Manifestations of Thermal Imbalance in Integrally Geared Compressors

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INTRODUCTION

- Case studies showing three distinct manifestations of thermally induced bow/imbalance in integrally geared machines:
  1. Spiral vibration with continuously varying amplitude.
  2. Hysteresis phenomenon.
Case I- Problem

- Integrally geared compressor with total of 4 sister units operating in the field
- All the machines trip during commissioning of the compressor (when started without specific process control procedures) due to continuously increasing 1X vibrations.
- Each machine requires a unique commissioning sequence to successfully reach operating conditions. (i.e. changing process flow, oil inlet temperature etc.)
- Once at operating speed, operating oil inlet temperature and operating load conditions, the machines will operate indefinitely.
- Controlled factory test for design duplicate number 5 could diagnose the issue and a corresponding solution found.
CASE I - Problem

- The pinion in question is a double overhang pinion (stg3,4) operating subcritical at 14500 with first critical at 31000 rpm. 4.5” L/D=1 brg.
- Controlled test showed a phase and time varying synchronous amplitude
- The spiral vibration pattern observed on the right is a hallmark of Morton effect

- The oil temperature was gradually increased and it was found the thermal instability went away beyond 107F. (Blue line in the figure denotes oil inlet temp).

Stg 3 and Stg 4 polar plot as observed on test stand showing spiral vibrations (1.6 mils scale)
CASE I - Analytical Predictions

- Texas A&M’s Morton effect code by Prof. Palazzolo et al. used for the analysis
- Transient Morton effect module predicted presence of Morton effect
- Temperature dependence wasn’t seen but load dependence was observed with Morton effect becoming evident at partial load cases.

Spiral vibration prediction confirms presence of Morton effect.
CASE I- Solution

- Literature suggested
  - Reducing overhang - unfeasible due to requirements of design duplicates
  - Heat barrier sleeve - unfeasible
  - Wider clearance brg – Attempted in field did not work
  - Shorter L/D bearing to create a more elliptical orbit
- A shorter pad width (L/D=0.72) bearing was used.
- The vibration signature after the design change (shown in the figure on the right) did not show cyclic vibration amplitude in the same temperature range and was acceptable. The vibration stayed constant to within 0.1 mils across the similar temperature range.
CASE I- Conclusion

• For a cold start (oil around 80°F) the machine would experience constantly increasing vibrations. Startup to operating conditions were achieved by varying the oil temperature and load quickly to keep the vibrations below trip level.

• The Operating oil inlet temperature of the machines was greater than 110°F. At these operating oil inlet temperatures the Morton effect is non-existent and at these conditions machine was found to work indefinitely.

• Machine startup issues were alleviated by using bearings with shorter pads (L/D=0.72).
CASE II- Problem

- Operating speed of this double overhung Pinion is 11800rpm. Machine is operating above its critical speeds.
- At operating speed the synchronous vibration amplitude for the high pressure stage was seen to increase from 0.6 mils to 1.2 mils with about a 40 degree change in phase. No change in vibration to low pressure stage end.
- The vibration does not oscillate but instead stays constant at this elevated level.
- Changing speed to both over and under operating conditions reduces the vibration level creating the hysteresis loop.
- This hysteresis loop is a remnant of thermal bowing. This also shows that thermal bowing happens within a narrow range of speeds, since the effects starts reducing at off-design speeds.
CASE II - Solution

- Once thermal equilibrium and operating lube oil supply temperatures were achieved the 1X vibrations were found to stay consistent at 0.65 mils. (circled on the plot)
- The machine vibrations were acceptable as is and did not need any design change.
CASE II- Conclusion

• Unlike Case I the vibration did not cycle.
• Hysteresis curve confirms presence of thermal bowing under certain temperature ranges.
• Changes to vibration with oil flow and oil temperature confirm the phenomenon originates in the bearing.
Case III - Problem

• The pinion in question operates at 10022rpm with predicted critical at 7500 rpm. High 1X vibration were not responding to field balance.
• Looking at trend data it was observed 1X vibration increased with time until it reached an elevated maximum with about 40 degree change in phase.
• When taken below design speed, the thermal bowing started to increase
• Vibration would initially reduce when taken above design speed but then went back up
• Unlike case 2 there was a broader range of speeds where the thermal bowing was still active
• Similar to case 2 only a 40 degree phase shift was seen and the pinion was seen to stay at a consistent elevated vibrations level at a given speed
Case III- Solution

• Prior success with short pad bearings to reduce thermal imbalance effect led us to attempt similar solution for this issue.
  • L/D = 0.7 considered appropriate based on the bearing load and speeds

• Using the new bearing, thermal unbalance was significantly reduced with synchronous vibration, staying consistent at 0.6 mils under operating temperature range

Vibrations stay below 0.6 with the new bearing design
Case III- Conclusion

• Unlike case II the original rotor bearing system was sensitive to thermal unbalances for a wider range of speeds
• Similar to case I, a short pad L/D=0.7 bearing found useful in reducing the thermal sensitivity of the rotor bearing system
Lessons Learned

• Rather than looking at synchronous vibration at an instant of time, a synchronous time trend for amplitude/phase change should be analyzed before high speed balancing to catch any underlying thermal unbalance phenomenon.

• Short pad bearings can be useful for getting rid of thermal bowing of shafts
Questions?