Gas Machinery Conference
Phoenix, Arizona
October 1-4, 2023

Dampers for Controlling Vibration in Reciprocating Compressor Systems

Jafet Lopez – Senior Manager
Energy Transfer
Jafet.Lopez@energytransfer.com

Josh Zeplin – Senior Project Manager
Energy Transfer
Joshua.Zeplin@energytransfer.com

Tom Stephens, P.E. – Senior Equipment Specialist
Ariel Corporation – Mt. Vernon, OH
tstephens@arielcorp.com

Troy Feese, P.E. – Senior Staff Engineer
Engineering Dynamics Incorporated – San Antonio, TX
tfeese@engdyn.com

Abstract

Viscous dampers have been used for many years to control torsional vibration in engines and to protect crankshafts from fatigue failure. Oil-filled couplings with leaf springs have also been used for controlling torsional vibration in engine driven compressor systems. That concept has been applied to an internal damper mounted at the auxiliary end of the compressor crankshaft.

This paper provides an overview of the compressor damper, how it works, and where it is mounted inside of the frame. The oil is supplied from the last bearing through a passage in the hollow shaft. This option was originally developed to protect the compressor crankshaft from harmful torsional vibration when operating over a large speed range (e.g., driven by engine or VFD motor). But testing has shown that the damper can also help to reduce lateral vibration of the compressor frame and attached components.

The case study in Section V outlines the torsional analysis procedure used to select a new motor and internal damper for an existing compressor unit. The system was verified to be acceptable through field testing.

The case study in Section IV demonstrates how effective the internal damper was at reducing torsional and lateral vibration. This system had a 16-cylinder, natural gas engine driving a 6-cylinder compressor in pipeline service. After nearly six months of operation, the internal flywheel in the compressor was replaced by an internal damper to improve reliability. While performing the modifications to install the damper, some damage was also found on the original lube oil pump, likely caused by high torsional vibration.

After the damper was installed, the torsional vibration was cut in half at the lube oil pump. The angular deflection in the damper was also confirmed to be below the limit. The internal damper lowered the torsional velocity to a level that no longer caused significant torsional-lateral interaction (TLI). That in turn reduced the compressor frame and cylinder vibration that previously occurred at 9× running speed.

Discharge nozzle pulsation could have been contributing to the previous elevated 5× torsional vibration. An acoustic resonance was measured near 71 Hz. A more restrictive orifice in the discharge flange could reduce the amplitude of the pressure peak and torsional excitation. However, this was not considered necessary to control the torsional vibration once the internal damper had been installed in the compressor.

I. Introduction

Torsional vibration problems have been occurring in reciprocating compressor installations for a long time. This is due in part to the fact that reciprocating equipment produces significantly more torsional excitation than rotating machinery such as turbines, electric motors, centrifugal compressors, and pumps. The purpose of this tutorial is to raise awareness of torsional vibration issues and how compressor dampers are now being utilized to control torsional and lateral vibration.

Torsional vibration involves the twisting of shafts while the machinery is rotating. Excessive torsional vibration can lead to failures of crankshafts, couplings, engine dampers, and compressor oil pumps. These failures typically occur at a 45-degree angle to the shaft axis (see Figure 1). Significant torsional vibration may cross-couple into noticeable lateral vibration in systems containing a gearbox or with a crankshaft [1]. Unfortunately, many torsional vibration problems are not discovered until after a failure has occurred.

Figure 1. Failed Engine Crankshaft without Damper

There are books [2, 3] and numerous technical papers written about torsional vibration, so the phenomenon is generally thought to be well understood. However, torsional vibration problems continue to be an issue with reciprocating [4] and rotating machinery [5]. One reason for this is the mating of equipment traditionally used in non-reciprocating applications (such as variable speed motors) with reciprocating compressors. Other causes of torsional problems include poor performance of engines and compressors, as well as improper application and maintenance of viscous dampers and elastomeric couplings.

Reciprocating machines produce torsional excitation at multiples of running speed (orders or harmonics), which is typically much higher in amplitude than rotating equipment. The GMRC document [6] provides typical values of torque variation for 2-throw, 4-throw, and 6-throw reciprocating compressors. When operated over a wide speed range, it is more likely that one or more of these torque harmonics will excite a torsional natural frequency (TNF) of the system. At resonance, torsional vibration can be amplified to a level that causes failures of shafts, couplings, mechanically driven oil pumps, and engine dampers.

Reciprocating compressors that operate over a wide speed range often require additional damping to limit torsional stresses and oscillations at resonance. For example, a system driven by a VFD motor may require a torsionally soft coupling with damping. Engine manufacturers have been using viscous dampers (filled with silicon fluid) to protect the engine crankshaft over the operating speed range. Although engine dampers affect the whole torsional system, they are not specifically designed for the compressor mode frequencies, nor are they at the ideal location for damping compressor modes.

There are several ways to address torsional resonances within the operating speed range. The most effective method depends on the torsional natural frequency (TNF) of concern and the torsional mode shape. Disc type couplings are relatively stiff and offer no additional damping. The flywheel or detuners need to be properly sized to avoid interference with high excitation harmonics. This resonant response is the main culprit for broken crankshafts, failed disc couplings, and broken motor rotors. Disc couplings are preferred due to lower initial cost and less maintenance later. In some cases, clocking of the compressor relative to the engine may be specified to reduce certain combined harmonics.

If the system is found not to be acceptable over the proposed speed range, the allowable operating speeds would need to be adjusted or an alternative design explored. For example, torsionally soft couplings can be used to isolate the driver from the compressor. Common types of couplings include rubber blocks in compression, elastomer in shear, coil spring, and leaf spring. These result in coupling modes that have high deflection across the coupling, while the driver and compressor shafts effectively oscillate as rigid bodies with very little angular deflection. This approach has been successfully used in the Dual Drive units consisting of engine, motor, and compressor.

Figure 2 shows an example of the first torsional mode shape for a system with a soft coupling containing rubber blocks. Stations 1 - 9 represent the motor core, stations 10 and 11 the rubber type coupling, and stations 12 - 16 a 2-throw compressor. Note the node crossing between stations 10 and 11 indicate the maximum twist is occurring through the coupling, whereas the motor and compressor crankshaft have little angular deflection (straight lines). The rubber coupling provides damping for this first torsional mode as indicated by the predicted amplification factor (AF) of 8.5.

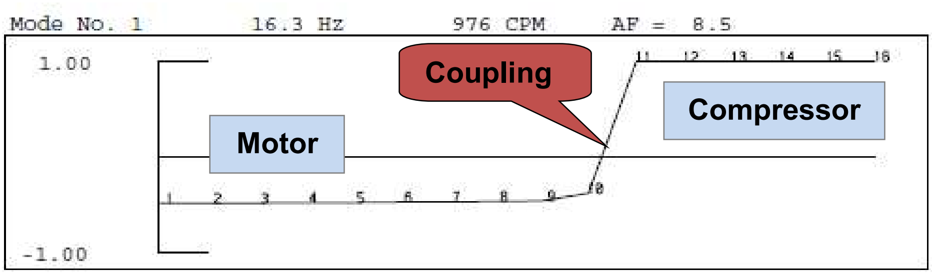

Figure 2. First Torsional Mode with Soft Coupling

Common practice is to tune the first TNF to be at least 70% below the minimum operating speed so that it will not be excited by the compressor. A generous separation margin is recommended due to variability of the torsional stiffness of the rubber, which is natural material and can therefore differ substantially from the catalog value due to load, frequency of vibration, temperature, age hardening, etc. Large flywheels (added inertia) usually accompany torsionally soft couplings to help lower the first TNF.

The electromagnetic (EM) effect [7] should be included in the torsional analysis for motor drives. This is particularly important for systems containing a torsionally soft coupling because the EM effect tends to increase the first TNF, which could then encroach on the minimum operating speed. Note that the EM effect can also be used in calculations that evaluate motor peak torque, sizing, efficiency, and current pulsation.

Some packagers will avoid using torsionally soft couplings because of increased initial cost. Maintenance costs could also be higher because the rubber elements will need to be routinely inspected and replaced. However, with the internal damper option for the compressor, the disc coupling can oftentimes be used even for a compressor system with a wide speed range.

II. Background on Dampers

Two types of dampers are discussed: viscous (untuned) and oil-filled leaf spring (tuned).

Viscous Dampers

Viscous dampers (Houdaille type) are used in engines to allow for operation over a wide speed range. The torsional damping could also help protect the engine crankshaft in case of abnormal operating conditions such as misfire and cylinder pressure imbalance. These dampers are normally intended to protect the engine crankshaft and not necessarily the driven machinery. To be effective, dampers need to be located at a point with high angular velocity, usually near the anti-node of the crankshaft mode. In most cases, this occurs at the front end of the engine.

Figure 3 shows the first two torsional mode shapes for an example system with a 20-cylinder, natural gas engine driving a 4-throw compressor through a disc pack coupling. The engine damper ring and housing are modeled by stations 1 and 2, the engine as stations 3 - 12, and the compressor as stations 15 - 24. Note that it is better to model the damper as two separate pieces instead of using the one-lump simplified assumption.

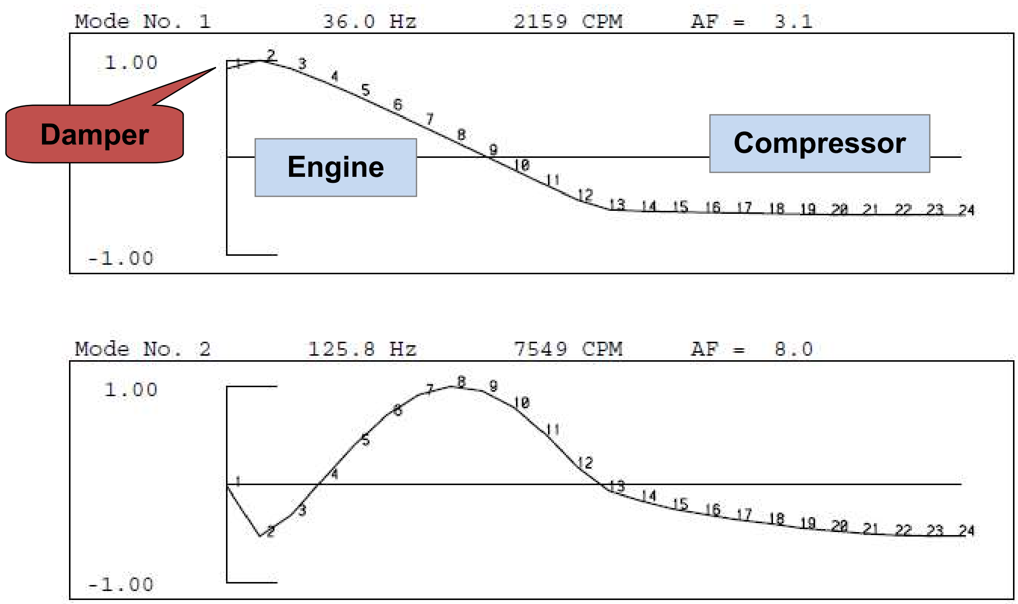

Figure 3. Engine Torsional Modes

The first torsional mode shape shows twisting through the engine crankshaft and one node near station 9. Note that the anti-node is near station 2 where the engine damper housing is located. The predicted amplification factor (AF) is approximately three, which indicates the mode is well damped. The second torsional mode shape shows additional twisting through the engine crankshaft and two nodes. The AF is approximately eight and still has significant added damping.

A viscous damper consists of a flywheel (station 1) that rotates inside the housing, which contains a viscous fluid such as silicon oil. An untuned damper does not contain an internal torsional spring. The shearing motion of the fluid between the internal flywheel and damper housing surfaces dissipates torsional vibration energy as heat. The damping characteristics can be adjusted by changing the internal clearances between the housing and flywheel.

According to Den Hartog [8], the optimum damping, c_{opt}, for maximum energy dissipation is given by the following equation:

$$c_{opt} = I_d \omega$$

where I_d is the inertia of the flywheel inside of the damper and ω is the frequency of oscillation.

Engineering Dynamics Incorporated
16117 University Oak, San Antonio, TX 78249
210-492-9100 www.engdyn.com

Engine manufacturers may offer different sized dampers as options depending on the service or application. For example, an engine driving a reciprocating compressor may require more damping than if it were driving an electric generator at fixed speed. Therefore, it is important to select a damper that will have nearly optimum damping and still not overheat. If a sufficiently large damper is unavailable, then it may be possible to use two dampers in place of one (see Figure 4) to provide the additional damping and heat capacity. This is provided the dampers are located external to the engine case. Some viscous dampers are located inside the engine case, which may not have sufficient room and/or cooling capability.

a) Single Damper **b) Double Damper**

Figure 4. Examples of Viscous Engine Dampers

Viscous dampers typically have limited-service life and require periodic checks. Maintenance and fluid changes are required to restore the damping properties. However, dampers located inside an engine case can be easily overlooked and forgotten. This could result in a reduction or complete loss of damping and protection of the engine. The fourth author has seen several cases where this happened and caused engine crankshaft failures.

Continuous heat absorption reduces the fluid viscosity over time. Simpson Industries (formerly a division of Holset) manufactures dampers and recommends changing them when the fluid viscosity has reduced 50% and the efficiency is approximately 80%. According to Simpson Industries, this occurs after approximately 25,000 to 30,000 hours of service [9]. Superior recommends replacement of dampers every 24,000 to 35,000 hours depending on the engine model [10]. Under extreme temperature conditions, the damper should be replaced more frequently since the silicon fluid will degrade more rapidly. Figure 5 shows how the color changes from clear to dark as the fluid degrades.

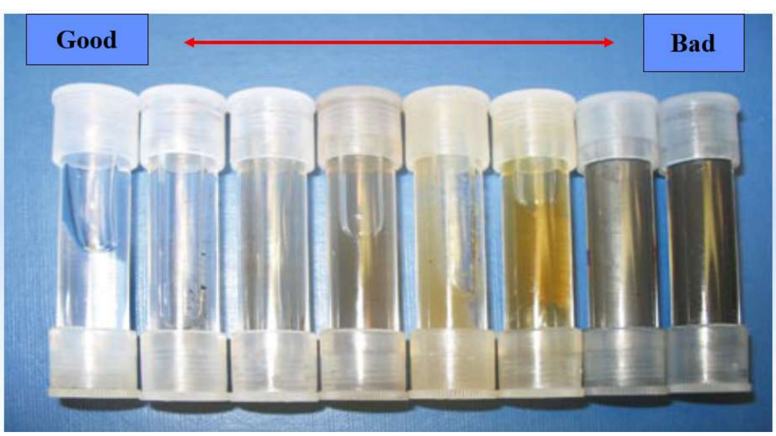

Figure 5. Damper Fluid Degradation [11]

Oil-Filled Leaf Spring Couplings and Dampers

This type of coupling or damper has radially arranged steel leaf springs with constant torsional stiffness and filled with oil. Damping due to oil flow through internal clearances of the device occurs as the leaf springs flex [12]. The oil-filled coupling has low torsional stiffness and high damping and normally requires pressurized oil supply through a hollow shaft. Figure 6 shows an example of a leaf spring damper.

Figure 6. Oil-Filled Damper [13]

The damper is tuned using the steel springs to significantly reduce torsional vibration at resonance and to protect the compressor crankshaft. It provides constant stiffness and high damping throughout its service life. The torsional vibration steel spring damper allows for lower life-cycle cost. By influencing the critical frequency and significantly reducing the torsional amplitudes, the damper can help to attenuate harmonic interferences within the operating speed range.

The internal damper is mounted on the auxiliary end of the compressor crankshaft and consists of a primary and a secondary section. Between these sections, groups of steel leaf spring packs are arranged. These spring packs, together with intermediate pieces and the secondary section, form chambers that are filled with pressurized engine oil. The steel springs are tuned to optimize the TNF of each system. Oil is used to reduce torsional vibrations by hydraulic damping.

Example torsional mode shapes are shown in Figure 7 for a 4-throw compressor driven by an induction motor through a disc pack coupling. The motor core is modeled as stations 1 – 9, the coupling hubs as stations 10 and 11, the compressor crankshaft as stations 12 – 23, and the outer housing of the damper as station 24.

The damper acts as a dynamic absorber and splits the TNF into two frequencies [12]. The first torsional mode shape involves primarily the damper section (lower damper mode). The second torsional mode shape is the first compressor crankshaft mode for which the damper was tuned (upper damper mode). Both of these torsional modes have significant damping (AF ≈ 5) and therefore do not pose a problem for the system.

The third torsional mode shape shows some twisting through the motor core but is primarily the second compressor crankshaft mode. Although there is an anti-node at the auxiliary end of the compressor, the damper is not as effective (AF ≈ 15) because it was tuned for the lower frequency. However, some additional damping is still being provided. Without the damper, the compressor crankshaft mode would likely have a much higher AF of approximately 40.

Induction motor rotors also have unique modes; however, there is relatively less excitation from the motor to excite them. Cracked rotor bars, broken motor shafts, and fan failures are all common failures seen in industry, but these failures are usually associated with the system modes for torsionally rigid couplings. The induction motor rotor should be modeled as multiple stations for improved accuracy.

Engineering Dynamics Incorporated
16117 University Oak, San Antonio, TX 78249
210-492-9100 www.engdyn.com

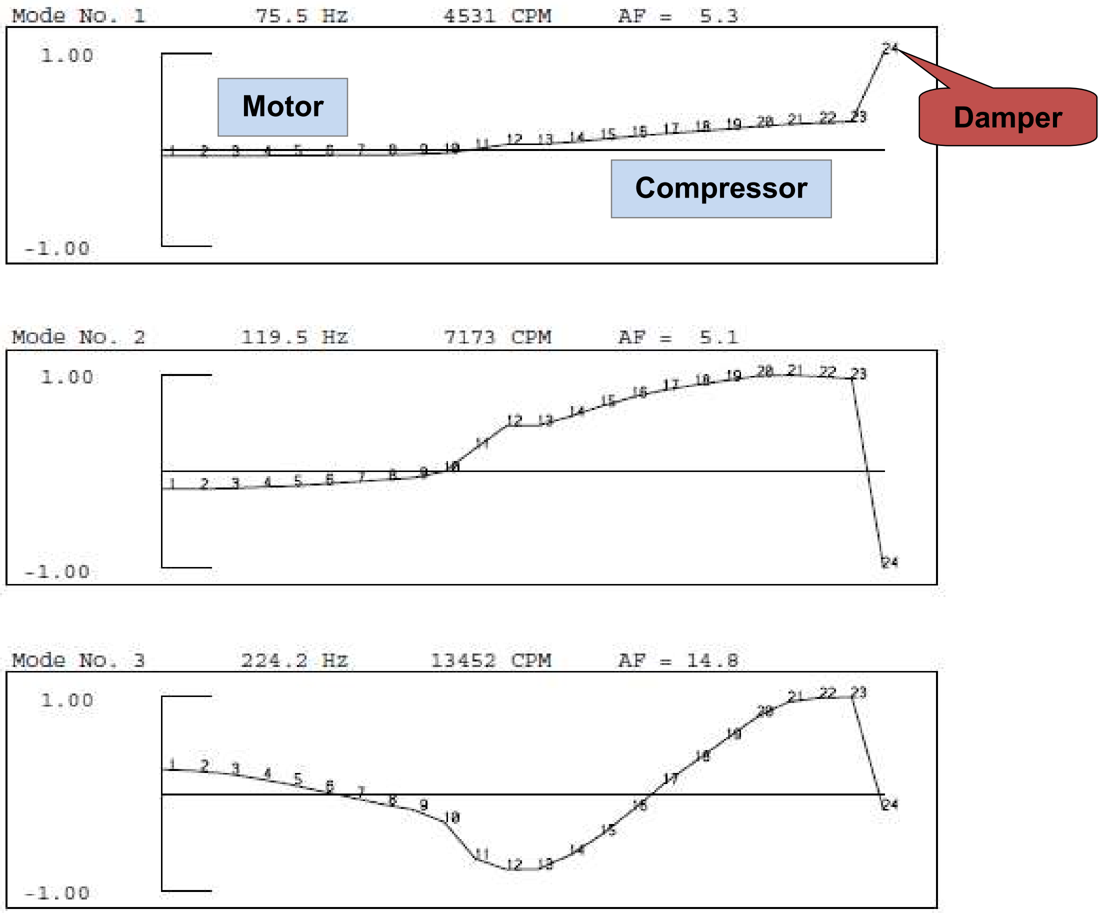

Figure 7. Compressor Torsional Modes with Damper

Since fresh oil is being supplied to the internal damper, the fluid will not degrade as happens over time with silicon fluid in sealed viscous dampers. However, the angular deflection must remain below a safe limit or the internal parts can be damaged. As described in reference paper [14], there was a case where damage indicative of high torsional vibration occurred inside of an oil-filled leaf-spring coupling (similar design to the damper). Damage to the internal components was initially indicated when oil samples showed an increase in copper content.

Compressors can also experience abnormal operating conditions such as valve failures, gas pulsation, and increased torque harmonics at certain load steps. For example, a single-acting compressor cylinder will typically produce higher dynamic torque at 1× running speed and a double-acting cylinder will have higher excitation at 2× running speed. Note that the maximum horsepower case will not necessarily correspond to the maximum torsional excitation. Therefore, a range of operating conditions (pressures, flows, gas mole weights) and load steps should be considered in a torsional analysis.

III. Options for Tuning TNFs

In some cases, it may be necessary to detune a TNF away from the running speed. There are several options for tuning TNFs. The coupling is often a good place to start. The first TNF or other torsional modes involving the coupling can often be tuned by changing the torsional stiffness of the coupling.

Adding inertia to the system tends to lower the TNFs. However, if inertia is added near a node (point of zero oscillation), then it will likely not affect that torsional mode. So based on the mode shapes, the torsional analyst will determine the optimum location for adding inertia, if needed. Figure 8 shows an example 6-throw compressor crankshaft.

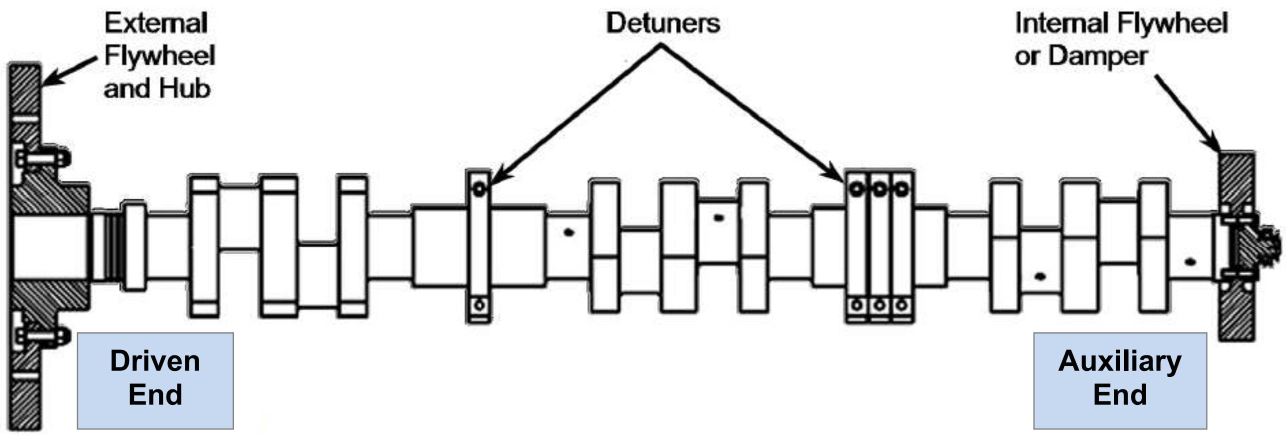

Figure 8. Compressor Crankshaft with Tuning Options

One option is to use an external flywheel or inertia ring added to the coupling hub. However, the overhung weight of the flywheel must not exceed the allowable limit. Depending on the torsional mode shape, another option is to add detuners to the spreader section(s) of the compressor crankshaft. These are sometimes referred to as “donuts” and must be ordered with the compressor serial number for proper fit. A flywheel can also be added to the auxiliary end of some compressors. Since it is located inside of the case, it is referred to as an internal flywheel.

If optioned, an internal damper could be installed at the auxiliary end of some compressor models where the internal flywheel is normally located. The internal damper is selected when a TNF cannot be adequately tuned away from the operating speed range or the operating conditions are severe. Not all of these options would be used at the same time and some compressor models may not accommodate certain options.

IV. Torsional Limits

Torsional limits have been developed over the years based on analyses, testing, and experience. For a reliable compressor system, several items should be checked in the analysis stage:

- Stresses in the engine or motor shaft.
- Oscillation and heat buildup in the damper (engine or compressor).
- Adequate service factor (SF) for the coupling considering rough operation of reciprocating machinery.
- Alternating torque and peak torque in the coupling.
- Heat load for a coupling with rubber element(s).
- Allowable alternating torque for the compressor stub shaft and other sections. The limits are based on the crankshaft material and include appropriate factors for design safety, endurance limit, stress concentrations, and specific shaft diameters.
- The speed variation on the auxiliary end should also remain below allowable values to prevent damage to the oil pump. The limit depends on the type of chain drive (single or double) and the frequency of vibration.
- It is recommended to limit the speed variation on the auxiliary end of the compressor crankshaft as low as possible to avoid torsional-lateral interaction [1].
- It is also recommended to check that the oscillation of the motor rotor is not excessive, which could cause problems with current pulsation, cooling fan failures, etc. For reference, the allowable speed fluctuation of a synchronous motor [15] should normally be limited to 1.5% p-p, which equates to torsional oscillation of approximately 0.9 deg p-p. Note that an induction motor may have a different limit, but this limit is usually not provided by the manufacturer, so it is up to the torsional analyst to use sound judgement based on experience when evaluating reciprocating compressor systems.

The following examples show torsional failures that occurred in compressor systems before the internal damper was available as an option.

Example of Failed Oil Pump

In the first example, a natural gas engine was driving a 6-throw compressor. Although the engine had a viscous damper and the system had a rubber coupling in shear, failures were still being experienced on the compressor. Figure 9 shows the failed lube oil pump.

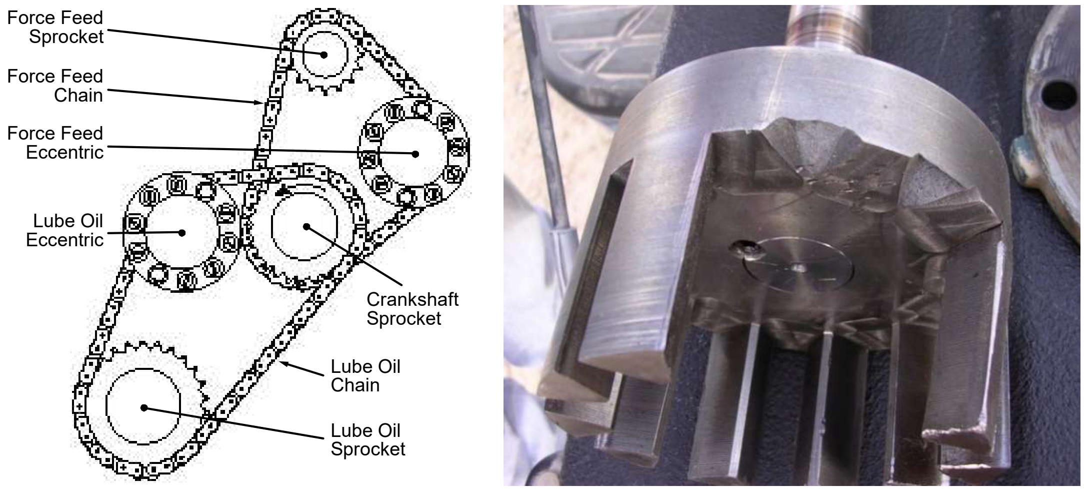

Figure 9. Lube Oil Pump Diagram (left) and Failed Part (right)

This model of compressor was an earlier version that had a single-chain driven oil pump. When the limit was exceeded, the chain and/or oil pump would break. It was determined that this unit was improperly operated at a condition outside of the original design point. Some newer compressor models have a dual-chain drive for the oil pump with a higher allowable limit.

Example of Failed Motor

In this second example, an induction motor was driving a 6-throw compressor through a disc coupling. The motor was rated for 4,200 HP at constant speed of 895 RPM. The skid mounted natural gas compressor was configured as a single-stage unit and could operate at seven load steps using valve unloaders on the head-end of each cylinder.

After being in service for approximately one year, a motor shaft failure occurred. The failure consisted of spiral cracks in the shaft and in the armature support webs (spider), as shown in Figure 10. The cracks occurred at a 45-degree angle to the shaft axis, which is indicative of torsional fatigue. The end user thought that the cracks might have been initiated at a stress riser caused by poor welds at the shaft-spider interface.

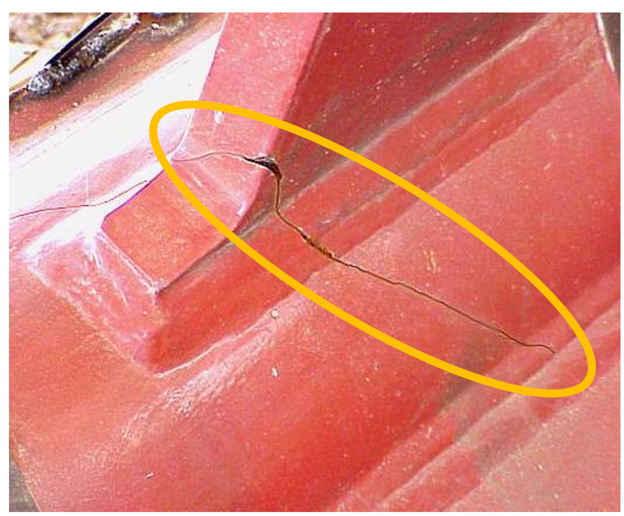

Figure 10. Failure of Motor Shaft

After the motor shaft was replaced, the unit operated for approximately three more months before experiencing loose coupling bolts. Super-nuts were added to prevent loosening of components and reduce maintenance. However, there was concern that this was only treating a symptom and not addressing the underlying torsional vibration problem.

Field measurements showed that the first TNF was coincident with 4× running speed (60 Hz). Because there was an insufficient separation margin, the overall alternating torque in the shaft was amplified and reached 250% of the full load torque (FLT) during operation at certain load steps. The shaft stresses were excessive, and the unit had to be taken out of service until modifications could be made. For comparison, GMRC [6] recommends that a motor driving a 6-throw compressor be designed for mean torque of 100% rated torque ± cyclic torque of 150% rated torque.

To detune the TNF from the 4th compressor order, five internal flywheels (sometimes referred to as "donuts") were recommended between throws 4 and 5. Follow-up field testing confirmed that the first TNF was lowered to 55 Hz as predicted. The recommended modification provided an adequate separation margin from the 3rd and 4th compressor harmonics for this constant speed unit and reduced the alternating torque.

There was also an acoustical resonance at 4× that increased the gas pulsations in the compressor cylinders, which resulted in increased excitation at that compressor harmonic. Cylinder flange orifice plates were installed to attenuate pressure pulsation at the nozzle acoustic resonance. This also helped to reduce torsional excitation.

V. Case Study – Existing Compressor Unit Needed New Motor

In this case study, the end user needed to retrofit an existing compressor unit with a new motor. The compressor unit had a history of coupling and motor failures since being commissioned. Any time a major change is made to a system, it is recommended to perform a torsional analysis. Two possible replacement motors were considered from different manufacturers. Calculations were made for both options, which determined either should be acceptable from a torsional standpoint. Therefore, the end user made the final selection based on pricing and availability of the motor.

This unit consists of an Ariel KBZ/6 reciprocating compressor in four-stage service. The components of the system are listed in Table 1. Based on the results of the torsional vibration analysis (TVA), using an internal compressor damper was the best option for maximizing reliability. There are actually two dampers (model 41 and 51) available for the KBZ/6. After analyzing both dampers, the smaller size was selected based on the predicted separation margins.

Table 1. System Description

Driver:	Induction Motor Rated 5,500 HP at 890 RPM, 8 Poles
Coupling:	TB Woods Size 92HT Keyless Straight Bore Motor Hub Est. Service Factor = 3.95
Compressor:	Ariel KBZ/6 Reciprocating Compressor 1st Stage: 26-1/2'' Bore (Throws 1, 3 & 5) 2nd Stage: 26-1/2'' Bore (Throw 6) 3rd Stage: 17-7/8'' Bore (Throw 2) 4th Stage: 12-1/2'' Bore (Throw 4) All Cylinders Are Double-Acting
Recommendations:	Ariel G-28705, Damper Part No. D-8520, Internal Damper 41 Installed on Auxiliary End of Compressor

A mass-elastic model of the system was developed. The motor core was represented by nine stations. The coupling was modeled as two stations. The mass-elastic model provided by the manufacturer was used for the compressor. The outer member of the internal damper was modeled as a separate station. In addition, an electric-magnetic (EM) spring [7] was applied from the center of the motor core to ground. The equivalent EM spring stiffness was estimated to be 5.5 million in-lb/rad for this motor.

Engineering Dynamics Incorporated
16117 University Oak, San Antonio, TX 78249
210-492-9100 www.engdyn.com

The calculated torsional natural frequencies are listed in Table 2. The torsional mode shapes are shown in Figure 11.

Table 2: Calculated TNFs with Compressor Damper

Mode Number	Frequency		Multiple of Running Speed
	Hz	CPM	
1	7.2	432	0.49×
2	67.7	4,060	4.56×
3	89.7	5,380	6.04×
4	137	8,191	9.20×

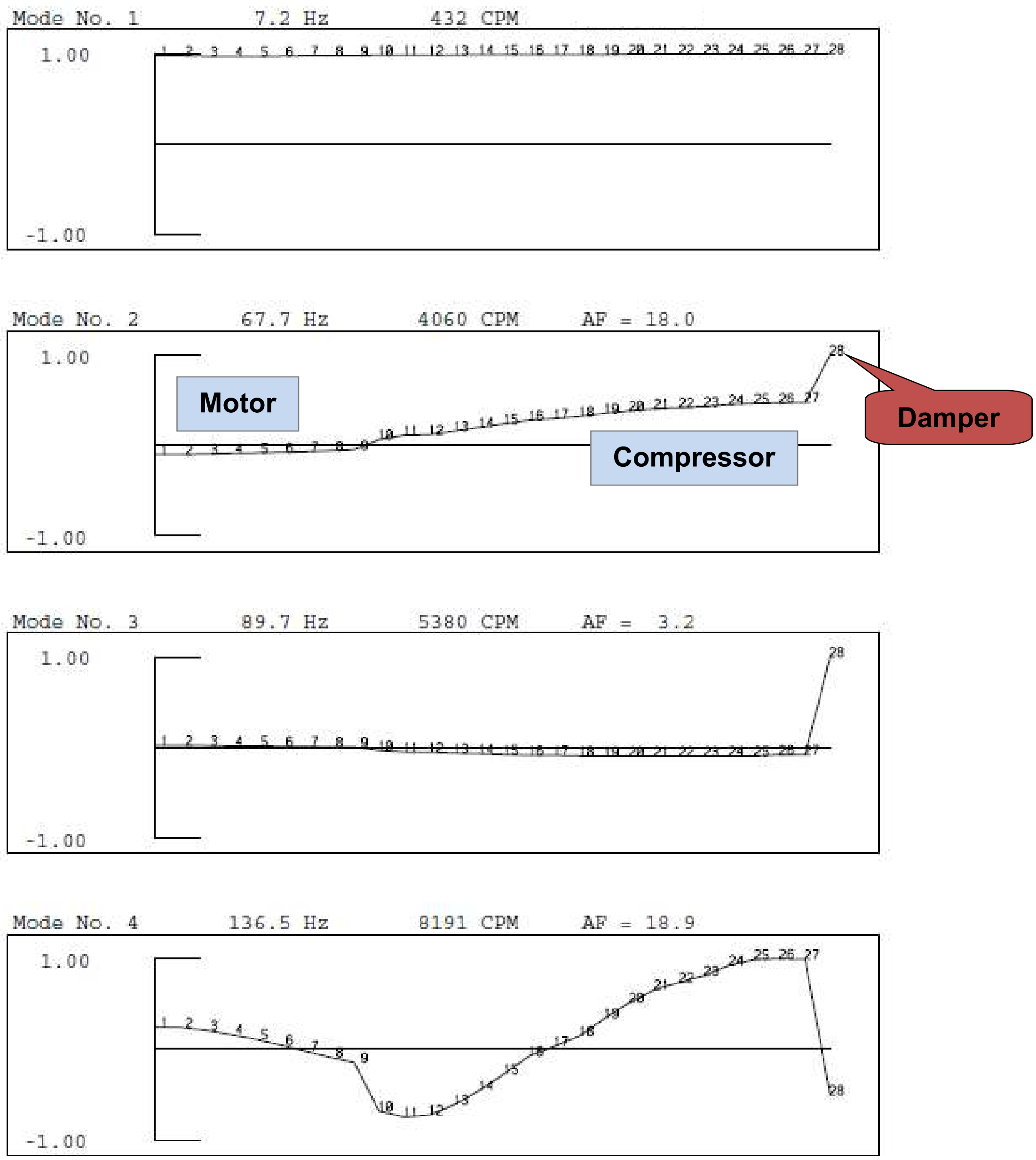

Figure 11. Predicted Torsional Modes with Compressor Internal Damper

The first mode is predicted at 432 CPM and is related to the EM spring effect from the motor. This mode involves rigid body motion and does not cause any alternating torque or stress in the system. Since it is predicted well below running speed, it should not pose a problem for the system.

The second torsional mode at 4,060 CPM occurs at 4.6× running speed and primarily involves twist through the coupling and compressor shaft, with the maximum oscillation occurring at the auxiliary end of the compressor. The 5× compressor harmonic intersects this mode at 812 RPM, which is 9% below the operating speed. The 4× compressor harmonic intersects this mode at 1,015 RPM, which is 14% above the operating speed. Note this torsional mode was tuned between the 4th and 5th compressor harmonics by selecting the smaller damper (model 41) instead of the larger damper (model 51) for satisfactory separation margins.

The third torsional mode at 5,380 CPM is due to the compressor internal damper. Since this mode only involves the outer member of the damper, it is well damped (AF ≈3) and does not pose a problem.

The fourth torsional mode at 8,191 CPM occurs at 9.2× running speed and again involves twist of the coupling and compressor, with the maximum oscillation occurring at the auxiliary end of the compressor. Although it is undesirable to have a TNF near an integer multiple of running speed, 9× is not expected to be a strong compressor harmonic during normal operation.

The interference or Campbell diagram plotted in Figure 12 compares the calculated TNFs with excitation frequencies produced by the motor and compressor. Intersections of the TNFs with the excitation frequencies define the torsional resonances or critical speeds of the system. For example, the torsional resonance indicated by the red circle where the 4th TNF is intersected by 9× running speed.

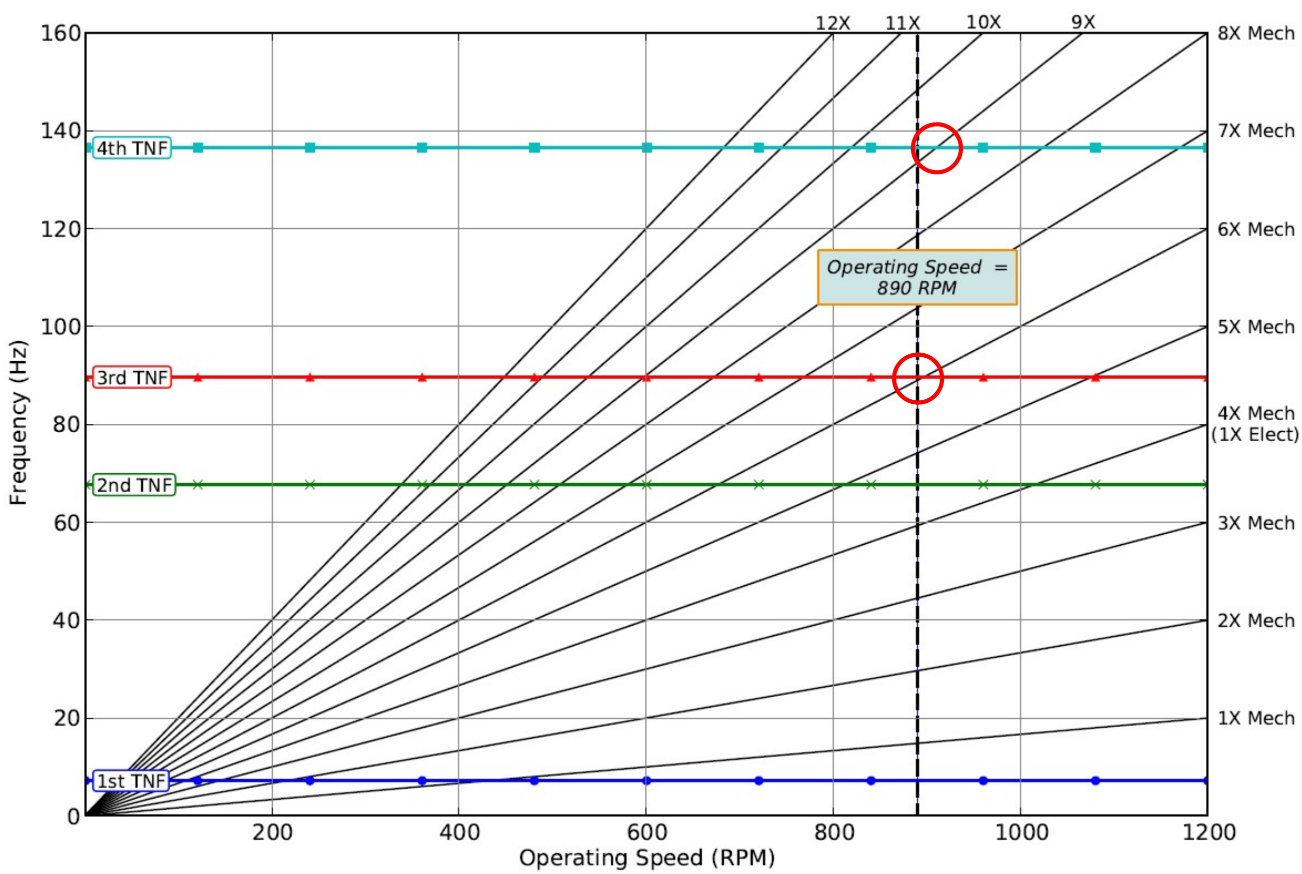

Figure 12. Campbell Diagram

Torsional resonances occurring near the operating speed could cause high alternating shaft stresses, coupling alternating torques, and torsional oscillations in the system. A separation margin (SM) accounts for variation between the calculated and actual values, which can differ due to inaccuracies in the supplied mass-elastic data. The American Petroleum Institute (API Standard 618 [16]) recommends a SM of at least 10% when possible. Since the SM was not met for all torsional resonances, forced response calculations are needed.

Allowable Limits

Unless specific allowable limits are provided by the manufacturer, the alternating shear stress should be compared to the estimated endurance limit of the shaft material. The ASME method [17] calculates the fatigue (endurance) limit based on the ultimate tensile strength (UTS). Fatigue modifying factors are then applied to account for surface condition, size, and reliability. The effects of mean stress are subsequently considered using a Goodman diagram [18].

For example, a motor shaft with an ultimate tensile strength (UTS) of 109 ksi would have an estimated allowable alternating shear stress of approximately 5,740 psi zero-peak (0-p). Including a safety factor (SF) of two allows for potential variations in the shaft properties, mass-elastic data, and the compressor torque-effort. Note that variations in torque-effort from the compressor could occur due to an unexpected change in operating conditions, gas pulsation in the cylinders, and/or unexpected valve losses or failures.

This allowable limit should be compared to intensified stresses. Stress concentration factors (SCFs) were included in the stress calculations to account for geometric discontinuities in the shaft sections. Peterson's Stress Concentration Factors [19] was used to determine the appropriate value. To minimize the SCF of the motor and compressor shafts, the keyless coupling hubs are used with an interference fit. For the motor, a SCF of 5 was assumed under the core area and a SCF of two was used for the stepped shaft sections.

Most coupling manufacturers will provide the allowable torque values. The rated torque for this coupling was 1,400,000 in-lbs. The full load torque (FLT) of this unit is 354,073 in-lbs (based on 5,000 HP at 890 RPM), which corresponds to a coupling service factor of 3.95. The allowable vibratory torque is 748,927 in-lb 0-p and accounts for the expected mean torque. The coupling service factor should always be verified to be sufficient for a reciprocating compressor application when performing the torsional analysis

Ariel provides allowable limits for their compressors in ER-83 [20]. The compressor crankshaft is upgraded AISI 4340 carbon steel. The vibratory torque limit for this crankshaft is 765,200 in-lb 0-p. Stress calculations are not required for the compressor.

The torsional velocity limit for the auxiliary end of the compressor depends on the chain drive system (single or dual chain). The KBZ/6 utilizes a dual-chain drive for the oil pump. The velocity limit at any harmonic is 40 RPM 0-p. The overall velocity limit is 55 RPM 0-p. Ariel also provides a guideline for torsional-lateral interaction (TLI), which decreases with frequency as shown in Figure 13.

For G-28705, Damper Part No. D-8520, Model Internal Damper 41 for KBZ/6 the permissible thermal load is 3,700 W. This should also be checked when performing the TVA.

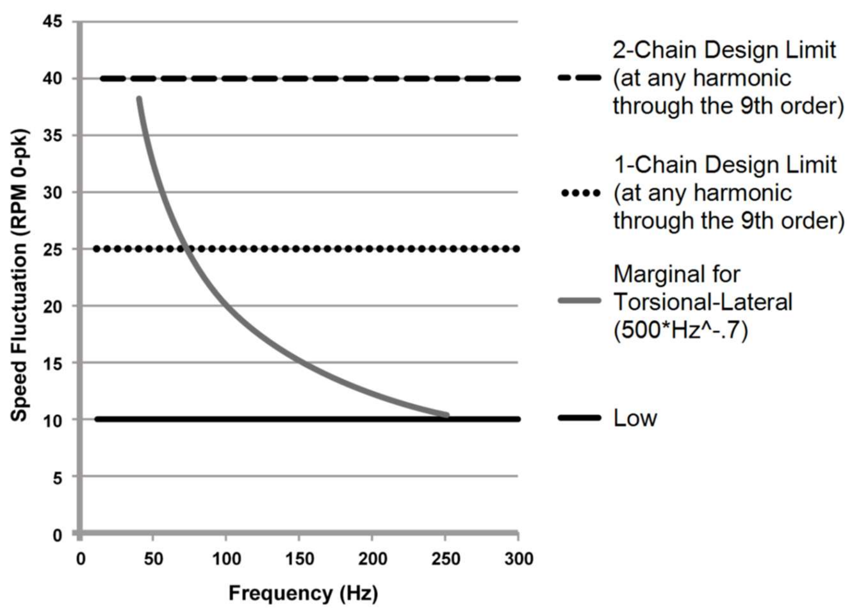

Figure 13. Allowable Speed Fluctuation at Auxiliary End of Compressor

Field Verification

Torsional vibration testing and a general vibration survey of the compressor unit with the new motor and internal damper were performed. A strain gage telemetry system was used to measure transmitted and alternating torque at the coupling. An encoder was used to measure angular oscillation at the auxiliary end of the compressor crankshaft.

Torque data were obtained during an aborted start and shutdown. As shown in Figure 14, the 2nd TNF was 70 Hz and the 4th TNF was 140 Hz. These measured frequencies were only slightly higher than computed in the torsional analysis, indicating good correlation.

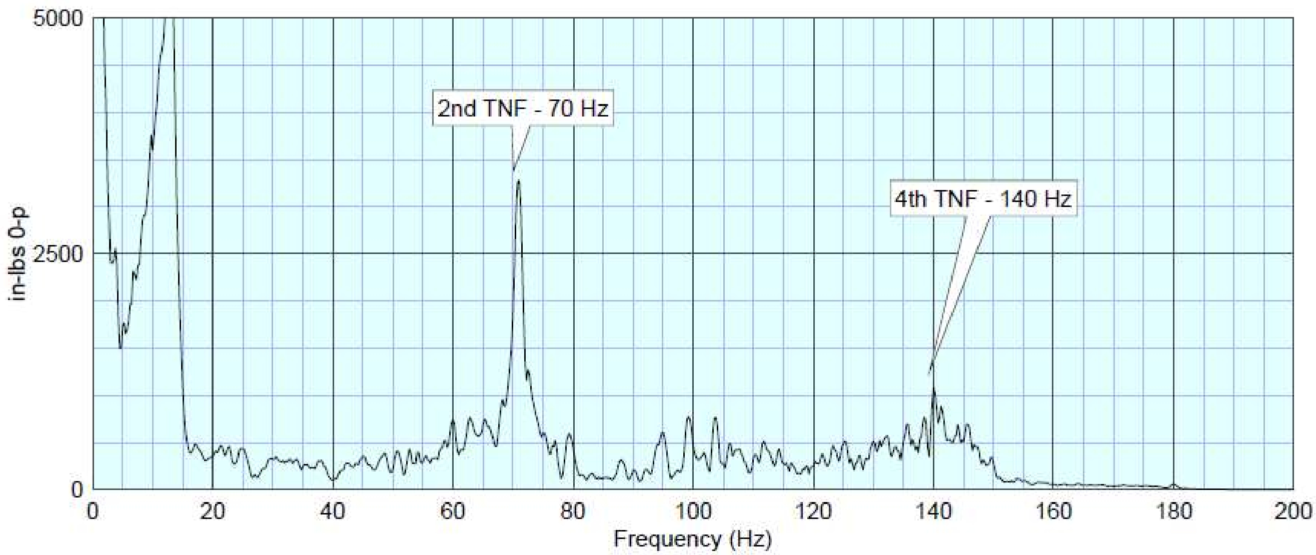

Figure 14. Measured TNFs of Motor-Compressor System

The speed fluctuation at the auxiliary end of the compressor was determined to be acceptable, as shown in Figure 15. Note that the values were also well below the TLI line.

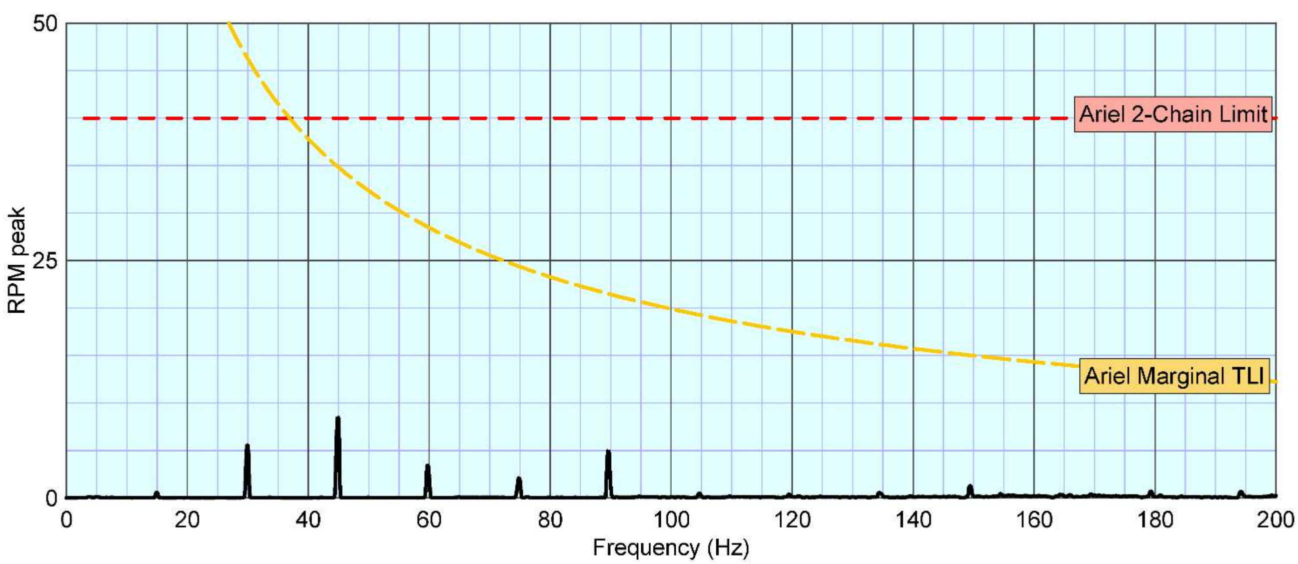

Figure 15. Measured Speed Fluctuation

Figure 16 shows that the compressor frame vibration was also acceptable. For reference, up to 0.5 ips 0-p is allowed for this compressor model / frame. Measured frame vibration levels were less than 0.3 ips 0-p.

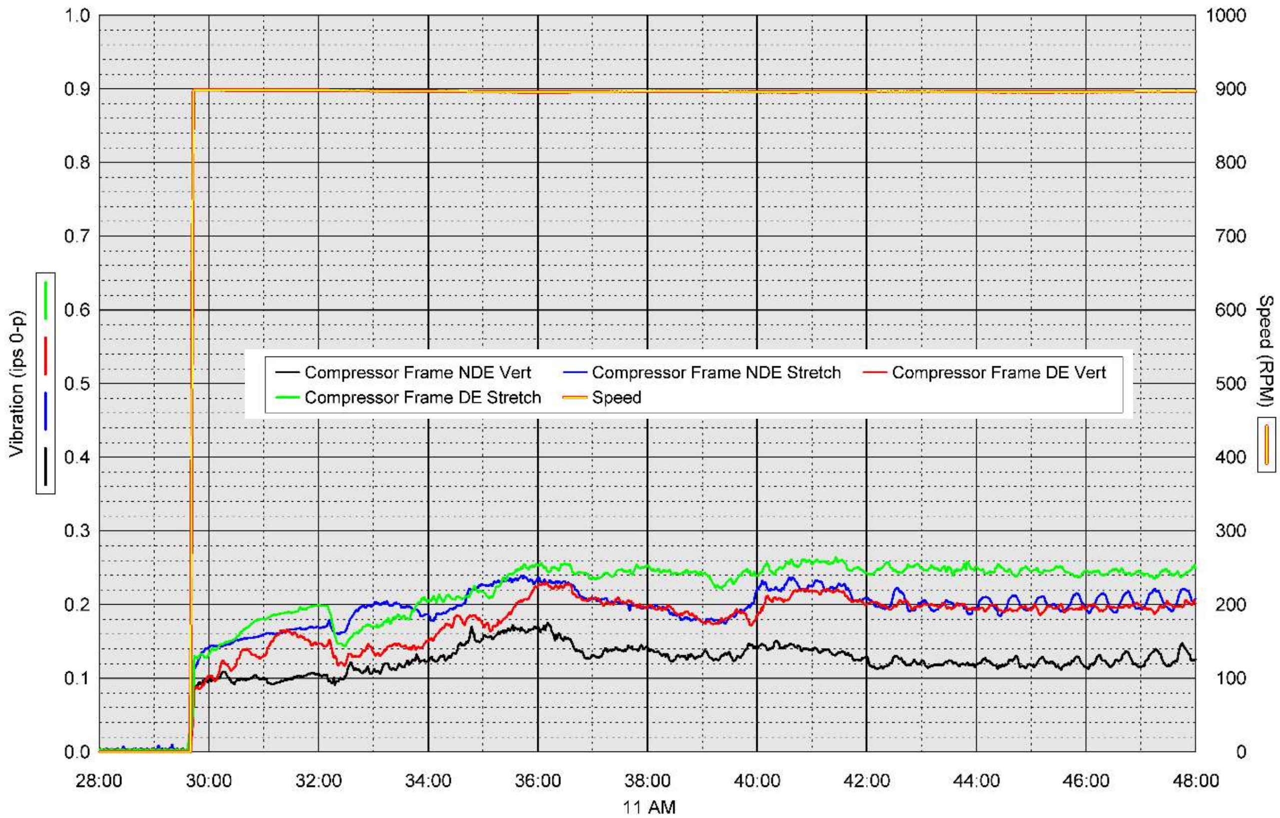

Figure 16. Measured Compressor Frame Vibration

The lateral vibration of the motor bearing housings was somewhat elevated, which can happen when driving reciprocating compressors as discussed in reference paper [21]. There may have been a structural natural frequency, not related to torsional vibration. A summary of guidelines and recommendations for motors driving reciprocating compressors is provided in the Appendix.

Summary

A steady-state TVA was performed to evaluate the system. Torsional excitation is relatively low from the motor compared with torque excitation from the compressor. The compressor has nine load cases, all of which were analyzed.

Although this compressor has all cylinders double-acting (DA), two additional cases for compressor valve failure were also analyzed. This was accomplished by using the maximum horsepower case and then single-acting (SA) throw 5 or 6 to simulate possible valve failure. However, it is recommended to routinely check performance to prevent the compressor from operating with failed valves.

Based on the end user's requirement to have a very reliable system and the results of the TVA, it was recommended to use the Ariel damper (G-28705, Damper Part No. D-8520, Internal Damper 41). The calculated alternating shear stresses, coupling and compressor vibratory torque, and compressor oscillations were predicted to be below the allowable levels for all operating conditions.

Field tests were conducted with the new motor, and the measured results confirmed that the TNFs and torsional response of the compressor system were acceptable. This case study demonstrates the successful implementation of the compressor internal damper.

VI. Case Study – Engine Driven Compressor Unit

System Description

This compressor unit is in single-stage, natural gas service as shown in Figure 17. It consists of Caterpillar 3616 engine, Rexnord CMR 925 coupling, and an Ariel JGC/6 compressor. All six of the compressor cylinders have a 12.5-inch bore. The operating speed range is 850 to 1,000 RPM. There are ten load steps (0 – 9) with HEVU's on throws 1, 3, 5 and HE FVCP's on all throws.


Figure 17. Engine Driven Compressor Unit

Background

Previous field testing identified the second TNF near 81 Hz. Speed fluctuation at the auxiliary end of the compressor crankshaft was just below the limit for a two-chain driven oil pump and of concern. In addition, the compressor frame and cylinder vibration were elevated in the stretch direction. The TVA was revised to include the Ariel damper D-8520. It was recommended to replace the internal flywheel with a damper as shown in Figure 18.


Figure 18. Internal Flywheel (left) and Internal Damper (right)

Torsional Lateral Interaction (TLI) can cause elevated compressor frame and cylinder vibration. TLI primarily occurs at ±1 order about the torsional vibration harmonic (5× in this case), which would be 4× and 6× running speed. However, there can also be TLI at ±1 order at 2× the torsional vibration order, which would be 9× and 11× running speed. The damper should protect the compressor oil pump and help to reduce the compressor frame and cylinder vibration occurring at 9× running speed.

Torsional Guidelines

The allowable vibratory torque and speed fluctuation of the compressor crankshaft were provided by the manufacturer. They also provided an allowable limit for the damper. For G-28705, Damper Part No. D-8520, Model Internal Damper 41 for JGC/6, the permissible thermal load is 3,700 W. The permissible continuous deflection is 11.2 mRad 0-p. This is measured between the damper housing and the compressor crankshaft.

The allowable velocity is dependent upon the chain drive system (single or dual). This compressor utilizes a dual-chain drive for the oil pump. Therefore, the velocity limit at any harmonic is 40 RPM 0-p and the overall velocity limit is 55 RPM 0-p. The compressor manufacturer also provides a guideline for torsional-lateral interaction (TLI), which decreases with frequency as was shown in Figure 13.

Torsional Testing

Before starting the field test, it was requested that phasing be checked between the engine and the compressor crankshafts. The outer dead center (ODC) was verified for compressor cylinder 1 by removing the crosshead cover. Next, from the engine flywheel it was found that engine cylinder 1 was within two teeth of top dead center (TDC).

Testing consisted of several speed runs of the unit as well as holding at a given speed for a vibration survey. At each compressor load step 0 - 9, speed sweeps were performed between the minimum (750 RPM) and maximum (1,000 RPM) speeds. The speed of the unit was slowly adjusted at the control panel in increments of 10 RPM.

The full load condition (LS 9) refers to all cylinders double-acting (DA) and all pockets closed. The minimum load condition (LS 0) refers to all pockets open and cylinders 1, 3, and 5 single-acting (head ends unloaded). The suction and discharge pressures were constant (Ps ≈ 700 psig and Pd ≈ 900 psig) during the speed sweeps.

Torsional vibration was measured using an encoder installed at the auxiliary end of the compressor. For the comparison, the previous values without the damper are shown in Figure 19. When operating near 974 RPM, the speed fluctuation at the auxiliary end of the compressor crankshaft reached 38 RPM 0-p. This was just below the Ariel limit of 40 RPM 0-p for a two-chain driven oil pump, and therefore of concern.

Figure 20 shows the measured speed fluctuations during a full load (LS 9) speed run of the unit with the new internal damper. The new values are well below 40 RPM 0-p, which is the speed fluctuation limit for this JGC/6 compressor. The measured response during full load operation was less than 20 RPM 0-p due to 3× running speed. There were also low amplitude torsional peaks near 80 Hz and 110 Hz. With the compressor internal damper, measured levels were below the TLI guideline (23 RPM 0-p at 81 Hz).

Note that the lower compressor orders (1× - 3×) remained the same since these are primarily forced and not amplified by a TNF. However, the peak at 5× running speed was greatly reduced in amplitude (from 38 to only 5 RPM 0-p) after installing the compressor internal damper.

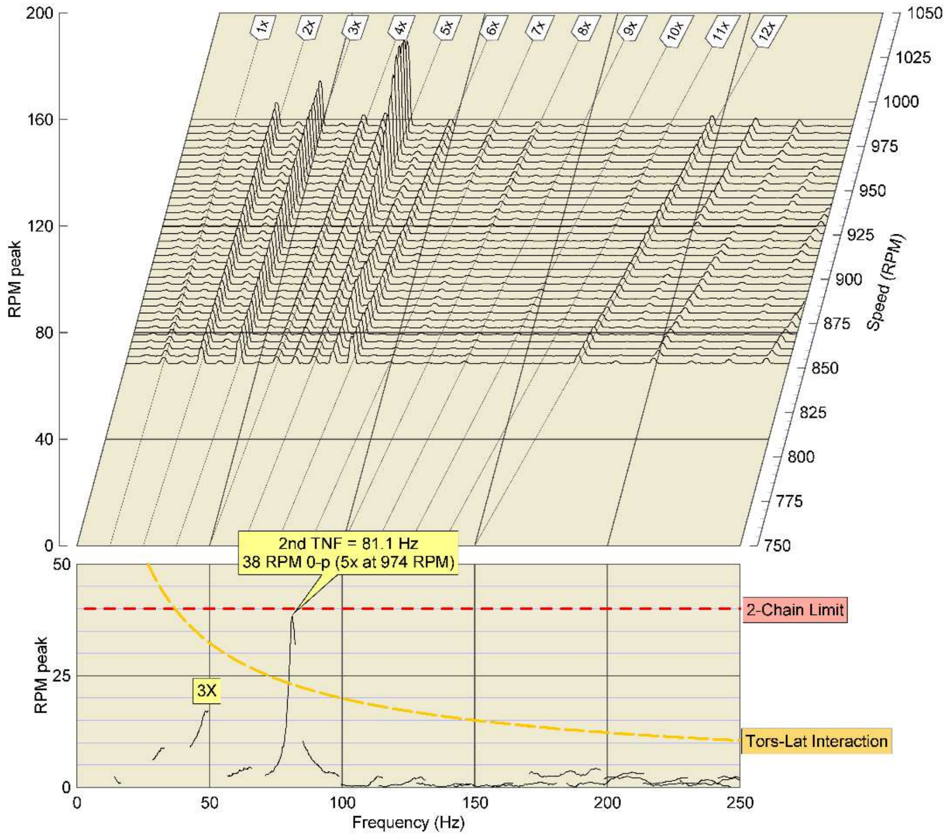

Figure 19. Speed Fluctuation at Auxiliary End without Damper

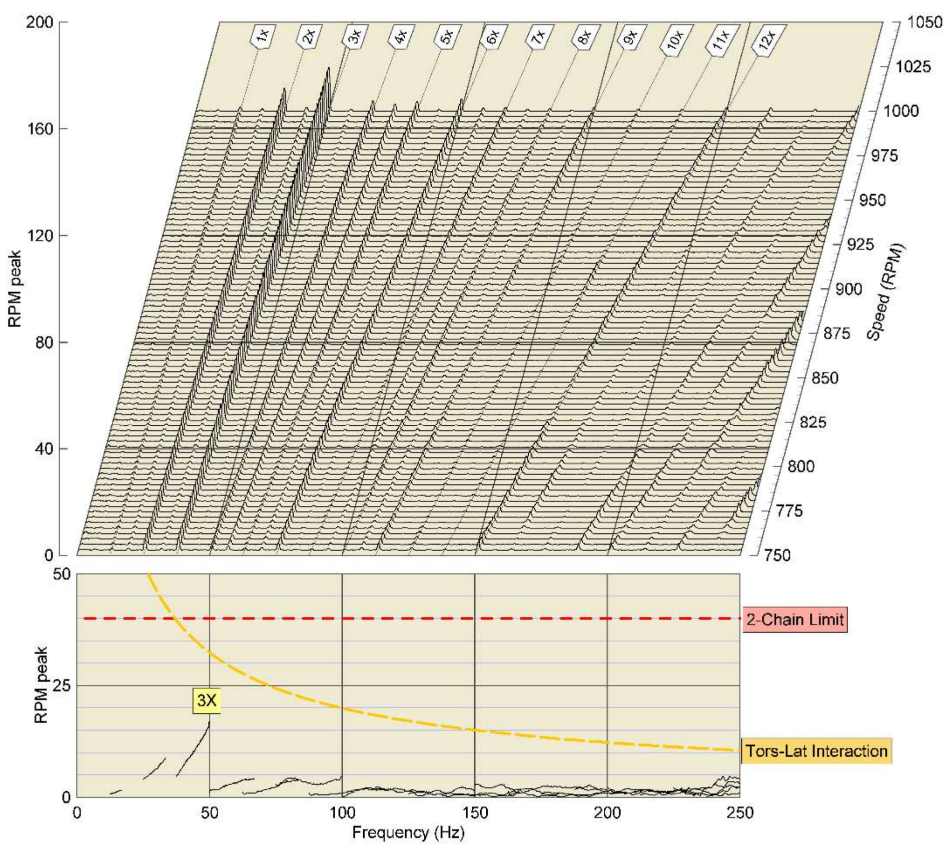

Figure 20. Speed Fluctuation at Auxiliary End with Damper

Engineering Dynamics Incorporated
16117 University Oak, San Antonio, TX 78249
210-492-9100 www.engdyn.com

The continuous angular deflection was checked between the internal damper housing and the compressor crankshaft. This was accomplished using a shaft encoder on the auxiliary end of the compressor crankshaft and an Ariel damper probe as shown in Figure 21. The Hilbert Transform [22] can be used to convert the pulses to angular velocity and then integrated to angular displacement. Other measurement methods are also available such as using a counter-card for the pulses. Subtracting the position of the damper housing from the crankshaft end determines the instantaneous deflection across the damper leaf springs.

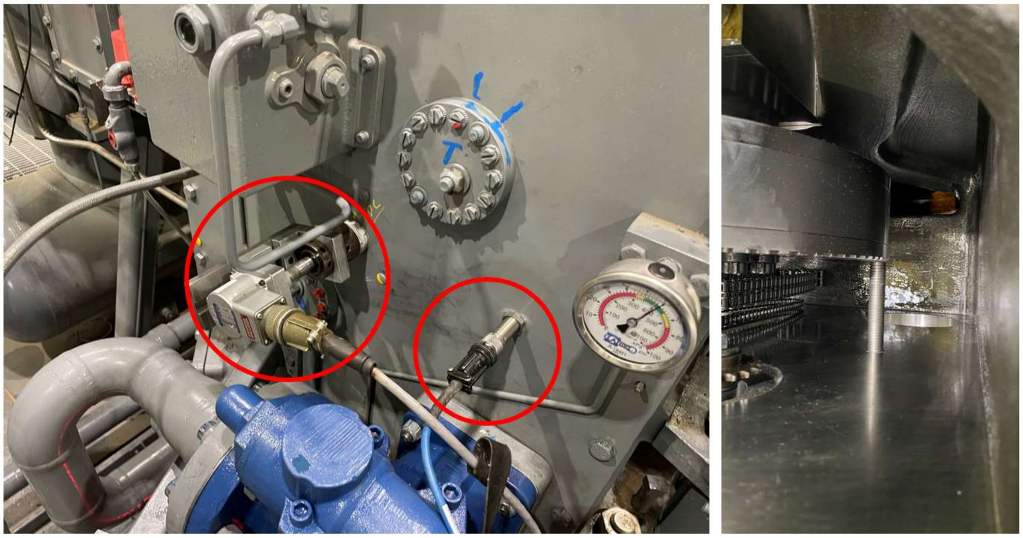

Figure 21. Shaft Encoder (left) and Damper Probe (right)

The measured levels were trended during operation through all compressor load steps. Figure 22 shows that the damper remained below the allowable limit of 11.2 mRad 0-p. Therefore, the oil pump and internal damper were confirmed to be acceptable.

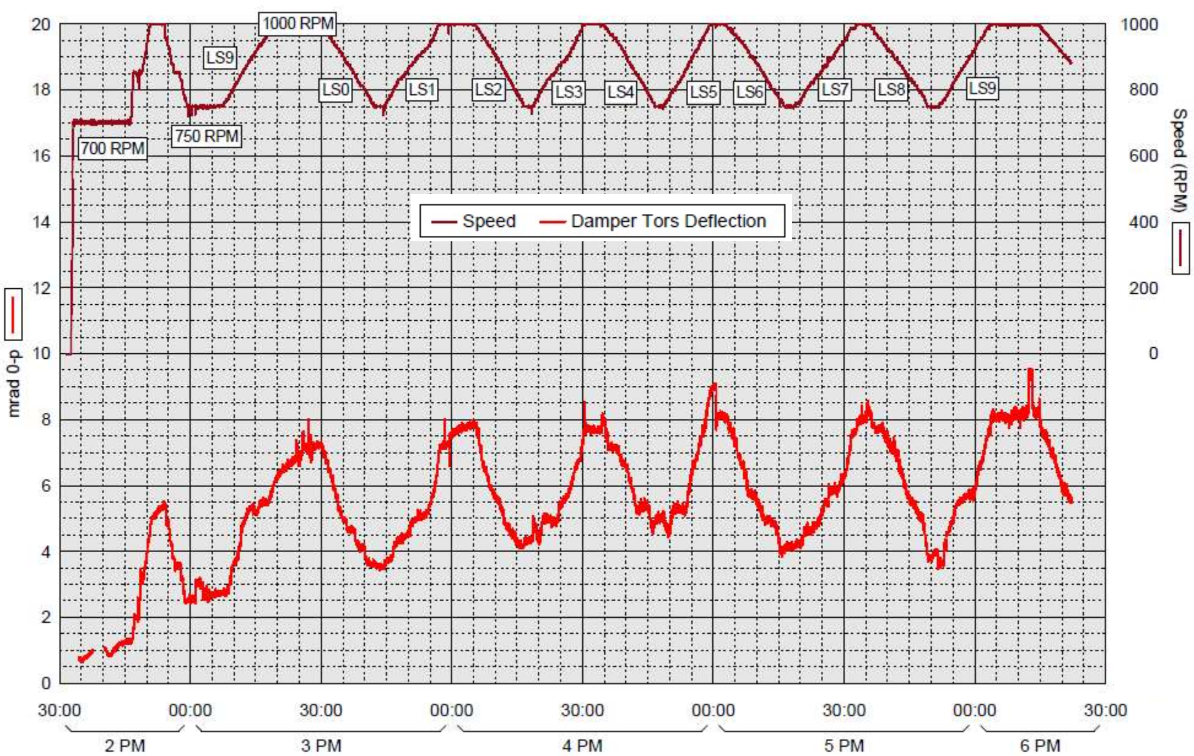

Figure 22. Trend Plot of Damper Torsional Deflection

Engineering Dynamics Incorporated
16117 University Oak, San Antonio, TX 78249
210-492-9100 www.engdyn.com

Pressure Pulsation

Previously, it was found that the measured torsional amplitude at 5× running speed was about an order of magnitude higher than that predicted by a third-party TVA. It was suspected that this discrepancy could be due to a nozzle resonance, which would not be accounted for in the ideal compressor torque-effort values.

During the previous test, pressure pulsation data were measured in the process piping, but not at the compressor cylinders. During this test, additional pulsation measurements were taken in compressor cylinder 1. Figure 23 shows the pulsation in the discharge side of compressor cylinder 1 during operation at LS 2. The discharge nozzle pulsation could have been contributing to the elevated 5× torsional vibration as shown by the peak in pulsation near 71 Hz.

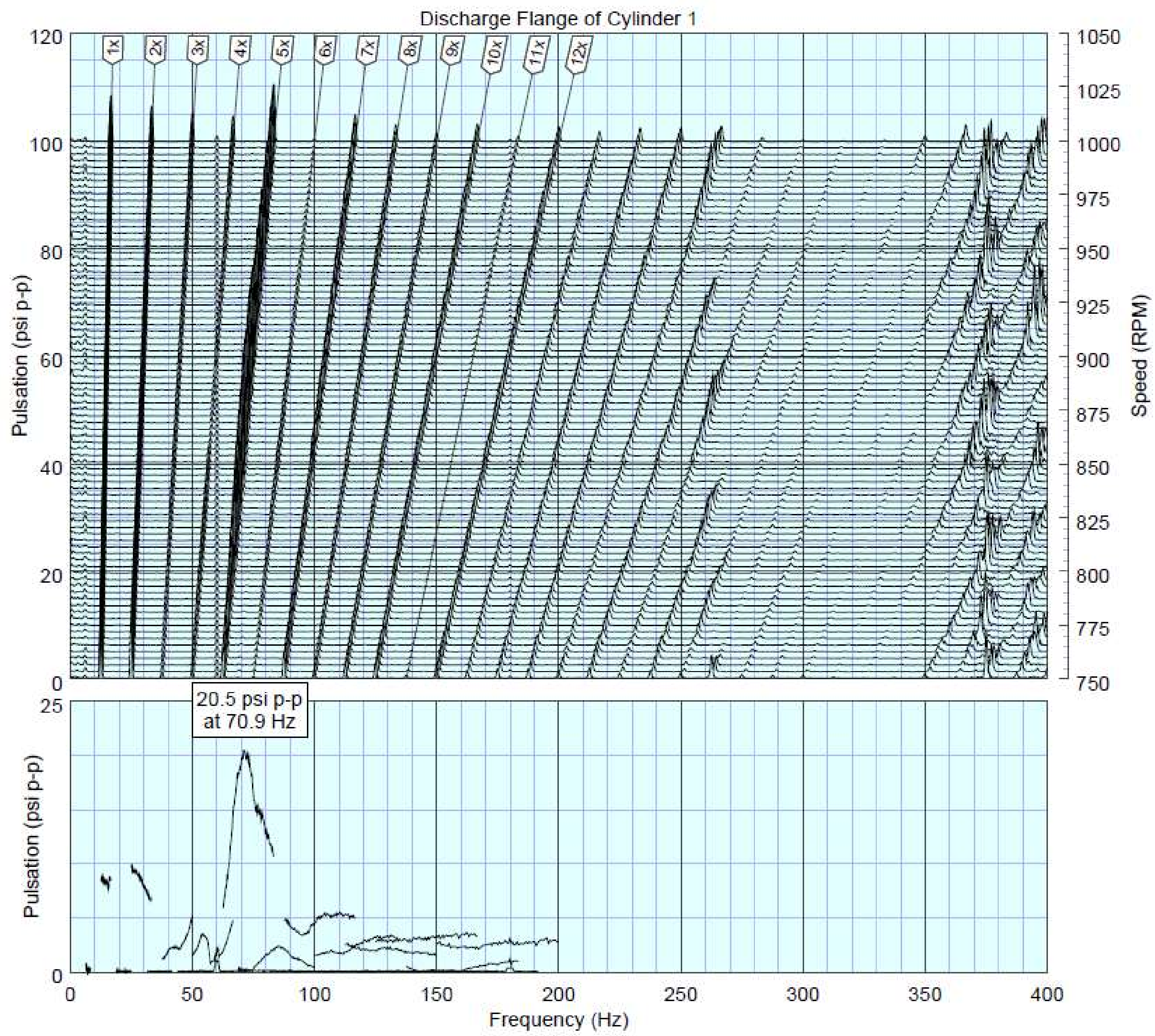

Figure 23. Pressure Pulsation in Discharge of Compressor Cylinder 1

Orifice plates are already installed at the cylinder flanges to help attenuate these resonances. The calculated pressure drop through the discharge cylinder flange orifice is approximately 0.3% of line pressure. This is typical for cylinder flange orifice plates in natural gas service and would be expected to adequately dampen the nozzle resonance. A more restrictive orifice in the discharge flange could help reduce the amplitude of this peak, but may not be required to control torsional vibration since the internal damper was so effective.

Compressor Frame and Cylinder Vibration

Compressor frame and cylinder vibration measurements were compared to the manufacturer's guidelines. Previously, the frame vibration exceeded the guidelines, reaching 1.1 ips 0-p while operating near 1,000 RPM without the damper. But with the damper installed, the compressor frame vibration in the stretch direction was 0.5 ips 0-p or less throughout the speed range and at all load steps. Before and after measurements are shown in Figures 24 and 25, respectively.

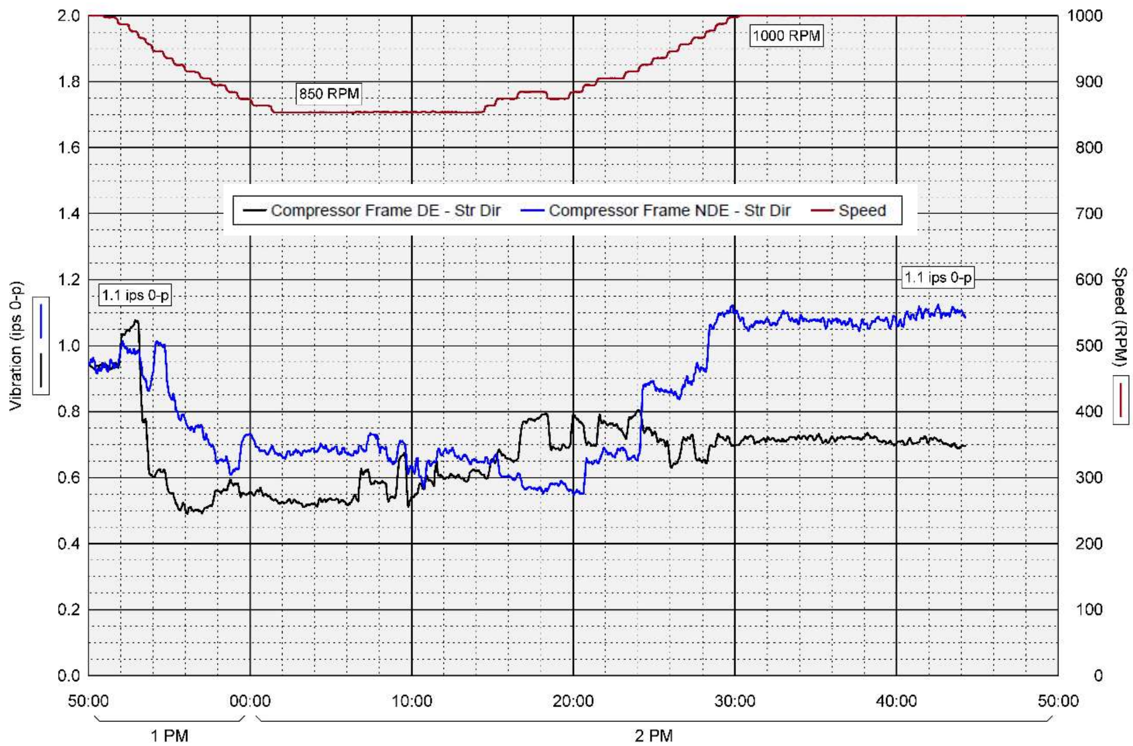

Figure 24. Overall Compressor Frame Vibration without Damper

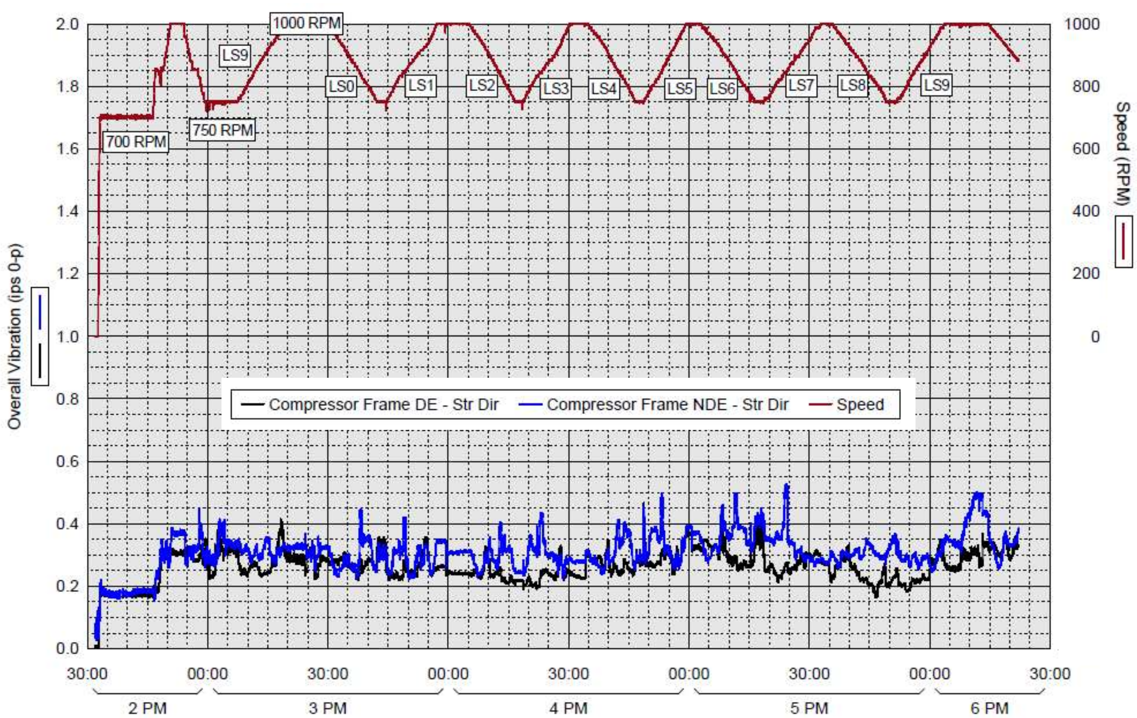

Figure 25. Overall Compressor Frame Vibration with Damper

Without the damper, the compressor cylinder vibration also exceeded guideline values at certain operating speeds. For example, the maximum overall vibration was 1.2 ips 0-p on compressor cylinders 4 and 5 while operating at full load (LS 9). After the internal damper was installed, the overall compressor cylinder vibration in the stretch direction was below the limit of 1 ips 0-p at all conditions tested (plots not shown).

The compressor internal damper lowered the torsional amplitude so that there was reduced torsional-lateral interaction (TLI). That in turn resulted in lower compressor frame and cylinder vibration that previously occurred in the stretch direction at 9× running speed and peaked at approximately 146 Hz.

There were still cylinder vibration levels above the 1 ips 0-p limit in the vertical direction. This may be related to shaking forces in the cylinders, nozzles, and /or bottles. Some of the highest cylinder vibration levels occurred while operating at LS 3. For example, compressor cylinder 1 reached 1.2 ips 0-p at 42.8 Hz, and compressor cylinder 5 reached 1 ips 0-p at 42.8 Hz. This occurred due to the 3× compressor harmonic when operating near 855 RPM as shown in Figure 26.

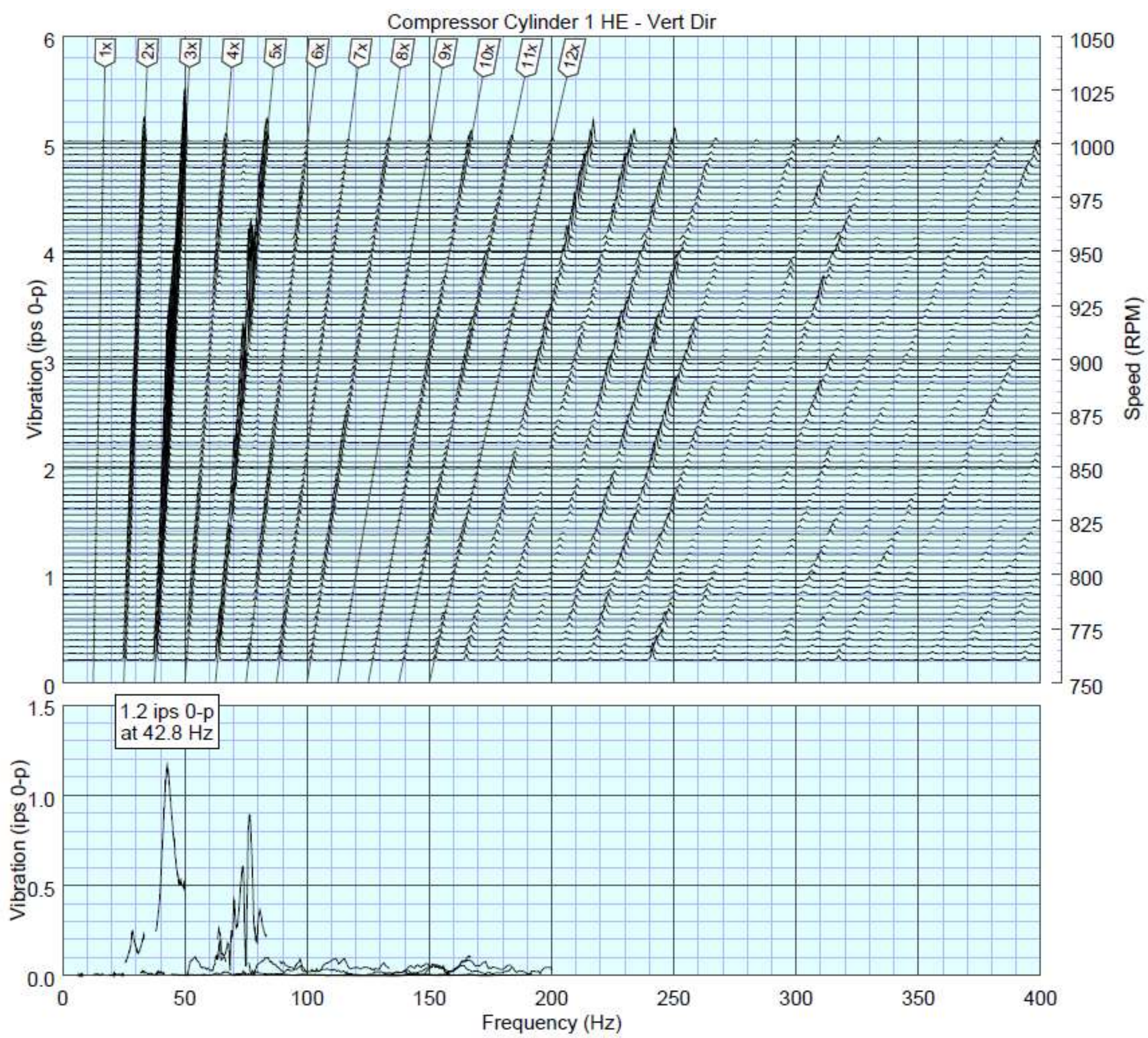

Figure 26. Waterfall Plot of Compressor Cylinder Vibration in Vertical Direction

Due to the elevated compressor cylinder vibration still occurring in the vertical direction, it was decided to conduct a vibration survey while operating at LS 3 and holding the speed near 855 RPM. Detailed vibration measurements were taken on the cylinder 1/3/5 side of the compressor as discussed in the following section.

Compressor Manifold Vibration

Elevated vibration levels were measured on the suction bottle, compressor cylinders, and discharge bottle in the vertical direction at 3× running speed (≈43 Hz). In addition, elevated vibration levels were found on the outlet of the discharge bottle and downstream elbow in the vertical direction at 5× running speed (≈71 Hz).

These vibration measurements were overlayed as shown in Figure 27. The maximum amplitudes were 1.7 and 1.9 ips 0-p at 3× and 5× running speed, respectively. For reference, the horizontal direction is parallel to the compressor crankshaft, and the stretch direction is parallel to the compressor cylinder.

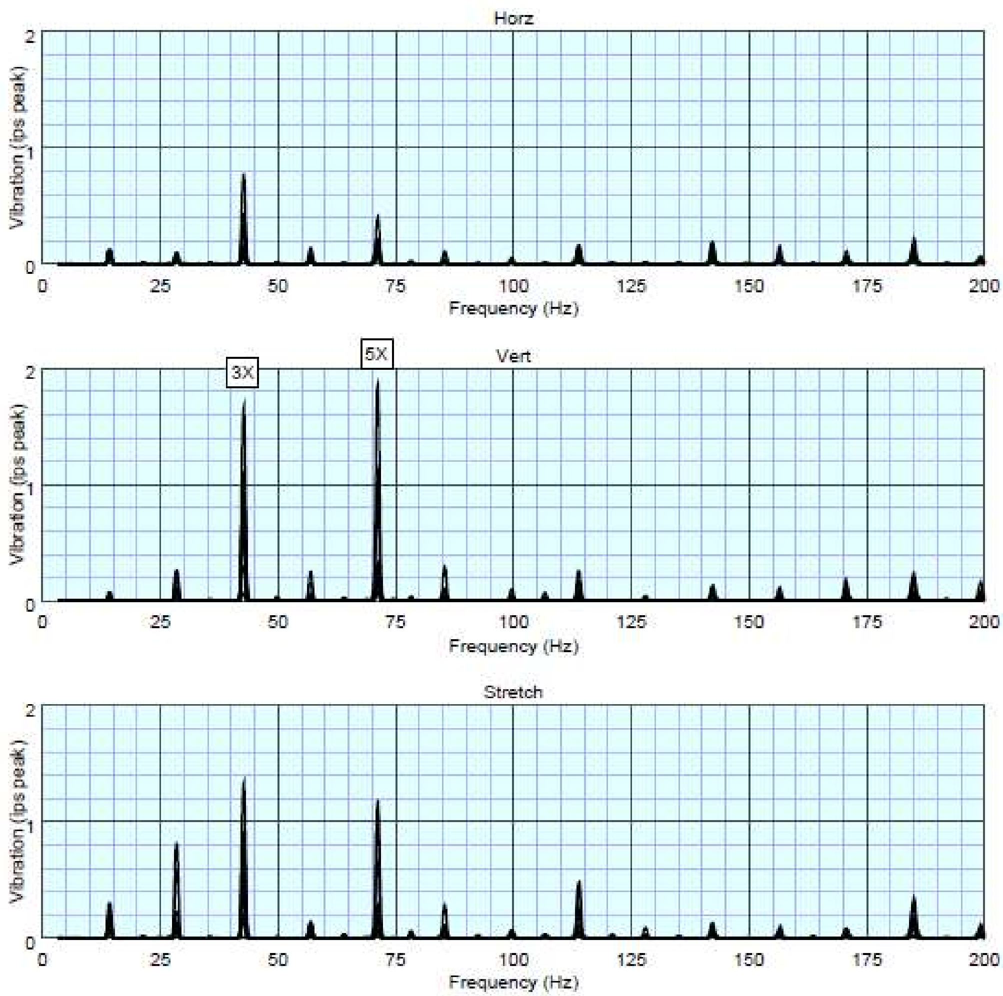

Figure 27. Summary of Vibration Measured at Load Step 3 and 855 RPM

Previously, variations in the tolerances of "identical" discharge bottle support straps were observed. The radius of curvature seemed to differ on some straps, resulting in a difference in contact length with the bottle. It is possible that the MNF of the failed straps was initially coincident with a "stronger" excitation order, resulting in high vibration and failure of the bottle strap.

It is recommended that the straps be replaced with new, properly fitting straps. A slightly thicker gasket material such as Fabreeka should be used between the strap and bottle to ensure maximum contact area. Also, the discharge bottle straps and wedge supports should be adjusted after the unit has been running and reached normal operating temperature.

Conclusions

The following conclusions were made based on the test results of the compressor with an internal damper.

1. The torsional measurements showed that the oil pump and internal damper were acceptable.
    a. The compressor crankshaft was reduced from 38 RPM 0-p to less than 20 RPM 0-p.
    b. The continuous angular deflection between the internal damper housing and the compressor crankshaft remained below the limit of 11.2 mRad 0-p.
2. The internal damper lowered the torsional amplitude below the torsional-lateral interaction level.
    a. That in turn reduced the compressor frame and cylinder vibration that previously occurred in the stretch direction at 9× running speed.
    b. Compressor frame and cylinder vibration levels were below allowable limits in the stretch direction.
3. Discharge nozzle pulsation could have been contributing to the previous elevated 5× torsional vibration. An acoustic resonance was measured near 71 Hz. A more restrictive orifice in the cylinder discharge flange could reduce the amplitude of the pressure peak and torsional excitation. However, this is not considered necessary with the internal damper in the compressor.
4. Straps and wedges need to be improved on the discharge bottles. Fabreeka or similar material should be used to provide good contact area. Adjustments should be made while the unit is warm.

VII. Case Study – Another Engine Driven Compressor Unit

A similar unit was tested later at another pipeline station. The unit consisted of a Caterpillar G3616 natural gas engine, Rexnord CMR 925 coupling, and an Ariel JGC/6 compressor. The operating speed range was 850 to 1,000 RPM.

The main difference from the compressor in the previous case was a slightly smaller cylinder bore of 12.25-inch. Compressor load steps were controlled by HEVU's on throws 1, 3, 5; and HE FVCP's on all throws. In summary, adding the Ariel D-8520 internal damper to the auxiliary end of compressor reduced torsional vibration at the oil pump and lateral frame vibration to acceptable levels.

Torsional Vibration

Torsional vibration was measured using an encoder installed at the auxiliary end of the compressor crankshaft. The measured speed fluctuations during the speed runs at compressor LS 0 are summarized in Figure 28. For example, the peak torsional amplitude at 5× running speed was reduced from 48 RPM 0-p (black line) to 5 RPM 0-p (magenta line) by installing the internal damper.

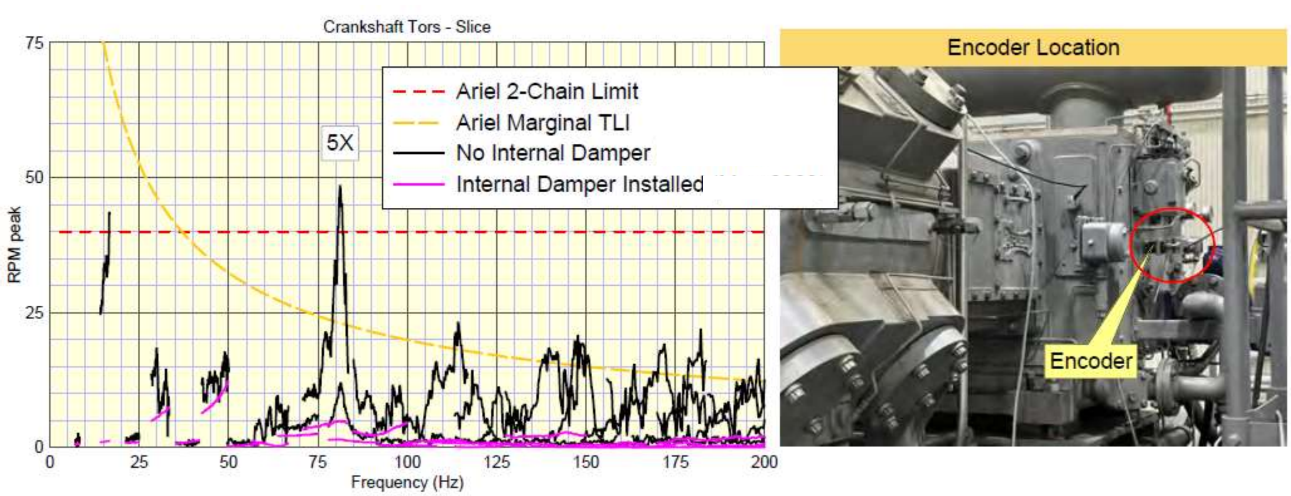

Figure 28. Comparison of Torsional Vibration Levels

The TNF near 80 Hz was significantly damped and the torsional vibration was reduced below the dual-chain design limit (40 RPM 0-p) and below the TLI guideline. The reduction in torsional vibration at 5× running speed also reduced the frame vibration at 6× running speed. Recall that TLI normally occurs at ±1 order of torsional vibration harmonic.

Compressor Frame Vibration

Figure 29 shows a trend plot of the overall frame vibration with the damper installed. The frame vibration was below the guideline of 0.5 ips 0-p during the speed sweeps at load steps 0 and 1.

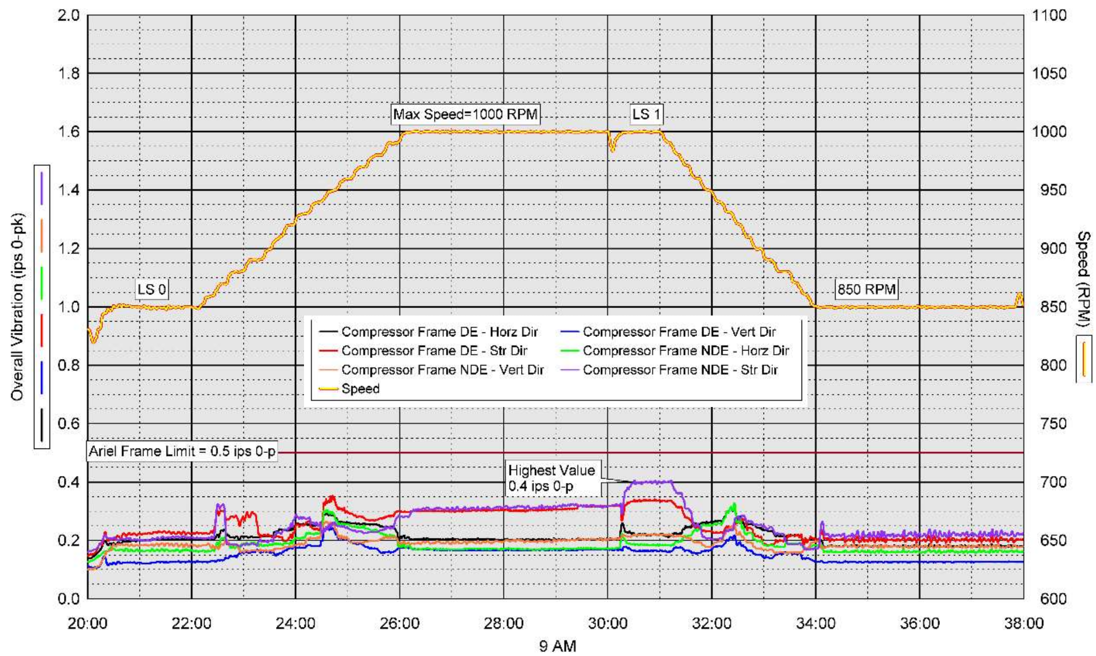

Figure 29. Overall Vibration - Compressor Frame

Figure 30 shows spectra slices extracted from the waterfall plots of the compressor frame in the horizontal direction. The plots compare the vibration with and without the internal damper. The vibration at 6× running speed was significantly reduced due to the internal torsional damper.

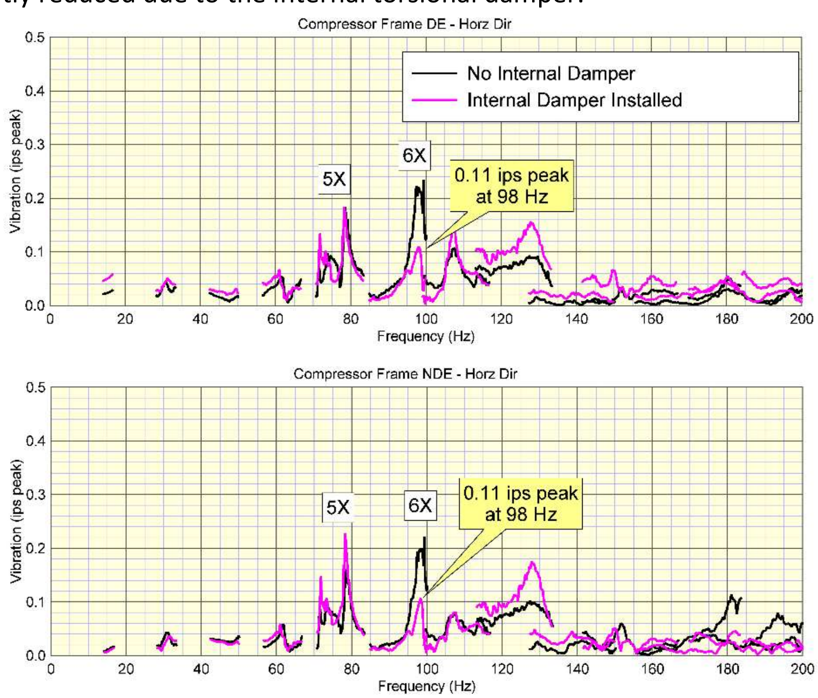

Figure 30. Slice Plots - Compressor Frame Vibration

Compressor Cylinder Vibration

Vibration was also measured on the compressor cylinders. The vibration levels of cylinders 2 and 6 slightly exceeded the guideline in the vertical direction, with a maximum amplitude of 1.1 ips 0-p while operating near 910 RPM. These amplitudes in the vertical direction were essentially unchanged compared with previous measurements taken before the damper was installed.

Figure 31 is a waterfall plot obtained on compressor cylinder 2. The peak response was 0.95 ips 0-p at 77 Hz (5× at 924 RPM) and indicates a resonance. Elevated 5× vibration was also observed at other locations on the compressor-manifold system and piping.

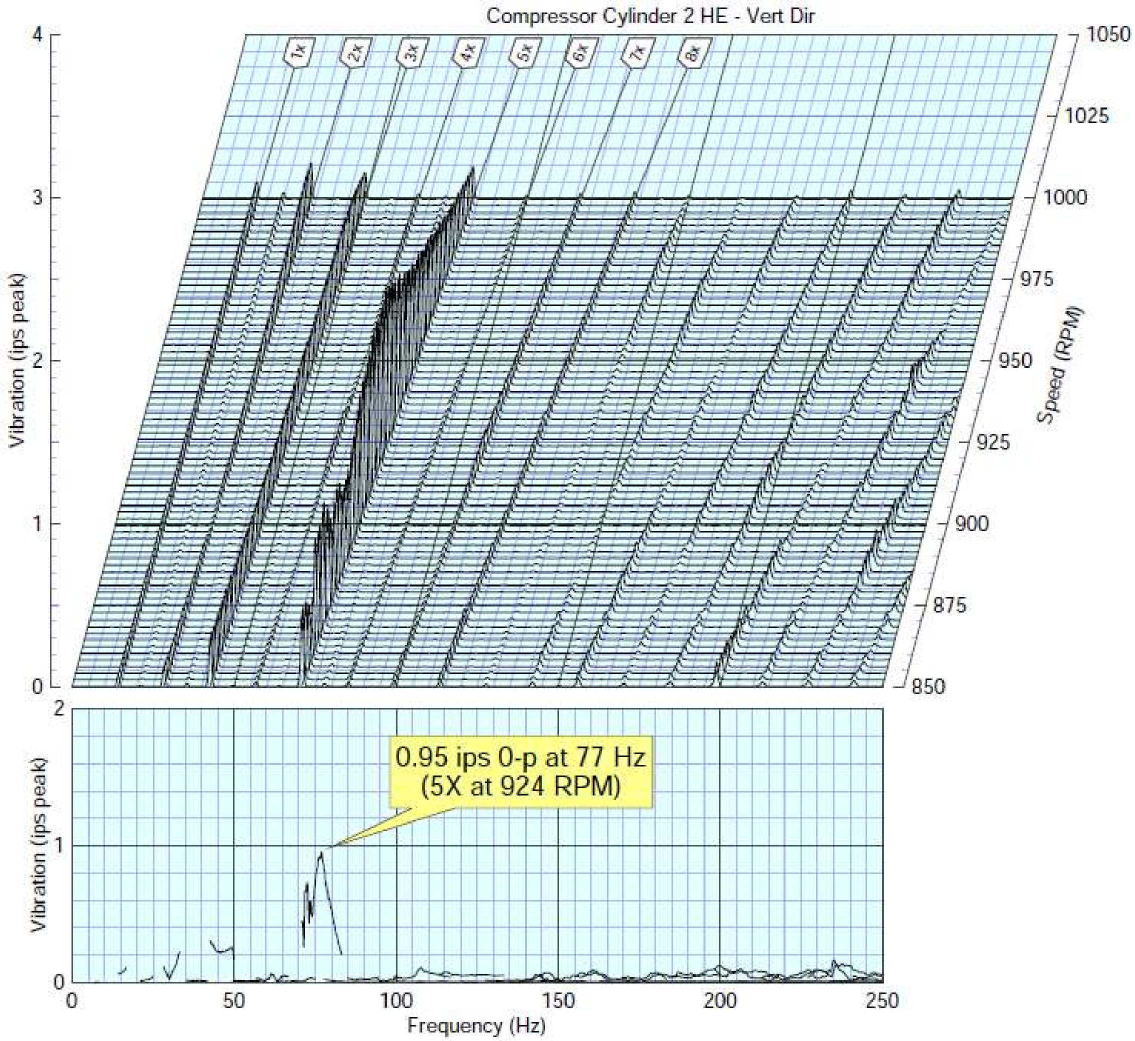

Figure 31. Waterfall Plot - Compressor Cylinder 2

The vibration peak at 77 Hz likely corresponds to the discharge nozzle pulsation resonance and mechanical natural frequency previously measured before the damper was installed. Tighter orifice plates were recommended to address this issue. The damper helped reduce torsional vibration and lateral frame vibration but did not help to reduce the vertical motion of the compressor cylinders, which is likely driven by acoustic energy.

Operating Deflection Shape

Further testing was performed to evaluate the vertical motion of the compressor cylinders. Operating deflection shape (ODS) data were obtained on the throw 2/4/6 side of the compressor while operating at 925 RPM, which was the speed that the peak vibration occurred (resonant frequency was 5x running speed). Clear Motion software [23] was used to animate the results.

Figure 32 shows a still image of the exaggerated deflected shape at 5× running speed (77 Hz). The deflected shape at 5× running speed involves out-of-phase twisting motion of the outer cylinders (throws 2 and 6). This is a typical mode shape for this cylinder configuration; however, it is normally difficult to excite. Pulsation induced shaking forces within the discharge nozzles are the likely source.

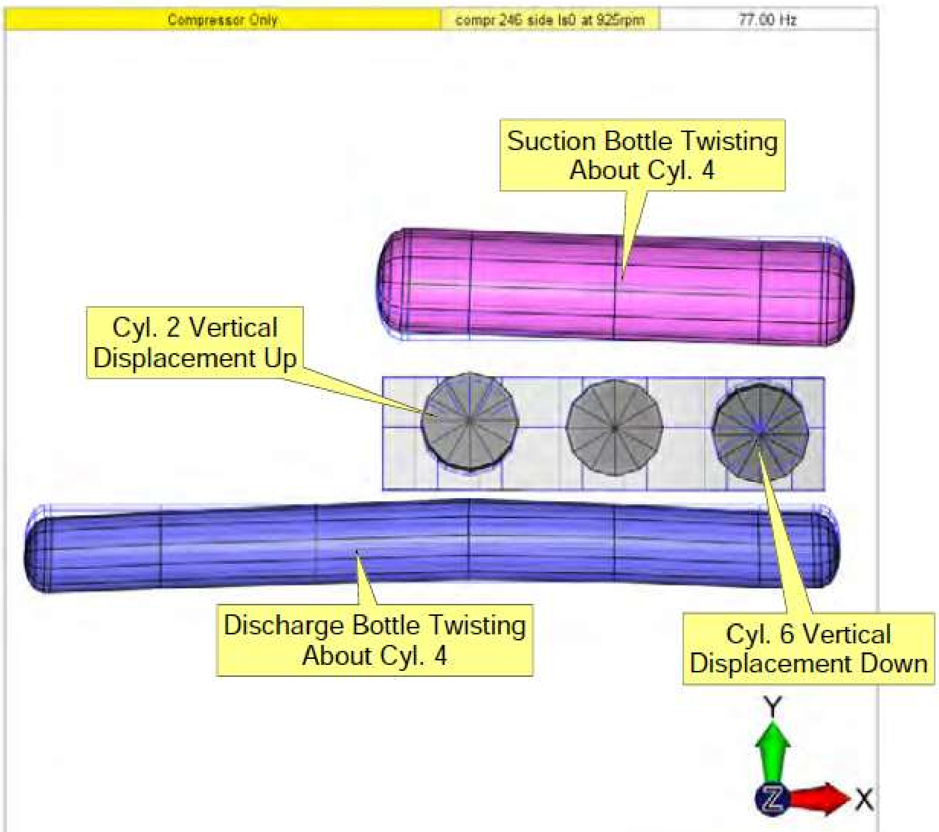

Figure 32. ODS at 5× Running Speed - Twisting of Compressor Manifold

Conclusions

1. Based on the torsional measurements of the unit with the damper installed, the oil pump and internal damper are acceptable, now below the allowable limits and the TLI guideline. The highest speed fluctuation of the compressor crankshaft was reduced from 48 to 5 RPM 0-p at 5× running speed.

2. Torsional-lateral interaction was also lower. Lateral vibration levels were reduced on the compressor frame in the stretch direction from 0.22 to 0.11 ips 0-p at 6× running speed.

3. Cylinder vibration in the vertical direction was still a concern at 5× running speed. Discharge nozzle pulsation could be the source of the excitation. A more restrictive orifice in the discharge flange was recommended to help reduce the amplitude of the pressure peak.

4. The discharge bottle supports could also be improved as discussed in the previous case study.

REFERENCES

[1] Newman, F., Stephens, T., and Harris, R., "Lateral-Torsional Vibration Coupling in Reciprocating Compressors," GMRC 2012. Also published at the Torsional Vibration Symposium, Austria, 2014.

[2] Wilson, W. K., Practical Solution of Torsional Vibration Problems, Volumes 1 - 5, New York, New York: John Wiley & Sons Inc.

[3] Nestorides, E. J. (Editor), A Handbook on Torsional Vibration, British Internal Combustion Engine Research Association (BICERA), 1958.

[4] Feese, T., and Hill, C., "Prevention of Torsional Vibration Problems in Reciprocating Machinery," 38th Turbomachinery Symposium, Houston, Texas, 2009.

[5] Wang, Q., Pettinato, B., and Feese T., "Torsional natural frequencies: measurement vs. prediction." *Proceedings of the 42nd Turbomachinery Symposium*. Texas A&M University. Turbomachinery Laboratories, 2013.

[6] Gas Machinery Research Council (GMRC), *Guideline and Recommended Practice for Control of Torsional Vibrations in Direct-Driven Separable Reciprocating Compressors*, 2015.

[7] Feese, T., and Aleksander K., "Electromagnetic effects on the torsional natural frequencies of an induction motor driven reciprocating compressor with a soft coupling." *Proceedings of the 45th Turbomachinery Symposium*. Turbomachinery Laboratories, Texas A&M Engineering Experiment Station, 2016.

[8] Den Hartog, J.P., Mechanical Vibrations, New York, New York: Dover Publications, 1985.

[9] Simpson Industries, "Torsional Vibration Dampers," Simpson Int'l (UK) Ltd., Accessed 12 January 2001.

[10] Superior, "Engineering Service Bulletin #272," May 1997.

[11] Hasse & Wrede, https://www.hassewrede.com/en/, Accessed 2009.

[12] Stephens, T., Yeiser, C., and Prenninger, K., "Torsional Damping Benefits for Reciprocating Compressors," GMRC, Pittsburg, 2017, and Torsional Vibration Symposium, Austria, 2022.

[13] https://www.geislinger.com/en/products/product/damper

[14] Feese, T., "Coupling Failure in Engine-Driven Pipeline Compressor System," Torsional Vibration Symposium, Austria, 2017.

[15] Feese, T., and Fanslow, M., "Applying API and NEMA Specifications to Limit Electrical Current Pulsation and Torsional Vibration of Synchronous Motors Driving Reciprocating Compressors," IEEE, PCIC-2019.

[16] API Standard 618, *Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services*, American Petroleum Institute, Washington D.C.

[17] ASME B106.1M 1985, Design of Transmission Shafting.

[18] Shigley, J. E., Mechanical Engineering Design, 5th Edition, McGraw Hill, 1989.

[19] Peterson, R. E., Stress Concentration Factors, John Wiley & Sons, New York, 1974.

[20] Ariel Corporation, Engineering Reference, ER-83, Revision 19, 2022. www.arielcorp.com

[21] Feese, T., and Schnitzler, W. "Special Considerations for Electric Motors Driving Reciprocating Compressors," Gas Machinery Conference (GMC), Louisville, Kentucky, 2021.

[22] Bendat, J. S., "The Hilbert Transform and Applications to Correlation Measurements," Bruel & Kjaer, 1985.

[23] Clear Motion Systems, www.clearmotionsystems.com

APPENDIX

Summary of Guidelines and Recommendations for Motor Driving Reciprocating Compressors

The GMCR guidelines [6] provide some best practices for motor rotor design, fabrication, and assembly. "Heavy Duty" motors should generally be selected for driving reciprocating compressors. A summary is shown below along with some other recommendations from reference paper [21].

- The motor supplier designs the motor for design torque applied to the shaft and spider as follows. Note that these multipliers may be reduced if a drive end flywheel, coupling inertia ring, or soft coupling is utilized in the driveline design.
  - For a 2-throw compressor, mean torque of 100% rated torque ± cyclic torque of 250% rated torque. This could be reduced to ±200% if an appropriate drive-end flywheel is specified.
  - For a 4-throw compressor, mean torque of 100% rated torque ± cyclic torque of 200% rated torque.
  - For a 6-throw compressor, mean torque of 100% rated torque ± cyclic torque of 150% rated torque.
- 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.
- To prevent possible slippage between the shaft and coupling hub, the interference fit needs to be calculated and sufficiently tight for all anticipated events.
- The motor shaft extension (including drive end bearing section) should have a diameter greater than or equal to the diameter of the compressor stub shaft.
- Use high strength material for the motor shaft. For example, using 4140 steel (or equivalent).
- Minimize stress concentration factors due to geometric discontinuities and welds.
- Minimize overhung motor arms to prevent flexing and vibration.
- Clamping force must be sufficient to hold the motor laminations and to limit shifting between arms and laminations.
- Anchor bolts should have sufficient clamping force to prevent slippage between motor feet and pedestal support. Use the correct thickness shims to avoid stacking too many shims. Keep the area free from oil.
- Use of a single cooling fan may create unbalanced axial thrust force on motor rotor. Thrust forces should be identified or dual opposing fans used.
- Motor cooling fan(s) should not have natural frequencies coincident with compressor harmonics.
- Use a stiff end bell design for the motor bearing housings. Natural frequencies should not coincide with compressor harmonics.
- Before using a VFD motor, need to verify there are no dangerous torsional resonances within the proposed speed range.
- Low-speed synchronous motor systems need to have sufficient inertia to prevent excessive torsional oscillation and electrical current pulsation.
- The electromagnetic effect (EM) should be included in torsional calculations, especially for a system with a torsionally soft coupling.