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Special Considerations for Electric Motors Driving Reciprocating Compressors

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Abstract

In the natural gas transmission, refining, and chemical industries, motors are commonly used to drive reciprocating compressors. However, the excitation produced by reciprocating compressor can be much greater than from centrifugal equipment. Therefore, motors driving reciprocating compressors need to have a more robust design to be reliable.

This paper presents several case studies on the following subjects:

- How Motor Mounting Can Affect Vibration
- Axial Vibration of Couplings
- Natural Frequencies of Motor Bearing End Bell Support
- Motor Cooling Fan Failures
- Torsional Failure of VFD Motor Shaft
- Excessive Motor Current Pulsation
- Motor Electromagnetic Effects when using Torsionally Soft Couplings

Some problems with motor driven reciprocating compressor systems may not be predicted in the design stage and could appear after commissioning the unit. Also, personnel normally dealing with electrical equipment driving centrifugal pumps or fans may not be familiar with the vibration characteristics of reciprocating compressors.

I. Introduction

There are many books and technical papers available on motors. For those just starting to learn about motors, PCIC / IEEE has a helpful five-part series of papers [1-5]. A Motor Primer (Part 1) presents theory and application information for induction motors typically used at a refinery, API Standard 541 [6]. Part 2 continues with some discussion on vibration due to rotor eccentricity and lateral critical speeds. Part 3 follows with how to size a motor, discussion on bearings, and what causes motor bars to break. Part 4 covers hydrodynamic journal bearings and how the stiffness and damping properties of the oil film affect the lateral critical speed and unbalance response of the motor rotor. Finally, Part 5 deals with the topics of torsional vibration, how unbalance and vibration affect motor life, and evaluating motor efficiency. These and similar papers concentrate more on the motor itself, and not how the driven equipment might affect the motor reliability once installed.

A literature search found another helpful paper [7] that covers common sources of motor vibration, flexible or resonant motor base, ways to measure vibration, troubleshooting procedures, diagnostic chart, and typical vibration alarm levels. However, these references do not specifically address reciprocating compressors and the effect higher shaking forces and alternating torque can have on the motor. Therefore, this GMC paper tries to add several considerations for a motor driving a reciprocating compressor.

*Note that only the case study dealing with the motor cooling fans in Section V was from Western Midstream.

II. Anchor Bolts

This first case study shows the importance of the motor being properly anchored. The induction motor was rated for 5,600 HP at 894 RPM and was driving a 6-throw, 3-stage compressor in CO_2 service. The motor is "soft started," and then transferred to constant speed. A soft start refers to the reduced motor acceleration and inrush current compared to an across-the-line start. Figure 1 shows photos of the unit.

Figure 1. a) Motor, b) Compressor

A baseline vibration check was requested on the motor after being put back into service following bearing repairs. During loaded operation of the compressor, the control panel indicated a vibration alarm on the motor. The vibration primarily occurred at 1× running speed, which can be related to imbalance [8] of the motor rotor, coupling, or shaking forces from the compressor.

To help diagnose the problem, vibration readings were taken at various elevations on the motor, pedestal, and skid. A profile plot was made that indicated possible rocking motion of the motor frame. Figure 2 shows that the centerline of the drive end (DE) and non-drive end (NDE) bearings had similar vibration levels in the horizontal direction, whereas the motor feet and top of the pedestal had the highest vibration in the vertical direction.

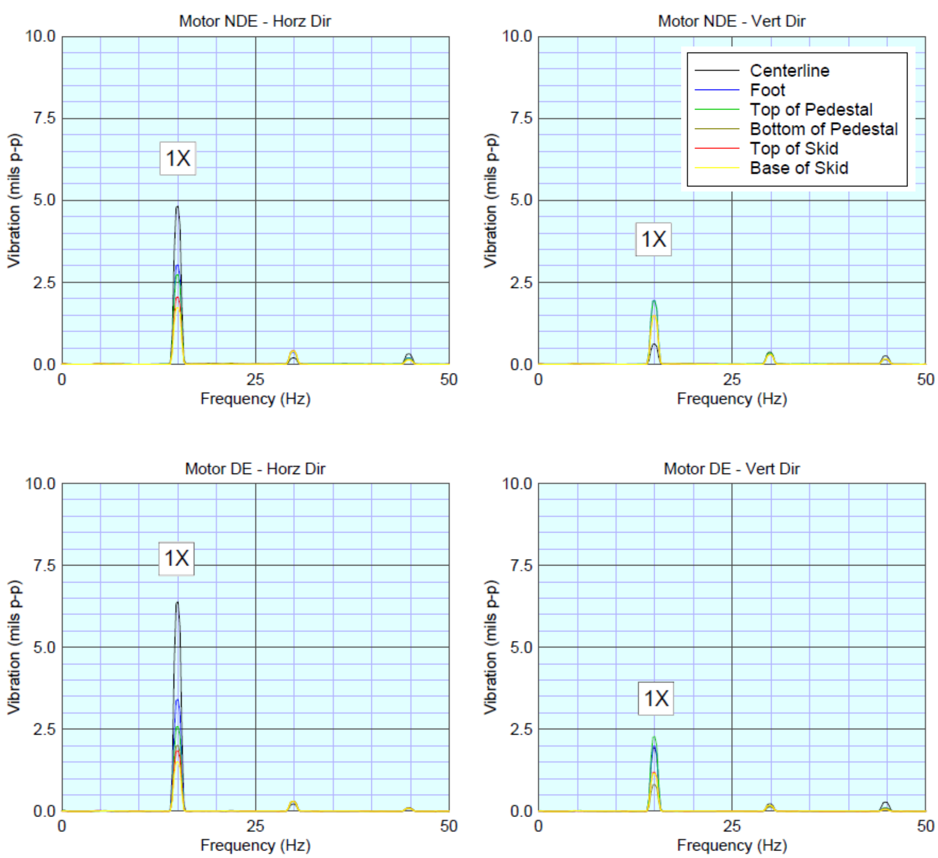


Figure 2. Vibration Measurements Taken on Motor, Pedestal, and Skid

To visualize the motion, operating deflection shape (ODS) data were obtained. The overall rocking motion of the motor frame was confirmed at 1× running speed, as seen in the exaggerated shape in Figure 3. There was also differential movement observed between the motor feet and pedestal support.

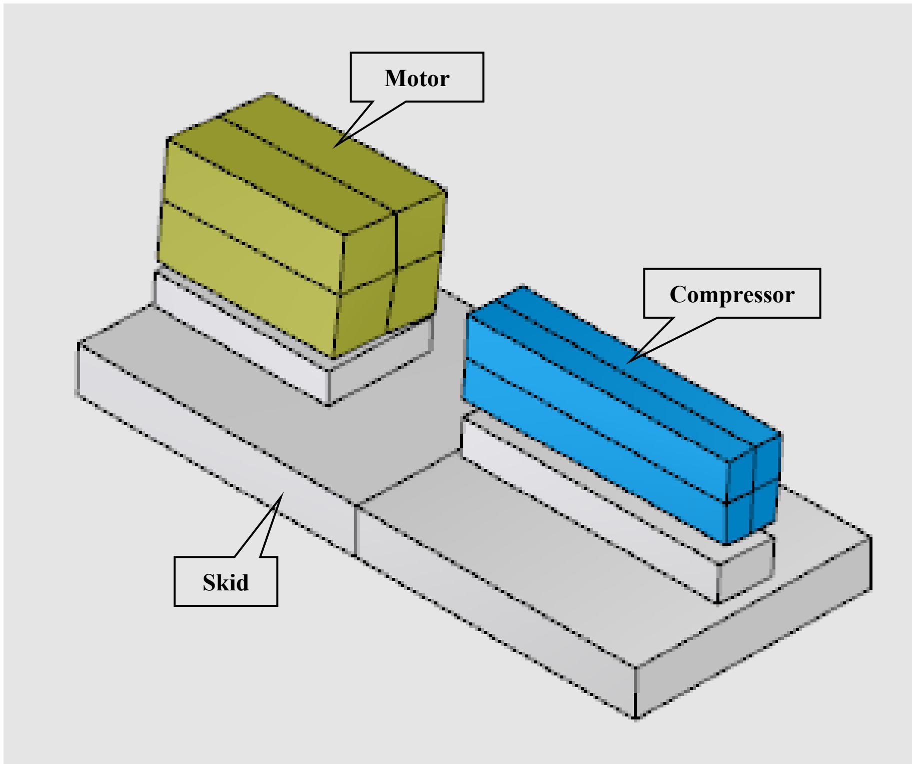


Figure 3. Operating Deflection Shape

ODS is a valuable tool for visualizing vibration associated with reciprocating machinery, piping, and structures [9]. The results from an ODS analysis can help identify areas of high vibration and help determine the best course of action. Possible cause(s) of equipment vibration could be bolted joint looseness, clamp or support looseness, insufficient foundation stiffness, etc.

To determine if the motor rotor was out of balance, a solo run was recommended. For the motor solo run, the center piece was removed from the coupling and the motor vibration was low. For example, motor bearing vibration on the drive end was 6.5 mils p-p at 1× running speed during the coupled run, but less than 1 mil p-p during the motor solo run. Therefore, satisfactory motor vibration was verified without the compressor.

Next, the unit was checked for soft foot on the motor and alignment between the motor and compressor was verified. It was thought that some unbalance could be caused by the coupling components. As a trial, the coupling was reassembled with the center member clocked 90 deg. This seemed to reduce the motor vibration slightly, but it was still near the alarm level.

Upon further inspection, it was observed that the anchor bolts for the motor were pulling the top washer through the slot and had reduced torque (see Figure 4). This could allow the differential motion measured with the profile plot and ODS animation.

Figure 4. Anchor Bolt at Motor Base

Thicker washers were fabricated in the field and the torque was increased on the anchor bolts. After these modifications, the indicated vibration on the control panel was reduced to approximately 0.16 – 0.18 ips, and the motor was no longer in alarm condition. Comparison vibration plots measured at the motor centerline are shown in Figure 5.

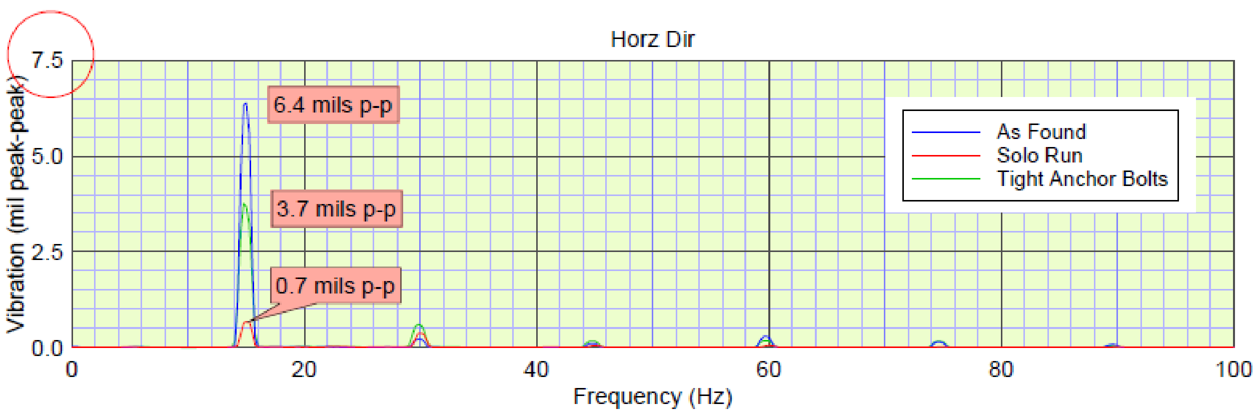


Figure 5. Comparison Vibration Plot

The motor vibration was shown to be sensitive to the mounting. If motor vibration increased in the future, it was recommended to improve the anchor bolts by using Supernuts, torque tubes for bolt stretch and to maintain preload, and adding gussets to the pedestal under the motor feet. See example in Figure 6. Bolted joints are covered in the book by Bickford [10] and a previous GMC paper [11].

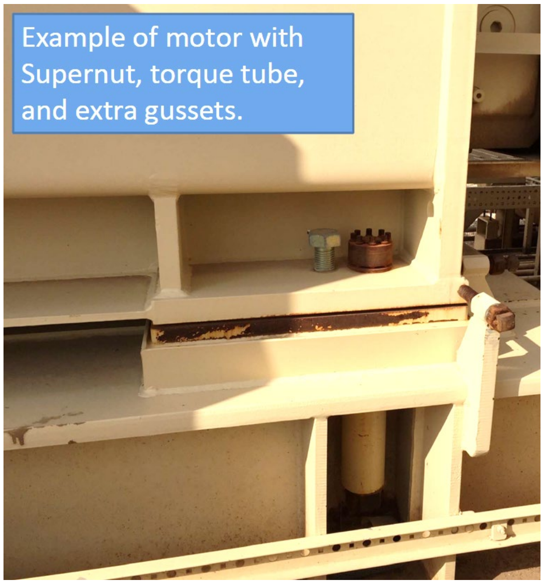


Figure 6. Example Anchor Bolt with Supernut

If shimming is required, it is recommended to use the proper size and thickness instead of a stack of thin pieces that could shift. These modifications would help reduce differential motion at the motor feet and increase stiffness of the support pedestal under the motor. Importance of proper tie-down shown is in reference [12]. Other helpful GMC papers cover grout and foundations [13,14,15].

III. Axial Vibration

Coupling disc pack failures occurred on the induction motors of two identical residue compressor units at a gas plant. The failures occurred within approximately 4000 hours of operation.

The motor rating was 9,300 HP at 717 RPM. The coupling used a keyless hub on the motor side and there was a flywheel adaptor on the compressor side. The reciprocating compressor was a 6-throw unit in 2-stage service. Photographs of one of the units are shown in Figure 7.

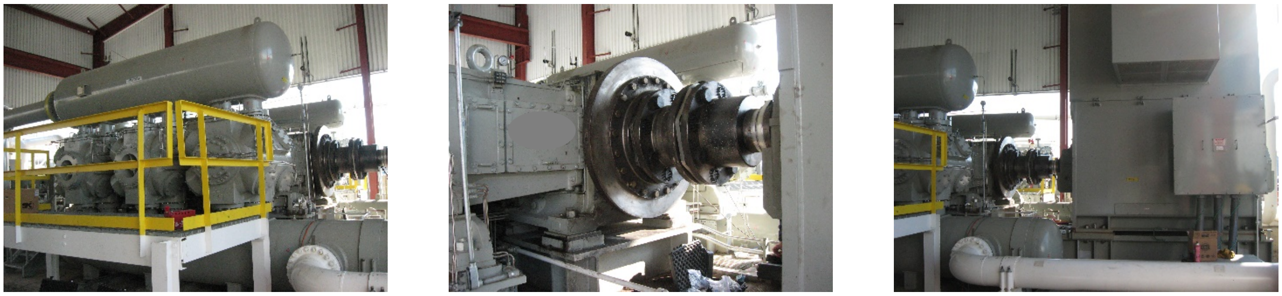


Figure 7. a) Compressor, b) Coupling, c) Motor

The initial hypothesis was that torsional vibration could be damaging the couplings. A torsional vibration analysis (TVA) was performed by a third party in the design stage, but no field tests had been performed to verify the torsional natural frequencies (TNFs). There was also concern since these were the first two couplings produced with such large disc packs.

Field testing included measuring torque in the motor shaft, torsional oscillation of the compressor crankshaft, and vibration of the motor bearings and compressor frame. Reference [16] provides details on the specialized instrumentation and methods needed for torsional field measurements. During the initial test run, axial motion of the coupling and motor rotor was observed so proximity probes were also added.

Test data were acquired at all 18 compressor load steps (LS 1 = full load, LS 18 = min load). Note that the compressor load steps were initially found to be incorrectly programmed (pockets open vs closed) during the first load sweep. The program was corrected for the remaining runs.

In summary, no torsional problems were found. Instead, high axial vibration of the coupling and motor rotor was observed and was believed to be causing the disc pack failures. The axial vibration was highest at certain compressor load steps. The motor had a single cooling fan mounted on the shaft, which could create a thrust force that interacts with the magnetic centering force and axial stiffness of the coupling. Axial alignment and installation procedures may have also been contributing factors.

Torsional Test

Torsional vibration was measured two ways: 1) strain gage telemetry system on the motor shaft and 2) encoder on the auxiliary end of the compressor crankshaft. The motor soft-starter takes approximately 15 - 16 seconds to reach full speed.

Based on data collected during startups and coastdowns, the first TNF was determined to be 52.5 – 54.0 Hz. The separation margin (SM) was approximately 9% from 4× running speed. The second TNF was approximately 88.5 – 89.0 Hz, which had a SM of approximately 5% from 7× running speed.

When the induction motor is unloaded, operation will be near synchronous speed of 720 RPM. As load compressor increases, the speed drops closer to 717 RPM. This means 4× running speed can vary slightly from 47.8 – 48.0 Hz and the 7× harmonic can vary from approximately 83.65 – 84.0 Hz. The separation margins were less than the API recommendation of 10%. However, these separation margins could still be considered acceptable based on measured torsional response (alternating torque / stress levels and oscillations).

The measured torque was well below allowable levels for all load steps. The coupling manufacturer provided allowable levels of +2.4 million in-lb (maximum) and -1.2 million in-lb (minimum) for continuous operation. Note that the coupling has an even higher limit for the peak transient torque. Torsional oscillation at the auxiliary end of the compressor crankshaft was also plotted for each load step. Values were then compared to the compressor allowable [17] and found to be acceptable for all cases tested.

Both units were tested and both units had similar TNFs. Torque and torsional oscillation results were also acceptable on the second unit. Therefore, torsional resonance or high alternating torque was eliminated as a cause for the coupling failures.

<u>Axial Test</u>

With the coupling guard removed, high axial motion was observed on the coupling and the motor shaft. Proximity probes were used to measure axial vibration. Since the surfaces were not smooth and treated for such measurements, runout compensation was included in the results. The motor was off from the magnetic center and moving axially during startup, which exceeded the gap range of the proximity probes. If the probe was gapped too close, it could rub and be damaged. However, if gapped too much the probe would be out of range and not sense any vibration. The main objective was to determine the frequency of the axial motion.

The highest axial motion on the motor side of the coupling hub was observed at compressor load steps 13 and 11 and exceeded the probe range of 50 mils peak-to-peak (p-p). Approximately 45 mils p-p was measured in the axial direction at 2.1 Hz. At load step 11, over 50 mils p-p was measured in the axial direction at 2.1 Hz. As shown in Figure 8, sub-synchronous frequencies of 2.1 Hz and 4.0 Hz were observed while operating at compressor load step 7.

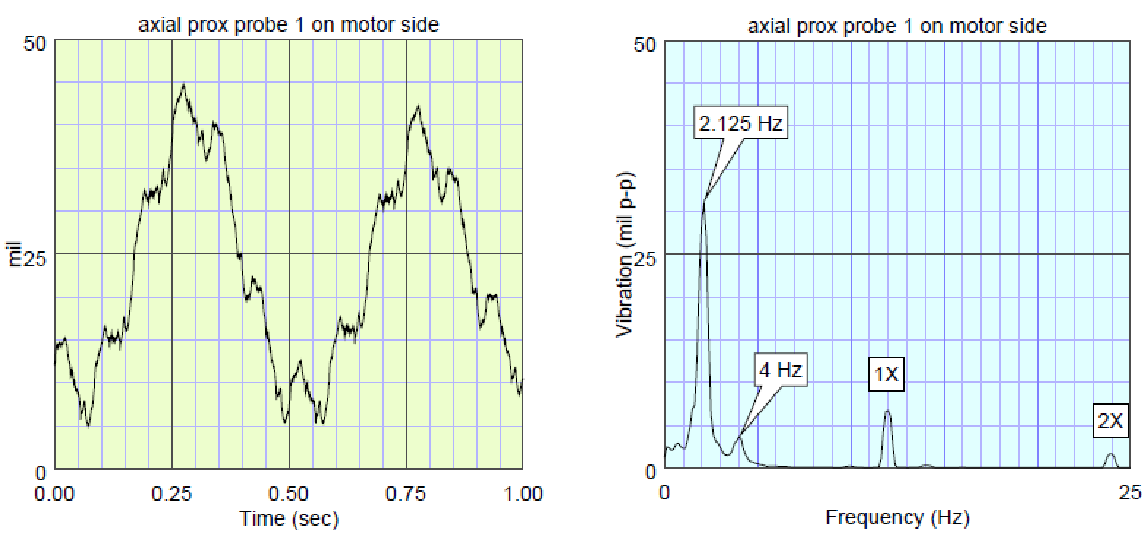


Figure 8. Axial Measurements

In summary, high axial movement was detected at compressor load steps 13, 11, and 8. Load step 7 was also unsteady. Load steps 12 and 3 were skipped because they were already known to have excessive axial vibration and did not want to damage the coupling. Since the proximity probe has limited range, the maximum amplitudes may not have been captured.

The second unit seemed to have worse axial vibration than the first unit. For example, axial movement was observed at almost every compressor load step. Another difference was that the motor was moving at 2.0 –

2.5 Hz at some load steps, while the center spacer piece in the coupling was visibly moving at a higher frequency, possibly 6 Hz. The frequency could not be verified since proximity probes were not installed on the center piece.

There were load steps that looked steady on the proximity probes for the motor and compressor flywheel, but axial motion was still observed on the coupling center piece. This happened at load step 18 just before the compressor shuts down. That may be why the motor came to rest away from the magnetic center and one disc pack was compressed while the other disc pack was stretched axially.

The axial position of the motor and compressor shafts were plotted, as shown in Figure 9. There was a large change in axial position for the motor during startup. There was another sudden shift in axial position between load steps 16 and 15.

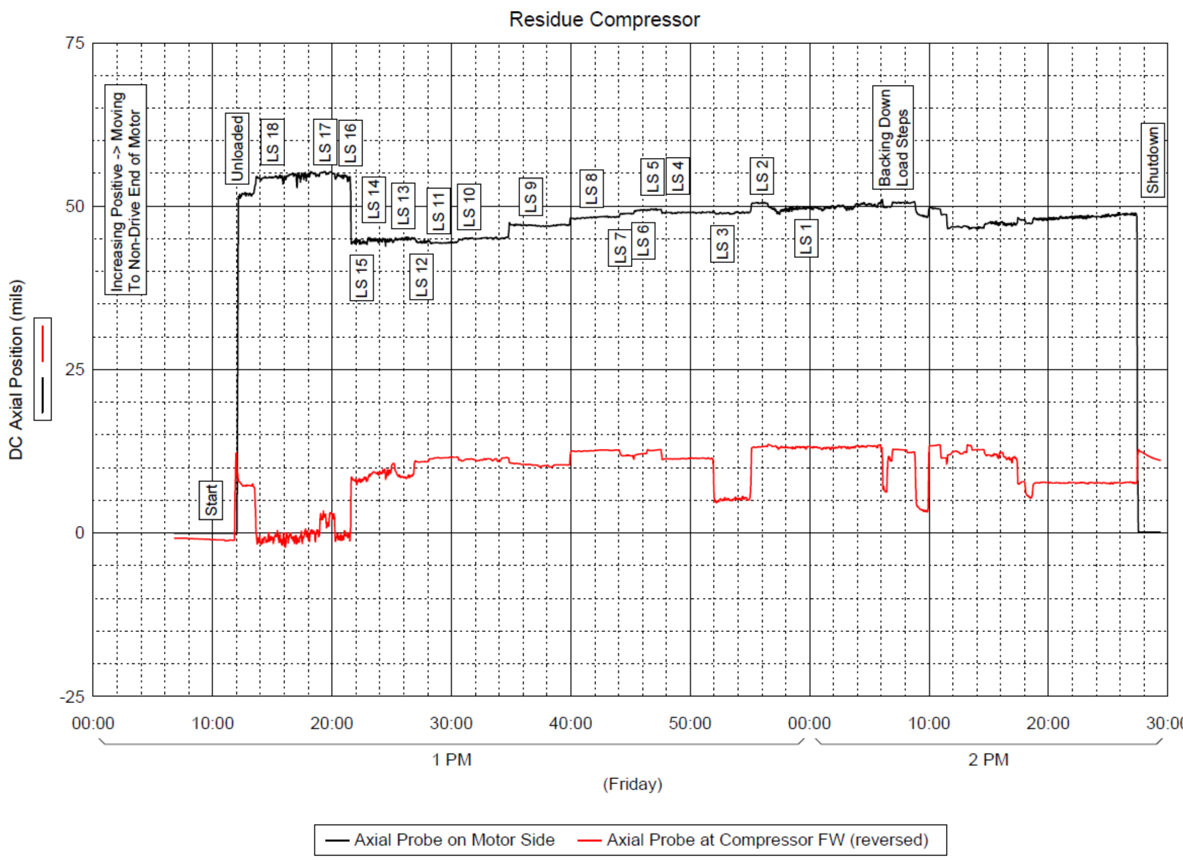


Figure 9. Trend Plot

There was significant axial motion of the motor for load steps 2 and 1. However, these were considered relatively good load steps for the first unit. Axial motion occurred at 2.5 Hz during load step 2 operation. At load step 1, additional low frequencies appeared: 4.88, 7.25, and 9.63 Hz. It is interesting to note that the spacing of the low frequencies was 1/5th of the 1× running speed of 12 Hz.

According to the coupling manufacturer, when one disc pack is elongated and the other disc pack is compressed, this is called “oil can.” If just the motor was shuttling back and forth, both disc packs would displace the same, either elongated or compressed.

Impact Test

With the unit down, an impact test was performed on the center spacer piece of the coupling. An impact ~~tests~~ test involves using an instrumented hammer with load cell and accelerometer to measure the response. A natural frequency is identified by a peak in amplitude along with a phase shift. For this case, with the unit at rest, the axial natural frequency was found to be approximately 27.5 Hz (1650 CPM) as shown in Figure 10.

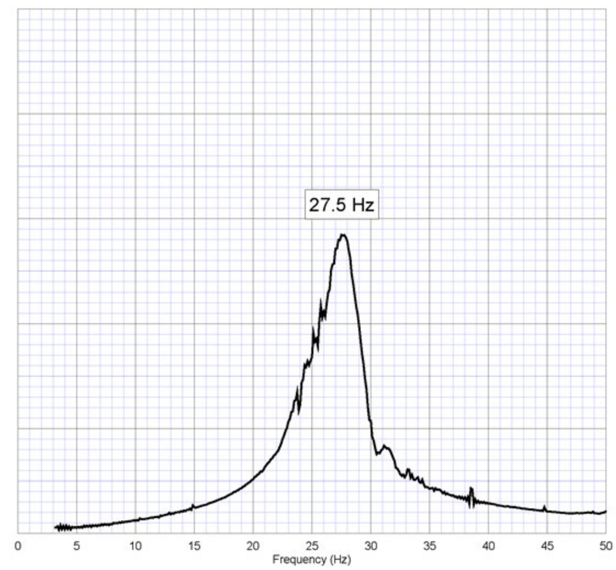


Figure 10. Impact Test

According to the coupling manufacturer, the spacer weighed 694 lbs. The manufacturer's calculated axial natural frequency for this coupling was 800 CPM (with zero axial deflection) to 1300 CPM (with maximum axial deflection of 0.1 inch). This corresponds to approximately 13 - 22 Hz, which was lower than measured with the impact test but well above the predominant frequencies of axial vibration observed. However, if the disc packs were deflected more than maximum allowable of 0.1 inch, this could account for the higher measured natural frequency in the field compared with theoretical predictions.

Verify Magnetic Center

One of the motors was run solo to check magnetic center. The motor was decoupled by removing a disc pack on the motor side of the coupling. The coupling spacer was left hanging on the other disc pack that was attached to the flywheel adaptor during the solo motor test.

A visible axial vibration of the motor shaft was observed during startup. This response settled out within approximately a minute. The motor ran at magnetic center when looking at the pointer on the drive end of the motor shaft.

Coastdown after the motor solo run took approximately half an hour because of high inertia and no load. During the coastdown, the motor shaft was pulled off magnetic center (away from compressor) by 5/8 inch. The motor rotor was not moved back to magnetic center position before reinstalling the disc packs in the coupling.

A similar test was performed on the motor of the other unit. It was reported that the motor ran at magnetic center during the solo run and then drifted 9/16 inch off magnetic center (away from compressor) during the coastdown.

This axial shift could be due to a thrust force produced by the single cooling fan. The motor uses sleeve bearings, which have no thrust capability to prevent an axial shift. It was reported that the motor rotor was not moved back to the magnetic center position before reassembling the disc pack coupling.

Axial Natural Frequency

In addition to the axial natural frequency of the coupling spacer, there is also an axial natural frequency of the motor versus compressor with the controlling stiffness being the coupling. As shown in Figure 11, there are several forces acting on the system. There was only one cooling fan on the motor shaft and it may have caused a thrust toward the non-drive end of the motor. There is also a magnetic centering force in the motor between the stator and rotor.

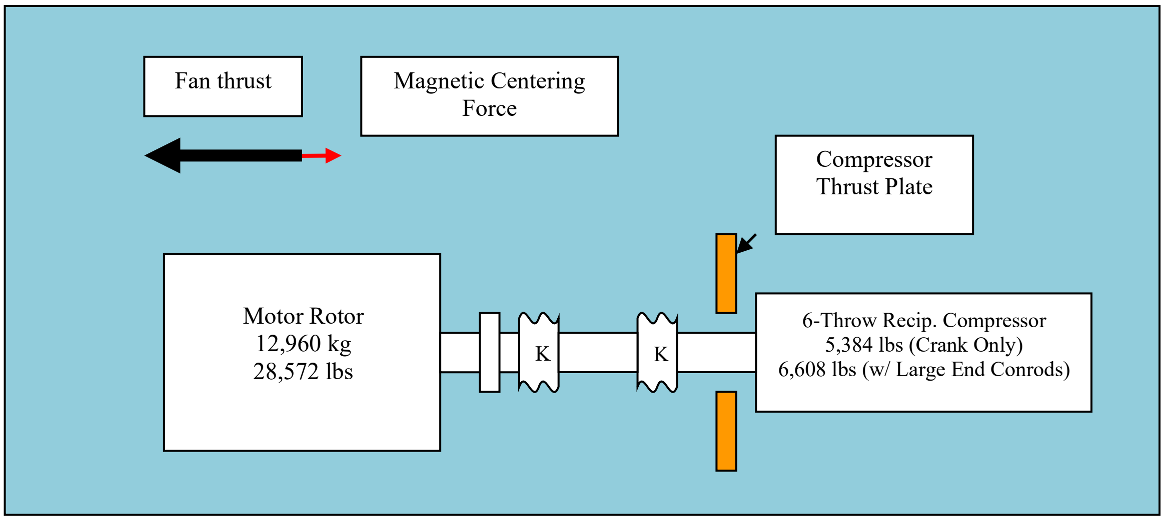


Figure 11. System Diagram

This compressor frame has the upgraded lube oil fed thrust plates on the driven side. Axial thrust was checked on one unit and found to be 0.026 inch from the frame to the flywheel with pry bars. The original thrust measured at compressor assembly was 0.040 inch. The other unit was listed as 0.038 inch. During the inspection, no wear was found on the thrust plates.

Backing out an axial stiffness from the impact test and calculated values from the coupling manufacturer shows that each disc pack could have a wide range of 6,300 - 27,000 lb/in. Assuming the compressor behaves as a fixed end, the axial natural frequency of the motor could range from 1.0 to 2.2 Hz. Using a 2-mass system and allowing the compressor to move, the axial natural frequency would range from approximately 2.5 - 6 Hz, all of which were seen in the data during the load sweeps.

Possible Causes for Coupling Failures

The coupling failures did not appear to have been caused by high torsional vibration. Measured torque was well below allowable limits for all compressor load steps.

According to the coupling technical field reference [18], there can be several causes of a coupling failure:

- Misalignment
- Loose locknuts
- Corrosion
- Torque overload
- Misapplication

Distorted disc packs were found in the new coupling before operation of one of the units, as seen in the photos in Figure 12. This may be caused by improper alignment and/or installation procedure.

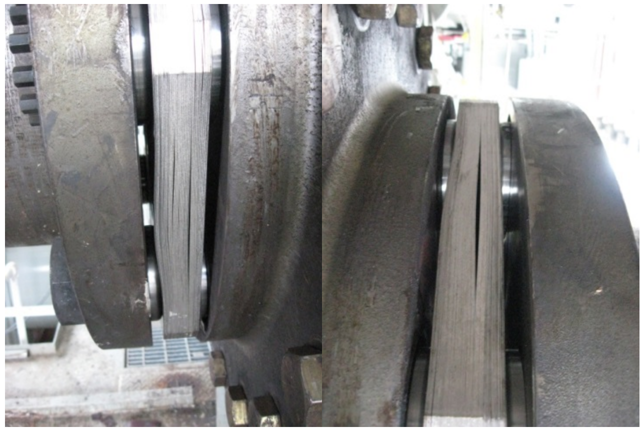


Figure 12. Coupling Disc Packs

High axial vibration of the coupling spacer and motor was observed and varied with compressor load steps. Axial vibration of the motor occurred at approximately 2 Hz to 2.5 Hz, which may coincide with the axial natural frequency of the motor based on simple hand calculations.

On one of the units, it was also noticed that the coupling spacer was vibrating in the axial direction at a higher frequency (near 6 Hz) than the motor (2 Hz) for most compressor load steps. Since the compressor units did not exhibit the same axial vibration for each load step, this makes locking out certain load steps impractical as a solution.

Visual inspection found no wear on the compressor thrust plates. This means that most of the axial vibration was likely occurring at the motor and through the coupling.

Magnetic center was verified for both motors during solo runs. One of the motor shafts moved 5/8 inch during coastdown while the other motor shaft moved 9/16 inch away from compressor. The motor rotors were not repositioned back to the magnetic center marker before re-installing the disc packs, which might have caused distortion.

Recommendations

- Since the couplings in both units were subjected rough operating conditions, inspections of the disc packs were recommended.
- It was recommended to verify alignment on both units. Previous experience of the compressor manufacturer showed that improving axial alignment solved the axial vibration problem.
- To set the axial distance between the motor and compressor:
    - Set the motor to magnetic center
    - Set the compressor to the center of its total thrust
    - Measure distance between hubs
    - By moving the motor, adjust distance to coupling OEM
- As a short-term solution, several compressor load steps were locked out.
- The long-term solution involved adding a thrust capability to the motor bearing to compensate for any forces created by the single cooling fan on the shaft. The motor does not usually have thrust capability. Therefore, the manufacturer was contacted to quantify magnetic centering force and thrust force from the external cooling fan so that the new motor bearing could be properly selected.

IV. Motor End Bell

For this example, the reciprocating compressor had 4-throws in 3-stage service and was driven by an induction motor rated for 3,150 HP at 894 RPM. The motor had a history of problems and previously experienced high bearing vibration in the horizontal direction due to rotor looseness.

It was reported that some of the spider arms "rang" at the motor shop indicating minimal contact with the laminations. These arms were selected to be welded to the laminations using bar stock. After reinstalling the repaired motor, impacting was no longer detected at the bearings during loaded operation.

During a follow-up test, high vibration was measured on the motor bearing housing in the axial direction, which exceeds the manufacturer's recommended levels of 0.3 ips peak (alarm) and 0.45 ips peak (shutdown). However, motor vibration is not normally trended in this direction, only horizontal and vertical directions.

A survey of the other units at the plant showed that the bearing housing vibration was similar. It was also reported that another motor at a different location was experiencing a similar problem. The vibration primarily occurred at higher frequencies (5× and 6× running speed) as shown in Figure 13.

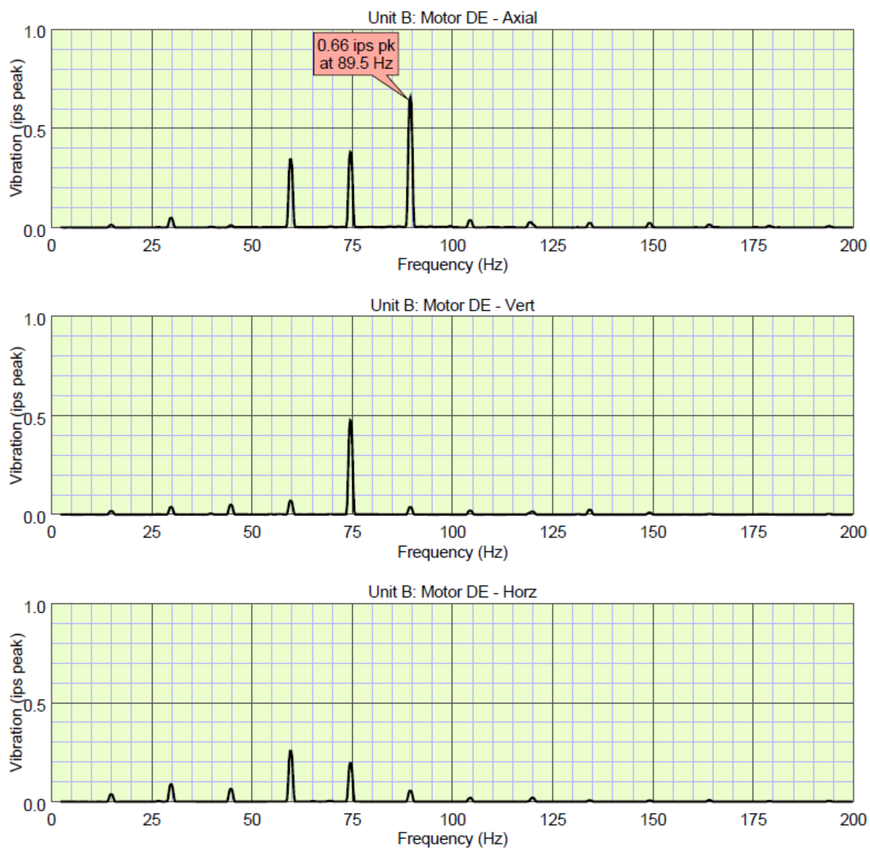


Figure 13. Motor Vibration

There was concern that the bolts around the end bell could loosen over time. Upon closer inspection, four of the eight bolts around the motor end bell were found to be loose.

After the bolts were tightened, a detailed operating deflection shape (ODS) was performed while the motor was running and coupled to the compressor. Based on this ODS measurement, the end bell is acting as a "drum head" and appeared to be resonant, Figure 14.

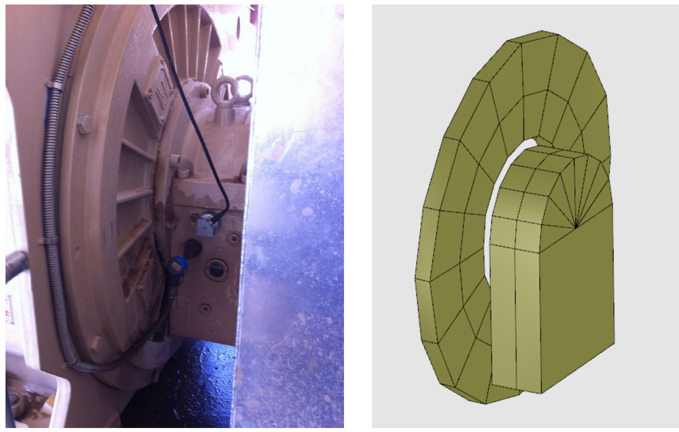


Figure 14. ODS of Motor End Bell

A shaker test was recommended to confirm the mechanical natural frequency (MNF) of the motor end bells. The shaker has a known imbalance of 0.24 lb-in, which produces a force that varies with speed squared. The shaker speed is controlled by throttling the supply air. The motor was tested while running (rotor on oil film) and decoupled from the compressor. It is important that the motor shaft be on the oil film to correctly measure the MNFs. Baseline vibration was low before starting the shaker test. A natural frequency was found at 91.7 Hz in the axial direction on the drive end (see Figure 15).

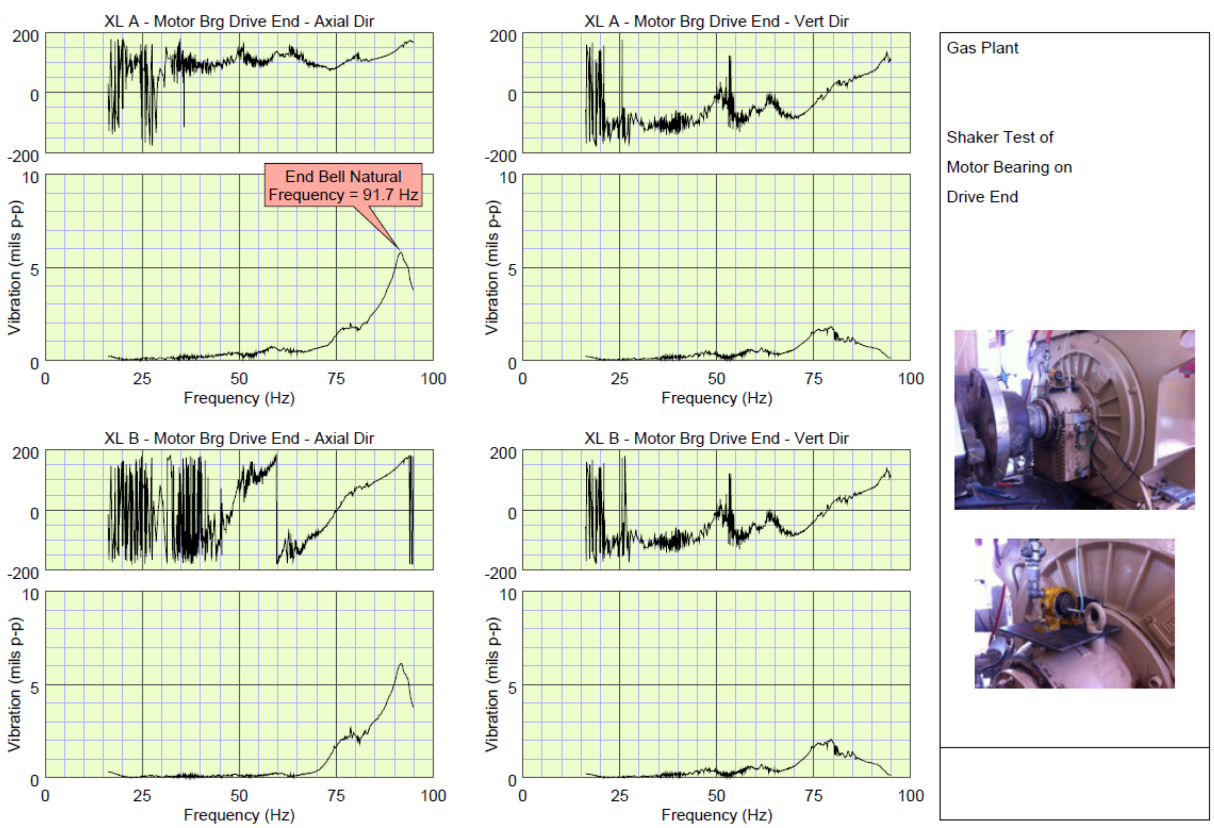


Figure 15. Results of Shaker Test

The shaker test was repeated on the non-drive end motor bearing, which had a natural frequency of approximately 97 Hz. Note that the drive end (DE) and non-drive end (NDE) bearings were different sizes, and the end bells were different designs with different size inspection windows. Vibration on the NDE bearing was lower because there was an 8% separation margin between the axial natural frequency and compressor harmonics.

There was discussion about how to stiffen the end bell to detune this resonance. The motor manufacturer already had an alternate end bell design with thicker plate specifically for reciprocating compressor applications. A finite element analysis (FEA) would be needed to fully evaluate the end bell design and possible modifications.

Conclusions and Recommendations

- Axial vibration on the motor drive end bearing of all of units tested exceeded the recommended allowable from the motor manufacturer. The vibration occurred at 5× and/or 6× running speed. This vibration likely contributed to the loosening of the bolts around the end bell.
- Operating deflection shape (ODS) measurements showed a “drum head” motion of the motor end bell in the axial direction. However, simple tests such as adding mass or increasing stiffness with wooden wedges were unsuccessful in reducing the axial vibration of the drive end motor bearing to an acceptable level.
- A shaker test of the drive end bearing identified an axial natural frequency near 92 Hz or 6× running speed. The axial natural frequency was 97 Hz on the non-drive end, which was 8% above 6× running speed. Based on the known shaking force applied to the drive end bearing, approximately 80 lbs 0-p would produce 1 mil 0-p or 0.6 ips 0-p of axial vibration at 90 Hz.
- The motor manufacturer provided thicker motor end plate for reciprocating compressor service.

V. Motor Cooling Fans

There are sixteen residue compressor units at two gas plants. Each unit consists of induction motor (rated for 5,000 HP at 896 RPM) driving a reciprocating compressor with 4-throws and 2-stages through a disc pack style coupling. The compressors have five load steps (load steps 0 through 4).

Figure 16. Motor – Compressor Unit

Figure 17 shows the motor rotor with two internal fans (NDE and DE) sitting on the shop floor.

Figure 17. Motor Rotor with Internal Fans

Some of these motors experienced failures of the internal cooling fans after commissioning. For example, the first two failures were detected when the cracked fan shroud contacted the winding causing a differential trip. This trip had a repeatable behavior (tripping at a certain point during the starting sequence). The plant found the differential protection reliable and likely prevented major damage from occurring.

Later the plant learned that when vibration levels were trending upward (although not yet at the alarm or trip levels) it could indicate an impending fan failure thus justifying a shutdown and visual inspection. The vibration trip level would not be reached until a motor fan blade fell completely off and caused noticeable imbalance (more severe than just cracking).

The fan failures are shown in Figure 18. Cracks appeared to start at the welds, which act as a stress riser and then propagate through the blades and shroud. The metallurgical study found fatigue fractures caused by reverse bending. This indicated vibration was likely causing stresses in the fan during operation.

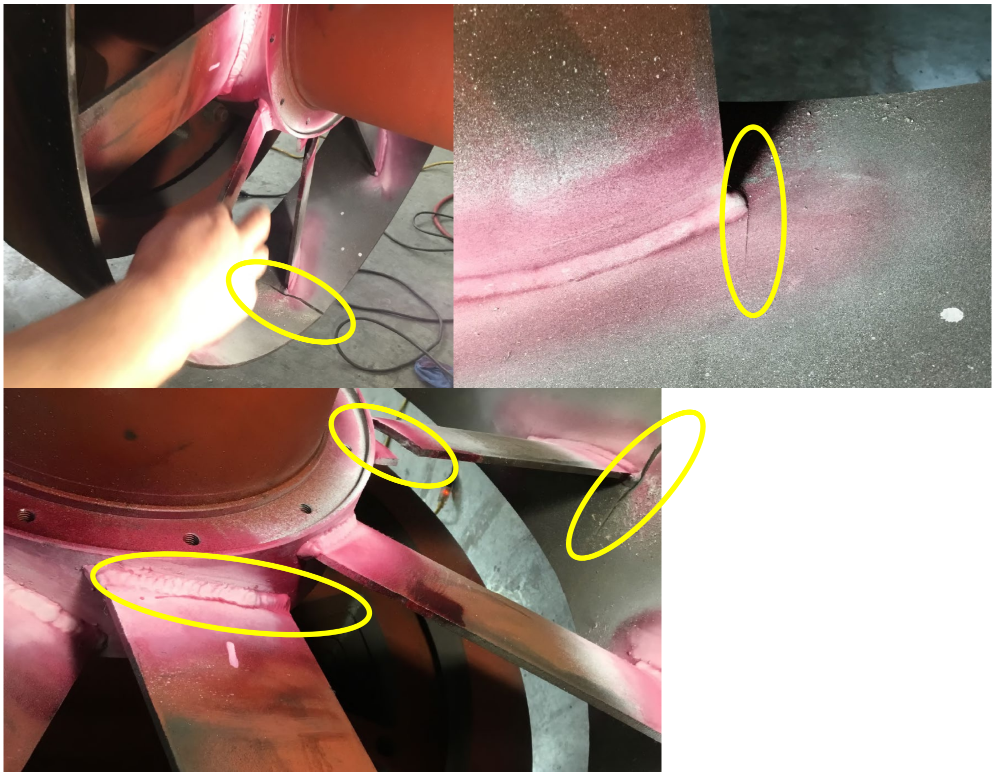


Figure 18. Cracks in Fan

To determine the root cause, several tests were proposed. Excessive vibration could occur if the fans have a natural frequency near a compressor harmonic or some other excitation source. Torsional vibration testing was performed on the unit. Impact tests of the fans were also performed to determine natural frequencies and mode shapes. Installing strain gages on the fan and using a telemetry system to record the strain levels during operation was discussed as the most beneficial test but was not possible due to time required and operational constraints at the plants.

Based on the initial field tests,

- The compressor systems had acceptable torsional and vibration characteristics throughout the load steps tested. The first torsional natural frequency (TNF) was approximately 83.0 - 83.5 Hz for both units tested. This corresponds to 5.6× running speed and has an acceptable separation margin. Therefore, no obvious problems were found that would immediately explain the failures of the motor fans.

- The only concern on one of the units was higher torsional velocity at 8× running speed, which is near 2× line frequency. This could potentially indicate an electrical issue with the motor. It was recommended that electrical data be reviewed, and a current analysis be performed [19].

- The fans could have a natural frequency that is coincident with a compressor harmonic or other excitation source (electrical frequency) resulting in high vibration and excessive stresses. However, this would not be detected from vibration readings taken on the outside of the motor frame and bearing housings.

Impact tests were recommended to determine the natural frequencies and mode shapes of the existing motor fans. The data could also be used to help normalize the finite element analysis (FEA) that the motor manufacturer was performing.

Impact response spectra were obtained using an instrumented hammer, a response accelerometer, and a strain gage. The impact tests were performed in the motor repair shop with the rotor removed.

Figure 19. Close-Up of Internal Fan NDE with Instrumentation

Frequency response functions (FRF) were plotted. A natural frequency is identified by a peak in response amplitude and a phase shift.

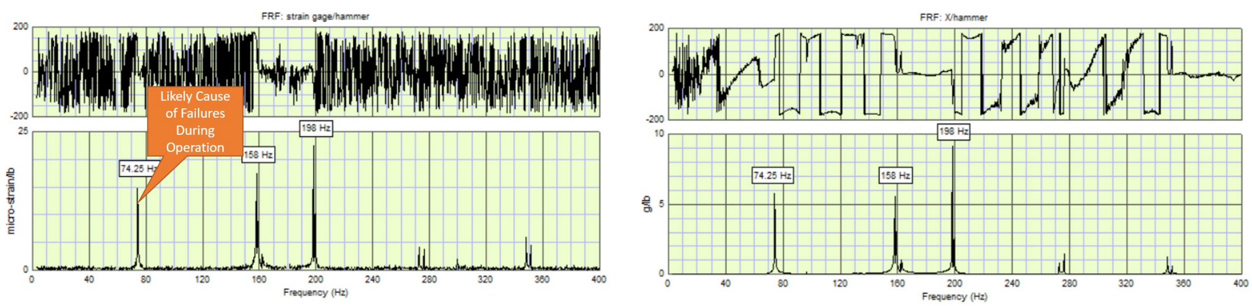


Figure 20. Impact Test Results

Two typical fan modes were identified by the impact data and mode shape software [20]. The first mode is like an umbrella folding with the fan blades flexing and the center hub fixed. Maximum stress would be near where the cracks formed. The second mode has the outer ring wobbling about the center hub with the fan blades flexing. Reference [21] gives more details about modal testing of a motor fan.

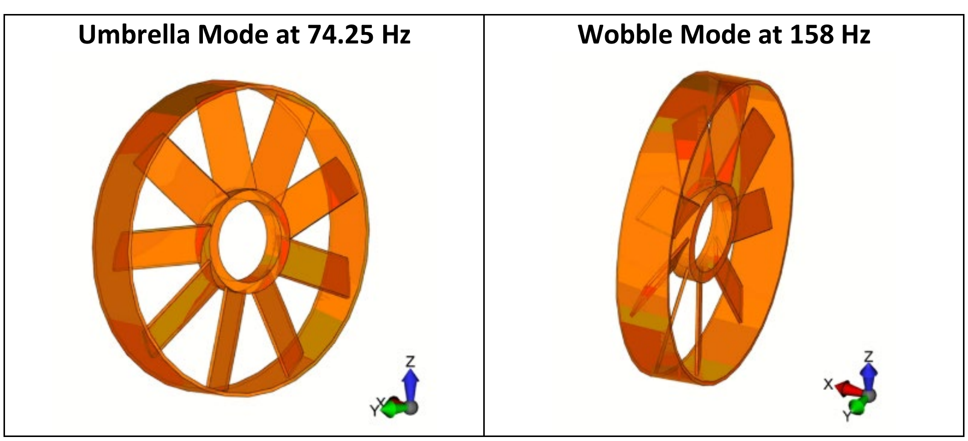


Figure 21. Fan Mode Shapes

The next step was to determine separation margins from compressor harmonics (see Table 1).

Table 1. Natural Frequencies of Original Fan

Mode	Nat. Freq. (Hz)	Mult. Run Speed
1	74.25	5.0
2	158.25	10.7
3	198.5	13.4
4	273.25	18.4
5	348.75	23.5
6	408.25	27.5
7	453.25	30.6
8	498.75	33.6

Insufficient Separation Margin

Conclusions from impact tests of original fan design performed at the motor repair shop:

- Fan has a natural frequency coincident with 5× running speed.
- The axial mode shape at 5× running speed resembles an umbrella opening and closing.
- Low damping means minimal excitation can cause vibration, alternating stresses, and fatigue failure.
- This would explain stresses near the failure location and reverse bending discussed in the metallurgical report.
- This resonance condition was likely related to the fan failures. Telemetry data during operation would be needed to confirm.
- Note that the fan tested had a reduced interference fit recently recommended by the motor manufacturer (approximately 0.005 inch).

The motor manufacturer was concurrently analyzing these fan failures as well as possible design changes. New fans were provided by the manufacturer. Modifications included:

- Interference Fit (Fan Hub to Shaft) Reduced to 0.005”
- Increased Outside Diameter of Fan Hubs. This resulted in shorter fan blades since the outer ring diameter remained the same.
- Thicker Fan Blades (3/8” versus 1/4”).
- Improved Weld Quality (which should reduce SCF).

The impact tests were repeated at the motor repair shop for the new fans. Both the NDE and DE fans were impact tested. The natural frequency results are summarized in Table 2.

Figure 22. Close-Up of Internal Fan NDE with Instrumentation

Table 2. Natural Frequencies of New Fans

New NDE Fan

Mode	Nat. Freq. (Hz)	Mult. Run Spd.
1	115.75	7.8
2	196.5	13.2
3	236.75	16.0
4	309 - 313	20.8 - 21.1
5	387.5	26.1
6	475.75	32.1

New DE Fan

Mode	Nat. Freq. (Hz)	Mult. Run Spd.
1	115.75	7.8
2	196.25	13.2
3	233.75	15.8
4	307.25	20.7
5	379 - 385.25	25.6 - 26
6	472.25	31.8

The first natural frequency of the fans increased from 74 Hz to 116 Hz due to the design modifications. Therefore, the new internal motor fans no longer had a natural frequency coincident with 5× running speed (now at 7.8× running speed).

Follow-Up Tests During Operation

The three motors were retrofitted with new internal cooling fans supplied by the manufacturer. As a follow-up test, the vibration levels were checked on all units at one of the gas plants. One of these units had been back in operation for approximately one month with the new fans. However, over that time period the vibration had been trending upward to approximately 0.2 ips 0-p overall on the motor NDE bearing. Plant personnel were concerned because vibration at this level previously indicated a failed motor fan (original design).

- When impact tested in place at the plant, the internal cooling fan on the NDE of the motor had similar natural frequencies in the field as measured in the motor repair shop. The new fan did not have any apparent cracks.
- The internal cooling fan on the NDE of one motor was found to be rubbing against the fiberglass shroud / baffle. Once corrected, the unit was put back into service. The motor vibration in the vertical direction decreased from approximately 0.21 to 0.15 ips 0-p while operating at compressor load step 4 (near full load). This demonstrates sensitivity of the motor vibration levels to problems with the fan such as a light rub.
- For one of the motors, vibration sidebands were measured around the rotor bar passing frequency possibly indicating an issue with the rotor bars. However, additional current signature analysis would be needed to confirm.

The plant had concern for some of the other motors that were experiencing an increase in vibration levels. Of the remaining motors with original cooling fans, one unit seemed to have the highest radial vibration of approximately 0.2 ips 0-p overall on the motor NDE bearing.

- All of the compressor frames had acceptable vibration throughout the load steps tested. Also, the radial vibration levels measured on the motor bearings were below the alarm level throughout the load steps tested.
- Although not continuously monitored, vibration levels in the axial direction on some of the motor bearings were found to be near the alarm level. For some compressor load steps, the motor end plate may be flexing at 6× running speed while the motor frame and foot had low vibration. The motor vibration was sensitive to the compressor loading. If not addressed, high axial vibration could possibly loosen the bolts holding the motor end plate and bearing housing thus increasing radial vibration.

It was recommended that:

- The electrical data be reviewed to determine if there were any indications of potential motor issues. High resolution FFT is required to evaluate the sideband surrounding the 60 Hz electrical frequency. A signature analysis of the motor current might be used to check for eccentricity, shorted laminations, broken rotor bars, etc.
- If vibration levels increased in the future on motors that have already been retrofitted with new internal cooling fans, check for possible rubbing on the shroud / baffle or cracks in the fan.
- Based on the measured data, the unit with the highest motor vibration levels was selected to be the next candidate for retrofitting the fans. If the vibration continues to increase, the compressor load might be reduced to try and prolong the life of the original motor fan.
- The plant may consider lowering the vibration alarm and trip settings for the motors based on operating experience. Previous failure events seemed to have occurred on the motors without the vibration levels reaching the alarm and trip limits. For example, lower values of 0.2 ips 0-p for the alarm level and 0.3 ips 0-p for the trip limit may be more reasonable based on historical trend data.

Unfortunately, the owner continued to experience problems with the modified fans, even on the DE side of the motor. Another option would have been to use a different type of cooling arrangement for the motor where the fans are driven by separate motors located on top. However, this was ruled out as too costly.

Suggestions were made to change the material to non-ferrous to prevent possible interaction with magnetic forces from the motor. Changing to cast aluminum and removing the shroud would also reduce mass and stress. Attachment would be with locking key and no heavy shrink fit. Essentially converting the axial fan design to traditional WP-2 motors. Figure 23 is a photo of the latest fan design.

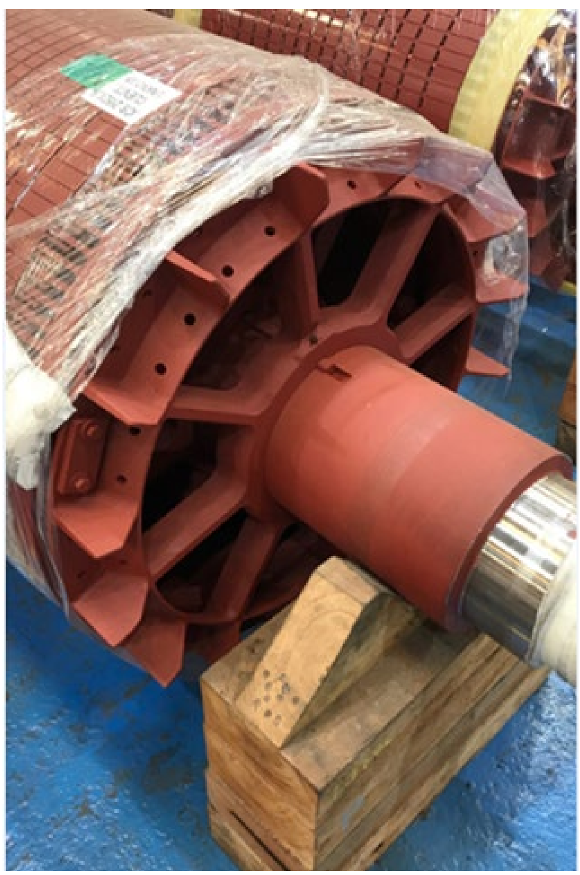


Figure 23. Aluminum Fan Design

VI. Torsional Failure of Motor Shaft

This case study [22] describes a vibration problem on a skid mounted high-speed reciprocating compressor package. The skid mounted package consisted of motor-driven reciprocating compressor with 4-throws and 3-stages. The induction motor was rated for 2750 HP and controlled by a VFD from 600 to 900 RPM. This type of package is commonly used in the gas gathering and gas processing services. These units may be re-purposed several times during the life of the equipment.

The initial concern was high vibration on the motor in the axial direction. When the coupling guard was removed for further testing, a 45-degree crack was noticed on the motor shaft (see Figure 24). This is a classic indication of a torsional fatigue failure.

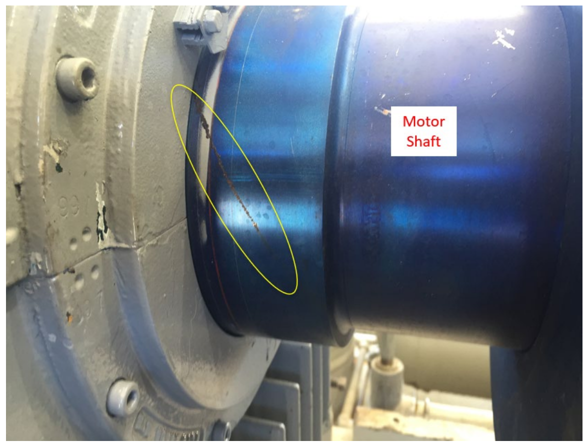


Figure 24. Failed Motor Shaft

A spare motor was installed, and torsional measurements were made. This confirmed the problem as a torsional resonance. The flywheel was modified to de-tune the resonance. Follow-up testing was conducted and confirmed the success of the modifications.

The torsional vibration analysis (TVA) report that was performed by a third party was subsequently reviewed. They had recommended a flywheel to de-tune the system so that 5× compressor harmonic would not be resonant at 900 RPM. The TVA also assumed constant speed operation and that the VFD would only be used for soft start. However, the unit was variable speed, so the torsional resonance was within the operating speed range and caused the motor shaft failure.

Once the motor was replaced, torsional measurements were taken using a strain gage telemetry system. The first TNF was found at 75 Hz (4500 CPM). This means that the resonance with 5× running speed would be at the maximum speed 900 RPM. The third-party TVA did not accurately predict the resonance. This can happen for many reasons as discussed in reference [23].

The short-term recommendation was to limit the operating speed range from 750 - 850 RPM to avoid the resonance at 900 RPM. The long-term solution involved trimming the flywheel to increase the TNF to approximately 80 Hz.

The TVA was normalized to match the field data. This involved adjusting the mass-elastic model to match the test data TNF. A parametric study was then performed on the computer to determine the optimal flywheel size. Follow-up field measurements with the modified flywheel confirmed good separation margin and acceptable alternating torque and stresses.

Figure 25 shows a comparison of the "before" and "after" torsional response measured in the motor shaft. A big improvement was made with the modified flywheel.

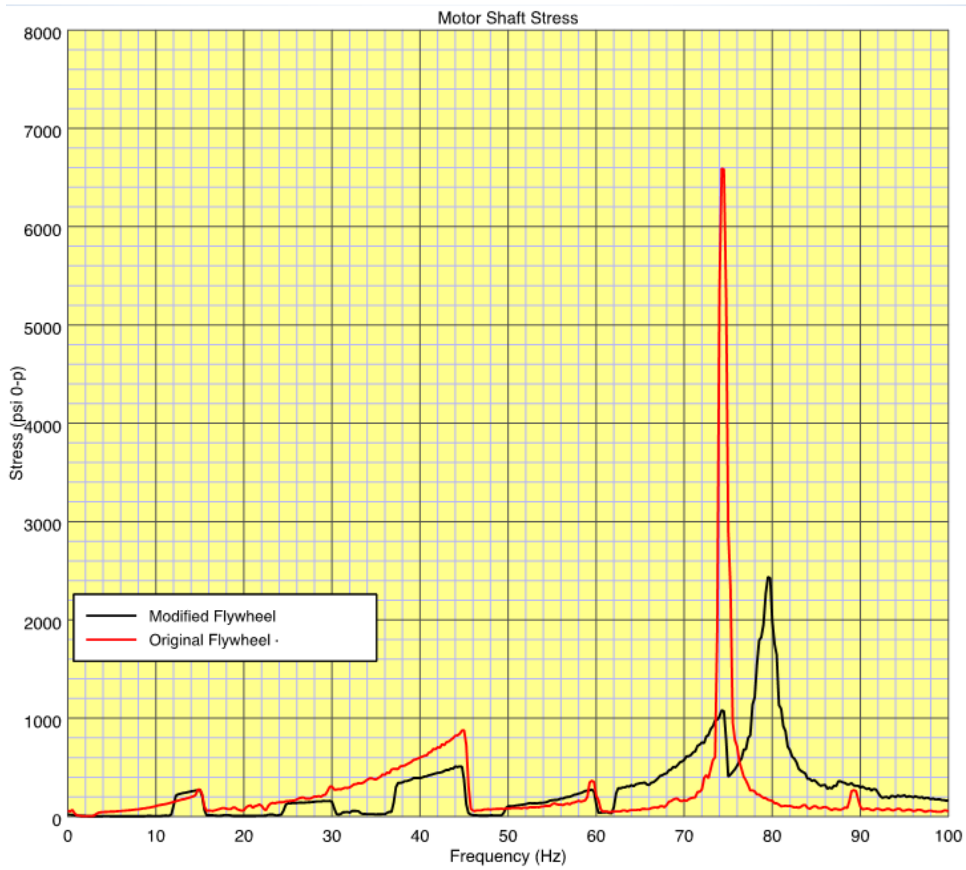


Figure 25. Comparison of Torsional Response

Lessons learned included:

- Really analyze the vibration data, which was the initial indication of the torsional problem.
- Accuracy of TNF calculations depend on the mass-elastic data provided by equipment OEM.
- Measuring TNFs during commissioning is often justifiable.
- VFD operation increases chances of encountering torsional resonances within the speed range.

VII. Motor Current Pulsation

Unsteady torque from a reciprocating compressor can cause angular oscillation of the motor rotor. The rotational system is electromagnetically coupled to the motor stator through the air gap flux. Proper sizing of a flywheel and the motor inertia are necessary to limit speed fluctuation, current pulsation, and avoid torsional resonance. Some current pulsation is to be expected; however, excessive levels can cause problems such as unstable current and power readings, failed motor synchronization during startup, nuisance trips during operation due to high current, and increased stator temperatures.

Review of Standards

The paper by Feese and Fanslow [24] discusses how to evaluate current pulsation in synchronous motors and compare them to allowable limits from the API and NEMA MG-1 [25] standards. API 546 [26] addresses brushless synchronous machines 500 kVA and larger while API Standard 618 [27] covers reciprocating compressors for the petroleum, chemical, and gas industries. These both specify that a torsional vibration analysis should be performed in the design stage. When performing a steady-state TVA, it is recommended that an equivalent torsional stiffness (Kem) be included between the motor rotor and stator (ground) to simulate electromagnetic (EM) effects.

API 546 and NEMA provide limits for current pulsation under the actual operating conditions. In many instances, lower current pulsation is recommended to reduce light flicker on power systems with weak short circuit capacity. To verify performance, it may be necessary to measure the current pulsation once the motor has been installed and is operating under full load.

API 618 specifies that TNFs of the complete driver-compressor system shall not be within 10% of any operating shaft speed or within 5% of any multiple of operating shaft speed in the rotating system up to and including the tenth multiple. For motor-driven compressors, TNFs shall be separated from the first and second multiples of the electrical power frequency by 10% and 5%, respectively.

For synchronous-motor-driven compressors, API has three additional requirements:

- *The combined inertia of rotating parts of synchronous motor-compressor installations shall be sufficient to limit motor current variations to a value not exceeding 66% of the full load current for all specified loading conditions, including unloaded operation with cylinders pressurized to their normal suction pressures.*
- *The inertial characteristics of the rotating parts of the compressor and of the drive train shall be such that rotational oscillations will be minimized. Undesirable oscillations include those that cause damage and those that result in harmful torsional and/or electrical system disturbances. For initial design purposes, peak-to-peak speed oscillation of the rotating system shall be limited to 1.5% of rated speed at full load and partial cylinder loads if step unloading is specified.*
- *The torsional stiffness and the inertia of all rotating parts shall provide at least a 20% difference between any inherent exciting frequency of the compressor and the torsional frequency of the motor rotor oscillation with respect to the rotating magnetic field.*

Evaluating Current Pulsation

Field measurements can be performed to evaluate the current pulsation for all compressor load steps, and to help identify any potential problems related to torsional vibration. Some plants require testing during startup of a new motor – compressor unit.

Unsteady load torque from the reciprocating compressor causes angular oscillation of the motor rotor which results in current pulsation. Since current is related to torque, the current amplitude will modulate instead of being constant.

One way to evaluate the motor current and pulsation is with the graphical method. The graphical method is performed on the time-wave form of the motor electrical current as shown in Figure 26.

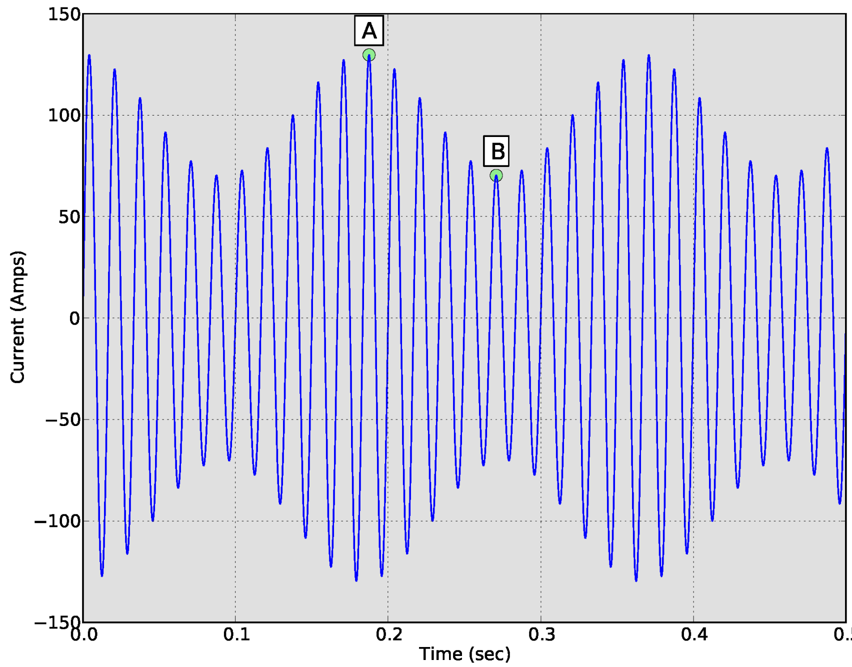


Figure 26. Electrical Current with Pulsation [24]

The formula for percent current pulsation is shown below.

$$\frac{(A - B) \times 100\%}{\sqrt{2} \times FLA}$$

For example, if FLA=71 amps RMS, A=130 and B=70 amps 0-p, then the current pulsation would be approximately 60%.

Example

The machinery had been in service for over 20 years, but after an upgrade of the switch gear, high current pulsation was detected causing the motor to trip. To avoid unexpected shutdowns, the averaging time for the current was extended. The synchronous motor (rated 4,000 HP at 277 RPM) used slip rings for static field excitation. The reciprocating compressor had a single load step (fixed clearance), all cylinders were double-acting, and atmospheric air was being compressed to 6,000 psi final discharge pressure.

Torsional oscillation correlated with the electrical measurements (stator voltage, stator current, field voltage, and field current). As shown in Figure 27, when the compressor was unloaded (blue traces), the current pulsation was low; however, when the compressor was loaded (red traces) the current pulsation increased. The percent current pulsation was found to be 92% of FLA, which exceeded the 66% limit.

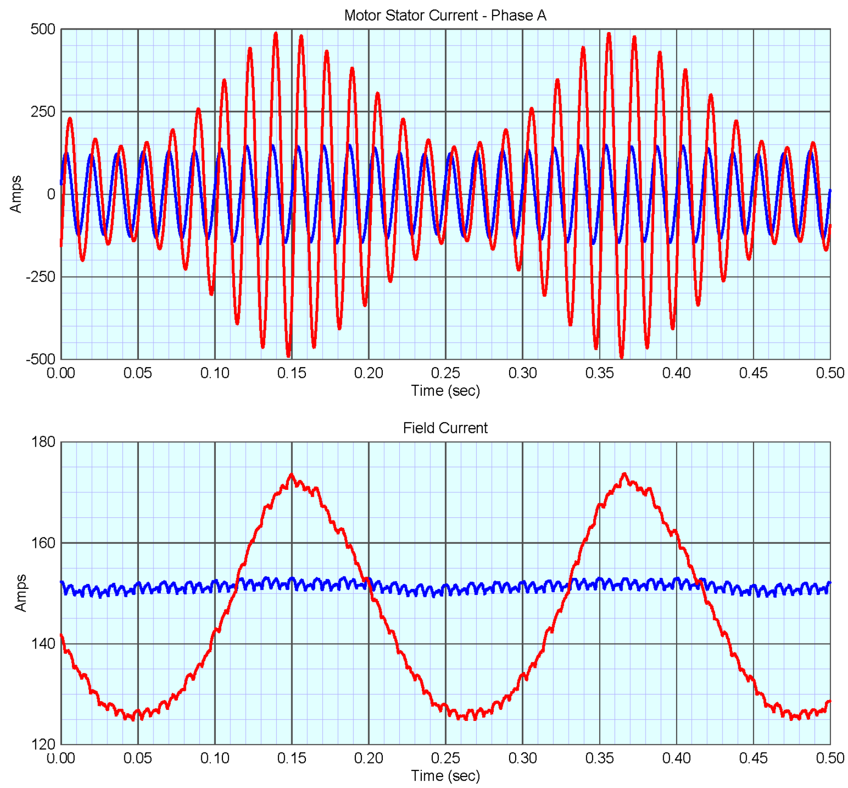


Figure 27. Measured Stator and Field Currents [24]

The angular oscillation of the motor rotor could be decreased by reducing the field excitation. Figure 28 shows the 1× harmonic decreasing from 1.75 to 1.3 deg p-p as the effective torsional spring to ground (EM) was reduced with a weaker field.

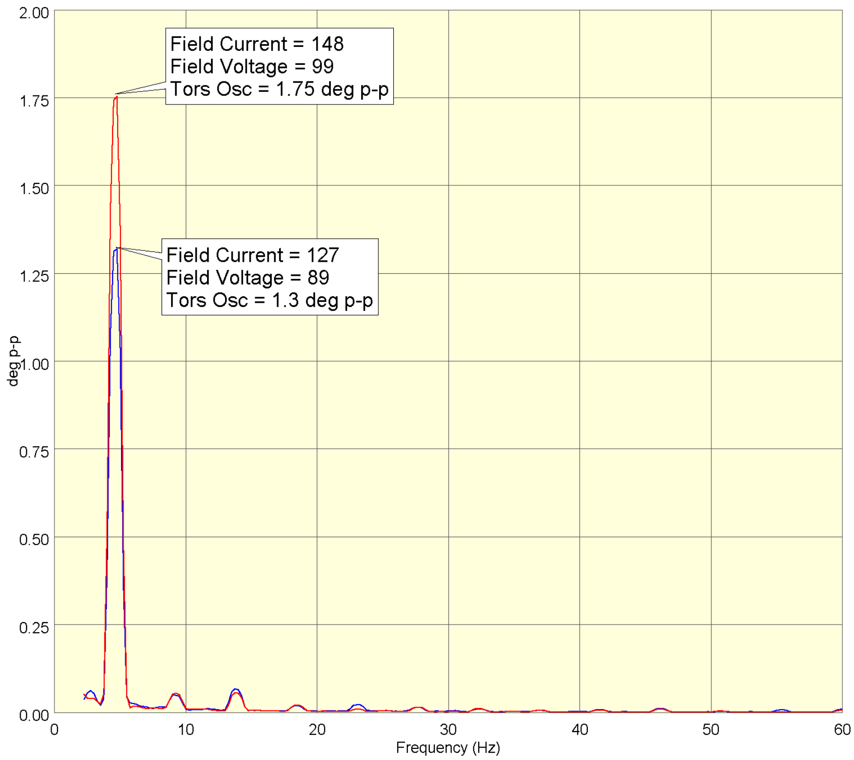


Figure 28. Angular Oscillation of Motor with Reduced Excitation [24]

For comparison, the adjacent compressor unit was tested which was operating at slightly higher speed of 300 RPM. It was found that the angular oscillation was only 0.5 deg p-p or less when operating the same type of compressor fully loaded.

<u>Conclusions</u>

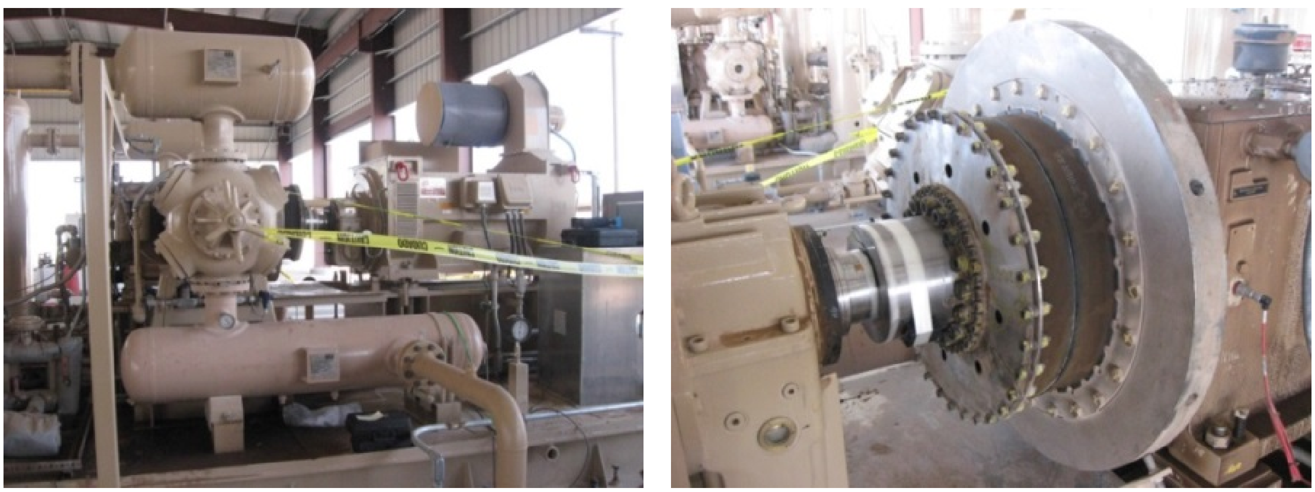
- The torque effort of the compressor can change depending on how it is loaded, which will affect the current pulsation in the motor. The torsional response can also be amplified if there is insufficient SM from the TNFs. Therefore, the complete motor – flywheel – compressor train must have sufficient inertia.
- It was recommended to check for failed compressor valves, which could affect the torque effort and possibly increase alternating torque levels at 1× running speed.
- Motor field excitation can also affect the current pulsation. The automatic PF control should not react to and amplify angular oscillation of the motor rotor. However, using constant excitation could be inefficient when the compressor is operating at reduced load. Ideally, the PF should be near unity.

VIII. Soft Coupling

For an induction motor driving a reciprocating compressor through a torsionally soft coupling, it is important to consider the electromagnetic (EM) effects on the torsional natural frequencies of the system. Not accounting for EM effects could cause substantial errors in the torsional vibration analysis (TVA) and insufficient separation margins (SM). Operating near a torsional resonance could then result in damage to the coupling, shafting, oil pump, etc.

Stiffening effects from the motor magnetic field can affect the torsional natural frequencies (TNFs) of motor driven compressor systems that utilize soft couplings. References [28,29,30] discuss how motor dynamics affect simulated results and present an approximate method for calculating the shift in TNF and damping due to electromagnetic effects. An induction motor driving a reciprocating compressor with a torsionally soft coupling can be sensitive to torsional stiffness across the motor air gap.

A specific example was provided in the paper by Feese and Kokot [31] where multiple compressor trains at a gas plant shared a common variable frequency drive (VFD). A VFD was used to “soft start” one motor at a time and then transfer that motor to across-the-line operation for constant speed (60 Hz electrical). Once three motors are running across-the-line, the VFD can actively control the speed of the fourth unit to adjust for varying plant operating conditions. Photographs are shown in Figure 29.

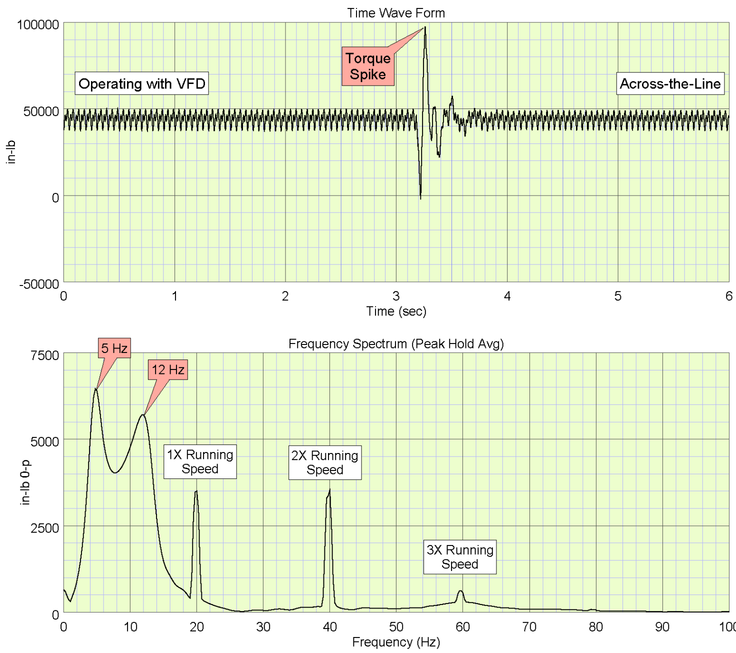


a) Reciprocating Compressor and Motor   b) Rubber-in-Shear Coupling and Flywheel

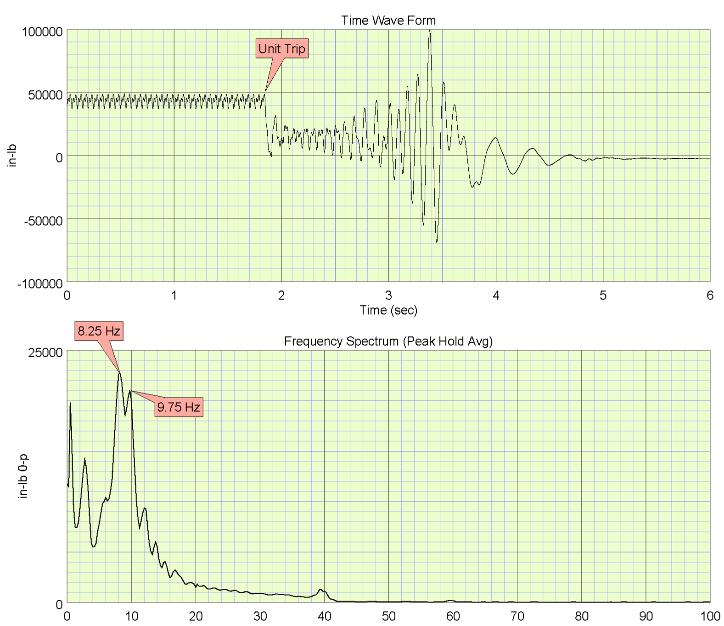
Figure 29. Photos of Compressor System and Coupling [31]

Field measurements of the unit during normal VFD startup determined the first TNF to be 12 – 14 Hz with a cold coupling, and 11 – 13 Hz in the warm condition. The difference in TNF was attributed to the rubber elements in the coupling being less stiff as they heat up.

A torque spike was captured when the motor was switched from the VFD to across-the-line operation. As shown in Figure 30, the highest torque spike was approximately 100,000 in-lb (twice the full load torque), which is still lower than the allowable peak torque for the rubber coupling. During the electrical switching event, non-synchronous response peaks were evident at 5 Hz and 12 Hz by using peak hold averaging. However, the 5 Hz frequency was not observed during startup with the VFD.



The motor was operating across-the-line (no VFD) just before a loaded shutdown of the compressor. The unit was stopped by cutting off electrical power to the motor, which quickly dissipated any EM stiffening effects. Load torque from the compressor stopped rotation in a few seconds. The response frequencies ranged from 8 – 10 Hz during the shutdown event (see Figure 31).



Neglecting the EM effect, the predicted TNF of 9.5 Hz would be below the minimum operating speed (see Figure 32). With the EM stiffness included, the TNF increases by 40% to 13.4 Hz and is within the speed range. The first torsional mode shape involves maximum twisting through the coupling with the motor and compressor having out-of-phase angular oscillation. If the alternating torque exceeds the coupling rating, the rubber elements could be damaged from overheating.

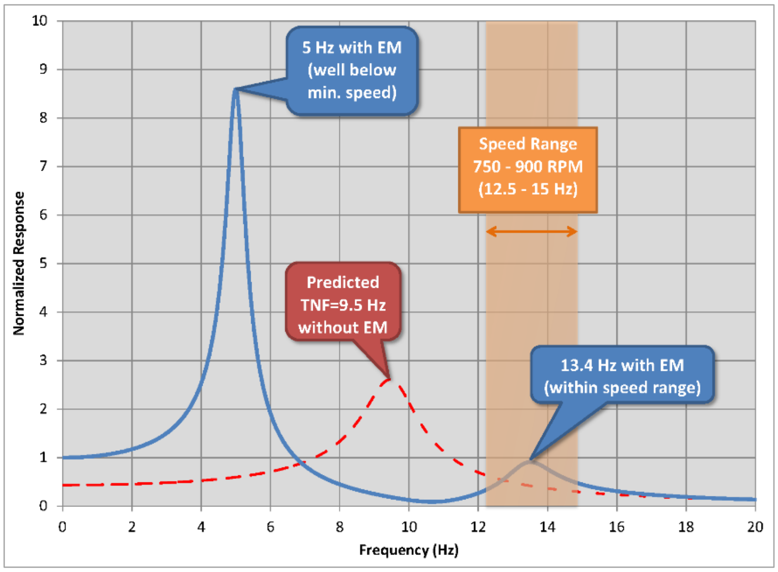


Figure 32. Comparison of Calculated Normalized Dynamic Torque in Coupling – With and Without EM [31]

With EM included, another TNF appears at 5 Hz. The mode shape associated with the TNF at 5 Hz has the entire motor-coupling-compressor system twisting across the EM spring to ground. Without the EM stiffness to ground, this would have been considered the zero mode for the entire system and typically not shown in the TVA report.

Since the TNF of 13.4 Hz fell within the operating speed range, it did not meet the API recommended separation margin. The TNF at 5 Hz had a higher peak response but was well below the minimum operating speed and not of concern. The TNF at 5 Hz likely would not have been noticed except for the switching from VFD to across-the-line that caused a small torque spike.

<u>Recommendations</u>

- Increase the minimum operating speed to avoid possible damage to the rubber elements in the coupling. This was accomplished by reprogramming the VFD in the field.
- Continuous operation of the compressor with single-acting cylinders or failed valves should be avoided since that would significantly increase exciting torque at 1× running speed and could result in premature wear of the rubber elements in the coupling.

IX. Summary of Guidelines and Recommendations

The GMCR guidelines [32] provide some best practices for motor rotor design, fabrication, and assembly. An abbreviated list is shown below along with other recommendations from this paper.

- "Heavy Duty" motors should generally be selected for driving reciprocating compressors.
- If there are multiple motor frame sizes that provide the required power and torque for a given application, the larger motor frame size should normally be selected.
- When possible, use keyless coupling hub and avoid keyway to reduce the stress concentration factor.
- The interference fit should be sufficient to prevent slippage between the shaft and coupling hub for all anticipated events.
- Motor shaft diameter should be equal or greater than compressor stub shaft (including diameter through the drive end bearing).
- Use high strength material for the motor shaft.
- Minimize SCF due to geometric discontinuities and welds.
- Minimize overhung motor arms to prevent flexing and vibration.
- Sufficient clamping force to hold the motor laminations, sufficient to limit shifting between arms and laminations.
- Anchor bolts should have sufficient clamping force to prevent slippage between motor feet and pedestal support. Avoid stacking too many thin shims. Clean oil away from foundation area.
- Use of single fan may create unbalanced axial thrust force on motor rotor. Thrust forces should be identified or dual fans used.
- Motor cooling fan(s) should not have natural frequencies coincident with strong compressor harmonics.
- Stiff end bell design for motor bearing housing. Natural frequencies should not be coincident with strong compressor harmonics.
- Before using a VFD motor, need to verify there are no dangerous torsional resonances within the proposed speed range.
- Low-speed motor systems need to have sufficient inertia to prevent electrical current pulsation.
- Electromagnetic effect (EM) should be included in torsional calculations for a system with a torsionally soft coupling.

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