Detecting a Hidden Lateral Rotor Natural Frequency in a Sewage Pump

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Presented by:

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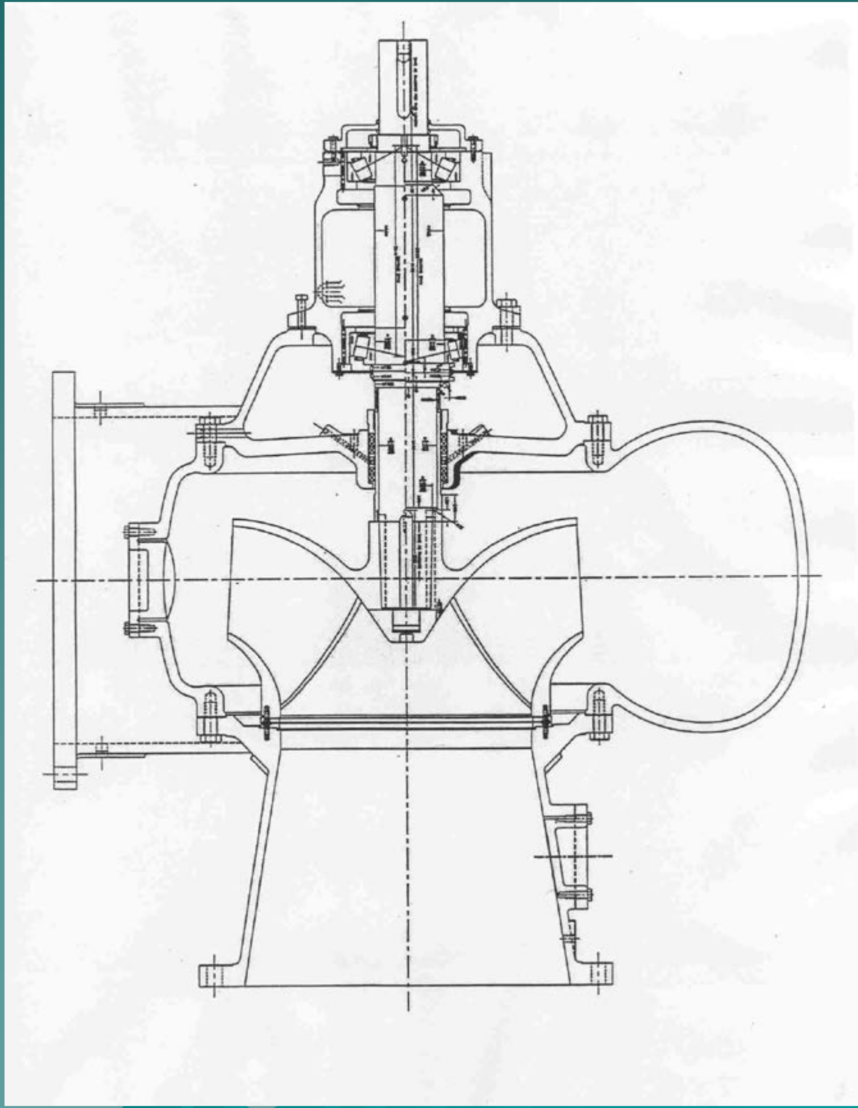
Problem Statement

- A VFD driven sewage pump with a long driveshaft was experiencing excessive vibrations at the bearing tower location at inter-mediate flow and speeds. The majority of the vibration was occurring at 3x running speed. The task was to determine the source of the vibration and propose a suitable solution without costly trial and error field fixes.



Pump Operating Conditions

- The operating speed range of the pump is 382 rpm to 509 rpm (6.4 Hz to 8.5 Hz) and is controlled by a VFD.
- The pump flow rate is 46,000 GPM and develops 56 Ft TDH at 800 HP.
- The pump impeller has three (3) vanes and is made from cast iron. The OD is approximately 40 inches.
- The peak vibration of 0.4 in/s at the bearing tower occurs at 84% flow (428 rpm = 7.1 Hz)

Analysis Methods

- Perform detailed modal and operating deflection shape (ODS) with multi-channel data acquisition equipment.

- Hammer impact modal testing as well as using an accelerometer attached to a stick riding on the shaft during operation were used to determine the natural frequencies of the pump.

Operating Deflected Shape
ME Scope Model

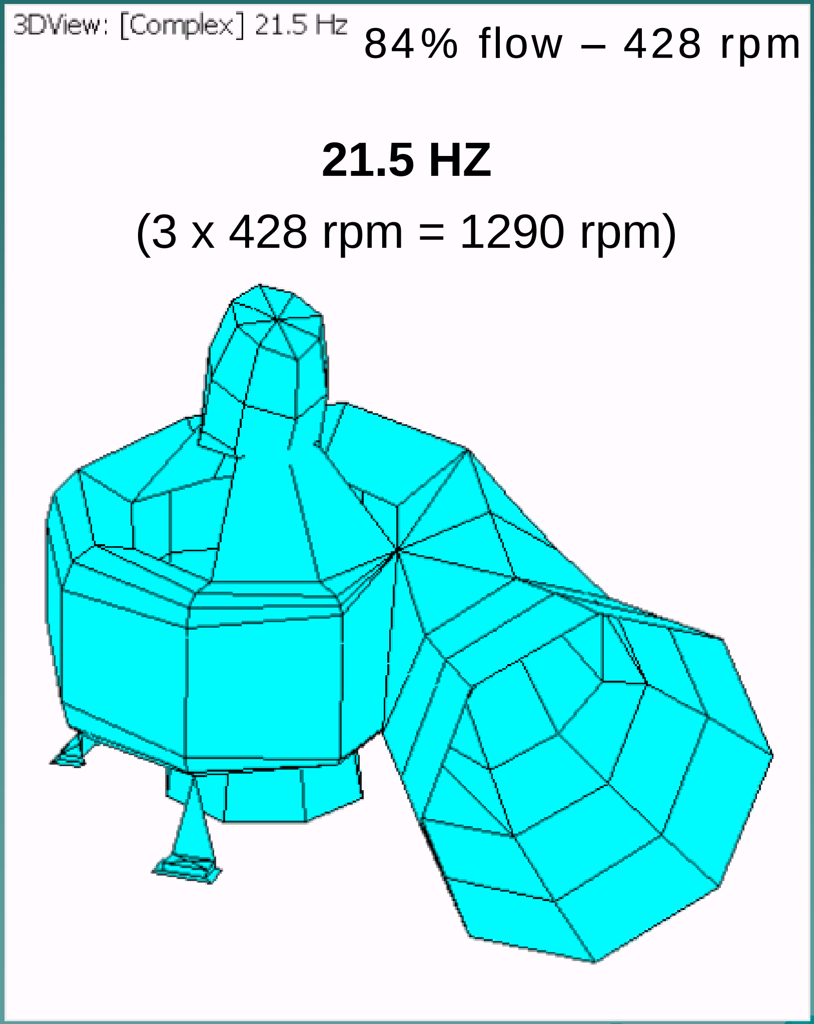


Figure 1

Hammer Impact Modal Response at Bearing Tower

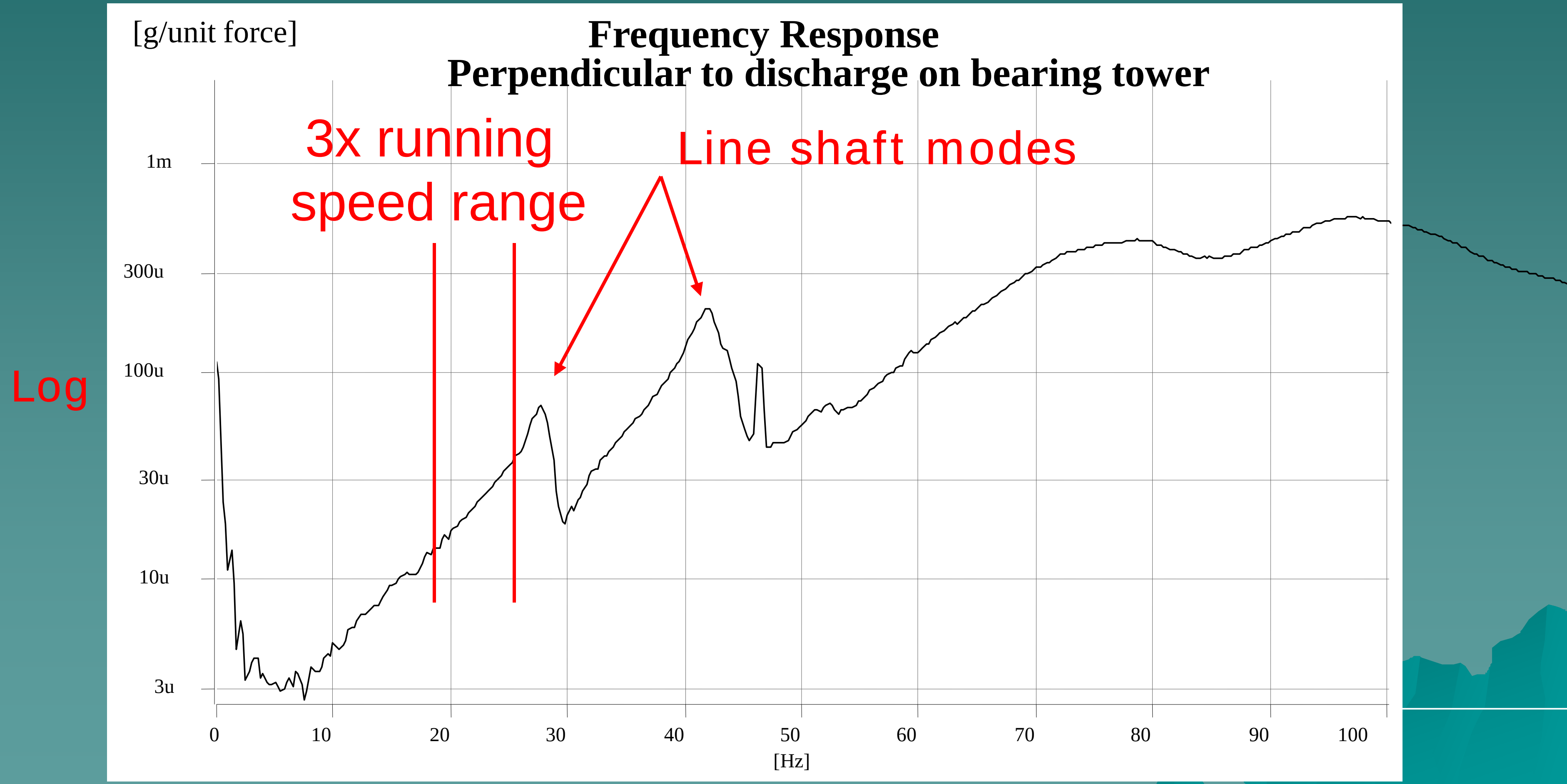


Figure 2

Shaft Stick Frequency Response

Harmonics of running speed

Log

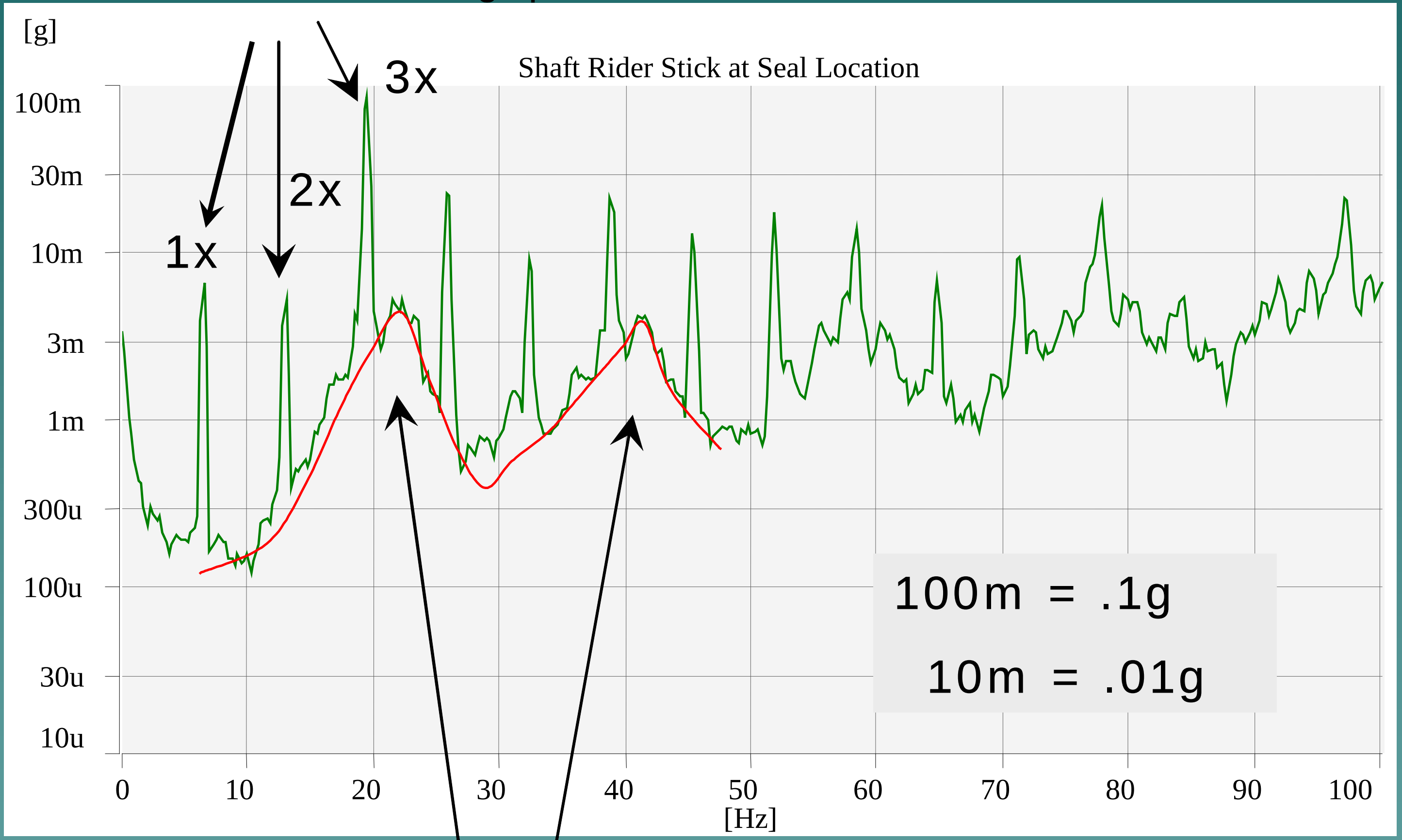


Figure 3

Underlying Natural Frequencies

Preliminary Conclusions

- Peak vibration was being excited by 3x vane pass at 84% flow (428 rpm).
- ODS clearly showed bearing tower motion at 21.5 Hz, but impact testing at bearing tower showed no natural frequency in 3x operating range.
- However, shaft stick measurements indicated a natural frequency at 22 Hz of unknown source.

Preliminary Conclusions

- The shaft natural frequency could not be excited by hammer impact test. The large overhung weight of the impeller (>1 ton) could not be impacted. Therefore, the nature and mode shape of the shaft frequency was not known.

- Thus, finite element analysis was needed to determine if a shaft lateral was indeed present and what would be required to eliminate the vibration problem.

Finite Element Model of Pump

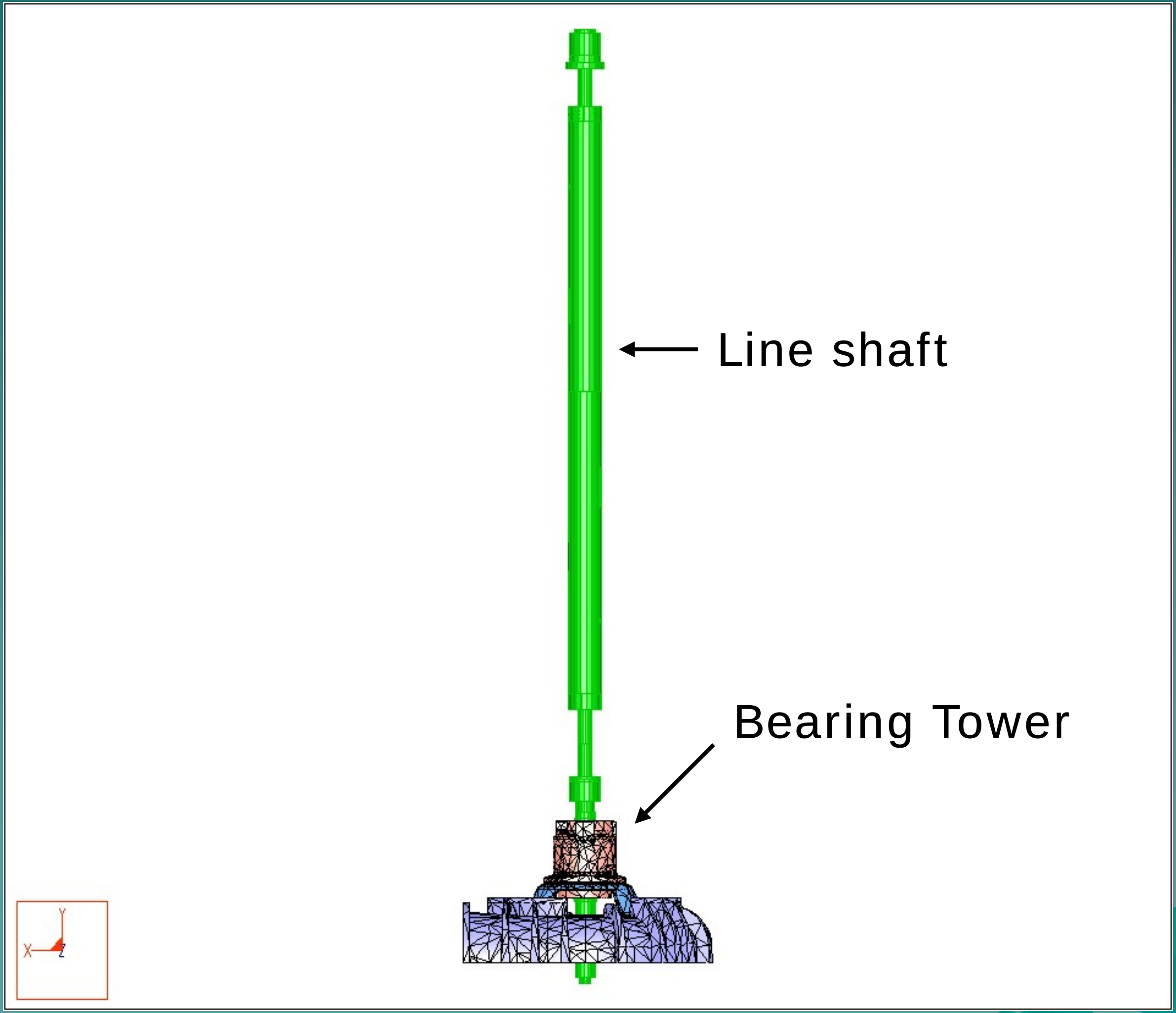


Figure 4

Enlarged View of FEA Model

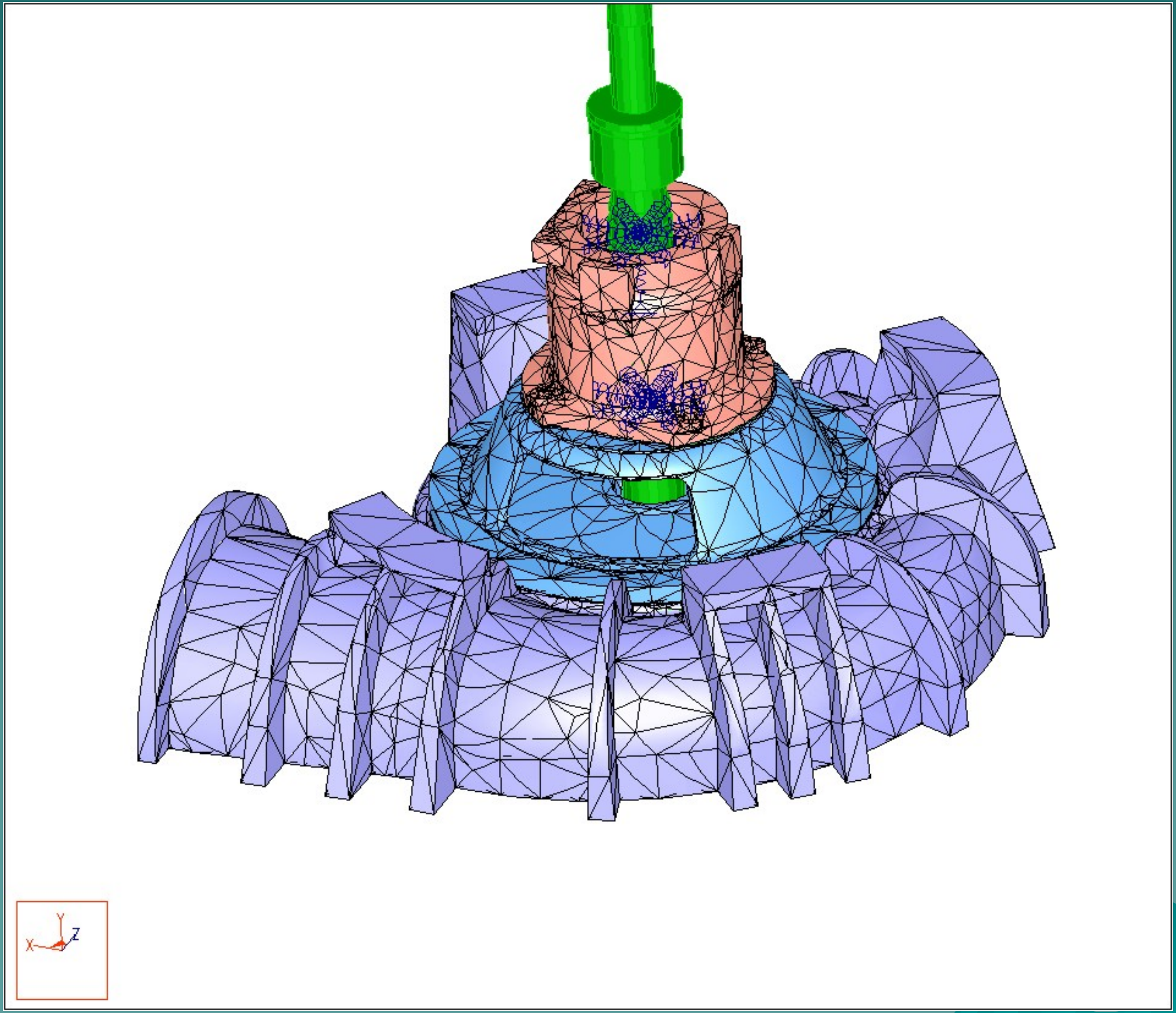


Figure 5

Solids Model Used to Calculate Impeller Properties

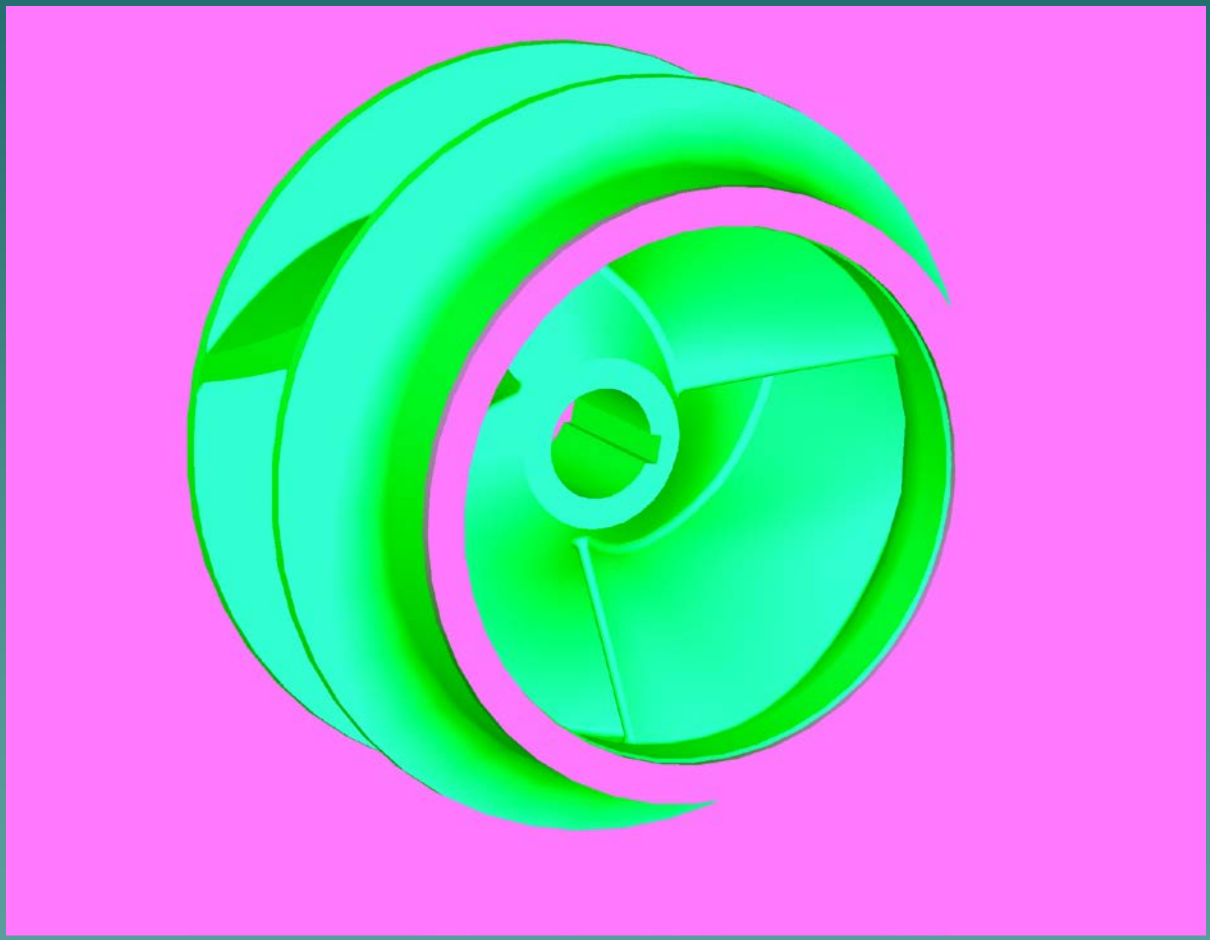
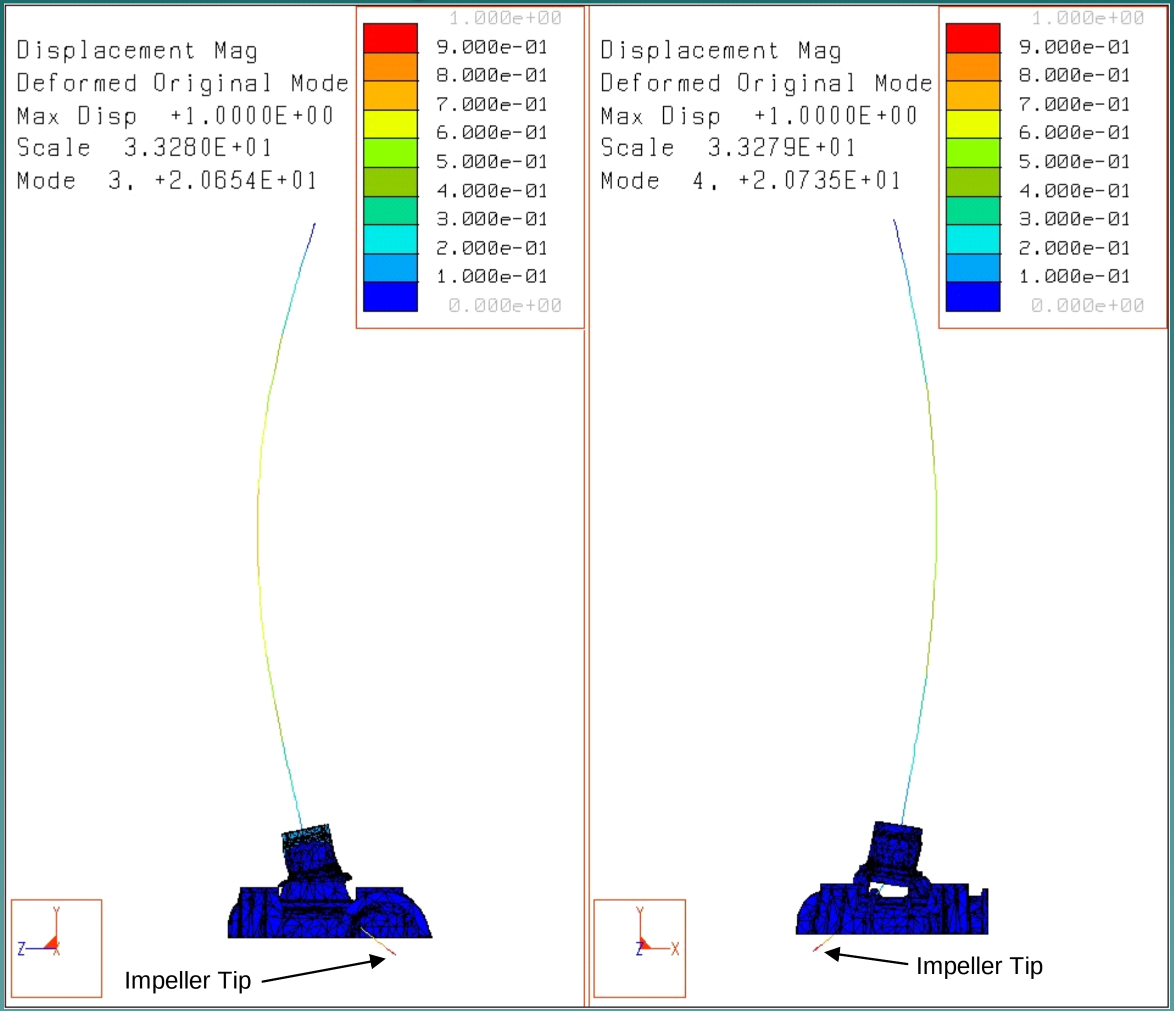


Figure 6

- Metal weight
- Center of gravity
- Transverse moment of inertia
- Entrapped water weight
- External water weight

Analysis Results



20.6 Hz 20.7 Hz

Figure 7

Analysis Results

Frequency – FEA complete pump	Frequency – Test	Description
8.1 Hz	8.5 Hz	1st torsional of rotor assembly
20.6 Hz	21-22 Hz	1st lateral of imp/shaft (parallel to discharge)
20.7 Hz	21-22 Hz	1st lateral of imp/shaft (perpendicular to discharge)
30.6 Hz	28-31 Hz	1st bending of drive shaft (parallel to discharge)
30.7 Hz	28-31 Hz	1st bending of drive shaft (perpendicular to discharge)
46.4 Hz	42-46 Hz	2nd bending of drive shaft (parallel to discharge)
46.9 Hz	42-46 Hz	2nd bending of drive shaft (perpendicular to discharge)

Conclusions

- A lateral rotor natural frequency was being excited by 3x vane pass at 21.0 - 22.0 Hz as was indicated by the underlying shaft stick data and corroborated with the FEA results.
- The rotor critical would have to be raised to 28.0 Hz to provide a 10% margin above the maximum running speed of 509 rpm (25.5 Hz).
- However, it was determined that even if the bearing tower were infinitely stiff, the natural frequency would still fall within the running speed. Therefore, no amount of external bracing would solve this problem.
- Attempts to lower the frequency were discouraged since this would weaken the pump.

Recommendations

- The modulus of the stuffing box and bearing tower were increased by switching from cast iron to ductile iron.
- The rotor shaft diameters were increased substantially to raise the shaft's stiffness.
- The impeller was re-located slightly higher on the shaft which reduced the moment arm thereby increasing the stiffness.
- From the FEA, all of the above were needed to raise the natural frequency to 28.0 Hz and provide a 10% margin above 3x running speed.

Final Results

- After the recommended fixes were implemented, the vibration levels decreased from 0.4 in/s to less than 0.1 in/s at the bearing tower.

- Follow up field test showed that the frequencies that were at 21-22 Hz were gone, and shaft stick measurements showed small peaks around 28 Hz just as predicted by the analysis.

Final Results

Log

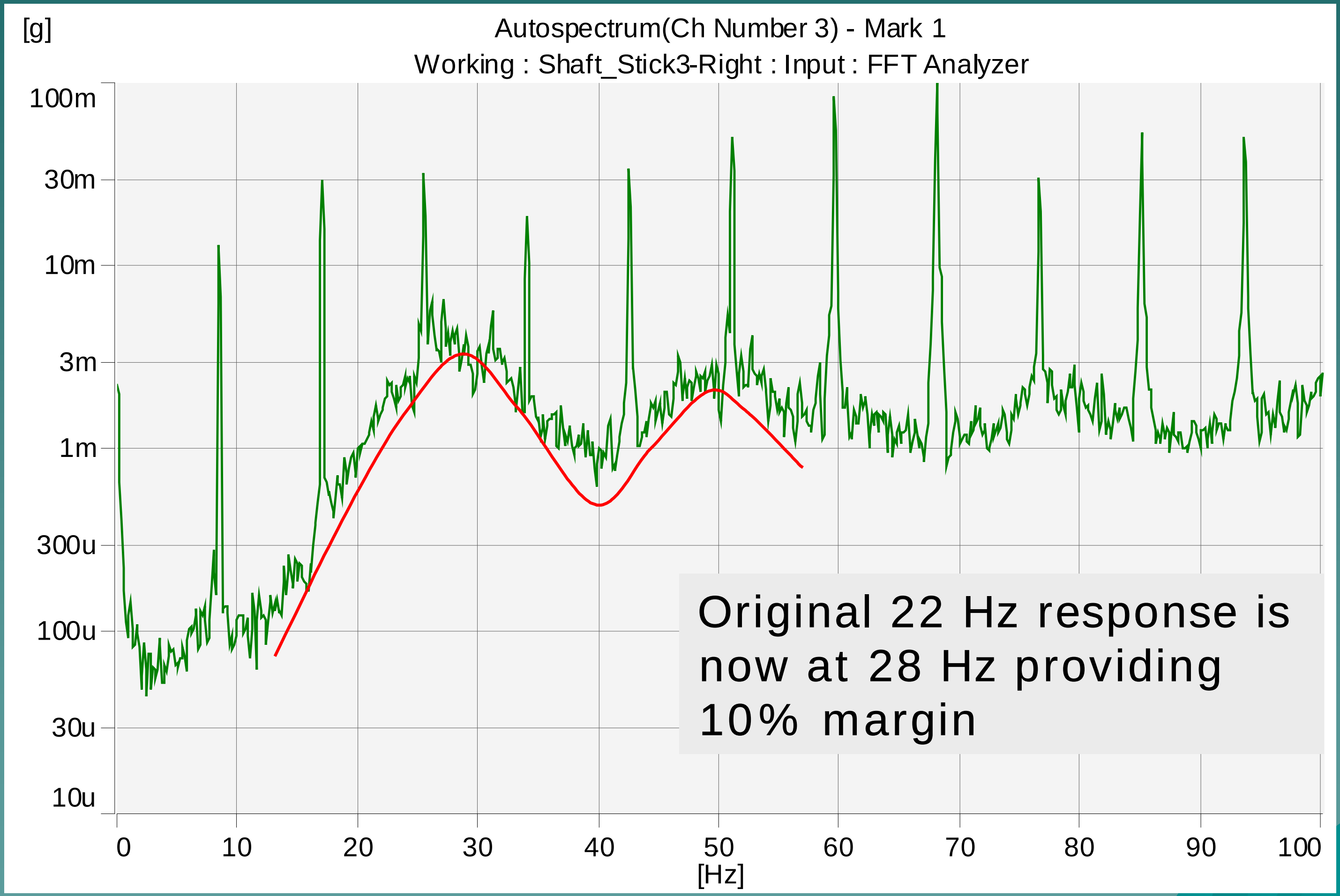


Figure 8