WORLD-CLASS OUTSTANDING INTERNATIONAL

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AXIAL VIBRATION FOR A SYNCHRONOUS MOTOR, GEARBOX, COMPRESSOR TRAIN

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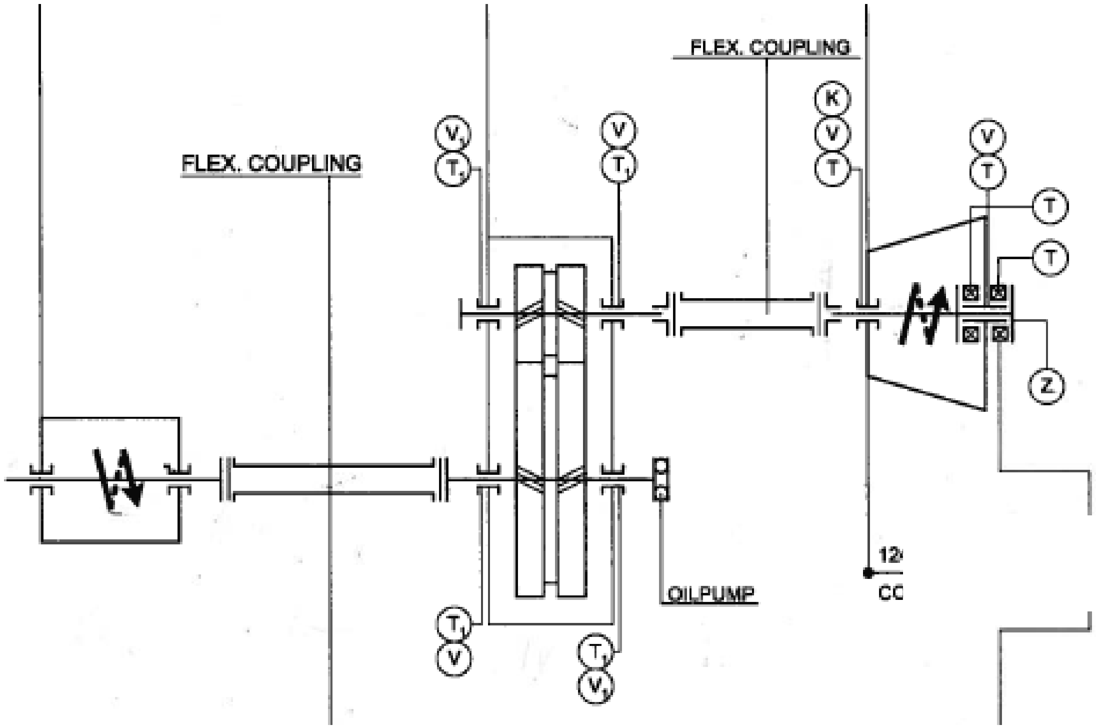
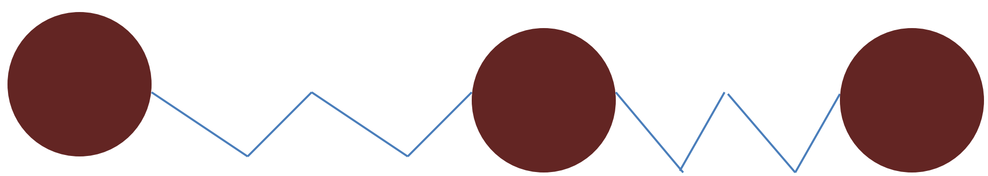
Problem Statement

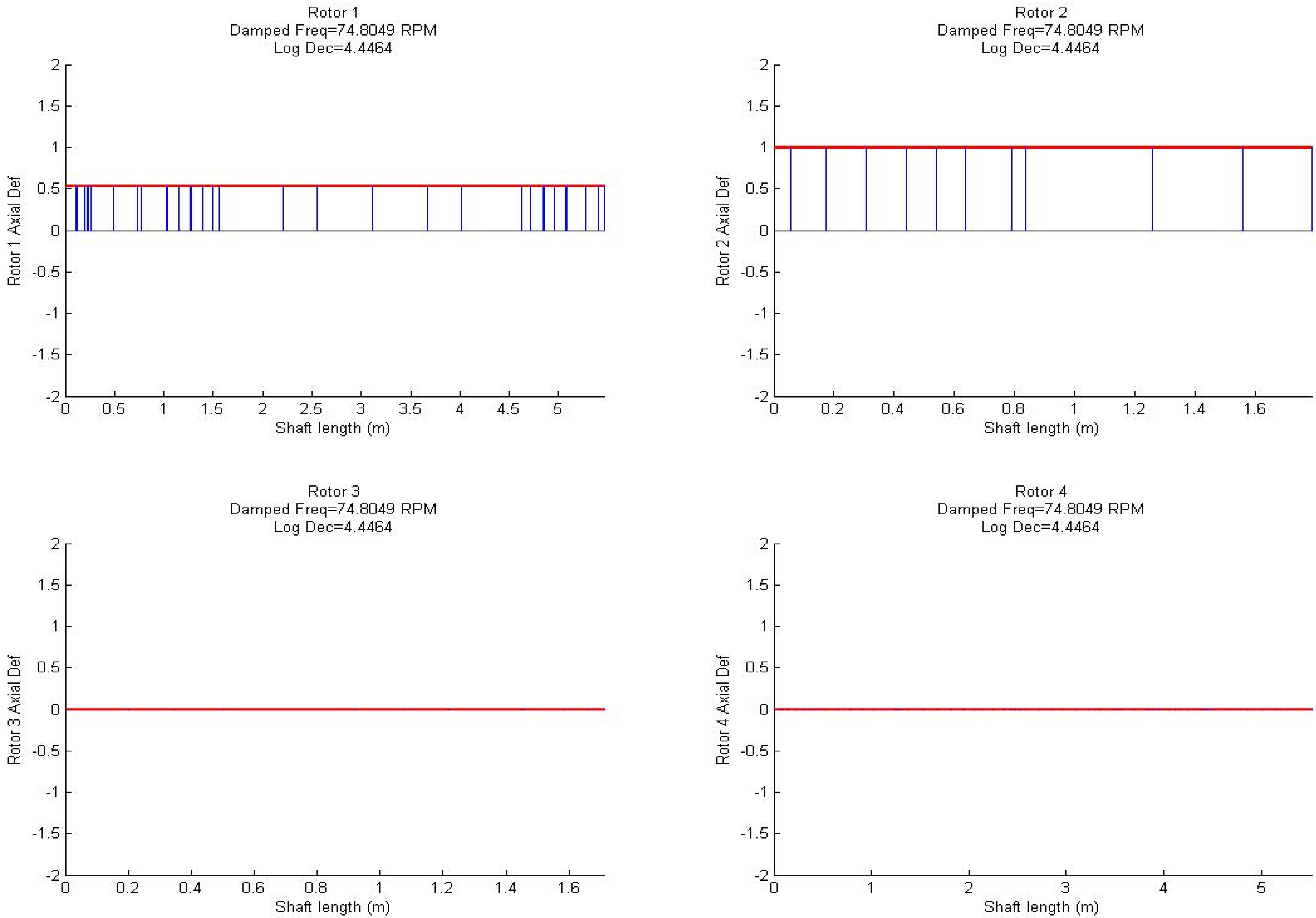
- Synchronous Motor (20KW), Gearbox, Inlet compressor
- Train installed 2007 with 2mm axial vibration mainly on the motor (low speed) side.
- Amplitude @ <u>2 mm</u> with a frequency of <u>2.8 Hz</u>(170/min). Gearbox running noisily.

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.