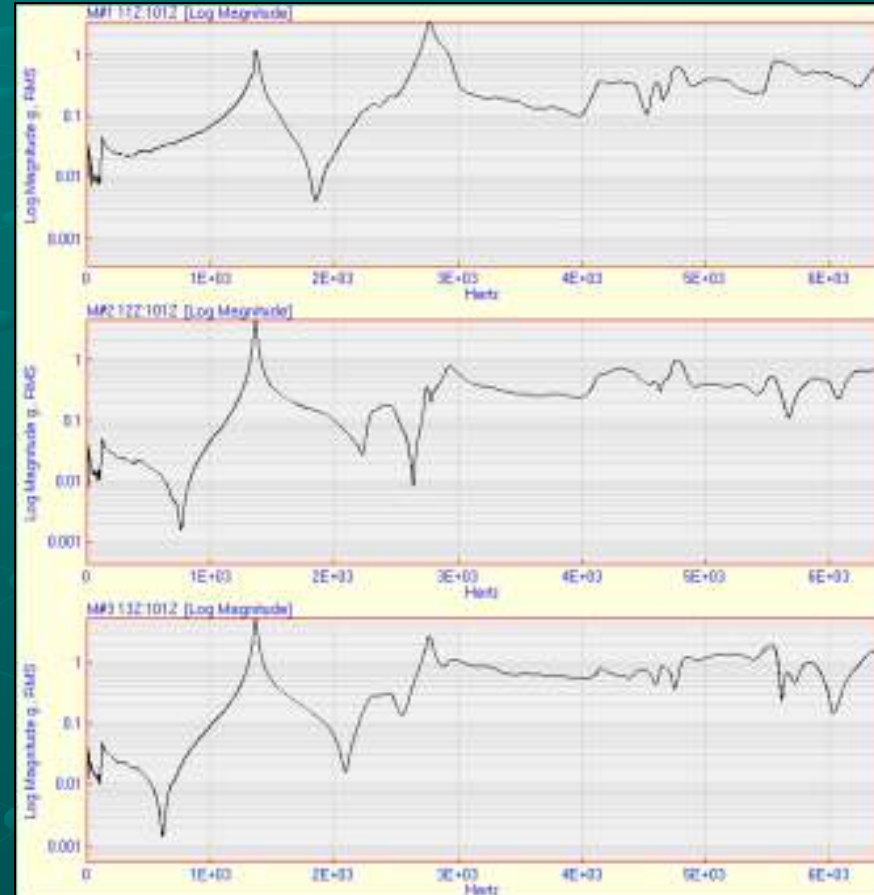


# Vibration EMA Test

## 11<sup>th</sup> Stage Impeller Modal Impact Test



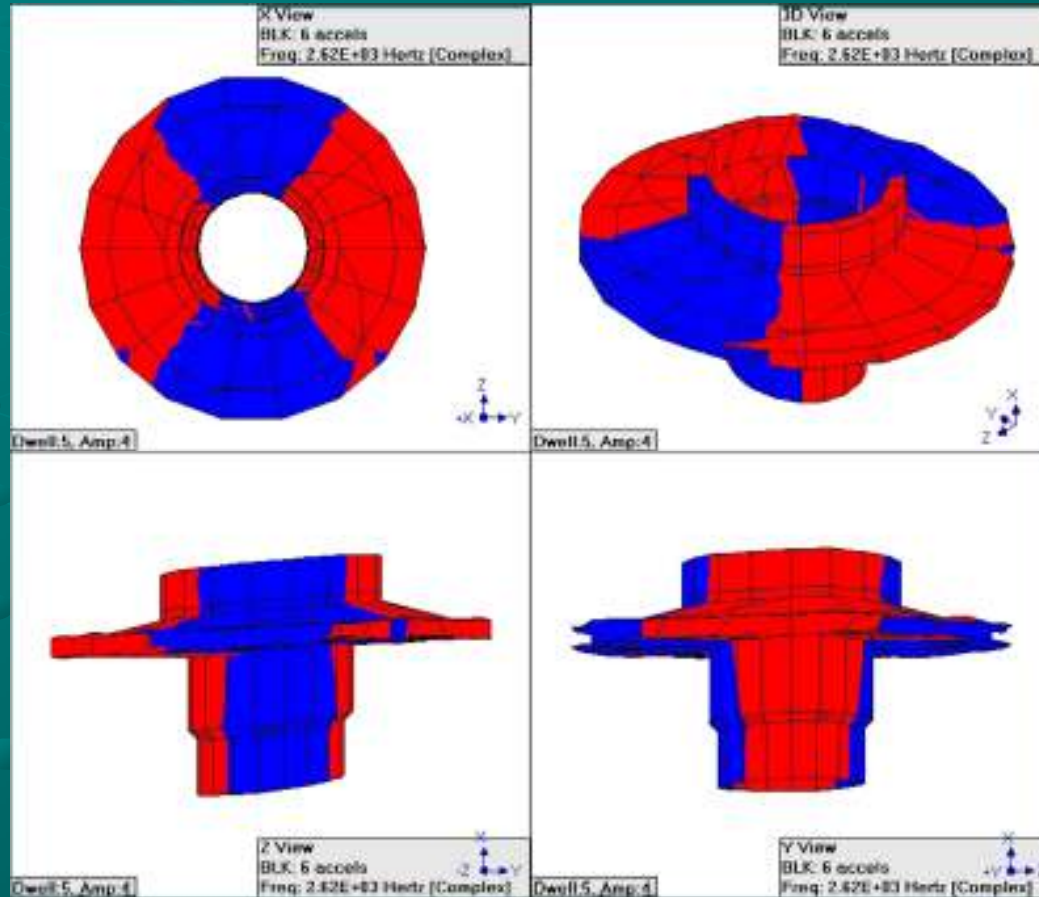
Impeller Model



FRF Wet

# Vibration EMA Test

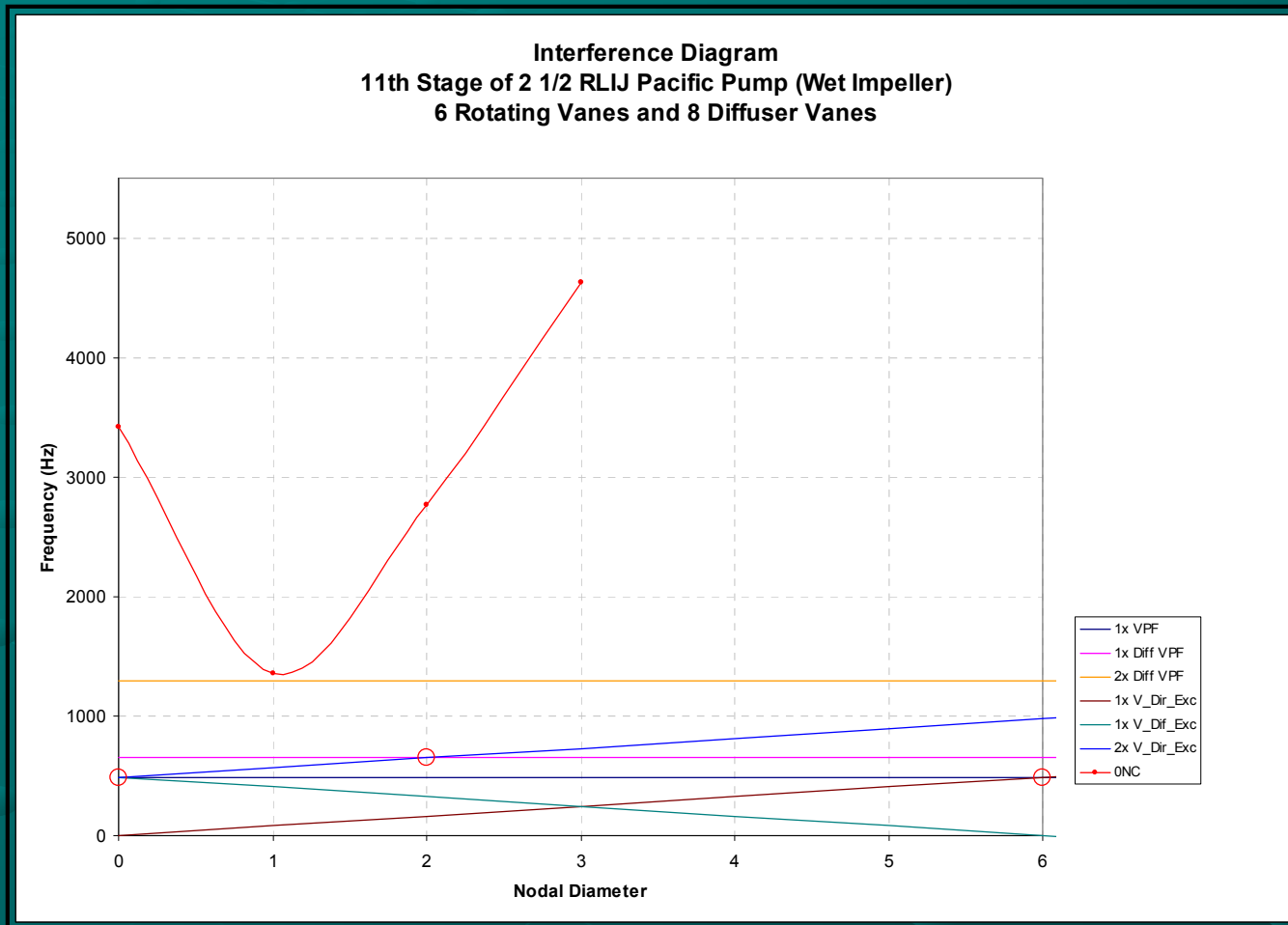
## 11<sup>th</sup> Stage Impeller Modal Impact Test



**2 Nodal Diameter / 0 Nodal Circle at 2620 Hz**

# Vibration EMA Test

## 11<sup>th</sup> Stage Impeller Modal Impact Test (Wet)

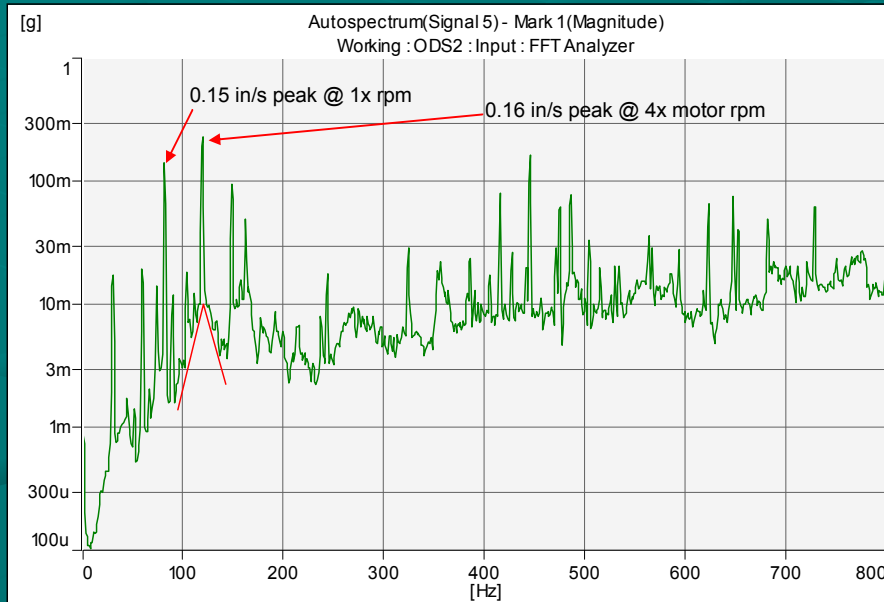


**Interference Diagram**

# Specialized Field Testing

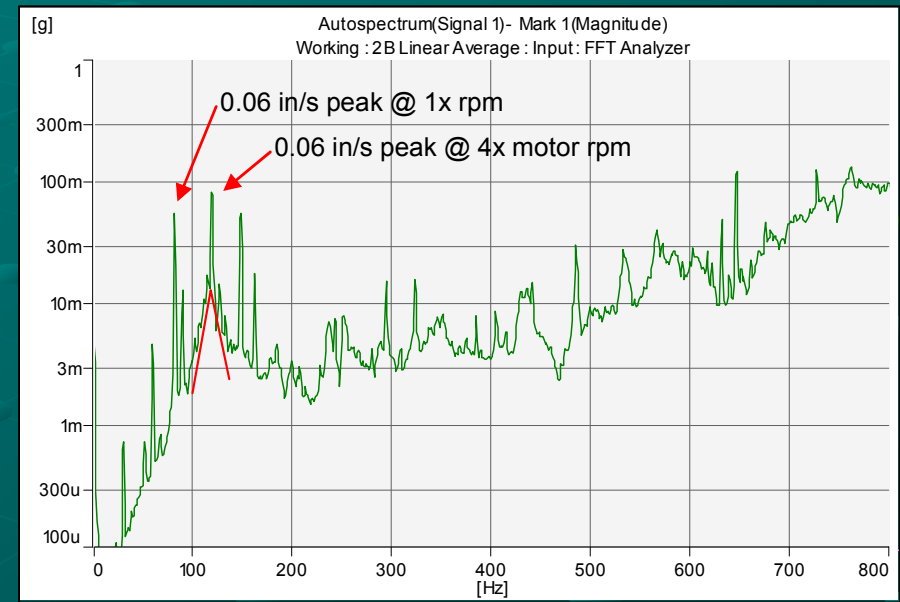
# Vibration Monitoring

Pump 1B Typical Spectrum OBB in the Vertical Direction



Pump Operating at 145 GPM

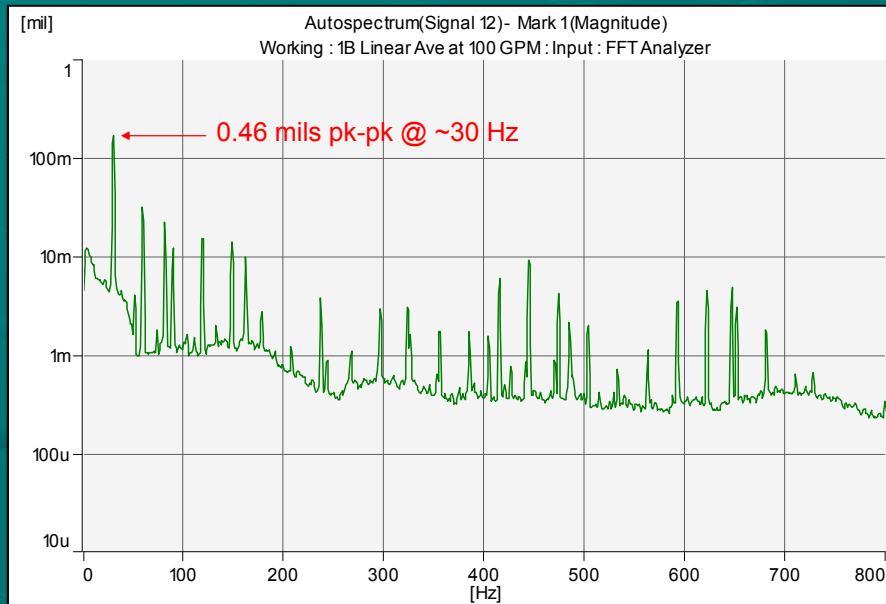
Pump 2B Typical Spectrum OBB in the Vertical Direction



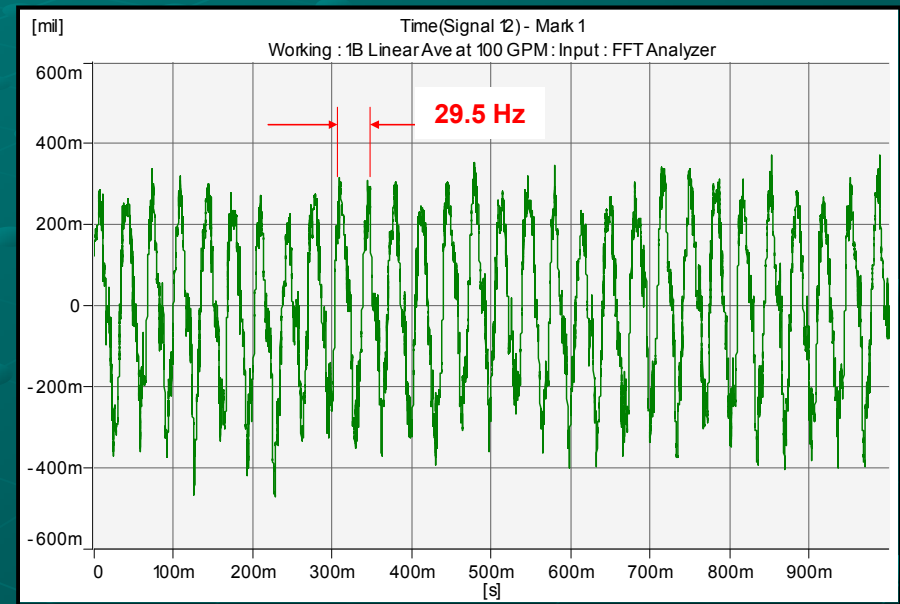
Pump Operating at 160 GPM

# Vibration Monitoring

## Pump 1B Axial Proximity Probe FFT at OBB



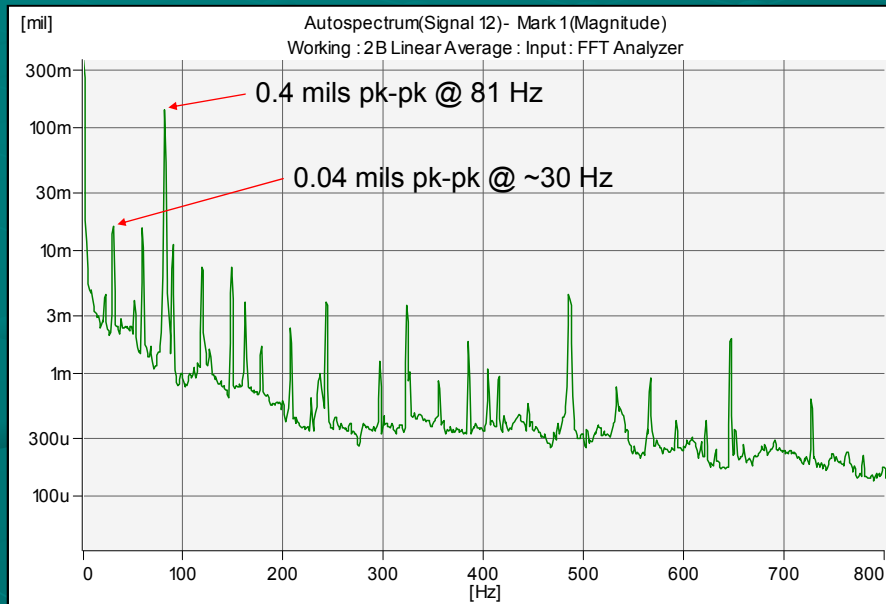
## Pump 1B Axial Proximity Probe Time Signature at OBB



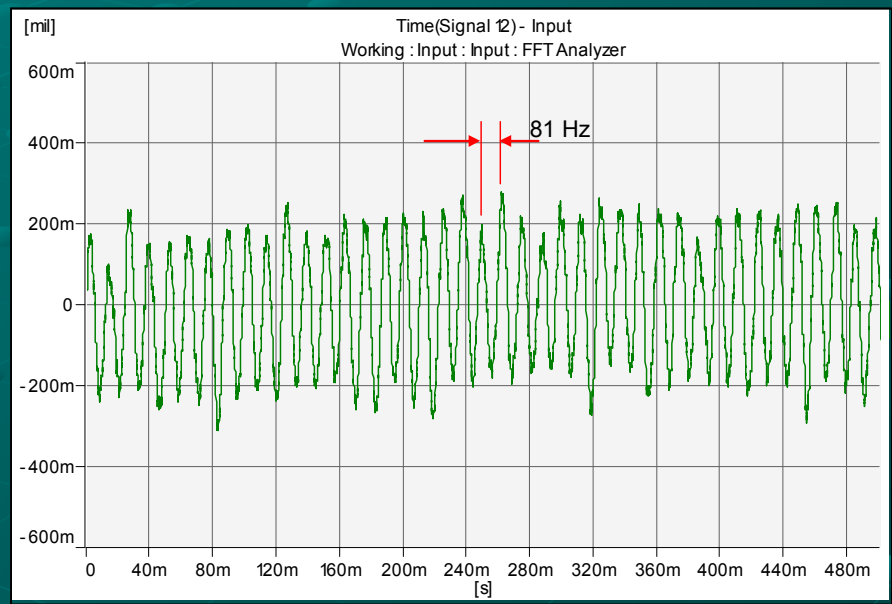


# Vibration Monitoring

## Pump 2B Axial Proximity Probe FFT at OBB



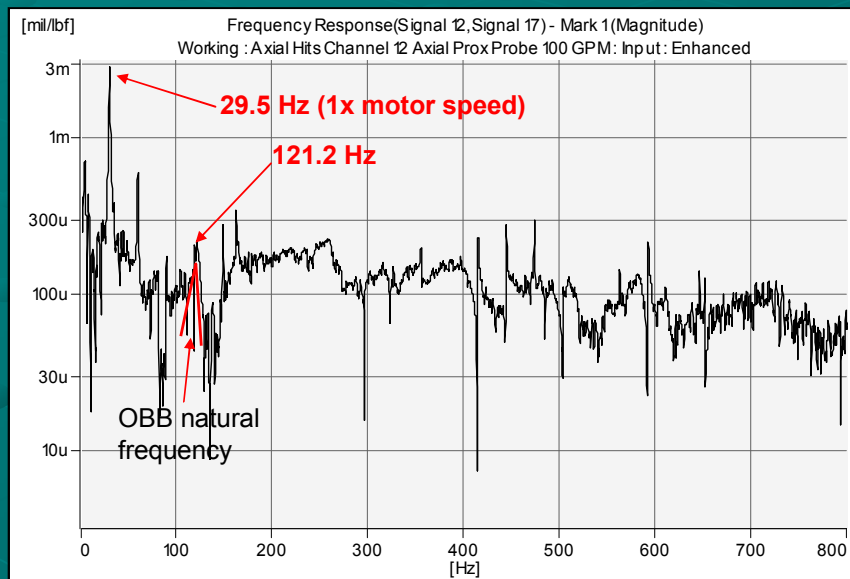
## Pump 2B Axial Proximity Probe Time Signature at OBB



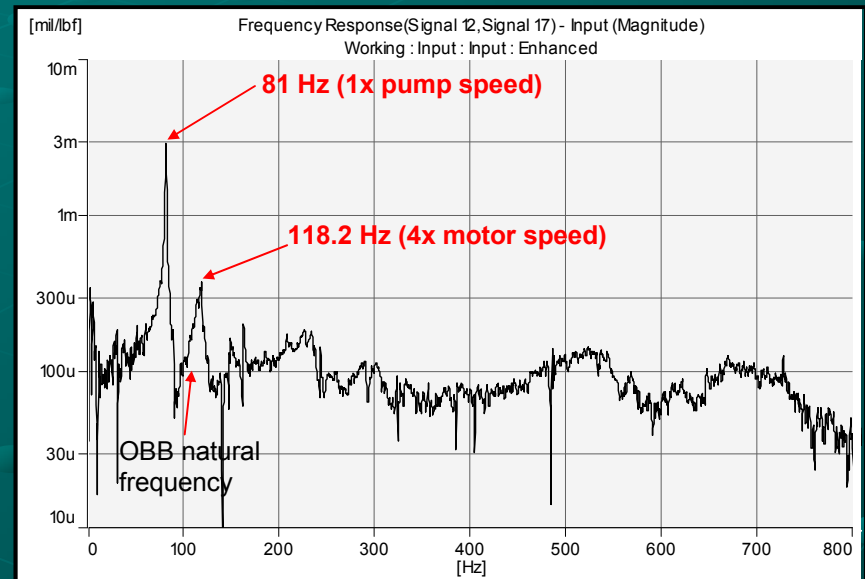
# Pump Rotor Modal Impact Testing During Operation

# Experimental Modal Analysis “Bump” Test

## Pump 1B Axial Proximity Probe FRF



## Pump 2B Axial Proximity Probe FRF

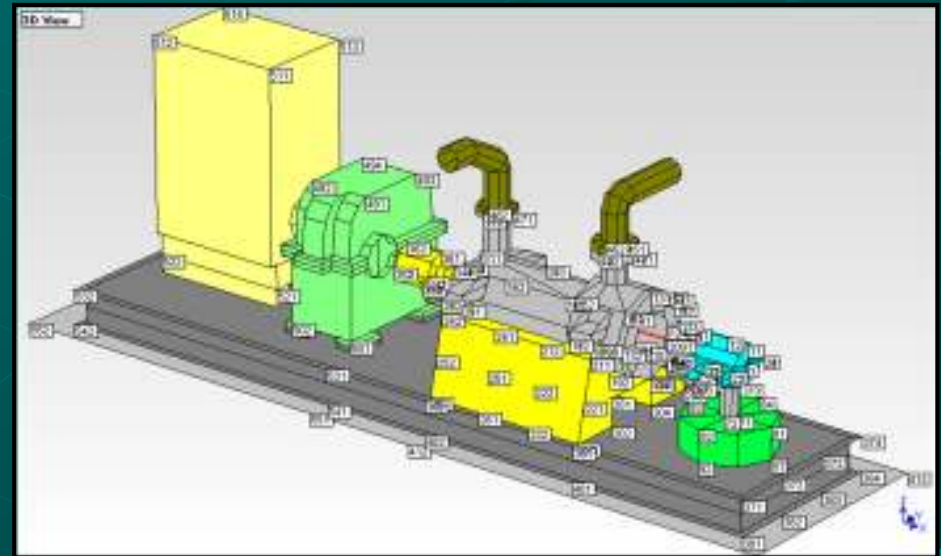
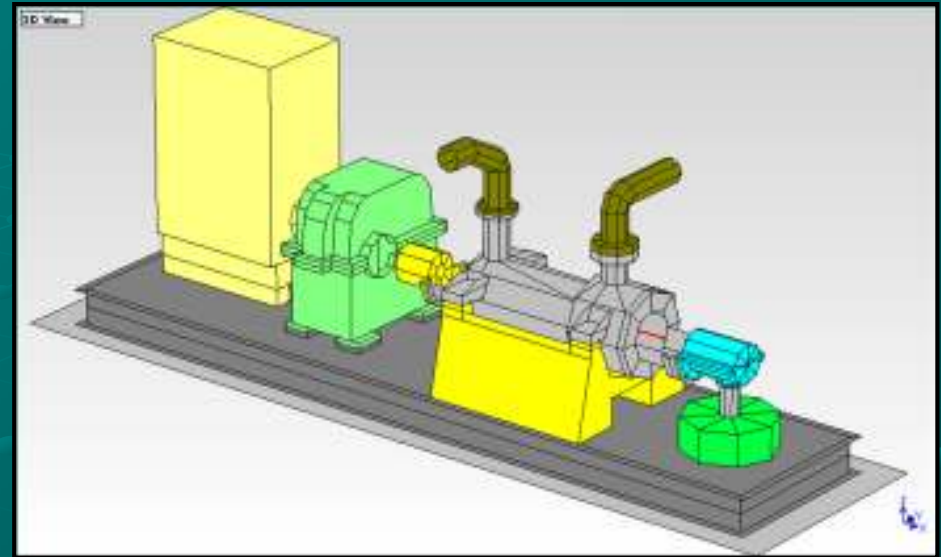


Pump operating at 160 GPM

# Operating Deflection Shape (ODS)

# Operating Deflection Shape

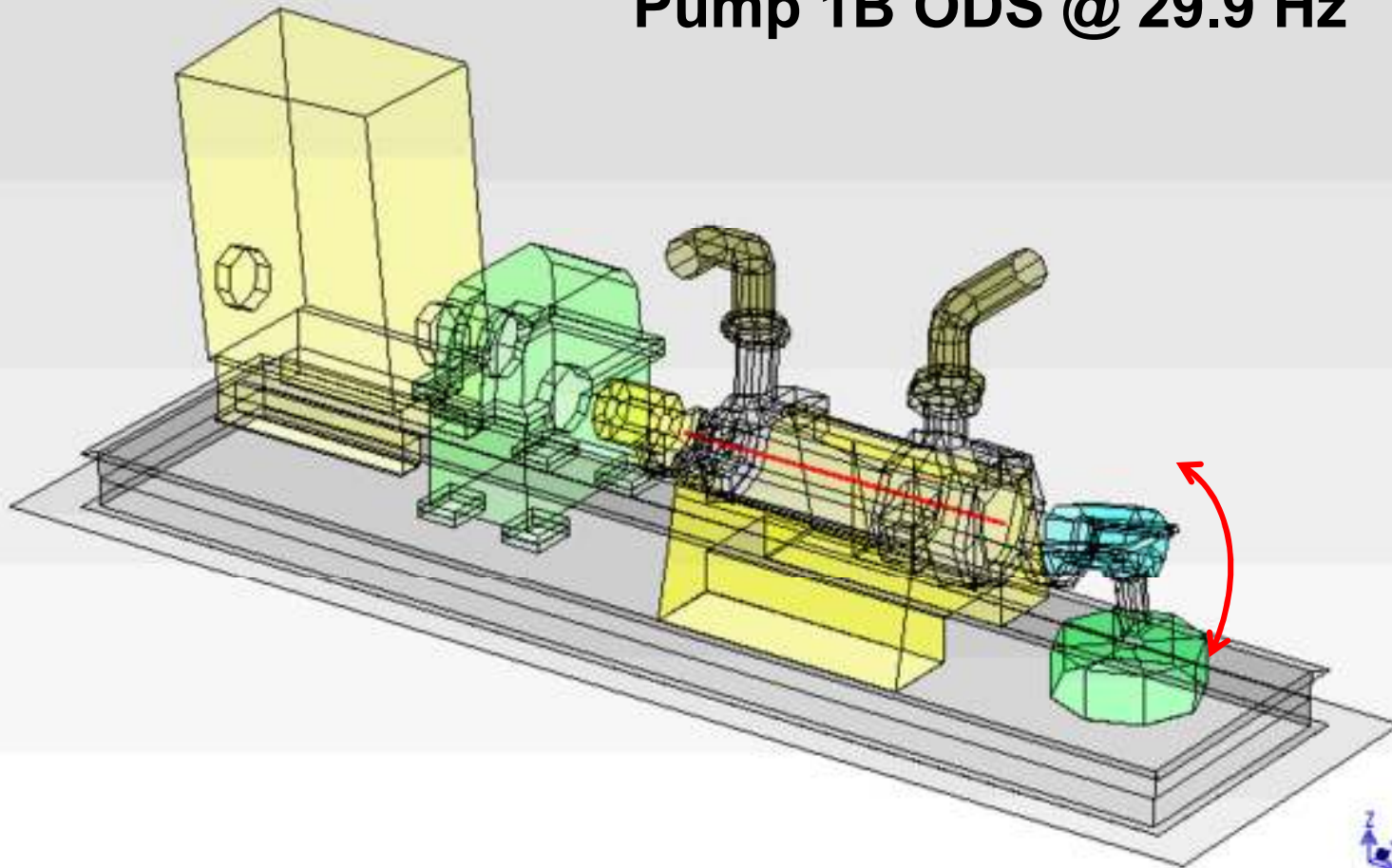
- Natural excitation signature of the pump train structure.
- Over 700 vibration measurements.
- Data base of amplitude vs. frequency and phase angle.
- 3-D CAD model assigning motion to each individual vibration data point.
- Create animations of the pump



# Operating Deflection Shape

3D View  
BLK: ODS01  
Freq: 29.9 Hertz [Complex]

Pump 1B ODS @ 29.9 Hz

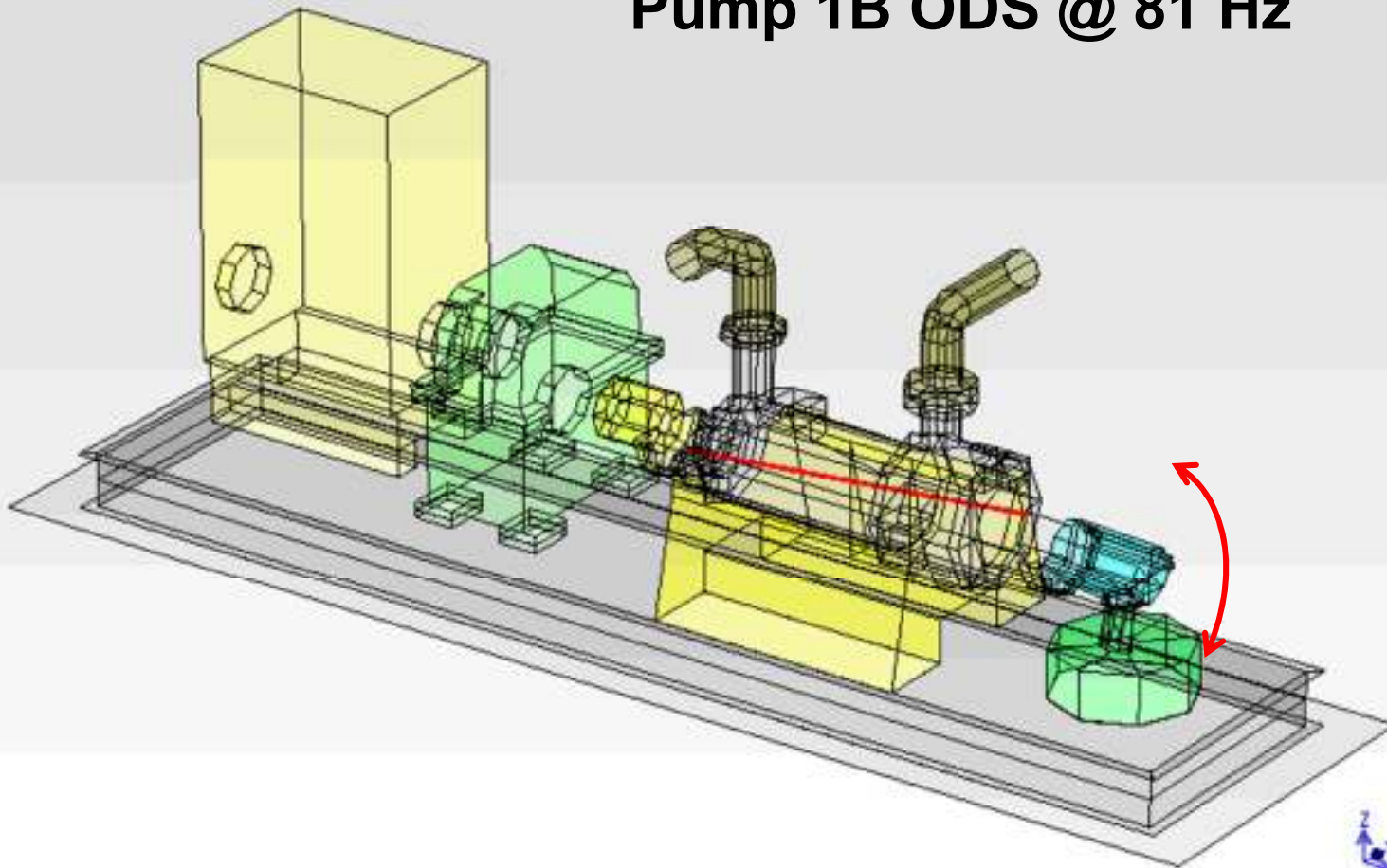


Dwell:5, Amp:5

# Operating Deflection Shape

3D View  
BLK: ODS01  
Freq: 81 Hertz [Complex]

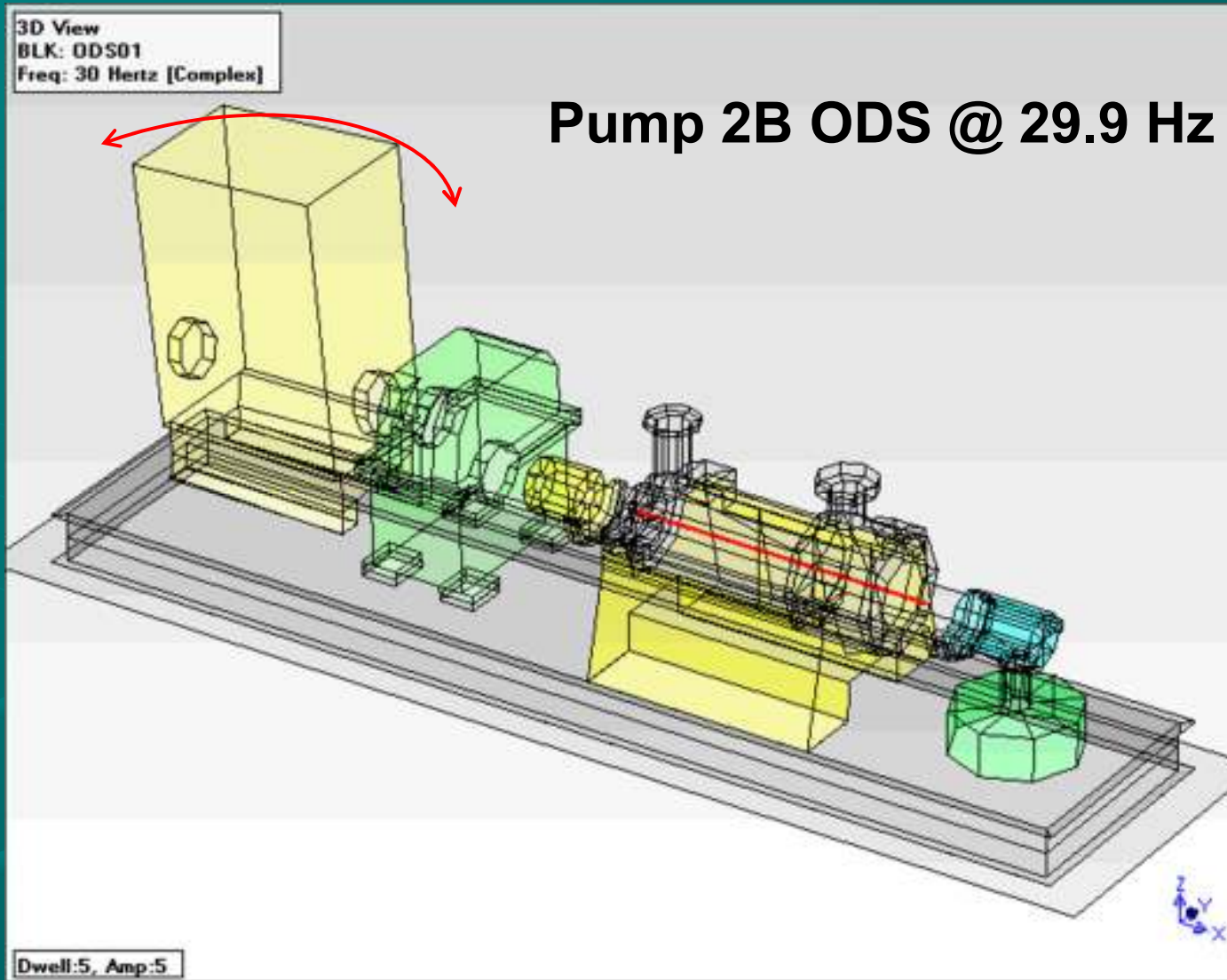
## Pump 1B ODS @ 81 Hz



Dwell:5, Amp:5

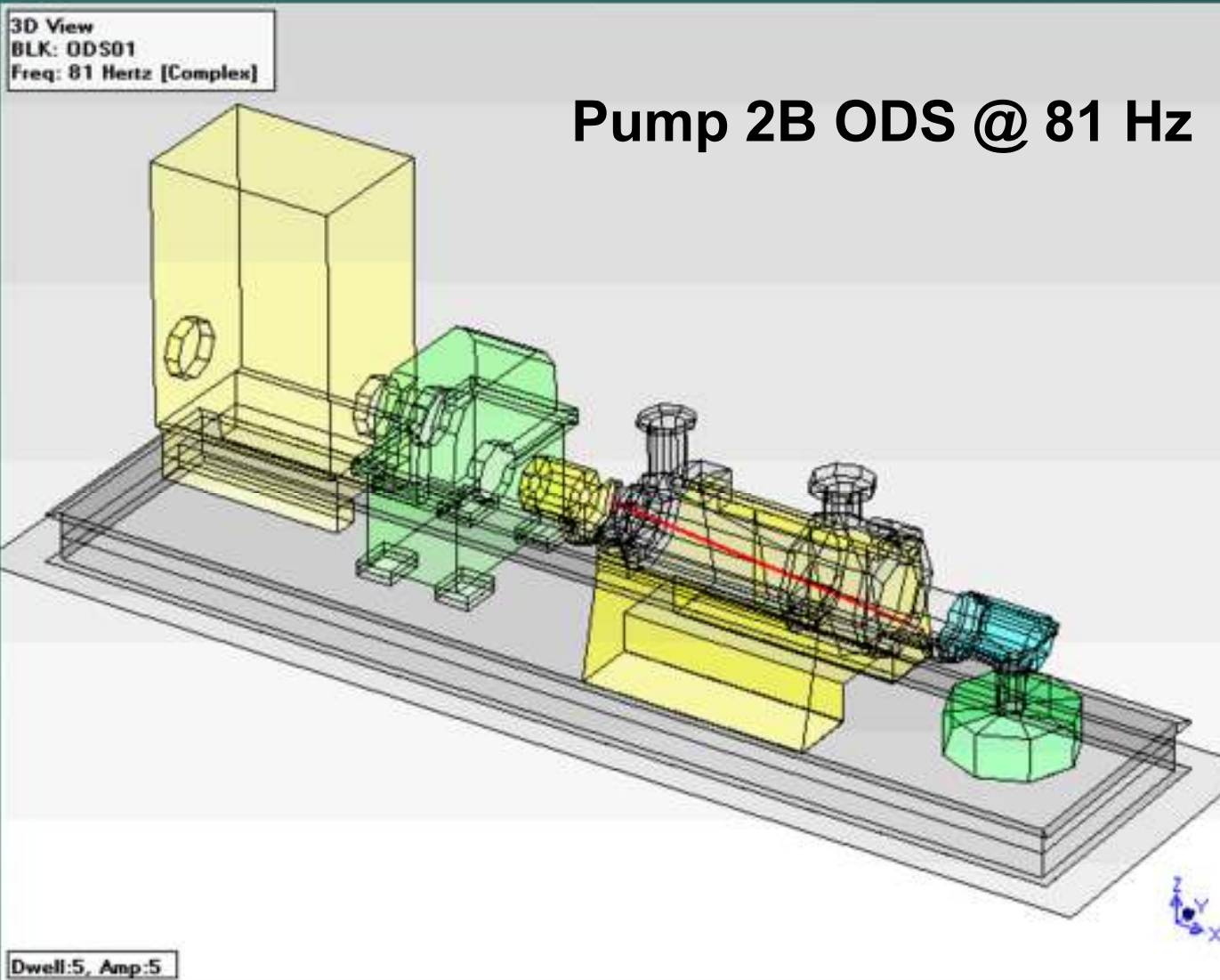


# Operating Deflection Shape





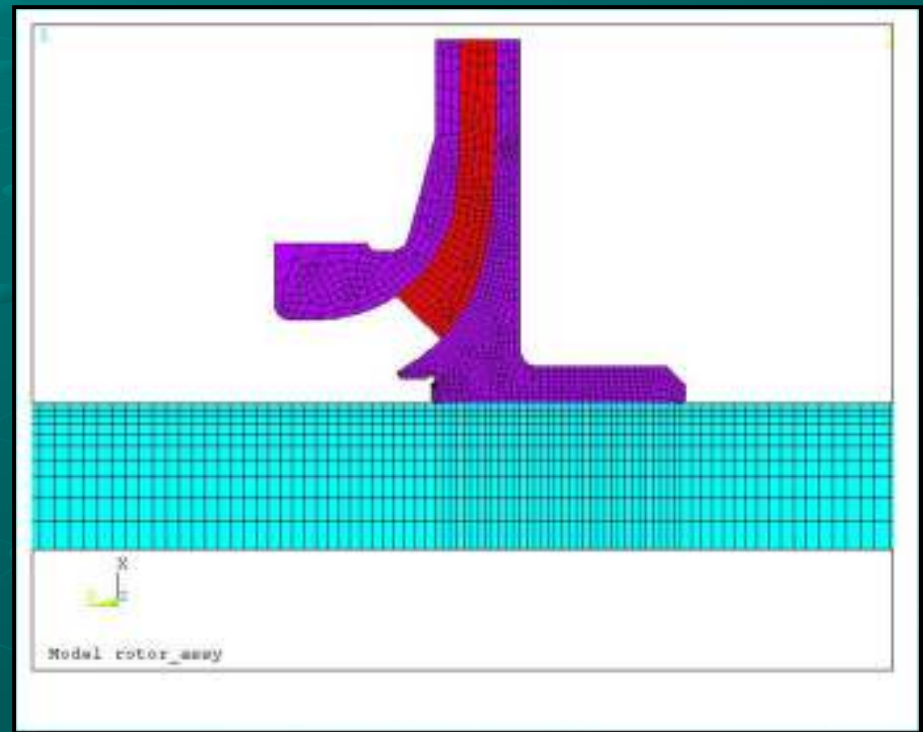
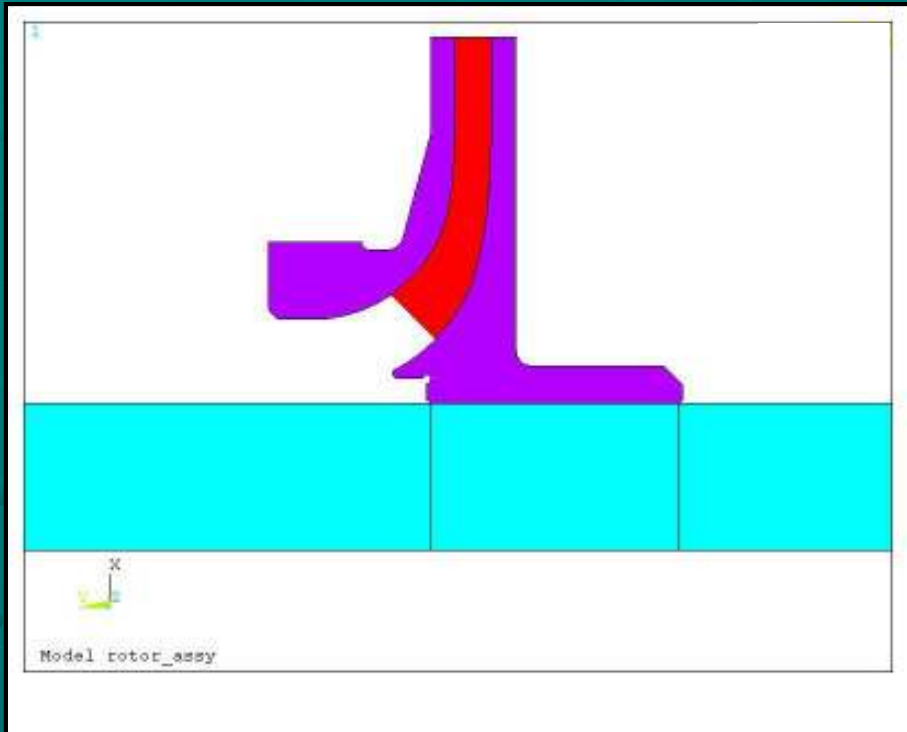
# Operating Deflection Shape





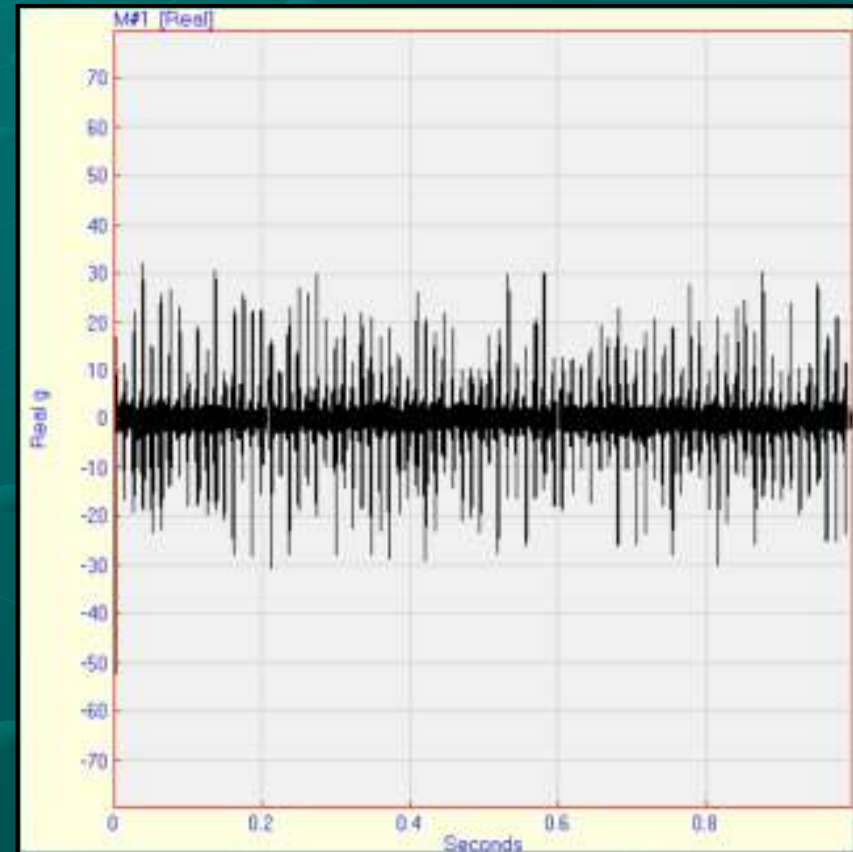
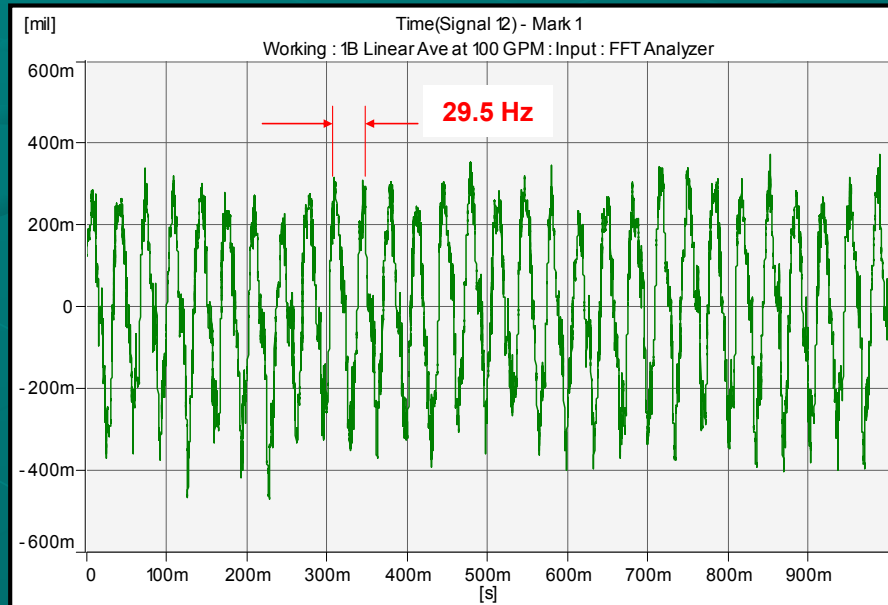
# **Finite Element Analysis (FEA)**

# 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.

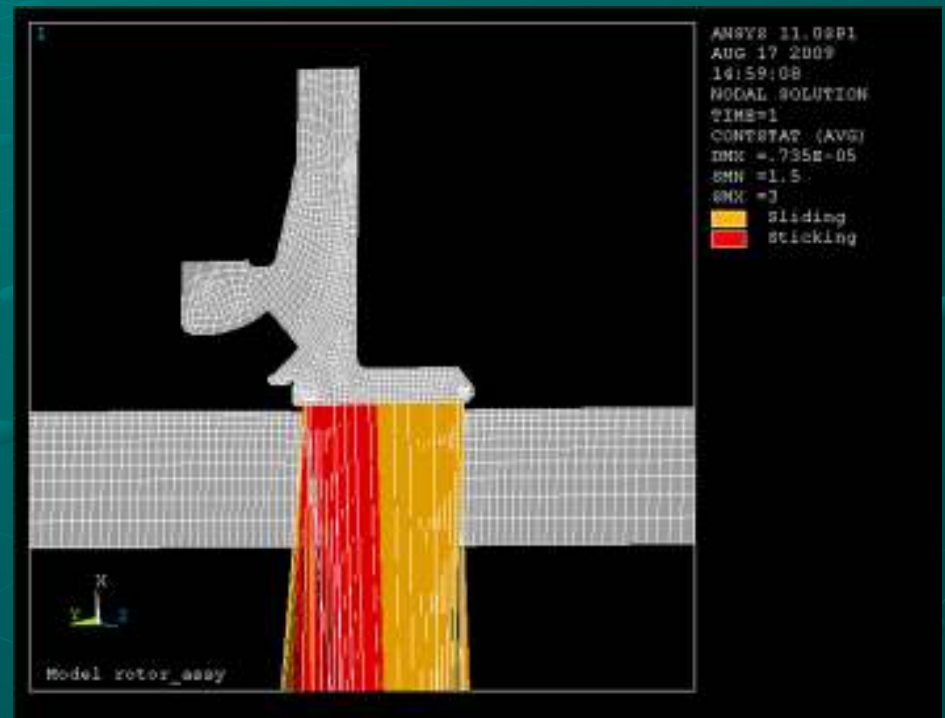
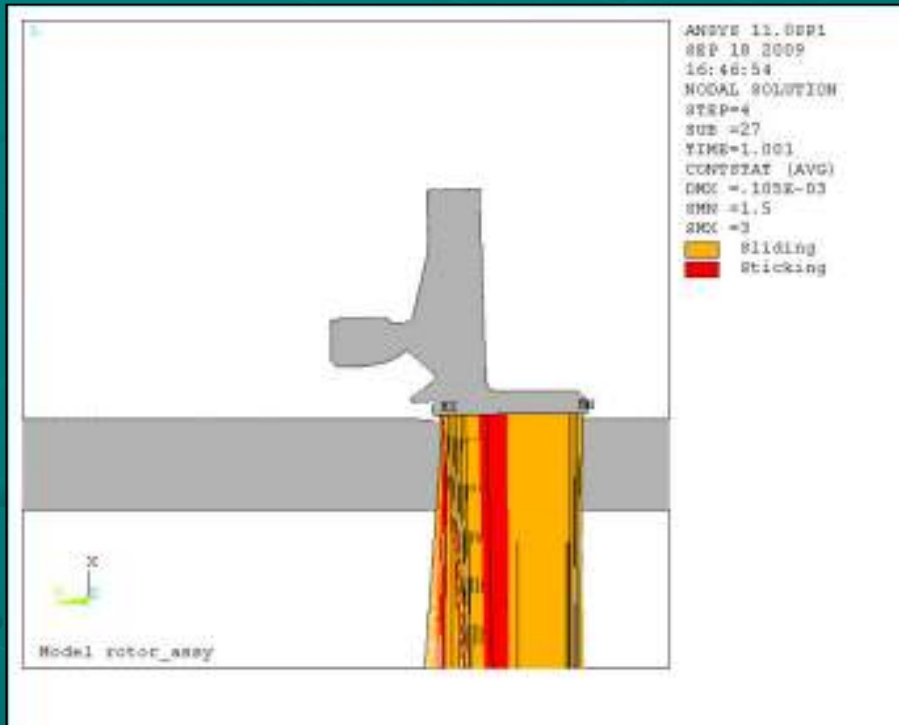
## Load Step 4:

- 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

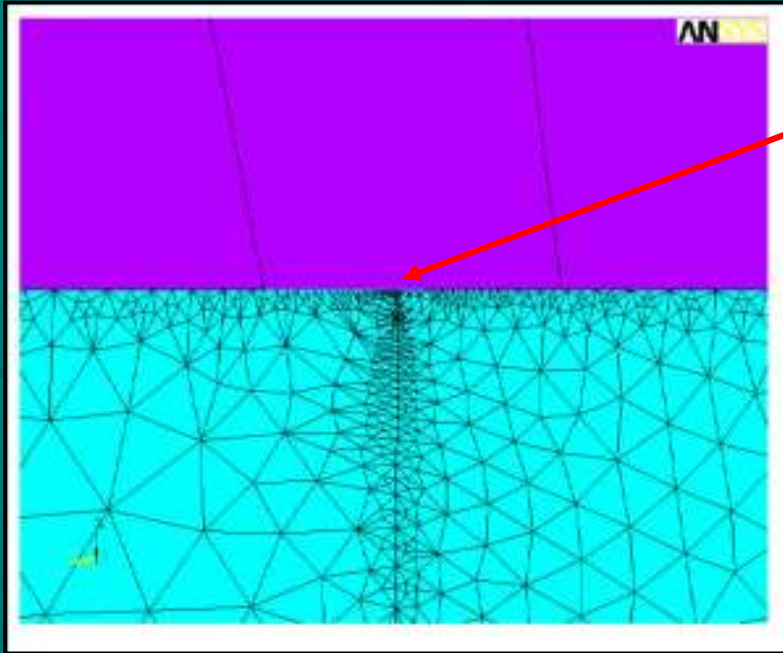


## **FEA-Based Fracture Mechanics Approach**

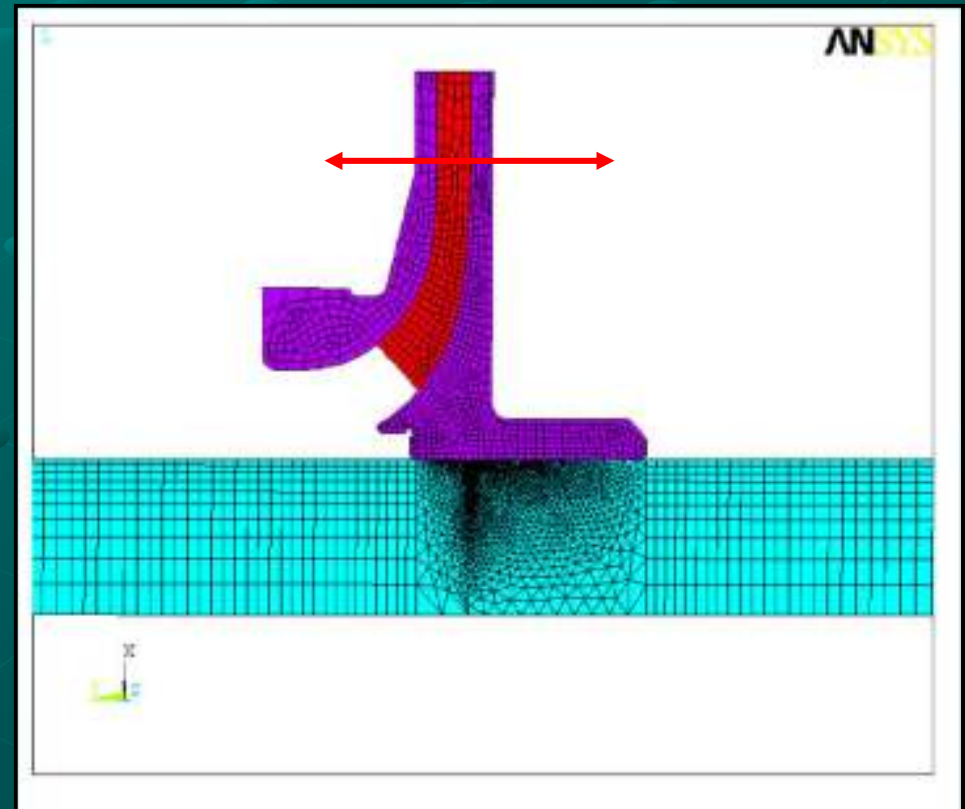
- Assume that relative sliding velocity between the impeller and shaft is equal to the peak axial shaft velocity measured on the end of shaft near the thrust bearing.
- Assume that fretting condition and asperities on shaft surface cause the impeller to axially lock-up against the shaft.
- Model the 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

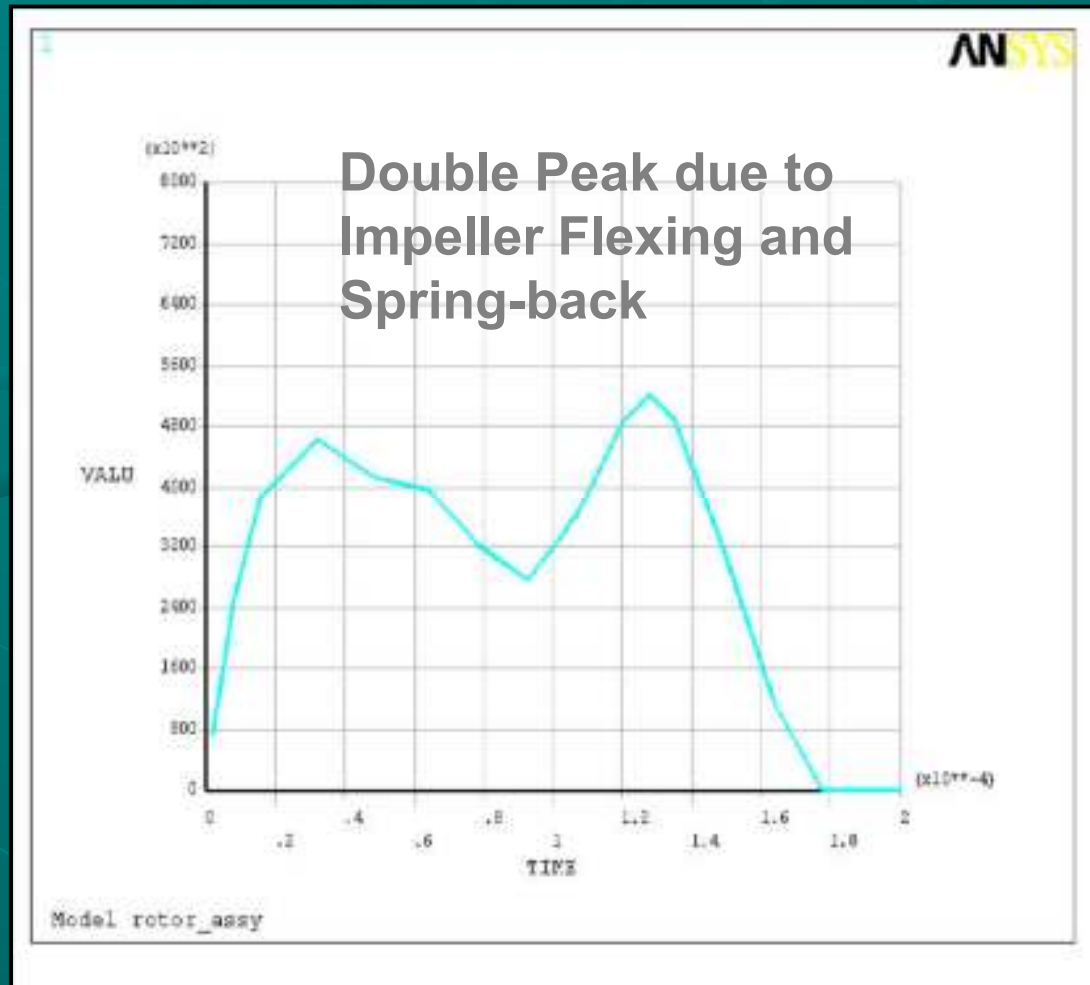


0.005 inch crack explicitly modeled



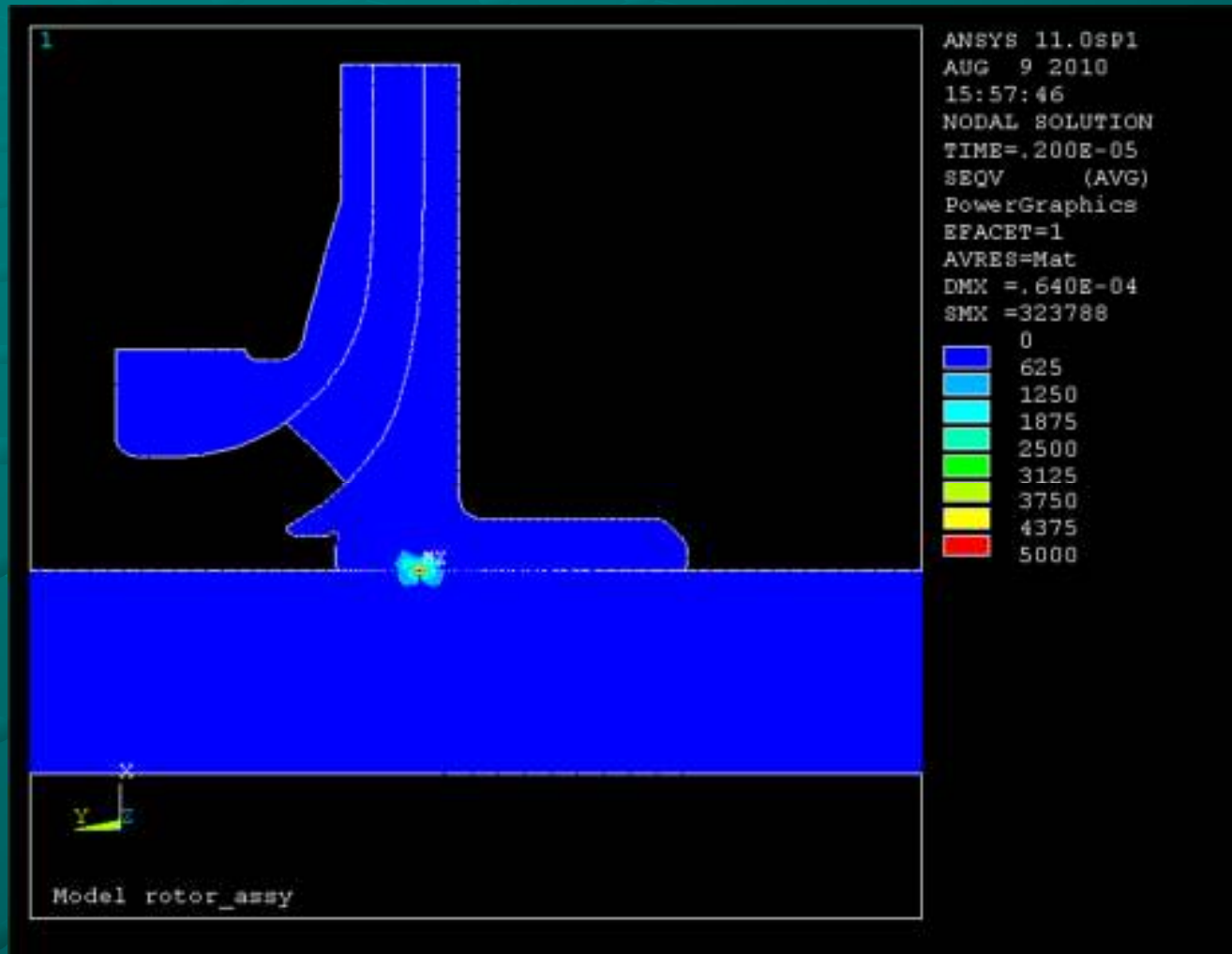
Impact Velocity 31 in/s pk  
(15 g's pk at 29.9Hz)

# FEA-Based Fracture Mechanics Approach



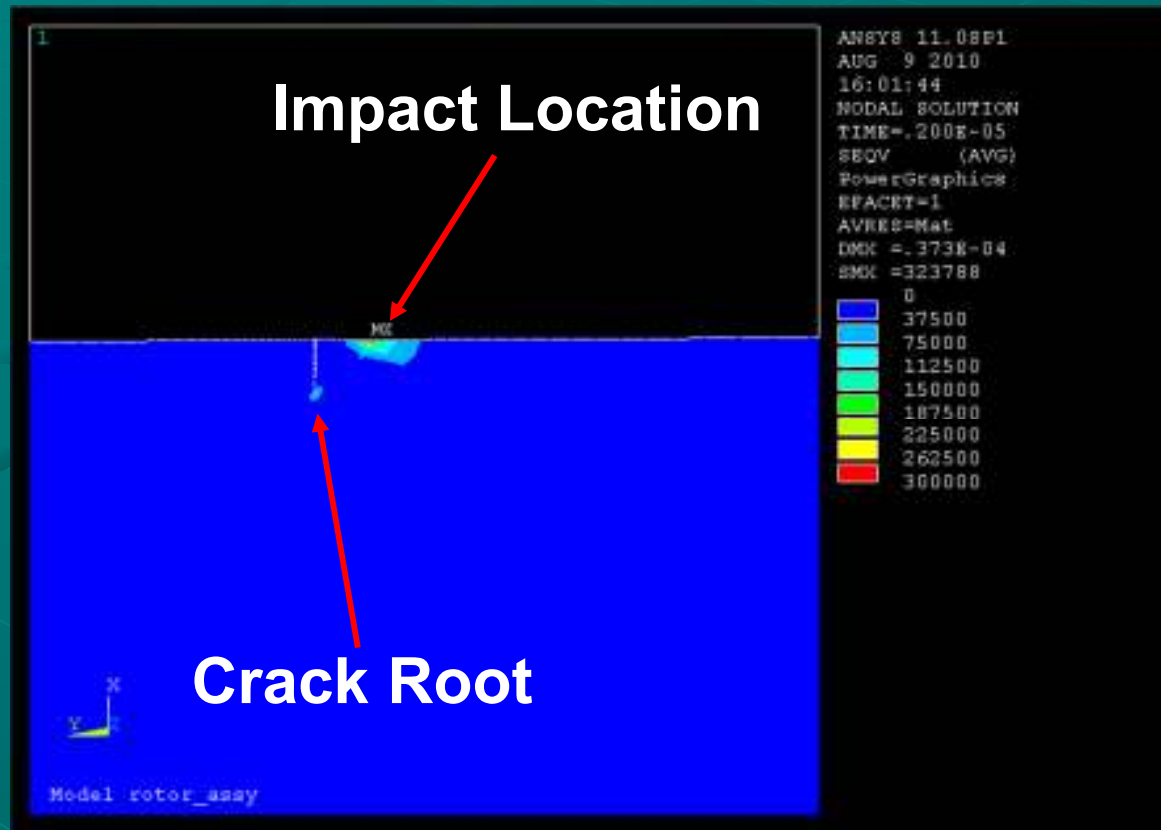
# FEA-Based Fracture Mechanics Approach

von Mises Stress Distribution (psi) Macro View

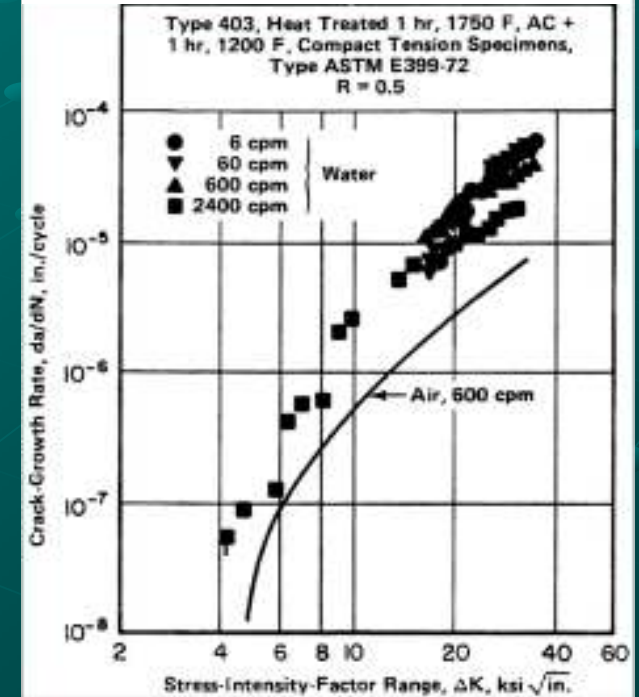


# FEA-Based Fracture Mechanics Approach

von Mises Stress Distribution (psi) In Vicinity of Crack



$$\Delta K = \Delta \sigma_z \sqrt{\pi a} (1.1)$$



ANSYS Calculated Stress Intensity Factor  $\Delta K_I = 16.3 \text{ ksi}\sqrt{\text{in}}$

# 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  $E_x = 0.29e+08$   $\mu_{xy} = 0.30$  at temp = 0.00.

$K_i = 16315. , K_{ii} = 7826.4 , k_{iii} = 0.0000$

## 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.

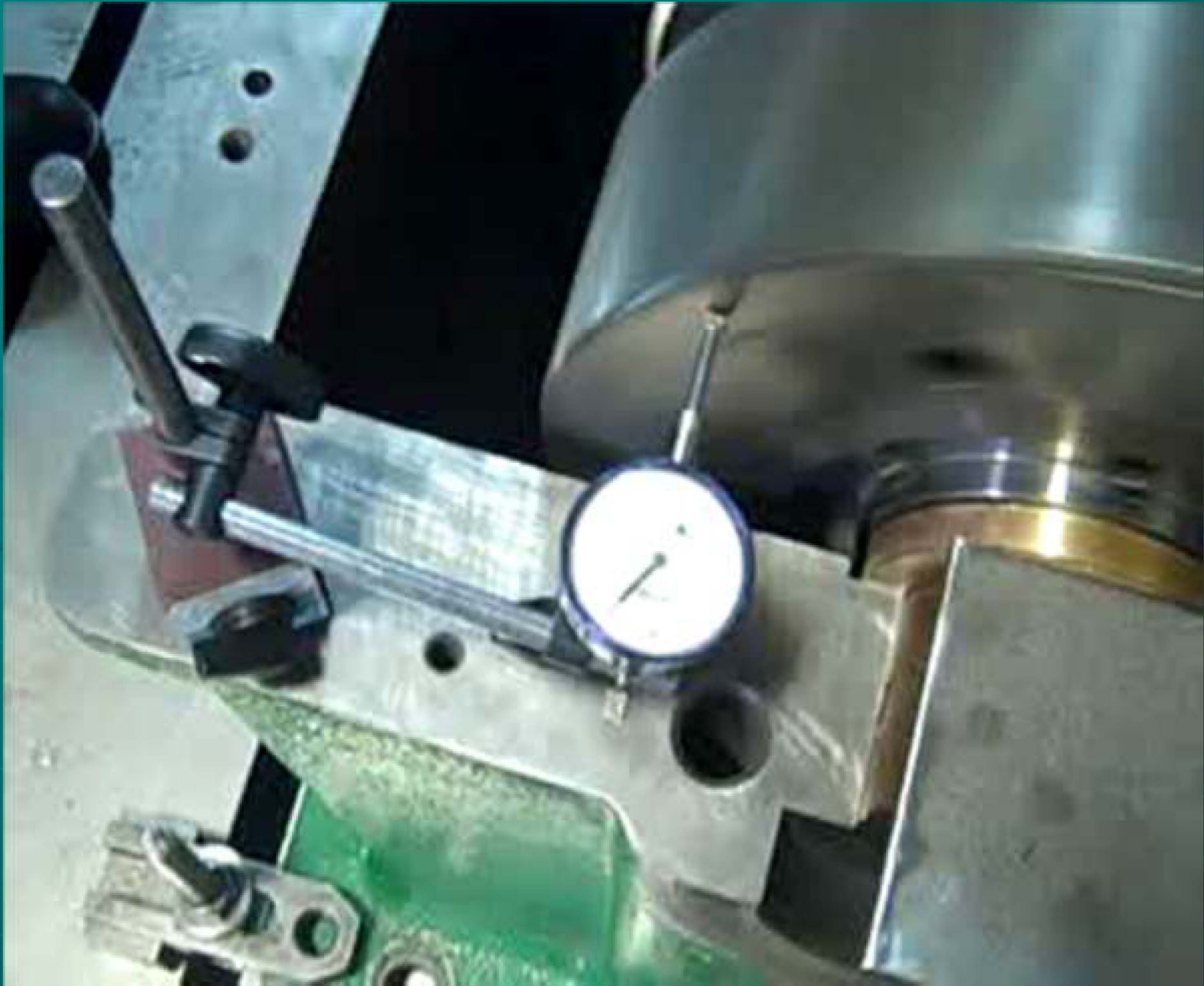
## Conclusions

- 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...?**