PREVENTION OF TORSIONAL VIBRATION PROBLEMS IN RECIPROCATING MACHINERY

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ABSTRACT

Torsional vibration problems continue to occur in reciprocating engine and compressor installations. This is due in part to the fact that reciprocating equipment can produce significantly more torsional excitation than rotating machinery such as turbines, centrifugal compressors, and centrifugal pumps. The purpose of this tutorial is to raise awareness of torsional vibration problems that can occur in reciprocating equipment, and to give guidelines based on experience with actual systems to avoid these problems in the future. A list of recommendations is provided to help achieve maximum reliability.

Case histories are presented where failures were linked to torsional vibration. The final solutions are given for these problems. Some of these solutions were based on practical considerations that could be retrofitted in the field. Of particular interest are the failures that would not have been predicted if only the "ideal" operating condition was analyzed. The results of these investigations emphasize the need for comprehensive torsional analyses in the design stage of critical systems.

INTRODUCTION

The purpose of this tutorial is to raise awareness of torsional vibration problems that can occur in reciprocating equipment, and to give guidelines based on experience with actual systems to avoid these problems. Case histories are presented of various failures and problems that were caused by torsional vibration. In most cases, the solutions to these problems were based on practical considerations that could be retrofitted in the field. A list of recommendations is provided to help attain maximum reliability. Various measurement techniques are also discussed.

Torsional vibration involves the twisting of shafts while the machinery is rotating. Excessive torsional vibration can lead to failures of crankshaft, couplings, engine dampers, and compressor oil pumps. These failures typically occur at a 45 degree angle to the shaft axis. In many cases, torsional vibration problems may not be apparent until after a failure occurs. This is because the machine usually does not vibrate unless it contains a gearbox, which can cause the torsional vibration to cross-couple into lateral vibration.

There are numerous books (Wilson, 1956; Nestorides, 1958) and technical papers (Wachel and Szenasi, 1993; Wachel, et al., 1995) written on torsional vibration, so the phenomenon is generally thought to be well understood and controlled. However, torsional vibration problems continue to occur in reciprocating and rotating machinery. One reason for this is the mating of equipment traditionally used in nonreciprocating applications (such as variable speed motors) with reciprocating compressors. Other causes include poor performance monitoring of engine and compressors, as well as improper application and maintenance of viscous dampers and couplings.

Reciprocating machines produce torsional excitation at multiples of running speed (orders or harmonics). When operated over a wide speed range, it is likely that one or more of these torque

harmonics will excite a torsional natural frequency (TNF) of the system. Some manufacturers and packagers mistakenly believe that engine-compressor sets do not have a potential for torsional vibration problems. However, at resonant frequencies, dynamic torque can be amplified to the point that torsional failures of couplings, mechanically driven oil pumps, and engine dampers can occur.

Various combinations of equipment, compressor cylinder sizes, load conditions, flywheels, couplings, etc., can affect the torsional characteristics of the entire system. For example, one of the case histories in this paper shows how changing out a motor with a "similar one" caused a failure due to high torsional vibration. Compressor frames that were acceptable in one type of service, may develop torsional problems or experience oil pump failures when the cylinder sizes are changed for another service or driven differently (i.e., slightly different operating speed, motor versus engine drive, etc.). Therefore, it is very important that reciprocating equipment be analyzed and critical applications tested.

As outlined in the API 684 tutorial, a complete torsional analysis should be performed (Hudson and Feese, 2006). The torsional analysis must include the whole system, not just one piece of machinery. Advanced analysis software will have a full range of capabilities such as:

- Steady-state and/or time-transient analyses.
- Calculation of torsional natural frequencies.
- Plots of torsional mode shapes.
- Campbell (interference) diagram.
- Calculated results in terms of: dynamic torque, alternating shear stress, torsional oscillation, and heat build-up.
- Nonlinear stiffness couplings.
- Damping due to rubber coupling elements and viscous dampers.
- Standard gearboxes and epicyclic gears.
- Variable frequency drives.
- Multiple compressor load steps and speeds.
- Combined forced response due to multiple orders or excitation sources.
- Comparing results to allowable levels.
- Consider "non-ideal" conditions such as: engine misfire and compressor valve failures.
- Transient events: synchronous motor start-up, electrical fault, and loaded shutdown of reciprocating compressor.
- Rainflow stress cycle counting, fatigue damage calculation, and estimated number of allowable events.

MASS-ELASTIC MODELING OF RECIPROCATING MACHINERY

The first step in analytically determining the torsional response is to calculate the torsional natural frequencies of the system. This requires mass-elastic data for the system (stiffness and mass inertia of the shaft and components being analyzed). For a round shaft, the mass moment of inertia and torsional stiffness can be calculated using simple formulas. For more complicated geometries such as crankshafts, the following procedure can be used if the manufacturer does not provide the mass-elastic data.

A crankshaft can be simplified into several main components: stub shaft that connects to a coupling or flywheel, journals where the bearings are located, webs, and crankpins. A mass-elastic model of the crankshaft is typically created by lumping the inertia at each throw and calculating the equivalent torsional stiffness between throws. Additional lumps may be created for the flywheel and oil pump.

Figure 1 shows a basic crankshaft throw. A throw consists of two webs and a crankpin. Depending on the type of crankshaft, there may be one or two throws between journals. The crankpin usually drives a connecting rod, cross-head (for compressors) and a piston and piston rod. Engines with power cylinders in a "V" arrangement may have two connecting rods at each crankpin by using an articulated rod design. Integral engine/compressor units can have two power cylinders articulated from the main connecting rod for the compressor cylinder for a total of three connecting rods per throw.

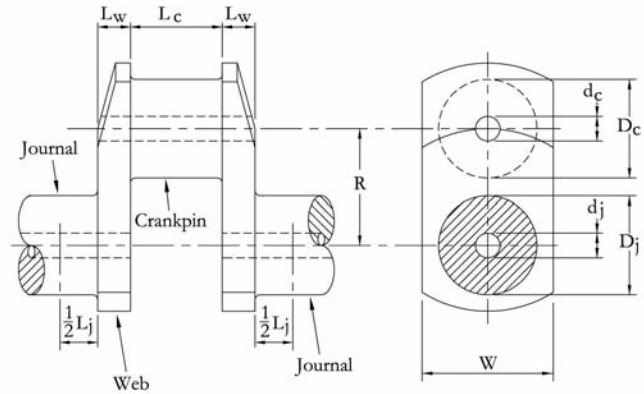


Figure 1. Typical Crankshaft Throw.

TORSIONAL STIFFNESS

Equations are given by Ker Wilson (Wilson, 1956) and BICERA (Nestorides, 1958) for calculating the torsional stiffness, Kt, of a crankshaft. The basic dimensions of the journals, webs, and crankpins are needed, as well as the shear modulus, G, of the shaft material.

Carter's formula:

$$K_t = \frac{G\pi}{32\left[\dfrac{L_j + 0.8L_w}{D_j^4 - d_j^4} + \dfrac{0.75L_c}{D_c^4 - d_c^4} + \dfrac{1.5R}{L_w W^3}\right]} \tag{1}$$

Ker Wilson's formula:

$$K_t = \frac{G\pi}{32\left[\dfrac{L_j + 0.4D_j}{D_j^4 - d_j^4} + \dfrac{L_c + 0.4D_c}{D_c^4 - d_c^4} + \dfrac{R - 0.2(D_j + D_c)}{L_w W^3}\right]} \tag{2}$$

Carter's formula is applicable to crankshafts with flexible webs and stiff journals and crankpins, while Ker Wilson's formula is better for stiff webs with flexible journals and crankpins. BICERA also developed curves based on test data for various types of crankshafts. The BICERA calculations can be more accurate, but they are also more complex and are not discussed here in detail.

When conducting a torsional analysis, Ker Wilson suggested using the average of his formula and Carter's formulas to determine the stiffness between throws. To calculate the torsional stiffness of the stub shaft to the centerline of the first throw, the torsional stiffness of the straight shaft section can be combined in series with twice the torsional stiffness between throws. For coupling hubs or flywheels with an interference fit, the one-third penetration rule should also be applied. The one-third rule assumes that one-third of hub length has the torsional stiffness of only the shaft, and for the other two-thirds of the hub length the shaft is fully engaged acting as a solid piece.

Equations (1) and (2) were developed before finite element analysis (FEA). A commercially available finite element program can be used to determine the torsional stiffness for a crankshaft section. The simple models shown in Figure 2 were developed from the basic dimensions and do not include small features such as fillet radii and oil holes.

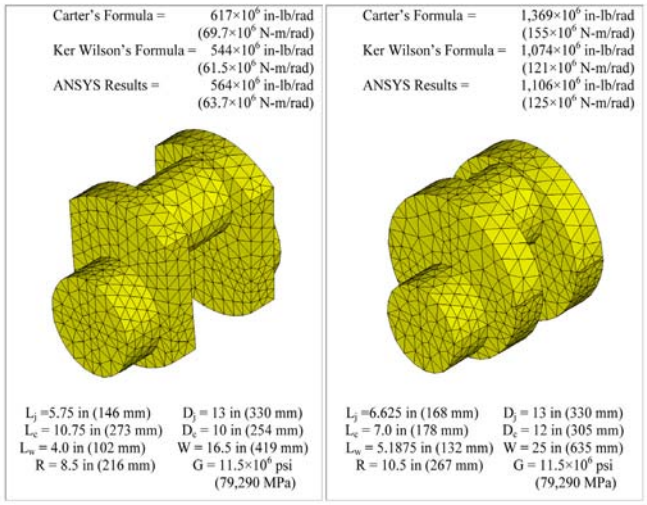


Figure 2. Finite Element Models Used to Calculate Torsional Stiffness of Crankshaft Sections.

The FEA models include the portion of the crankshaft between journal centers, which is equal to the distance between compressor throws. One end was fixed while a known moment was uniformly applied at the other end of the model. The calculated torsional stiffness is equal to the moment divided by the angle of twist at the free end where the moment was applied. It is interesting to note that for both cases, the calculated torsional stiffness using the commercial software fell within the range of values from the Carter and Ker Wilson formulas.

Polar Mass Moment of Inertia

The polar mass moment of inertia (commonly referred to as WR^2) at each throw depends on the rotating inertia and the reciprocating mass. The rotating inertia is constant, but the effective inertia of the reciprocating parts actually varies during each crankshaft rotation. This effect is considered to be negligible in most engines except in the case of large slow-speed marine applications (Pasricha, and Carnegie, 1976).

The equivalent inertia, I_{eqv}, can be approximated by adding the rotating inertia of the crankshaft section, I_{rot}, to half of the reciprocating mass, M_{recip}, times the throw radius, R, squared:

$$I_{eqv} \approx I_{rot} + \tfrac{1}{2} M_{recip} R^2 \tag{3}$$

The rotational inertia of the journal and crankpin can be calculated using the equation for a cylinder. Since the crankpin rotates at the throw radius and not about its center, the parallel axis theorem must also be used. The inertia of the webs can be estimated with an equation for a rectangular prism. Any rotating counterweights that may be bolted to a web should also be included. Due to the complexity of these calculations, inertia values are usually computed using FEA or three-dimensional computer-aided design (3D CAD).

The connecting rod is generally heavier at the crankpin (rotating end) and lighter at the reciprocating end. If the weight distribution of the connecting rod is unknown, assume two-thirds of the weight is rotating and one-third is reciprocating. The rotating mass of the connecting rod is multiplied by the throw radius squared and added to the crankshaft rotating inertia to obtain the total rotational inertia, I_{rot}.

The total reciprocating mass includes the small end of the connecting rod, cross-head (for compressors), nut, piston and piston rod. Half of the reciprocating mass is multiplied by the throw radius squared and added to the rotating inertia to obtain the equivalent inertia, I_{eqv}, as shown in Equation (3). An example calculation is given in Table 1 for a three-stage compressor.

Table 1. Example Inertia Calculation for Reciprocating Compressor.

Compressor Dimensions			
Stroke	4.5	in	(114 mm)
Throw Radius	2.25	in	(57 mm)
Connecting Rod Weight	96	lb	(43.5 kg)
	1st Stage	**2nd Stage**	**3rd Stage**
Reciprocating Weight (lb)			
Connecting Rod, Small End	32	32	32
Crosshead Pin Assembly	30	30	30
Piston & Rod Assembly	235	200	110
Crosshead Assembly	75	105	190
Balance Nut	6	10	18
Total Reciprocating Weight	**378 lb** (171 kg)	**377 lb** (171 kg)	**380 lb** (172 kg)
Mass Moment of Inertia (lb-in^2)			
Crank Throw	1,150	1,150	1,150
Connecting Rod, Big End	324	324	324
Reciprocating Weight	957	954	962
Total Equivalent Inertia	**2,431 lb-in^2** (0.711 kg-m^2)	**2,428 lb-in^2** (0.711 kg-m^2)	**2,436 lb-in^2** (0.713 kg-m^2)

TORQUE-EFFORT CONSIDERATIONS

Reciprocating compressors and engines produce unsteady torque. This torque variation is normally much higher than in rotating equipment (centrifugal compressors, fans, etc.). The torque excitation produced by reciprocating machinery can occur at multiple frequencies and have significant amplitude, so they must be considered to avoid torsional problems.

Torque Variation Due to Inertial and Gas Forces

From a torsional standpoint, there are two types of forces that cause torque variation at each throw: inertial and gas load. The total force times the distance between the crankshaft centerline and the effective throw centerline is equal to the moment imposed on the crankshaft.

Figure 3 is a sketch showing the typical rotating and reciprocating components of a two-throw compressor. The components for an engine would be similar, although the piston would normally be directly attached to the connecting rod (no crosshead or piston rod).

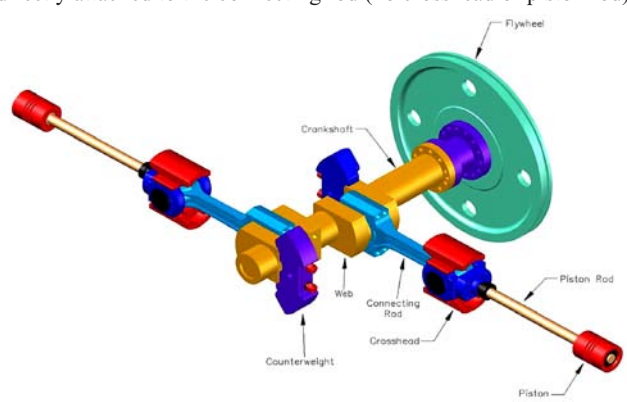


Figure 3. Rotating and Reciprocating Components of Two-Throw Compressor.

The total force varies depending on the crank angle. At bottom dead center (BDC) and top dead center (TDC), the throw is in-line with the connecting rod and piston so that no moment can be imposed on the crankshaft. At 90 degrees from BDC and TDC, the moment arm is at the maximum length (full crank radius).

Inertial Force

The inertial forces are caused by the reciprocating mass of the connecting rod, crosshead, piston and piston rod. From BICERA (Nestorides, 1958), the inertia force due to these reciprocating parts (F_i, in lbs) at a particular crank angle is:

$$F_i = 0.0000284 \times N^2 R w \times \left[\cos\theta + \frac{R}{L} \cos 2\theta \right] \tag{4}$$

where N is the speed (rpm), R is the crank radius (inch), w is the total reciprocating weight, L is the connecting rod length (inch), and θ is the crank angle (deg). Note that the inertia force varies with the speed squared and is zero at bottom and top dead center.

Gas Force

The gas force is equal to the differential pressure across the piston times the cross-sectional area of the bore. The swept volume for each cylinder is the bore area times the stroke. Next, the pressure versus crank angle must be determined for each cylinder over 360 degrees for compressor or two-stroke engine and 720 degrees for a four-stroke engine. From the *Diesel Engine Reference Book* (Lily, 1984), the torque due to the gas forces is determined by:

$$T_g = pAR\left[\sin\theta + \frac{\sin 2\theta}{2\sqrt{n^2 - \sin^2\theta}}\right] \tag{5}$$

where T_g is the instantaneous torque, p is the gas pressure at angle θ, A is the piston area, R is the crank radius, and n is the connecting rod to crank ratio. Any distortion in the pressure waveform will affect the dynamic torque and torsional response (Szenasi and Blodgett, 1975).

A third type of curve that is often seen is called tangential effort (or tangential pressure), which is the torque normalized for unity crank radius and piston area. The advantage of this type of curve is that it could be applied to different sized machines.

Once the inertia and gas forces have been determined, they must be correctly summed for each cylinder. The throws must then be properly phased for the entire machine as shown in Figure 4. A Fourier analysis can then be performed to represent the torque wave forms as harmonics. Figure 5 shows the torque amplitudes at each compressor order. Results of the Fourier analysis are complex numbers, which can be represented as amplitude and phase or as coefficients of sine and cosine functions for each harmonic.

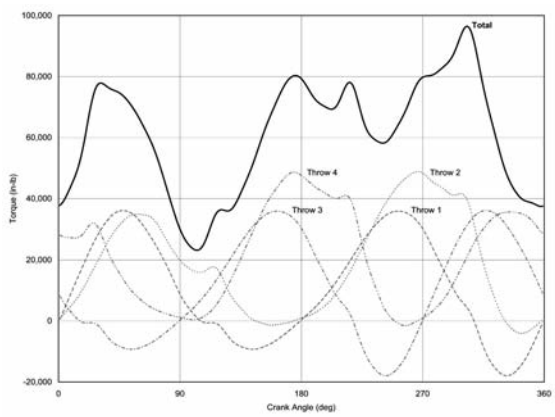


Figure 4. Compressor Torque Variation Versus Crank Angle.

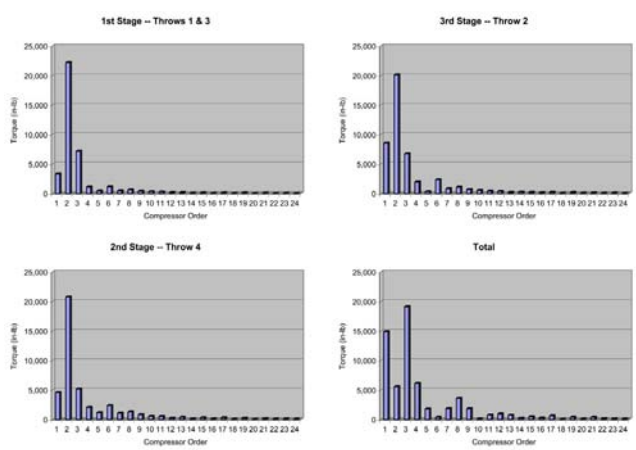


Figure 5. Fourier Analysis of Compressor Torque Excitation.

Compressors and two-stroke engines will have integer harmonics of running speed while four-stroke engines produce both integer and half orders. Depending on the cylinder phasing, certain orders may cancel out while others become dominant. A torsional analysis should include up to at least the 12× harmonic when examining the overall torque output from the machine.

Compressor and engine manufacturers will often provide this torque information in various forms with the performance calculations. To use their data in a torsional analysis, it is very important to understand the sign convention and if the values are only for gas forces or if the effect of reciprocating mass has also been included. Tangential effort curves for engines have also been published by Porter (1943) and *Lloyd's Register of Shipping* (2000).

Compressors

For compressors, suction valves allow the gas to enter when the piston approaches BDC and the cylinder pressure momentarily dips below the suction line pressure. As the piston compresses the gas and approaches TDC, the gas is released when the pressure in the cylinder is higher than the discharge line, causing the discharge valves to open. Single-acting compressors use only one side of the piston while double-acting cylinders use both the crank and head ends. Unloaders and clearance pockets can also affect the harmonic content of the gas pressure forces.

Ideal pressure cards are often used for analysis since they can be computed from the compressor and gas properties. However, ideal cards do not include valve/manifold losses and gas pulsation, which can significantly affect the resulting torque harmonics. High pulsation can occur in the cylinder due to various acoustic resonances associated with the compressor-manifold system (cylinders, bottles) and attached piping.

Figure 6 shows an ideal P-T card (pressure versus angle) superimposed over a measured P-T card for both ends of a compressor cylinder. As seen, there is an over/under-pressure and oscillating response during opening of both the suction and discharge valves. This changes the harmonic content and can cause an increase in torsional excitation by the compressor.

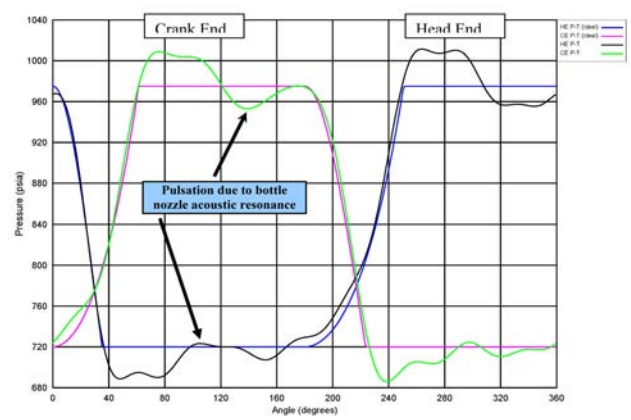


Figure 6. Comparison of Actual Versus Predicted Pressure-Angle of Compressor Cylinder.

Torsional vibration problems with reciprocating compressors can be caused by:

- *Valve failure*—A compressor valve failure can be simulated in a computer analysis by unloading one end of a cylinder. For example, a double-acting cylinder (both ends loaded) should be analyzed as single-acting, even if the cylinder does not have unloaders.

- *Gas pulsation*—To include the effects of gas pulsation (dynamic pressure), the P-T card calculated from an acoustical analysis could be used in the torque-effort calculation. For existing systems, the actual P-T card should be measured in the field and used in the analysis.

• *Load steps*—Some reciprocating compressors have multiple load steps to account for varying flow conditions. Certain load steps could significantly affect the harmonic content and influence the torsional responses. For example a single-acting cylinder will produce higher dynamic torque at 1× and a double-acting cylinder will have higher 2×. 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.

Engines

Combustion in engines occurs from either auto-ignition in diesels or spark-ignition in gas engines. Engines may use two or four-stroke cycles. For a two-stroke engine, intake, compression, expansion, and exhaust occur during one revolution of the crankshaft. However, with four-stroke engines, these cycles occur over two revolutions, which causes half-order excitations (once every two rotations).

Some engines have a choice of firing orders, which can change the excitation levels of strong harmonics. In critical systems, the firing order with the minimum excitations should be chosen to reduce the torsional response.

When performing a torsional analysis of an engine-driven system, identical P-T cards are normally used for every cylinder, although the actual cylinder pressures will vary. Using identical P-T cards in a computer analysis may cause some orders to appear to cancel out or have low amplitude. Likewise, if the reciprocating weights are not well balanced some of the engine orders could be affected.

Figure 7 shows large variation in peak pressures among the cylinders of an engine with multiple performance problems. Poorly maintained engines will tend to operate at non-ideal conditions that can cause high torsional vibration. The authors have seen where an older engine was retrofitted with an electronic fuel injection system to improve horsepower and exhaust emissions, but as an added benefit the torque excitation was reduced and the speed control became more stable.

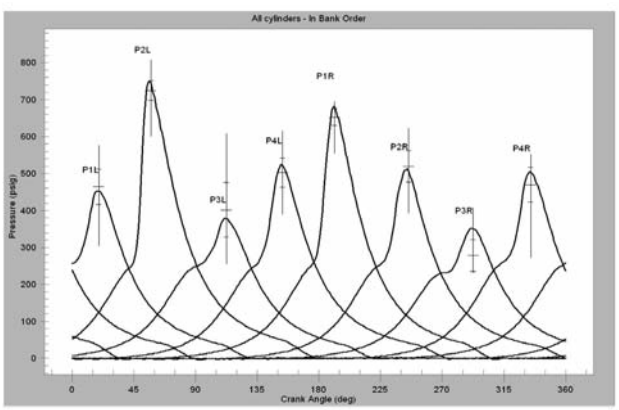


Figure 7. P-T Parade for Engine with Performance Problems.

There are several situations that can lead to torsional vibration problems in engines:

• *Engine misfire*—A misfire condition should be analyzed by assuming at least one cylinder does not fire. Misfire is common when the fuel is inconsistent, such as biogas from waste treatment or landfills. Peak cylinder pressure is greatly reduced when detonation does not occur.

• *Pressure imbalance*—This occurs when ignition timing is mismatched between the power cylinders.

• *Ignition problems*—Improper spark can occur if the capacitor is connected with reversed polarity or if the wires are bad.

• *Leaks*—Peak pressure can also be affected by leaks in fuel valves or compression pressure.

• *Instability*—The engine control system could inadvertently excite a torsional natural frequency. For example, the authors have seen where a diesel engine used for ship propulsion damaged a gearbox. An engine manufacturer may perform a stability analysis of the control system and change filter/gain settings as necessary to prevent the engine from "hunting" and causing this type of torsional problem.

Case History—Coupling Failure

• Equipment:
  Diesel engine driving a six-throw reciprocating compressor through an oil-filled, leaf spring coupling
  Rated 7,860 hp (5,861 kW) at 775 rpm
  Speed range = 575-775 rpm
  Compressor has 22 load steps

• Problem:
  Premature coupling damage

• Cause:
  First TNF was coincident with 1× running speed. Original torsional analysis used improper coupling damping and did not evaluate full range of compressor excitation.

• Solutions:
  Short-term: Speed and load step restrictions
  Long-term: Detune first TNF by installing a compressor flywheel

An engine-driven compressor unit at a gas transmission pipeline station experienced coupling failures. The unit had a torsionally soft coupling between the engine and compressor that utilized radial leaf springs. The coupling was supplied with pressurized oil from the engine through a central bore in the crankshaft extension. Damping due to oil flow through internal clearances of the coupling occurs as the leaf springs flex.

The damaged coupling is shown in Figure 8. Note that all of these cracks occurred at a 45 degree angle to the shaft axis. This is indicative of damage produced by high torsional vibration. Oil samples showed an increase in copper content before coupling failure.

Figure 8. Failed Leaf-Spring Coupling.

Field Testing

The damaged coupling was replaced before the field testing was performed. To help diagnose the cause of the failure, torsional vibration and compressor P-T data were acquired.

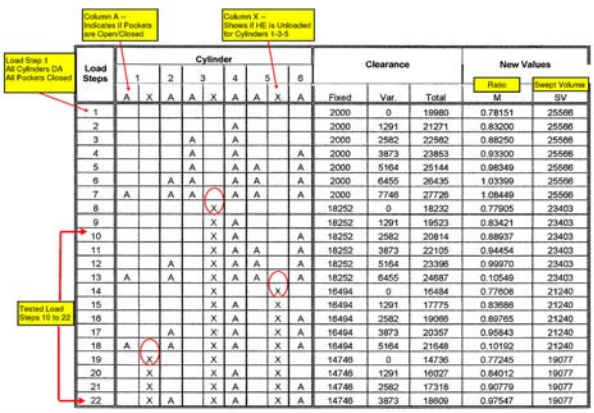
Torsional oscillation of the engine was obtained using a proximity probe monitoring the gear teeth on the engine flywheel. Assuming the teeth are equally spaced and the lateral vibration is low, variations in time between tooth passing will indicate torsional vibration. The demodulated signal was then converted to angular velocity and integrated to angular displacement.

Torsional oscillation of the auxiliary end of the compressor was measured using a torsiograph, which directly attaches to the rotating shaft. The torsiograph directly measures torsional vibration in degrees.

The angular deflection across the coupling was measured using an optical sensor. A strip of paper that was printed and laminated with alternating black and white squares was glued to the circumference of the coupling flange on the compressor side. Each square was approximately 1 inch wide for a total of 124 squares (62 black and 62 white). These pulses were converted to torsional vibration using the Hilbert transform (Randall, 1990). Knowing the torsional stiffness of the coupling, the dynamic torque in the coupling could be inferred based on the measured angular deflection.

Cylinder pressure-time (P-T) data was measured using dynamic pressure transducers installed in all six compressor cylinders on the head-end (HE) plus one transducer in cylinder #1 crank-end (CE). A once-per-rev tachometer signal referenced to engine cylinder #1 TDC, and a once-per-rev tachometer signal referenced to compressor cylinder #1 TDC were also measured.

All compressor cylinders had pockets that could be opened and closed. In addition, cylinders 1-3-5 had head-end unloaders. The HE of each compressor cylinder was unloaded by holding the suction valves open. The compressor had a total of 22 load steps, as shown in Figure 9.



Load Steps	Cylinder 1 A	1 X	2 A	3 A	3 X	4 A	5 A	5 X	6 A	Clearance Fixed	Var	Total	New Values Ratio M	Swept Volume SV
1										2000	0	19980	0.78151	25566
2						A				2000	1291	21271	0.83200	25566
3				A		A				2000	2582	22562	0.88250	25566
4				A		A			A	2000	3873	23853	0.93300	25566
5				A		A	A		A	2000	5164	25144	0.98349	25566
6			A	A		A	A		A	2000	6455	26435	1.03399	25566
7	A		A	A		A	A		A	2000	7746	27726	1.08448	25566
8					X					18252	0	18232	0.77905	23403
9					X	A				18252	1291	19523	0.83421	23403
10					X	A			A	18252	2582	20814	0.88937	23403
11					X	A	A		A	18252	3873	22105	0.94454	23403
12			A		X	A	A		A	18252	5164	23396	0.99970	23403
13	A		A		X	A	A		A	18252	6455	24687	0.10549	23403
14					X			X		16494	0	16484	0.77608	21240
15					X	A		X		16494	1291	17775	0.83688	21240
16					X	A		X	A	16494	2582	19066	0.89765	21240
17			A		X	A		X	A	16494	3873	20357	0.95843	21240
18	A		A		X	A		X	A	16494	5164	21648	0.10192	21240
19		X			X			X		14746	0	14736	0.77245	19077
20		X			X	A		X		14746	1291	16027	0.84012	19077
21		X			X	A		X	A	14746	2582	17318	0.90779	19077
22		X	A		X	A		X	A	14746	3873	18609	0.97547	19077

Figure 9. Compressor Load Steps.

All of the test signals were continuously monitored and digitally recorded over the full speed range and at as many load steps as possible. However, available load steps were limited based on the pipeline conditions at the station. Initially, the engine speed was slowly varied from 775 down to 575 and back up to 775 rpm. The compressor load steps were allowed to automatically change as needed at the various speeds.

Figure 10 shows the speed profile and compressor load steps versus time. Load steps were obtained from the station's computer monitoring system and notes added to the trend plot. As shown, the torsional vibration amplitude at 1× speed significantly increased to 1.75 deg peak-to-peak when the load steps were changed and the speed was reduced.

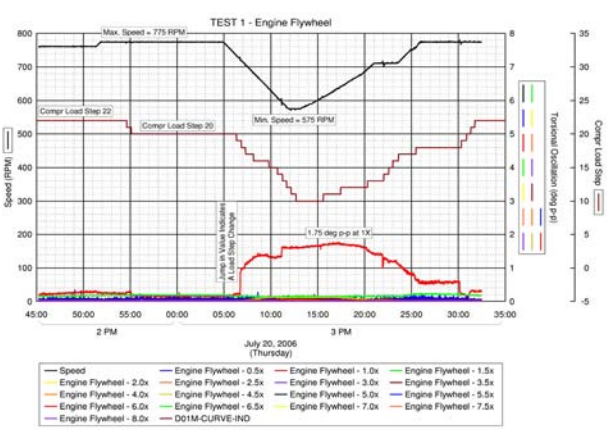


Figure 10. Trend Plot of Speed and Compressor Load Steps.

Data measured during the speed sweep tests were plotted and curve fit to determine the peak response frequency and damping values. The first three torsional natural frequencies of the system are listed in Table 2. The corresponding amplification factors (AF) were estimated using the method described in the *Testing for Damping Value* section.

Table 2. Measured Torsional Natural Frequencies and Amplification Factors (Damping).

Mode	Hz	CPM	AF
1	10.75 – 11.35	645 – 681	6
2	46.9	2813	9
3	86.6	5196	40

The first TNF was identified at 10.75 Hz (645 cpm) when the torsional oscillation was low. For other conditions, the torsional oscillation was much higher and the leaf springs may have been contacting the internal stops. The torsional stiffness of the coupling varied depending on the amplitude and frequency of vibratory torque. As a result, this raised the first TNF of the system to as high as 11.35 Hz (681 cpm).

The torsional oscillations at the engine flywheel and coupling flange at the compressor were measured. From these values, the differential angular displacement across the coupling was plotted in Figure 11. In addition to the various orders, the overall angular oscillation across the coupling is also shown in the figure. The coupling manufacturer provided a static torsional stiffness of the coupling. By multiplying this stiffness by the differential angular displacement, the dynamic torque in the coupling can also be determined. In this case, the maximum dynamic torque amplitude was 36,141 ft-lb zero-peak (49 kN-m 0-pk).

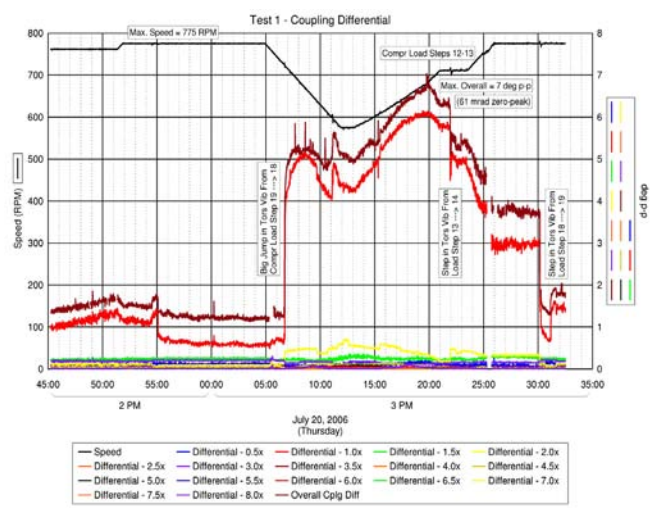


Figure 11. Coupling Differential Angular Displacement.

The worst conditions were measured while operating near 675 to 680 rpm and at two different compressor load steps. Both of these cases involve only one compressor cylinder single-acting (SA), which resulted in high dynamic torque at 1× running speed. The maximum measured value was 7 degrees peak-to-peak or 3.5 degrees zero-peak, which equates to 61 milli-radians zero-peak.

Note that the average transmitted torque in the coupling normally causes 80 milli-rad of angular displacement. Adding the constant angle of 80 milli-rad to the maximum dynamic angle of 61 milli-rad yields a peak of 141 milli-rad. This instantaneous

angular displacement exceeded the coupling allowable limit of 125 milli-rad as given by the manufacturer.

The field measurements clearly showed that the torsional vibration and torque levels exceeded the coupling limits while operating at speeds near the first torsional natural frequency and using load steps involving single-acting compressor cylinders.

Torsional Analysis

The torsional analysis was originally performed by a third party and predicted that the system design would be satisfactory. In the analysis, the first TNF was calculated to be within the operating speed range, but was assumed to be well damped due to the oil filled coupling. However, during the field tests it was determined that the coupling damping was actually much less than had been used in the torsional analysis. As a result, the actual dynamic torque in the coupling was approximately double the level predicted by the torsional analysis and exceeded the allowable limit.

Upon review, the torsional analysis was found to be incomplete because only three compressor load conditions were evaluated. In hindsight, all of the 22 load steps should have been analyzed. The results from the additional cases would have shown that the coupling limit would be exceeded for some of the compressor load steps with single-acting cylinders.

Summary

Test results showed that the design limits of the coupling were exceeded in terms of angular oscillation and vibratory torque. Physical evidence and oil analysis also supported this finding. The worst conditions were measured while operating near 675 to 680 rpm, which was near the first torsional natural frequency, and was strongly excited while operating at compressor load steps involving single-acting compressor cylinders.

For the short-term solution, speed and load step restrictions were implemented. The recommended long-term solution was to detune the first TNF below minimum running speed by adding a flywheel to the compressor.

Case History—Motor Failures and Loose Coupling Bolts Caused by Torsional and Acoustical Resonances

- Equipment:
  Constant speed induction motor driving a six-throw, single-stage reciprocating compressor through a flexible disc coupling
  Rated 4,200 hp (3,132 kW) at 895 rpm
- Problem:
  Motor shaft failure and coupling damage
- Causes:
  Coincidence of first TNF with 4× running speed. Acoustical resonance at 4× increased the gas pulsations in the compressor cylinders, which resulted in increased dynamic torque.
- Solution:
  Installed internal compressor flywheel to detune TNF. Installed cylinder flange orifice plates to attenuate pulsation at nozzle acoustic resonance.

The skid mounted natural gas compressor was configured as a single-stage unit and could operate at seven load steps through the use of 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 12. Note that this is a photo of a section that was cut out of the failed motor shaft. 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 initiated from poor welds at the shaft-spider interface.

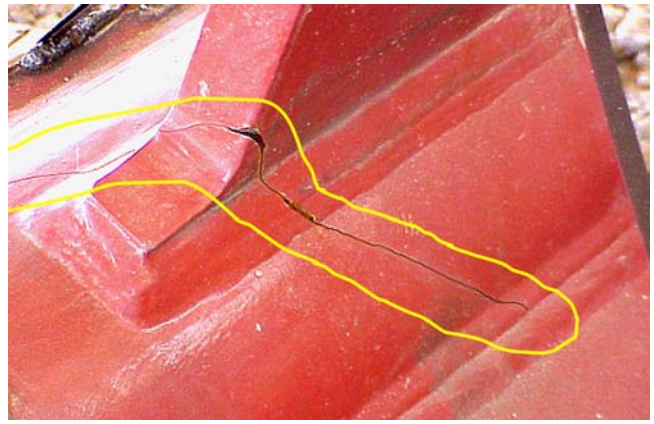


Figure 12. Failed Motor Shaft.

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

Field Testing

Torsional vibration measurements were obtained using a strain gage telemetry system installed on the motor shaft near the coupling hub. Data were acquired during an unloaded shutdown and at all seven load steps while operating at normal conditions. The first TNF was identified during the unloaded coastdown near 60 Hz (Figure 13), which was coincident with 4× running speed.

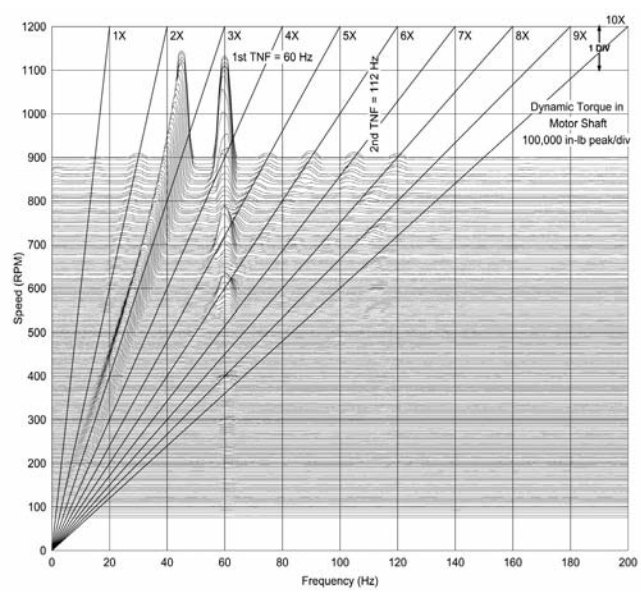


Figure 13. Waterfall Plot of Compressor Shaft Torque During Coastdown—Original System.

As shown in Figure 14, the maximum overall dynamic torque in the shaft was 750,000 in-lb zero-peak (84,739 N-m 0-pk) and occurred during operation at load step 3. This dynamic torque amplitude was approximately 2.5 times the full load torque and corresponded to a stress of 18,100 psi zero-peak (125 MPa 0-pk) including an SCF of 2. While operating at load step 3, compressor cylinders 1-2 are double-acting, and cylinders 3-6 are single-acting (head-ends unloaded). For this unit, the cylinders were numbered from the motor end. The shaft stresses were considered excessive and the unit was taken out of service until modifications could be made.

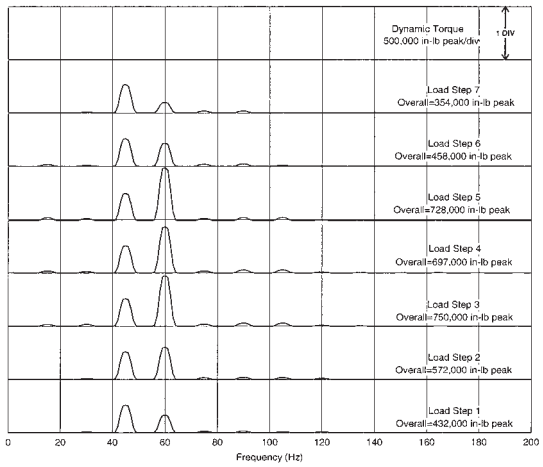


Figure 14. Frequency Spectra at Each Load Step—Original System.

Torsional Analysis

A torsional analysis had been performed by a third party prior to installation of the unit. The original analysis predicted the first TNF to be near 70 Hz, which was approximately 17 percent higher than the measured frequency of 60 Hz.

A second torsional analysis was performed to evaluate modifications to detune the system. It was determined that the computer model was sensitive to the simulation of the motor. Using a multi-mass motor model and including the effects of the core stiffening in the motor, the first TNF was computed to be 60 Hz, which agreed with the measured frequency.

To detune the first torsional natural frequency from the fourth compressor order, internal flywheels (five "donuts") were recommended between throw #4 and throw #5. Follow-up field testing verified that the first TNF was reduced to 55 Hz as shown in Figure 15. The recommended modification provided an adequate separation margin from the third and fourth compressor harmonics.

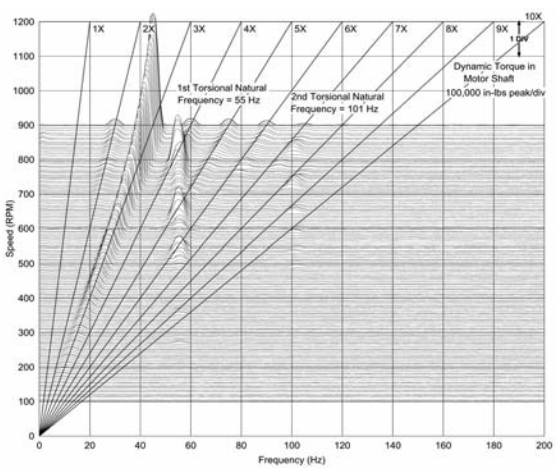


Figure 15. Waterfall Plot of Compressor Shaft Torque During Coastdown—Modified System.

Follow-up Field Testing and Analysis

A significant discrepancy was noted between the calculated and measured fourth order response amplitudes at certain load steps. The calculations were made using an ideal P-V card for the torque-effort calculations, which did not include performance degrading phenomena such as gas pulsation and valve losses. This caused the originally computed torques to be approximately 1/17 of those measured at load step 3. To improve the accuracy of the analysis, P-T cards were measured in each cylinder and used in the torque-effort calculations.

The existing and proposed (compressor with internal flywheels) systems were reanalyzed using the measured P-T card data. The correlation between the measured and calculated torsional amplitudes of the existing system was greatly improved. The actual P-T data revealed significant fourth order pulsation, which was later determined to be caused by an acoustical resonance associated with the nozzle connecting the bottle to the cylinder. To reduce this pulsation at 4× running speed, more restrictive orifice plates with higher pressure drop were recommended at the cylinder-to-bottle flanges of the suction and discharge bottles.

Summary

- Systems containing reciprocating compressors should be tuned to avoid operation on significant compressor excitation orders. Proper motor modeling techniques must be used to maximize accuracy of the natural frequency calculations (refer to the TORSIONAL MODELING OF MOTORS section).
- Using ideal P-V cards in the torque-effort calculations showed that certain compressor excitation orders were low amplitude. However, the actual excitation was significantly higher due to valve losses and/or acoustic resonances, which created pulsation in the cylinder. Therefore, P-V cards based upon actual measured data should be used whenever possible.
- The cylinder loading sequence could be optimized to reduce torsional excitation for single-staged systems. This unit was loaded 1-2-3-4-5-6 (starting from the motor end), which loaded across the machine. Loading one side of the machine first, such as cylinders 1,3,5, then 2,4,6, could have resulted in lower 4× excitation at load steps 2 and 6.

VISCOUS DAMPERS

Viscous dampers (Houdaille type) are often used in reciprocating engines to help limit torsional vibration and crankshaft stresses (Brenner, 1979). 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.

Equivalent Damping and Inertia

A viscous damper consists of a flywheel that rotates inside the housing, which contains a viscous fluid such as silicon oil (Figure 16). An untuned damper does not contain an internal torsional spring.

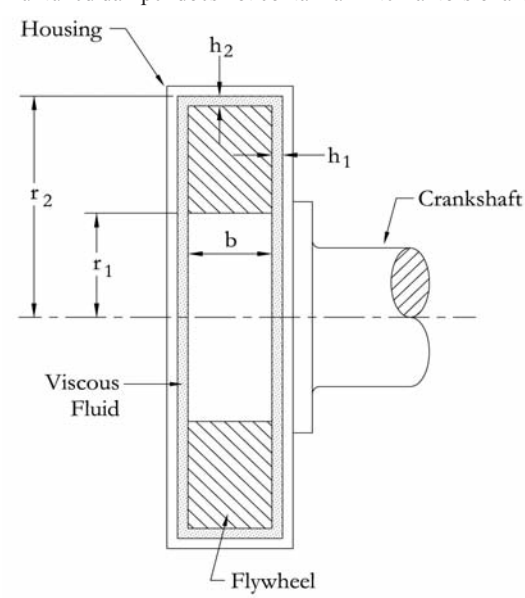


Figure 16. Untuned Damper.

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, h_1 and h_2, and/or the fluid viscosity, μ. From the *Shock and Vibration Handbook* (Harris, 1996), the damping constant is:

$$c = 2\pi\mu\left[\frac{r_2^3 b}{h_2} + \frac{r_2^4 - r_1^4}{2h_1}\right] \text{ in-lb-sec} \tag{6}$$

According to Den Hartog (1985), the optimum damping, c_{opt}, for maximum energy dissipation is:

$$c_{opt} = I_d \omega \tag{7}$$

where I_d is the inertia of the flywheel inside of the damper and is the frequency of oscillation. The equivalent damper inertia can be determined from:

$$I_{eqv} = \frac{I_d}{1 + (I_d \omega / c)^2} + I_h \tag{8}$$

where I_h is the inertia of the damper housing. For optimum damping, Equation (7) is substituted into Equation (8) such that the equivalent damper inertia is equal to half of the flywheel inertia plus the housing inertia.

$$I_{eqv} = \tfrac{1}{2} I_d + I_h \tag{9}$$

Engine manufacturers may provide a lumped inertia value for a damper. As shown in Equation (9) for optimum damping, the equivalent damper inertia is equal to half of the flywheel inertia plus the housing inertia. If the lumped inertia is used for the damper, the system natural frequency calculations will be satisfactory, but will not predict the internal flywheel resonance, which is usually well damped and not of concern. The damping ratio of the internal damper flywheel mode can be checked by hand using vibration equations for a single degree of freedom (SDOF) system.

The preferred method is to model a damper using separate inertias for the internal flywheel and housing. These two inertias are connected by equivalent damping and stiffness properties. The torsional analysis program can determine the effective damping by calculating the damped eigenvalues or using the half power point method (Meirovitch, 1986; Thomson, 1993) on the response curves. Also, the heat build-up inside the damper should be calculated and compared to the allowable value provided by the manufacturer.

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. 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 to provide the additional damping and heat capacity. This is provided the dampers are located external to the engine case. Some dampers are located inside the engine case, which may not have sufficient room and/or cooling capability.

Service Life

Viscous dampers have a 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.

Continuous heat absorption reduces the fluid viscosity over time. One manufacturer of dampers recommends changing them when the fluid viscosity has reduced 50 percent and the efficiency is approximately 80 percent. According to that manufacturer, this occurs after approximately 25,000 to 30,000 hours of service (Simpson Industries, 2001). Another manufacturer recommends replacement of dampers every 24,000 to 35,000 hours depending on the engine model (Superior, 1997).

Under extreme temperature conditions, the damper should be replaced more frequently since the silicon fluid will degrade more rapidly. If overheating occurs, the fluid could turn black as shown in Figure 17. Loss of or solidification of the fluid will cause the internal damper flywheel to seize to the housing and would no longer provide damping. Burnt or discolored paint on the outside of the damper is another sign of overheating.

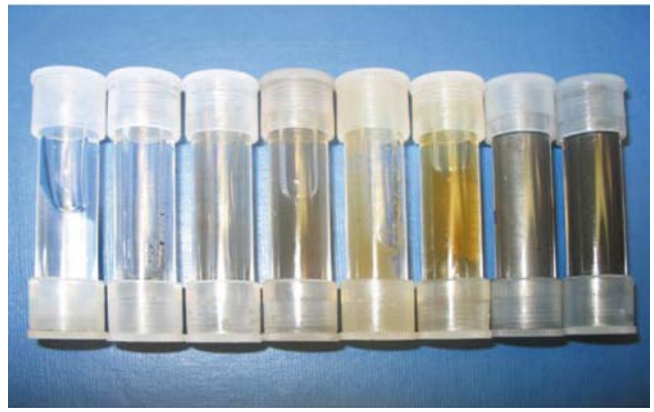
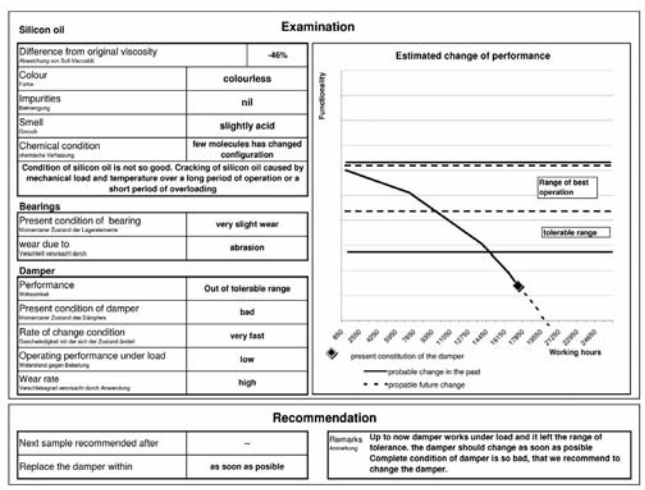


Figure 17. Damper Fluid Colors. (Courtesy Hasse & Wrede, 2009)

There are several companies that can test and remanufacture viscous dampers. A small amount of silicon oil is drained from the damper and sent to a lab where the fluid is then analyzed to determine the damper condition. Periodic checks help determine the rate at which the silicone fluid is degrading. A sample test report is shown in Figure 18.



Silicon oil — Examination

Difference from original viscosity	-46%
Colour	colourless
Impurities	nil
Smell	slightly acid
Chemical condition	few molecules has changed configuration

Condition of silicon oil is not so good. Cracking of silicon oil caused by mechanical load and temperature over a long period of operation or a short period of overloading

Bearings	
Present condition of bearing	very slight wear
wear due to	abrasion

Damper	
Performance	Out of tolerable range
Present condition of damper	bad
Rate of change condition	very fast
Operating performance under load	low
Wear rate	high

Estimated change of performance

Recommendation

Next sample recommended after	–
Replace the damper within	as soon as posible

Remarks: Up to now damper works under load and it left the range of tolerance. the damper should change as soon as posible Complete condition of damper is so bad, that we recommend to change the damper.

Figure 18. Sample Damper Fluid Test Report. (Courtesy Hasse & Wrede, 2009)

Testing for Damping Value

By varying the engine speed and tracking an order that passes through the first crankshaft mode, the torsional natural frequency (TNF) and system damping can be determined from the experimental data. Instrumentation and torsional measurement techniques are discussed in the TORSIONAL MEASUREMENTS section.

The torsional natural frequency of concern was excited by 4.5× engine speed near 470 rpm. The TNF was determined to be 35.2 Hz with a maximum torsional response on the damper end of 0.30 degrees peak-to-peak. A response curve was fit to the measured data shown in Figure 19.

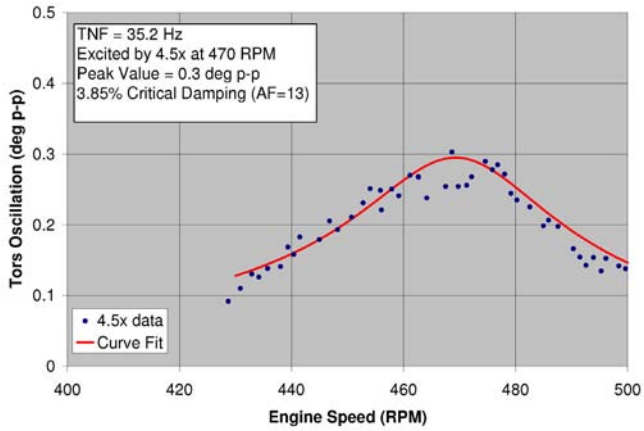


Figure 19. Damping Value.

At resonance, the response peak is equal to the normal response from excitation multiplied by the amplification factor. The response curve was determined using the magnification factor:

$$\left|G(i\omega)\right| = \frac{1}{\sqrt{[1-(\omega/\omega_n)^2]^2 + (2\zeta\omega/\omega_n)^2}} \tag{10}$$

where $|G(i\omega)|_{max} = AF$.

Damping values were guessed until the curve closely matched the measured data points. From visual inspection, this seems to be more accurate than using the half-power bandwidth method, which would be difficult to apply due to the scatter in the measured data points.

The AF was estimated to be 13. This corresponds to a damping ratio, ζ, of 3.85 percent where:

$$AF = \frac{1}{2\zeta\sqrt{1-\zeta^2}} \approx \frac{1}{2\zeta} \tag{11}$$

The damped natural frequency, ω_d, is related to the undamped natural frequency, ω_n, as:

$$\omega_d = \omega_n\sqrt{1-\zeta^2} \tag{12}$$

Note that for lightly damped systems ($\zeta < 0.05$ or $AF > 10$), the damped and undamped natural frequencies will be very similar (Meirovitch, 1986). In this case, the undamped frequency was 470 cpm and the damped frequency was 469.7 cpm. Most torsional systems are considered to be lightly damped.

The damper had just been replaced in this engine and was considered in new condition. By comparing test results from follow-up measurements, the damping value can be compared to the baseline value.

As the damper deteriorates, the amount of damping will decrease. The resonant frequency could also shift if the damper flywheel locks up with the damper housing in which case no damping would be provided. Note that the maximum torsional oscillation alone is not a good indicator of damper health since the peak value also depends on the load conditions.

TORSIONALLY SOFT COUPLINGS

A torsionally soft coupling can be used when a flexible connection is required between components. The coupling is termed "soft" because it typically has a lower torsional stiffness than the shafts that it is connecting. Table 3 gives a brief description of six types of soft couplings sometimes used in reciprocating applications. Possible advantages and disadvantages of each type are listed. Note that a torque shaft has been included as a soft coupling. Although not technically a coupling, it consists of a long, flexible shaft installed between components and has a low torsional stiffness.

Table 3. Torsionally Soft Couplings.

Type	Description	Advantages	Disadvantages
Rubber-in-Compression	Transmits torque through rubber blocks compressed between two steel spider flanges. Has a variable torsional stiffness that increases with mean torque.	Capable of high mean torque and large shock loads. High damping.	Limited life of rubber elements due to generated heat. Variable stiffness can be higher than other couplings and can pose a problem in reciprocating machinery with varying torque. Can cause TNF to track with speed.
Rubber-in-Shear	Uses rubber-in-shear elements with a constant torsional stiffness.	Low torsional stiffness and high damping.	Limited life of rubber elements due to generated heat. High service factor required for reciprocating applications.
Helical-Spring	Uses compressed steel springs with constant torsional stiffness	Wide range of torque and stiffness values. Optional damper pads.	Helical-spring couplings can be massive and expensive. Special tools are required for spring replacement.
Leaf-Spring	Radially arranged steel leaf springs with constant torsional stiffness. Usually filled with oil for damping.	Low torsional stiffness and high damping.	Normally requires pressurized oil supply through a hollow shaft.
Magnetic	Conductor Assembly with copper rings (on driver-end) transmit torque via air-gap to high energy permanent magnets (on driven-end)	Transmitted torque and stiffness are controlled by width of air-gap. Variable speed control for some applications. Torque-overload protection.	Limited horsepower range and limited reciprocating machinery installations (as of 08/2002).
Fluid Filled	Coupling or torque convertor that contains fluid, fluid levels can be adjusted for various slip	Isolates machines, does not allow dynamic torque to be transmitted, may add some damping	Heavy, possible unbalance, over-filling can cause seals to rupture, some inefficiency due to speed slip
Torque Shaft	Typically long shaft with reduced diameter. Can be used in conjunction with hollow quill shaft and flywheel.	Very predictable torsional stiffness. All steel construction.	Can be very long, making system layout awkward. No damping. Could be subject to fatigue damage or lateral critical speeds.

There are several instances when using soft couplings could be beneficial in controlling torsional vibration:

• *Isolate excitation between components*—Soft couplings are often used when an engine drives a gearbox or generator. The coupling can protect the driven components by absorbing the dynamic torque generated by the engine Feese, 1997).

• *Detune a torsional natural frequency*—A variable speed reciprocating compressor may have several damaging torsional resonances within the speed range due to the compressor excitation harmonics intersecting the first torsional natural frequency. With the use of a soft coupling, the first natural frequency can be tuned below minimum speed. In many instances, this is the only way to achieve the full desired speed range. Otherwise, the speed range has to be limited, or certain speed bands have to be avoided.

• *Add damping to the system*—Some soft couplings can attenuate high torsional amplitudes that are the result of a resonant condition. For couplings with rubber elements, this is accomplished through hysteretic damping. Hysteretic damping (internal friction) dissipates heat when the shaft or coupling material is twisted due to torsional vibration (Wright, 1975). Note that this differs from Coulomb damping that can take place when differential motion occurs in shrink fit hubs, keyed, splined or bolted connections and sliding surfaces that are not properly lubricated. The leaf spring type of coupling provides viscous damping since it is oil-filled. As described previously, viscous damping dissipates energy by shearing of a fluid film. The dynamic magnifier for most resilient couplings, which is a measure of the damping properties, typically ranges from 4 to 10. Dynamic magnifiers for standard flexible-disc and gear type couplings are approximately 30.

Several factors must be considered when choosing a soft coupling. For rubber couplings, the tradeoff is usually increased maintenance, since the rubber degrades over time due to heat and environmental factors. Most coupling manufacturers state that the life of the rubber elements can be four to five years, assuming ideal operating conditions. However, the actual life may be significantly less if the coupling is subjected to heat or a harsh environment (e.g., oil mist, ozone, etc.). Special silicon blocks may help to extend life in high temperature environments.

In many cases, excessive torques can occur in a soft coupling during a start-up or shutdown, particularly loaded shutdowns. Therefore, it may be necessary to perform a time-transient analysis of the system, so that the peak torques can be compared to the coupling allowable levels. This analysis may also include calculating fatigue damage in a torque shaft. Alternating shear stress above the endurance limit would cause a torque shaft to have a finite life.

Variable Stiffness

When using a rubber-in-compression coupling, the torsional stiffness of the rubber blocks varies with mean torque and temperature. The actual coupling stiffness can vary by as much as ±35 percent from catalog values, which should be considered in the analysis. Figure 20 shows an example of how the torsional stiffness of a compression type coupling could vary with load and temperature. This nonlinear stiffness can make it difficult to tune natural frequencies between orders. Also, transient events such as synchronous motor start-ups require special analysis techniques when a nonlinear stiffness coupling is involved (Feese, 1996).

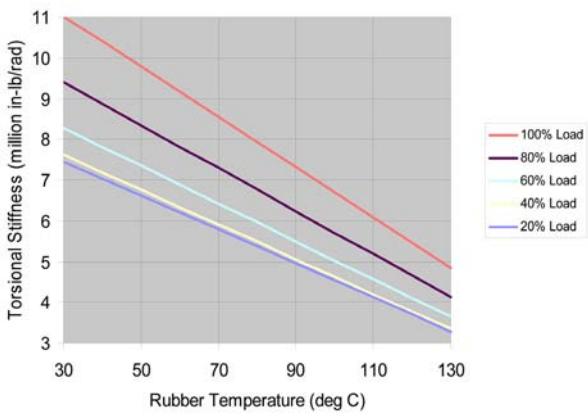


Figure 20. Variable Torsional Stiffness of Coupling with Rubber Blocks in Compression.

One may think that a system containing a variable stiffness coupling can never continuously operate on a torsional natural frequency that causes high dynamic torque in the coupling since the rubber blocks would heat-up and lower the torsional stiffness thus automatically detuning the resonance. However, this concept is incorrect. Running at or near a resonant condition for just a few minutes could elevate the temperature beyond the melting point of the rubber blocks and damage the coupling. If the torsional natural frequency was just above the operating speed, a temperature rise in the coupling could actually lower the torsional stiffness and TNF, creating a resonant condition, and making the dynamic torque more severe. Also, the torsional natural frequencies of a system could be different for cold starts and hot restarts.

Service Factors

When using a rubber-in-shear coupling, an appropriate service factor should be used to allow for possible torque overload during start-up, shutdown, or any other unexpected condition. To ensure an adequate safety factor, several coupling manufacturers require that the catalog nominal torque rating of the selected coupling size be at least 1.5 to 2 times the transmitted torque of the system. This would correspond to an actual service factor of 4.5 to 6 since the catalog rating is usually 3 times the torque being transmitted. The coupling manufacturer should always verify the selection. Other factors such as end-float and allowable misalignment should also be addressed with the coupling manufacturer.

The vibratory torque and heat dissipation (for damped couplings) must be calculated and should also be reviewed by the coupling manufacturer for acceptability. The allowable vibratory torque amplitude is typically 20 percent to 30 percent zero-peak of the coupling rated torque. The heat dissipation is a function of the vibratory torque and frequency and is normally specified in terms of power loss (Watts).

Detuning Torsional Resonances

In instances where a very soft coupling is required to achieve an acceptable system, a multi-row coupling (two or three rows of soft elements in series) can sometimes be used; however, this is not normally recommended since a new torsional resonance of the coupling occurs that can result in damaging dynamic torque and heat loads. Instead, a standard soft coupling in conjunction with a flywheel is preferred.

Inertia can usually be added with an integrated flywheel at the coupling hub or by using internal flywheels sometimes called "donuts" inside some compressor frames. Rotating counterweights can also provide additional inertia similar to flywheels.

Since a soft coupling is often used to tune a torsional natural frequency, either below running speed or between orders, care must be taken to ensure all appropriate cases are analyzed. For example, misfire in a four-stroke engine can result in increased torque excitation at 0.5× and 1× running speed. If the torsional natural frequency is near these harmonics, coupling damage could occur. Similarly for reciprocating compressors, all load steps (e.g., pockets, unloaders, deactivators) should be analyzed throughout the operating envelope, as the loading will influence the torque excitations. In systems with variable frequency drives (VFD), torque harmonics generated by the VFD can excite the resonance during start-up, and, therefore, must be evaluated. Also, the ramp rate or frequency at which the VFD controls the speed of the units should not be coincident with a torsional natural frequency.

Case History—Gearbox Failure after Installation of a New Motor

- Equipment:
  Constant speed induction motor driving a two-throw reciprocating compressor through a speed reducing gearbox
  Motor rated 400 hp (298 kW) at 1,780 rpm
  Compressor speed = 450 rpm
- Problem:
  Catastrophic gearbox shaft failure
- Cause:
  TNF near 2× running speed was excited by high dynamic torque generated by two-throw compressor.
- Solution:
  Installed torsionally soft, damped coupling

An existing compressor system at a refinery experienced several gearbox and low speed coupling failures after the original motor was replaced with a new and different motor. A picture of a failed gearbox is shown in Figure 21. A torsional analysis was not performed on the unit before the replacement motor was installed. Problems were not reported with the sister units, which still had the original motors.

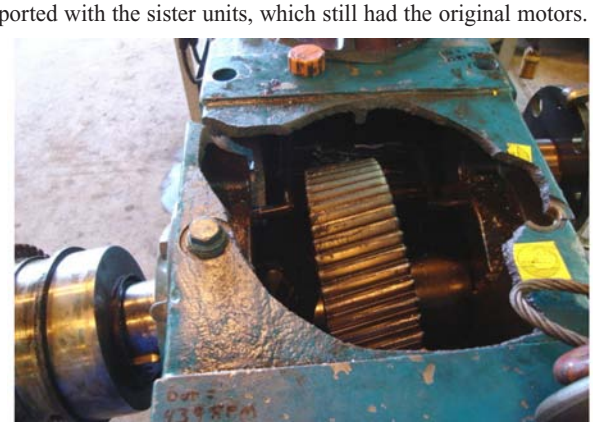


Figure 21. Failed Gearbox.

The system consisted of an induction motor driving a two-throw, two-stage reciprocating compressor through a speed reducing gearbox. A standard disc-pack coupling was used between the motor and gearbox, while a steelflex style coupling was used between the gearbox and compressor. The motor was rated for 400 hp at 1,780 rpm. The compressor speed was 450 rpm.

Field testing and a torsional analysis were subsequently performed on the system to determine the cause(s) of the failures and to determine a permanent solution.

Field Test

The field testing consisted of acquiring the following measurements:

- A strain gage telemetry system was mounted to the gear output shaft to measure mean and dynamic torque.
- Triaxial accelerometers were mounted on the gearbox to measure displacement (mils peak-to-peak).
- High-frequency accelerometers were glued to the gearbox case to monitor acceleration (g's) at gear-mesh frequency and its multiples.
- Motor current readings were obtained using a wrap-around current transformer. Current measurements were obtained on the three phases at the electrical junction box at the motor.
- Two optical keyphasors were used, one for the motor speed, and the other for the compressor speed.
- Dynamic pressure transducers were added to measure pressure time waves in the compressor cylinders. Pressure taps were located on top of the compressor cylinders for the head and crank ends.

A plot of the gear shaft torque during startup is shown in Figure 22. The time from rest to operating speed was approximately 4.4 seconds. The peak torque value was 110,000 in-lb (12,428 N-m). When a system is started, the first TNF will normally be excited. The period between torque cycles was 0.0683 seconds. Using the period between torque cycles, the corresponding frequency would be 14.6 Hz. This corresponds to approximately 2× compressor speed.

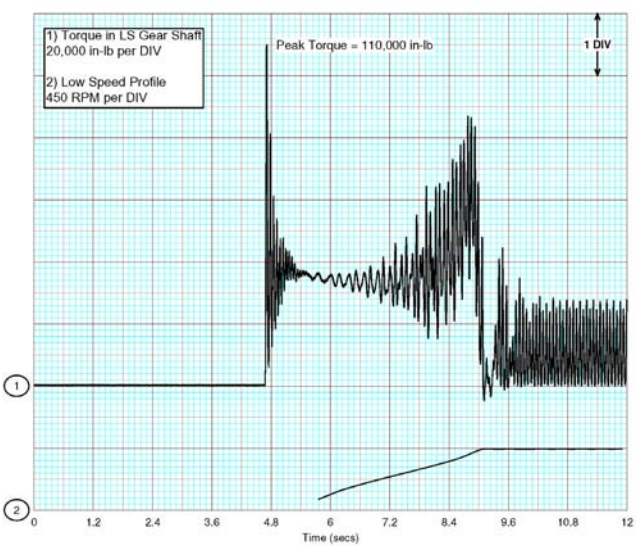


Figure 22. Start-Up Torque.

The dynamic torque varies depending on the compressor loading. Frequency spectra were taken of the torque readings for each load condition as shown in Figure 23. As summarized in Table 4, the majority of the dynamic torque is caused by energy at 2× compressor speed. Although a two-throw compressor inherently produces a significant dynamic torque at 2× compressor speed, the amplitudes are being amplified due to the close proximity to the first torsional natural frequency.

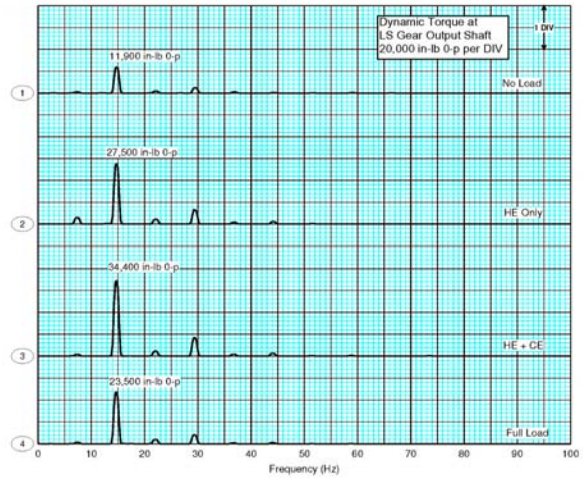


Figure 23. Dynamic Torque at Various Load Conditions.

Table 4. Summary of Torque Measurements—Original System.

Operating Condition	Compr. Speed (RPM)	Motor Current (amps)	Transmitted LS Torque (in-lb)	Dynamic Torque at 2× Compr. Speed (in-lb 0-pk)	Overall Dynamic LS Torque (in-lb 0-pk)	Percent Dynamic Torque
No Load	442.5	n/a	10,000	11,900	13,500	135%
HE Loaded	441.6	45	24,000	27,500	32,000	133%
HE + CE	440.7	55	31,000	34,400	36,500	118%
Full Load (HE+CE+Pockets)	440.2	64	36,000	23,500	27,500	76%

Gear manufacturers typically recommend that the dynamic torque at the gear mesh be limited to no more than 25 percent to 40 percent of the transmitted torque. As shown in Table 4, the system exhibits 76 percent to 133 percent dynamic torque at certain load conditions. This results in torque reversal and backlash of gear teeth. In addition, this is a single helix gearbox so this alternating torque also creates high thrust loads. It has been reported that the thrust bearings were damaged in the original gearbox.

The field measurements identified a torsional natural frequency of the system near 2× running speed of the compressor. High dynamic torque was measured between the low speed coupling and gear that would explain the previous failures. The system as configured would be expected to fail again, so changes were needed to detune the TNF away from 2× compressor speed.

Torsional Analysis

To evaluate possible solutions, a steady-state torsional analysis of the existing compressor/drive train was performed. The analysis showed that the first torsional natural frequency of the system (13.4 Hz) was in close proximity to 2× compressor speed (14.7 Hz). The highest torsional excitation produced by the two-throw compressor occurs at 2× compressor speed. The first torsional mode shape shows that the majority of the twisting occurs through the low speed coupling resulting in high dynamic torque in the coupling as well as excessive shear stresses in the gearbox low speed shaft.

A parametric study was performed to detune the first torsional natural frequency away from 2× compressor speed. Several different coupling and coupling/flywheel configurations were analyzed in an attempt to provide adequate separation margin. The optimum option selected was to isolate the compressor torsional excitation from the drive train and lower the first torsional natural frequency below 1× compressor speed. To accomplish this required utilizing a torsionally soft coupling. A rubber-in-shear type coupling was selected, which resulted in acceptable torsional response of the system based on the analysis results.

Follow-up Field Test

Follow-up field testing was performed with the new coupling installed between the gearbox and compressor flywheel (Figure 24). The measurements showed it was successful in de-tuning the first

torsional natural frequency away from 2× compressor speed. The measured first torsional natural frequency of 4.5 Hz compared well with the torsional analysis which predicted 4.2 Hz with the soft coupling.

Figure 24. New Rubber-in-Shear Coupling.

As shown in Table 5, the steady-state dynamic torque was significantly lower with the soft coupling installed compared to the torsional response with the original coupling. With the exception of the compressor unloaded condition, the dynamic torque was less than 25 percent of the transmitted torque, especially when both compressors cylinders were double-acting.

Table 5. Gearbox Dynamic Torque.

Compressor Load Condition	Gearbox High Speed Shaft Torque in-lb zero-peak (N-m 0-pk)	
	Original Coupling	New Coupling
Unloaded	5,000 (565)	2,100 (237)
HE Loaded	12,500 (1412)	2,500 (282)
HE + CE Loaded	7,000 (791)	750 (85)
Full Load (HE+CE+Pkts Closed)	7,200 (813)	750 (85)

The system has been running satisfactorily after the modification and no further gearbox failures have been reported.

TORSIONAL MODELING OF MOTORS

Motor manufacturers will normally provide the total rotor inertia, which will include the shaft, core, and any added components (fans, etc.). The shaft stiffness is also provided and is usually given from the coupling (drive) end of the shaft to the center of the rotor, or from the coupling end of the shaft to approximately one-third the length into the motor core. The shaft stiffness is normally based on the base shaft diameter only. Some torsional analysts use this single mass-spring system for the torsional analysis.

Experience has shown that modeling a motor as a single mass-spring system can result in inaccurate and/or missed torsional natural frequencies because the flexibility and inertial distribution through the motor core is not considered. Stiffening effects due to the motor core construction can also influence the torsional natural frequencies. In some cases, small differences may be sufficient to result in torsional problems since the system may be operating on or very near resonance, although the analysis using the single mass-spring motor model may have indicated an adequate separation margin.

There are several types of motor core construction:

- Machined or welded webs/spiders attached to the shaft
- Keyed or shrunk-on laminations
- Squirrel cage—forged shaft with rotor bars in grooves

Torsional Stiffness

The stiffness of the motor shaft can be influenced by the various types of construction. In general, the machined or welded webs can add significant stiffness (typically 10 percent to 40 percent over the base shaft diameter stiffness), while keyed on laminations typically add minimal stiffness. The effects of shrunk-on laminations can vary depending on the actual interference fit.

Several procedures have been developed to approximate the stiffness of spider motors, including API (API 684, 1996) and Frei, et al. (1996). The methods basically involve determining an equivalent shaft diameter based on the properties of the base shaft, the spider arms (number, length, thickness, etc.), and the construction (integral or welded arms). Any effects due to the laminations around the spiders are not considered in these methods.

A finite element analysis of the webbed motor shaft can provide even more accurate results. Although this may initially require additional analysis time to build the finite element model, a template can be created so that future motors can be evaluated more quickly. The required input parameters include the shaft length and diameter, the number of webs, and the length and width of each web. The laminations could also be included. The torsional stiffness is determined by rigidly fixing one end, applying a known moment to the free-end, and calculating the resulting twist. This stiffness can then be converted to an equivalent cylindrical shaft diameter over the motor core length.

Figure 25 shows a finite element model of an induction motor with six welded spider arms (webs) that was created with commercially available software. Table 6 summarizes some of the motor core dimensions and lists the resulting equivalent diameters based on the finite element analysis, and the API (API 684, 1996) and Frei, et al. (1996), methods. The equivalent diameter determined from the finite element analysis was 7 percent larger than the base shaft diameter, which corresponded to a 33 percent increase in torsional stiffness (K_t is proportional to D^4). Note that different calculations would need to be performed to determine the equivalent flexural diameter for a lateral critical speed analysis.

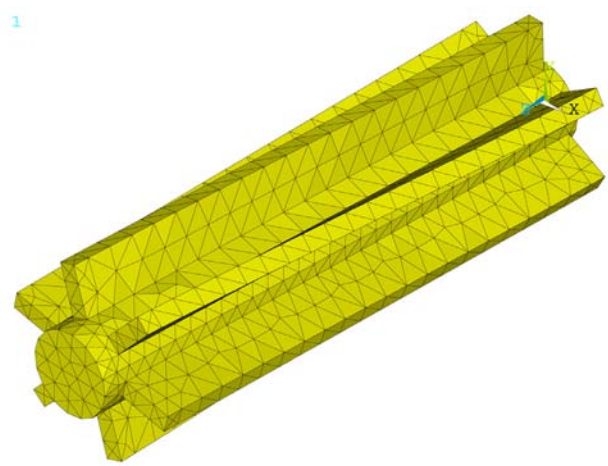


Figure 25. Finite Element Model of Induction Motor Core.

Table 6. Equivalent Shaft Diameter for Six Arm Motor Core.

Motor Core Length	64 in (1626 mm)
Length of Arm Above Shaft	6 in (152 mm)
Arm Thickness	2 in (51 mm)
Base Shaft Diameter	9.5 in (241 mm)
Equivalent Diameter using FEA	10.2 in (259 mm)
Equivalent Diameter using API [23]	10.1 in (257 mm)
Equivalent Diameter using Frei [24]	9.6 – 10.4 in (244 – 264 mm)

Case History—Effects of Motor Model on Torsional Natural Frequencies

- Equipment:
  Variable speed motor driving a six-throw, single-stage, natural gas reciprocating compressor through a flexible disc coupling
  Rated 5,500 hp (4101 kW) at 1,200 rpm
  Speed Range = 720 to 1,200 rpm
- Problem:
  Compressor crankshaft torsional failure
- Cause:
  High shear stress due to operation on second TNF
- Solution:
  Reduced flywheel inertia to increase second TNF. Modified speed range to 850-1,200 rpm to avoid operation on first TNF.

A torsional analysis was originally performed using a single mass-spring motor model. The first TNF was calculated to be 69 Hz (4,140 cpm), while the second TNF was predicted to be 124 Hz (7,446 cpm). Based on the results of that analysis, the allowable speed range was determined to be 720 to 1,200 rpm.

After approximately one year of operation, a failure occurred in the compressor crankshaft. The failure consisted of a spiral crack that originated from the coupling keyway, which is indicative of high torsional (shear) stress.

To investigate the cause of the failure, a torsional field test was performed. The first TNF was measured to be 70 Hz and the second TNF was 120 Hz (Figure 26). The first TNF was within 1.5 percent of the calculated frequency, and the second TNF was within 3.5 percent, which are typical margins of error. However, the 6× compressor harmonic, which is a strong excitation order for this single-stage, six-throw compressor was coincident with the second TNF while operating at the maximum speed (1,200 rpm). This resulted in unintensified crankshaft stresses of 10,000 psi zero-peak (69 MPa 0-pk). Since the coupling hub was keyed to the crankshaft, the intensified stresses at the base of the keyway were over 30,000 psi zero-peak (207 MPa 0-pk). This stress level was well above the endurance limit of the shaft material and was determined to be the cause of the failure.

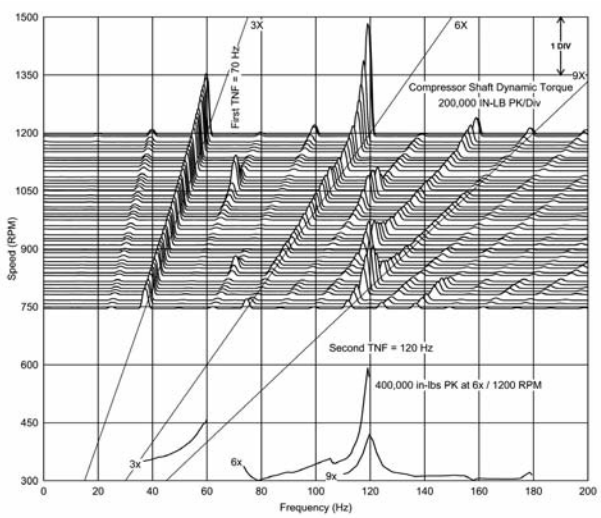


Figure 26. Waterfall Plot of Compressor Torque—Original System.

A new torsional analysis was performed to evaluate modifications to the system that would provide a safe separation margin between the second TNF and the operating speed of 1,200 rpm. The motor was modeled with multiple masses and the stiffening due to the motor webs was considered using the finite element model shown in Figure 25.

The more accurate model of the motor shaft without other normalization predicted the first TNF at 70 Hz while the predicted second TNF was 120 Hz. These correlated exactly with the measured frequencies. It is interesting to note that, although the second TNF was primarily a compressor crankshaft mode, the frequency was still sensitive to the effects of the motor model.

To detune the system, the original compressor flywheel was replaced with a smaller internal flywheel located inside the compressor frame. As shown in Figure 27, this reduced the inertia of the compressor and increased the second TNF to 130 Hz, which allowed safe operation at 1,200 rpm. However, the minimum speed had to be increased to 850 rpm because this modification also increased the first TNF to 76 Hz. Intersection of this resonance with the 6× compressor harmonic near 760 rpm had to be avoided to prevent excessive motor shaft stresses and coupling chatter.

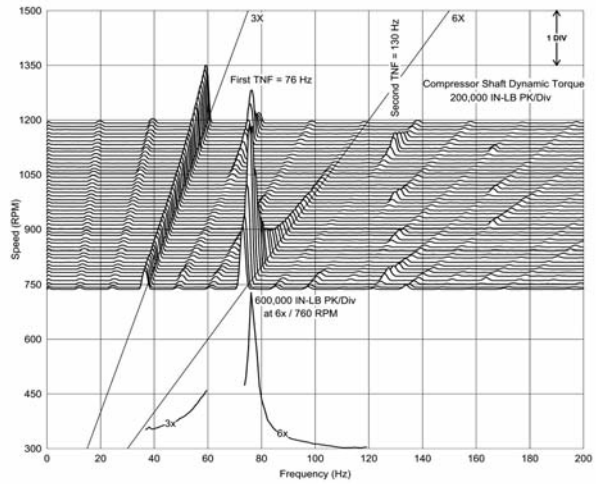


Figure 27. Waterfall Plot of Compressor Torque—Modified System.

The motor model can affect the calculated torsional natural frequencies of a system, even modes that are primarily associated with the driven equipment. In some instances, the difference in frequency may only be a few percent, but can have detrimental results, especially when tuning between strong excitation orders. A multi-mass model should always be used to represent a motor with a long core. The stiffening effects of the motor core should also be included in the torsional model.

Case History—VFD Added to Motor-Driven Two-Throw Compressor

- Equipment:
  Induction motor driving a two-throw, two-stage compressor through a flexible disc coupling
  Originally operated at constant speed of 890 rpm, but installed a new variable frequency drive (VFD) with a proposed speed range of 600 to 900 rpm.
- Problem:
  Operation over new speed range resulted in excessive torsional vibration.
- Cause:
  VFD was added without performing a torsional analysis. Torsional natural frequencies were excited within the new speed range.
- Solution:
  The speed range was modified to avoid operation on resonance.

An existing constant speed reinjection compressor had been in service for a number of years and had operated satisfactorily from a torsional perspective. To increase operational flexibility, the

owner needed the compressor to operate over a broader range of flows so a variable frequency drive was selected. The original operating speed of the unit was 890 rpm and the proposed speed range with the new VFD was 600 to 900 rpm.

A torsional analysis, which should be mandatory for a variable speed compressor, was not performed prior to ordering the VFD. Since reciprocating compressors can produce significant excitation at multiple frequencies (refer to the TORQUE-EFFORT CONSIDERATIONS section), operation over a wide speed range can be damaging to the unit. The owner was advised that a torsional analysis and/or field testing should be performed prior to commissioning the unit with the new VFD to determine the acceptability over the speed range.

Although VFDs can produce dynamic torque at multiples of electrical frequency, newer pulse width modulation (PWM) drives are generally smooth, especially compared to the dynamic torque levels produced by a two-throw reciprocating compressor. Therefore, the focus of this field study was on the possible interference of the major compressor orders with the torsional natural frequencies of the system within the proposed operating speed range.

For the field test, a strain gage telemetry system was installed on the motor shaft near the coupling. An optical tachometer was used to monitor rotational speed. The compressor was operated at multiple conditions, including an unloaded start-up/shutdown, and at various load steps over the operating speed range.

The VFD was used to vary the operating speed from 900 to 600 rpm with the compressor loaded. Figure 28 is a time-domain plot showing the motor speed and the overall vibratory torque in the motor shaft during the speed sweep. Figure 29 is a speed waterfall plot of the motor dynamic torque during the same speed run. A torsional natural frequency can be identified at 71 Hz and is excited by 6× running speed while operating at 710 rpm.

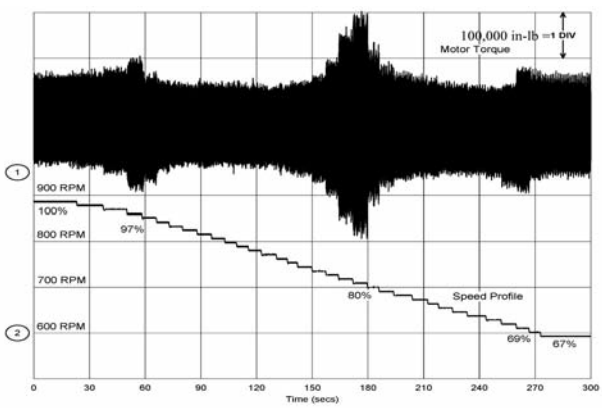


Figure 28. Time Domain Plot of Motor Torque and Speed During Speed Run.

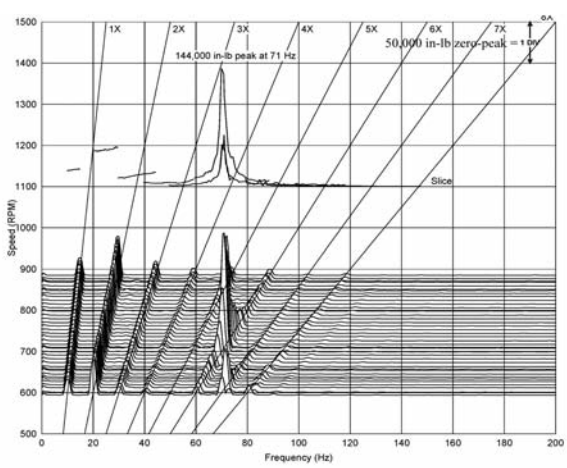


Figure 29. Waterfall Plot of Motor Torque During Speed Run.

The peak torque amplitude was 144,000 in-lb zero-peak (16,270 N-m 0-pk), which corresponds to an unintensified shear stress of 4,710 psi zero-peak (32.5 MPa 0-pk) in the shaft. The intensified shear stress (due to the keyway at this location) was 14,130 psi zero-peak (97 MPa 0-pk). This was well above the allowable shear stress limit of the motor shaft of 10,600 psi (73 MPa), so continuous operation of the unit near 710 rpm would have resulted in a shaft failure. Lower amplitude, but still significant, response peaks also occurred near 609 rpm (due to excitation at 7× running speed) and near 852 rpm (due to excitation at 5× running speed).

A torsional analysis was subsequently performed to evaluate the response of the entire train. The measured torsional data was used to normalize the model. The analysis confirmed that operation in the vicinity of 710 rpm would be damaging to the motor shaft and the flexible disk coupling as well. To ensure reliable operation, a restricted speed range was required. Alternatively, a torsionally soft (e.g., rubber-in-shear) coupling and/or different flywheel would be required to detune the system. In the end, the owner opted to program the VFD to block-out the dangerous speeds.

OTHER CONSIDERATIONS

There are numerous other things to consider from a torsional perspective when evaluating a system with a reciprocating engine and/or compressor.

Shaft Properties

The shaft size, material strength and stress concentration factors all have significant effects on the long-term reliability of a system. Therefore, it is important to use good engineering practices when designing shafts and selecting materials.

- Experience has shown that some rotating equipment shafts (e.g, motors, generators, centrifugal pumps, etc.) may not be adequately sized for use with reciprocating equipment, which can produce significant dynamic torque modulations. A good "rule-of-thumb" for motors driving reciprocating compressors (or generators driven by engines) is to make the minimum diameter of the motor shaft between the core and coupling (including the drive-end bearing journal) at least equal to the compressor crankshaft diameter. Another guideline for sizing a shaft in reciprocating service is to limit the mean stress level (excluding stress concentration) to 3,000 psi or less.

- A stress concentration factor (SCF) is a stress multiplier that should be minimized, but is unavoidable since shaft steps, welds, and/or keyways are inevitably required. A common reference for determining SCF's is Peterson's *Stress Concentration Factors* (1974). The worst-case SCF occurs with a square cut fillet radius and would be approximately 5 for a keyed shaft and 2 for a stepped shaft. When possible, keyways should be avoided, particularly at coupling hubs, since the shaft diameter in this area is often reduced. Instead, an interference fit (shrink or hydraulic) should be considered.

- Torsional shaft stresses (predicted or measured) must be compared to the endurance limit in shear for each shaft material. Shear endurance limits are not often readily available for many steels. A method for determining the endurance limit is provided by Shigley and Mischke (1989), which converts the ultimate tensile strength (UTS) to an endurance limit and include fatigue factors such as surface, size, and reliability. Note that although the ASME method (ASME B106.1M, 1985) has been withdrawn (meaning no active committee), it is similar to Shigley and Mischke (1989) and still considered valid.

Mean stress is subsequently considered using a Goodman diagram. For a steel shaft with 3,000 psi mean stress and typical values for surface, size, and reliability factor, the shear endurance limit will be roughly 12 percent (0-pk) of the UTS. Therefore, if the combined intensified shaft shear stress (total stress amplitude including any stress concentration factor) exceeds this value, a

fatigue failure could occur. A safety factor of at least two is recommended for design, particularly for reciprocating applications. With this safety factor, the allowable alternating shear stress is reduced to approximately 6 percent of the UTS. Therefore, it is important to use quality shaft material with high UTS values.

Systems with a Gearbox

Typically, torsional vibration problems are difficult to diagnose without special instrumentation because there are often no symptoms until after a failure has occurred. The exception to this is a system containing a gearbox. If torque modulation is sufficient to unload the teeth, then "gear hammer" can occur, producing an audible noise. Torsional vibration can also couple into lateral vibration due to variation in torque at the gear mesh (Wachel and Szenasi, 1980). Severe torsional vibration can often be detected by shaft proximity probes and/or accelerometers attached to the case. The problem is distinguishing between torsional and lateral vibration in these readings.

Special consideration should be given to gearboxes used with reciprocating equipment. Several gearbox manufacturers recommend limiting the dynamic torque at the gear mesh to 30 percent to 40 percent of the rated torque during steady-state operation. To prevent backlash at low load conditions, the dynamic torque should not exceed the transmitted torque.

Transient events such as startups and emergency loaded shutdowns (ESDs) can also cause high dynamic torque at the gear mesh. Depending on the speed ramp rate during startup, engines and synchronous motors can cause peak torques at the gear mesh that are much higher than recommended for continuous operation. If the peak torque exceeds the catalog rating of the gearbox, then the manufacturer should be contacted to discuss potential gear tooth damage from repeated starts.

Case History—Integral Compressor Converted to Motor Drive

- Equipment:
  Integral engine-compressor converted to motor-driven compressor. A gearbox was also added.
  Motor rated 1,250 hp (932 kW) at 1,189 rpm
  Compressor speed = 303 rpm
- Problem:
  Excessive gearbox vibration
- Cause:
  The first TNF was coincident with 4× compressor speed (20 Hz), resulting in excessive dynamic torque in gearbox.
- Solution:
  Detuned first TNF by modifying low speed coupling

Background

An integral engine-compressor at a natural gas plant was converted to a motor drive. The integral engine-compressor was modified by removing the engine portion (eight power cylinders), but leaving the existing four compressor cylinders. The compressor was originally configured with two cylinders in gas service and two in propane refrigeration service, but was converted to all propane service during the modification. A speed reducing gearbox was installed and disc-pack type couplings were used on both sides of the gearbox, with the low-speed coupling bolted to the compressor flywheel. Fully loaded, the rated motor speed was 1,189 rpm and the compressor speed was 303 rpm.

Upon startup, the vibration levels were reportedly low, but after approximately two weeks of operation, vibration levels increased on the gearbox. The unit began experiencing frequent shutdowns due to high gearbox vibration. The unit was inspected multiple times, but no damage to the gearbox or other components was observed; however, the high vibration continued.

Various analytical studies, including an acoustical analysis, inertial unbalance study, and a torsional analysis, were performed by third parties prior to modifying the unit. A review of the torsional analysis showed that excessive dynamic torque was predicted in the gearbox, but had not been flagged as a potential problem. It was then decided that field testing would be performed to fully evaluate the dynamic characteristic of the unit.

Gearbox Vibration

Vibration data were obtained on the motor, gearbox, and compressor as the unit was operated at multiple load conditions and during startup and shutdown. The highest vibration was measured on the gearbox bearing at the LS gear output in the axial direction (parallel to the shafts). As shown in Figure 30, the maximum amplitude was 0.8 ips zero-peak (20 mm/sec 0-pk) at 237.5 Hz, which converts to 1 mil peak-to-peak (25 μm peak-to-peak) of displacement or 3 g's pk of acceleration. A secondary frequency was found at 360 Hz.

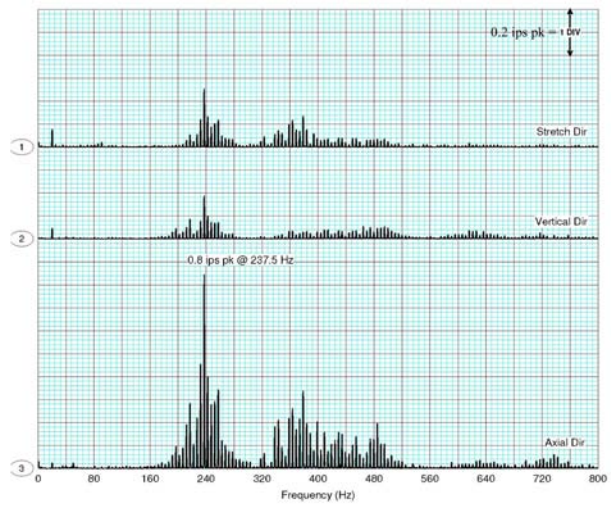


Figure 30. Vibration Spectra on Gearbox.

The gearbox has 40 pinion teeth and 157 gear teeth. Therefore, the gear-mesh frequency would be approximately 793 Hz, which is much higher than the measured frequency responses. Note in Figure 30 that the vibration levels at the gear mesh frequency were low.

While the unit was down, impact tests of the gearbox and LS gear shaft were performed to measure the mechanical natural frequencies. A natural frequency of the gearbox case was found in the axial direction near 245 Hz. Another natural frequency of the output shaft/LS coupling was found near 335 Hz. These two natural frequencies are near the peaks measured while running. Note that these natural frequencies could shift slightly when the gearbox is loaded.

Multiple side-bands having 5 Hz spacing were observed about the 237 and 360 Hz response peaks. Note that the compressor speed is 303 rpm, or approximately 5 Hz, indicating the side-bands are related to the low speed gear shaft and/or compressor. The sidebands near the two natural frequencies may indicate impacting (Taylor, 1994).

Time averaging was performed on the data and it was determined that the vibration signal was phase synchronized with the low-speed tachometer signal referenced to cylinder #1 outer dead center (ODC). The repeated impacts at 5 Hz excited these two natural frequencies of the gearbox case and shaft near 240 and 360 Hz. Possible sources of impacts could be the coupling hubs making contact or loss of lubrication in one spot of the gearbox bearing creating a contact zone. Another possibility would be that the compressor crankshaft was shuttling back and forth in the axial direction.

Torsional Vibration

Torsional vibration readings were obtained using a strain gage telemetry system and a laser vibrometer on the high speed coupling (refer to the TORSIONAL MEASUREMENTS section for a discussion of the measurement devices). The data were used to evaluate the torsional response and to compare with the torsional analysis previously performed. Compressor performance measurements were also obtained to determine the torque effort of the unit and to check for bad valves, leakage, etc. The speed could not be varied during operation because the motor did not have a variable frequency drive.

When a motor is initially started, a sudden torque is applied to the system, which can excite torsional natural frequencies of the system. By counting the number of cycles within a set time period, the first torsional natural frequency can be estimated. For this system, 10 cycles occurred within 0.5 seconds and the calculated TNF was 20 Hz. This is likely the first TNF. The energy at 4× compressor speed will excite the first TNF at 20 Hz when operating at 303 rpm. This is undesirable, as there should be a sufficient separation margin between a TNF and any significant excitation harmonic to prevent excessive dynamic torque from occurring.

Figure 31 shows that the torsional vibration at 4× compressor speed increased by a factor of 6 when the compressor went from being partially loaded to fully loaded. Note that the other orders or harmonics of torsional vibration stayed approximately the same and did not vary with load. Therefore, it was primarily the 4× compressor order that was very sensitive to compressor loading, which is coincident with the first TNF of the system.

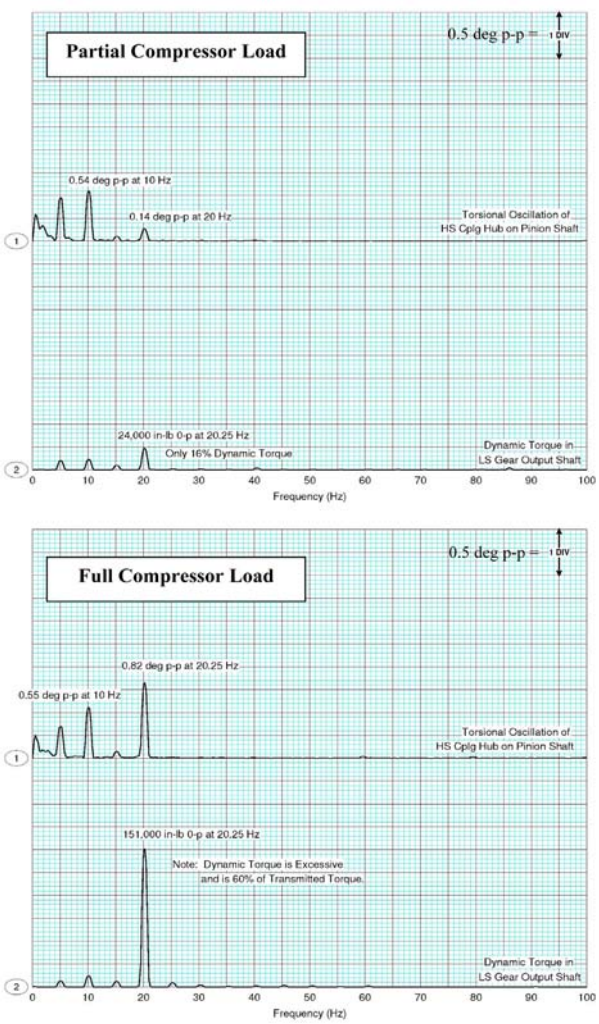


Figure 31. Torsional Response Measurements.

At full load, the dynamic torque amplitude was approximately 60 percent of the transmitted torque and reached amplitudes as high as 420,000 in-lb zero-peak (47,454 N-m 0-pk). The gearbox manufacturer has a baseline allowable for the alternating torque transmitted through the gear mesh. Alternating torques as high as 25 percent to 30 percent were considered acceptable, as long as the transmitted torque is sufficient to prevent reversing torque and backlash of gear teeth. It was concluded that the torsional vibration was excessive and could damage the coupling and/or gearbox.

Current pulsations were observed in the motor current readings, with sidebands occurring at ±20 Hz of the 60 Hz electrical frequency. Figure 32 shows an example of motor current in the frequency domain. At 60 Hz, the motor current was 168 amps or approximately 96 percent of full load (175 amps). These sidebands were likely due to torsional vibration of the motor core occurring at 4× compressor speed (20 Hz). The sidebands were 8 to 9 amps or approximately 5 percent of the fundamental.

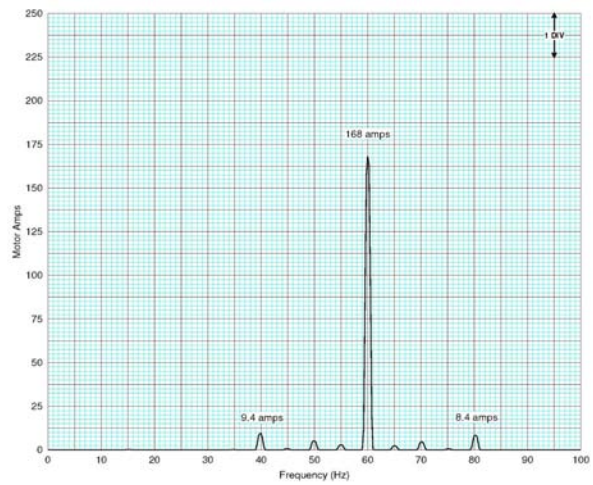


Figure 32. Frequency Spectrum of Motor Current.

The compressor torque-effort curves were developed from pressure data taken during the performance measurements. The strongest torque harmonics, including the effects of the reciprocating masses, were the second and fourth compressor orders. The significant fourth order torque (29 percent of transmitted torque) excited the first TNF.

Torsional Analysis

The third party analysis originally predicted a first TNF of 16.4 Hz, which did not match the actual TNF of 20 Hz measured in the field. In addition, the measured dynamic torque in the gearbox was approximately 60 percent of the transmitted torque and was much higher than predicted by the analysis. The difference between the predicted and actual values was due to several modeling deficiencies that were found in the analysis.

For example, the torsional stiffness values were taken directly from the drawing for the pinion and gear shafts without correcting for the coupling hub lengths (one-third penetration rule). This caused the torsional stiffness to be too low. It is not known how the motor was modeled or the motor shaft stiffness was calculated because the motor manufacturer stated that a shaft drawing was not produced for this project. For more accurate results, a shaft drawing was needed to model the motor rotor as multiple stations instead of only one lumped inertia value (as discussed in the TORSIONAL MODELING OF MOTORS section).

An improved torsional model was developed to more accurately represent the actual system. The predicted results were normalized to match the field data. The TNFs and mode shapes of the system with various modifications were then evaluated. The changes under consideration included a larger flywheel inertia, a rubber-type low-speed coupling, and modifications to the original coupling.

The most practical of these options was modifying the low-speed coupling. The modifications included replacing the spool piece and bull gear side disc pack, which resulted in a 40 percent reduction in torsional stiffness of the coupling. The lower stiffness reduced the first TNF from 20 Hz to 16.6 Hz (996 cpm), which is 3.3× compressor running speed, and 21 percent below 4× compressor running speed. The second TNF for the modified system was predicted at 54 Hz (3,266 cpm), which is 11× compressor running speed (or 2.7× motor operating speed).

Steady-state forced response calculations were made for compressor speeds from 200 to 400 rpm to evaluate the system torsional response when excited by the compressor. Compressor torque harmonics were based on the pressure/pulsation data from the field testing. At running speed, the stresses in all shafts were predicted to be below the allowable limits. The gear mesh torque was predicted to be 64,653 in-lb zero-peak (7,305 N-m 0-pk) at the operating speed. This is 25 percent of the rated full load torque.

Summary

With the new low speed coupling installed, the gearbox vibration was reduced from approximately 0.43 ips (11 mm/sec) to 0.11 ips (2.8 mm/sec), which is a significant reduction. Although detailed follow-up testing was not performed, it is believed that a significant improvement in torsional vibration response was achieved by the coupling modification, which reduced the overall gearbox vibration and solved the problem.

Variability in Systems

Several of the previous topics discussed how the predicted torsional response of a system can vary due to modeling techniques, such as the core of a motor, and the interaction with other system components. In addition, acoustical resonances can amplify pressure pulsation and increase dynamic torque excitation in the system. Uncertainties with the mass-elastic data of each component of the train can also have a significant effect and should be considered during an analysis. For example, most coupling manufacturers state that the torsional stiffness of a coupling can vary 10 percent to 35 percent. The different methods for calculating crankshaft stiffness could also affect the torsional analysis, particularly if a system has been tuned between strong excitation orders.

Another source of variability in the system is the manufacturing and fabrication tolerances and procedures. Testing has shown that the torsional natural frequencies of "identical" motor-driven compressor trains can vary by 5 percent or more. This may be due to minor differences in the motor core (welds, pole bolt torques, windings, laminations, etc.), coupling bolt torque, coupling hub penetrations, interference fits, and the forged and machined compressor components (crankshaft, crosshead guides, crankpins, etc.).

Most API specifications recommend a minimum 10 percent separation margin between a calculated torsional natural frequency and any significant excitation order. In many cases, a minimum 15 percent separation margin should be used to account for possible variation between the predicted and actual system. This separation margin can be less when the natural frequencies have been measured.

As mentioned previously, unequal peak firing pressure, valve failures, and other maintenance problems can result in actual conditions that vary significantly from the theoretical conditions. It is imperative to use an adequate safety factor and to consider these off-design conditions in the design and analysis phases.

Case History—Pole Bolt Failures in a Synchronous Motor

- Equipment:
  A single bearing, synchronous motor directly connected with a rigid flange to a two-throw, two-stage reciprocating compressor
  Rated 6,000 hp (4,474 kW) at 327 rpm
- Problem:
  Motor pole bolt fatigue failure
- Cause:
  Coincidence of first torsional natural frequency with 10× running speed. Improper pole bolt torque.
- Solution:
  Modified flywheel to detune torsional resonance. Increased pole bolt torque.

A refinery purchased two new units consisting of a synchronous motor directly driving a two-throw reciprocating compressor at 327 rpm. The motor had a single outboard bearing design and was directly coupled to the compressor flywheel through a bolted flange (no coupling). Although these units had not been officially commissioned, there was concern because pole bolts had failed on two identical motors at a different refinery after only one month of operation. The pole bolt failures caused one of the 500 lb (227 kg) poles to contact the stator, destroying the entire motor. A metallurgical analysis determined that the motor pole bolts had failed due to low-stress, high-cycle fatigue.

Field testing was performed to determine if a TNF of the system was causing high torsional oscillation of the motor. If so, this could result in high forces on the bolts due to varying angular acceleration of the motor poles. Torsional oscillations were measured at the free-end of the motor shaft with a torsiograph. In addition, a strain gage telemetry system was installed on the motor shaft between the flywheel and core to measure shear stresses and torques.

A synchronous motor produces mean and pulsating torques during startup. The frequency of the pulsating torque varies from twice electrical line frequency to zero at synchronous speed. When the pulsating torque frequency coincides with a TNF below 120 Hz, resonance occurs and severe torsional vibrations can be generated. Therefore, the startups were monitored to determine the peak torque in the motor shaft. From the time-domain data, the first TNF was determined to be approximately 48 Hz.

Data were also acquired during a coastdown of the unit with the compressor discharge valves removed to provide a slower deceleration. The waterfall plot in Figure 33 shows the first 14 compressor orders as diagonal lines. The first TNF peaks when various compressor orders pass through resonance. A peak appears near 46 Hz as the tenth order excites the first TNF. However, as the compressor continued to coast down in speed, the first TNF appeared to shift higher toward 49 Hz. For a direct-coupled motor/compressor system, the torsional natural frequency should not vary with speed or load. This variation indicated some nonlinearity sensitive to centrifugal force, such as looseness in the system.

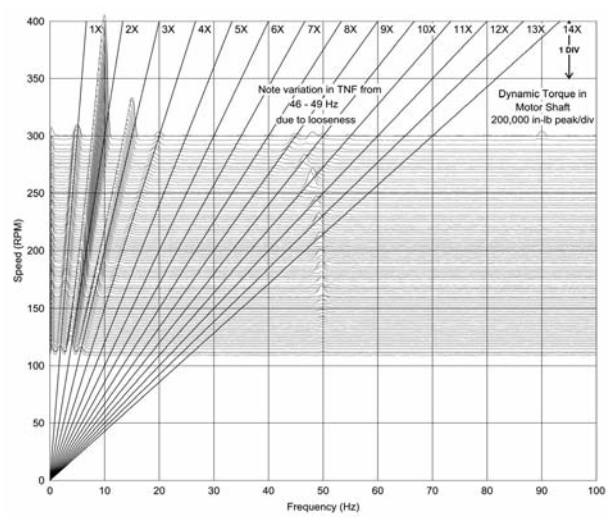


Figure 33. Waterfall Plot of Motor Shaft Torque During Unloaded Coastdown (As-Found Condition).

The flanged connection between the compressor crankshaft and motor shaft at the flywheel was checked and the bolt torques increased. The unit was retested and the variability in the first TNF still existed. Therefore, the motor was sent to the shop for inspection. Several of the pole bolts were found to have insufficient torque. The exact cause of the loose bolts was never determined. It is possible that using a manual torque wrench could have caused inaccuracies of ±25 percent in the torques (Donald, 1981).

Centrifugal force caused the motor poles with insufficient bolt torque to no longer be rigidly attached to the spider. A sensitivity study was performed which showed how this reduced stiffness between the motor poles and spider could lower the first TNF. Figure 34 shows the computed TNFs and mode shapes for different motor assumptions.

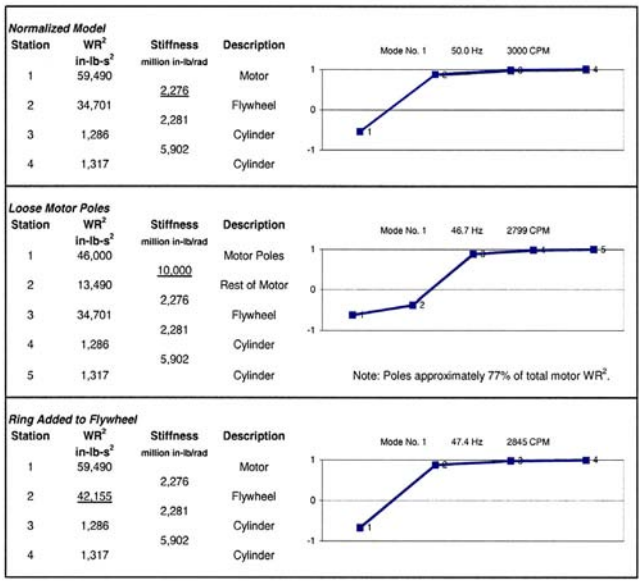


Figure 34. Computed Torsional Natural Frequencies and Mode Shapes for Various Motor Assumptions.

If the bolt preload is insufficient and joint separation occurs during operation, all of the centrifugal force from the motor poles will act on only the bolts. This looseness of the poles will also allow bending stress to occur in the bolts due to the tangential motion at the large outer radius caused by the torsional oscillation of the motor shaft. The tensile stress combined with high-cycle, low-bending stress on a bolt with insufficient preload agrees with the failure mode reported by the metallurgist.

Two-throw compressors normally produce significant dynamic torque at 1× and 2× running speed. However, higher orders can be greatly amplified by torsional resonances in torsionally stiff systems without couplings. The half power point method was used to determine the amplification factor of approximately 80, indicating little damping in this system at this frequency. While this observed AF value is high, similar AFs of 100 to 110 have been documented by others (Grgic, et al., 1992; Tripp, et al., 1993).

Since the first torsional natural frequency was coincident with 10× compressor speed, the system needed to be further improved. By adding an inertia ring to the compressor flywheel, the first TNF was tuned between the ninth and tenth compressor orders at synchronous speed. The system was retested with the motor reassembled and additional flywheel inertia. The measured first TNF agreed with the target frequency of 47.5 Hz and no longer varied with speed (Figure 35). All of the compressor load steps were tested and the modified system was found to be acceptable.

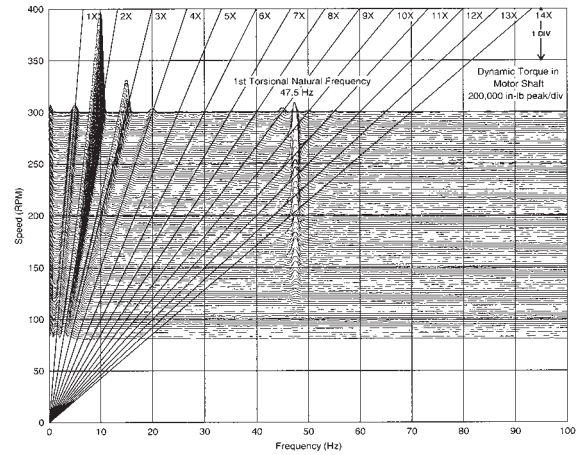


Figure 35. Waterfall Plot of Motor Shaft Torque During Unloaded Coastdown—Modified System.

TORSIONAL MEASUREMENTS

Torsional vibration is often "silent" and, in many cases, it may not be obvious there is a torsional vibration problem. This is because conventional monitoring equipment, such as accelerometers and shaft proximity probes, will not detect torsional vibration (with the exception of systems with a gearbox). Coupling chatter, damper failures, and compressor oil pump failures can be indicators, but many times, torsional problems are not detected until a major failure occurs. Therefore, special equipment is needed to measure torsional vibration.

Situations where torsional testing may be required could include the following:

- *Previous torsional failures*—If an unexpected or premature failure of a component occurs, testing of the repaired system is often the best method to investigate the cause(s) of the failure.
- *Critical applications*—If a system poses unusually high risks to life, other machinery, or plant processes, testing should be performed to ensure reliable operation. This could include a prototype machine or an existing model operating at higher speeds or pressures than previously designed.
- *High chance of variability*—This could include systems with torsionally soft couplings and/or wide speed ranges or operating conditions. If many assumptions had to be made in the analysis phase due to lack of drawings and technical information, testing should be used to confirm the results.
- *Product development*—Newly designed systems that will be mass produced should be tested under load. It is much easier to correct a problem with an initial unit at the factory than it is to retrofit many units that have already been shipped to customers.
- *Used systems*—Compressor systems that have been modified or put into a different service, such as restaging and/or changing operating conditions, should be reanalyzed or tested.
- *New unit for municipality*—Many municipalities have specifications that require torsional vibration testing of new units by a professional engineer.

There are multiple devices currently available to measure torsional vibration. Descriptions of some of the preferred methods are discussed below.

Strain Gage Telemetry System

A strain gage telemetry system can be used to directly measure transmitted and dynamic torque in a shaft or coupling spool piece. The full bridge arrangement with four gages shown in Figure 36 can measure torsional strain while negating or minimizing strains due to bending, axial and temperature effects. The voltage signal

produced by the bridge is proportional to strain and can be converted to torque or shear stress. The measured stress can then be compared to allowable levels.

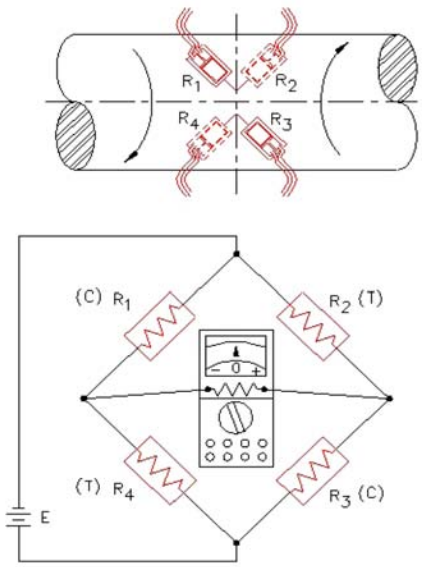


Figure 36. Full Bridge Arrangement with Four Gages.

Since the strain gages are mounted directly on the rotating shaft, it is necessary to use a wireless telemetry system to transfer the measured strain signals to the recording equipment. Various systems are commercially available. Some require batteries while others use induced power via a radio frequency signal that is inductively coupled from the receiving to the transmitting antenna. The resulting signal is digitized and the digital data stream is reconstructed into an analog signal at the receiver. Resolution and frequency ranges may vary. Telemetry systems can have 16-bit resolution and a 0-500 Hz frequency range.

The strain gages should preferably be located where maximum twist occurs, which requires knowledge of the torsional mode shapes of the system. Depending on the particular mode shape, this installation location may not be feasible. Installing the gages on the shaft near the coupling hub (but away from any keyways) is usually adequate, or directly on the coupling spool piece (Figure 37). Installation of the gages and telemetry system normally requires several hours with the unit down.

Figure 37. Strain Gage Telemetry System Installed on Coupling Spool Piece.

Calibration of Strain Gage System

For checking torsional vibration, shunt calibration is normally satisfactory. However, for accurately measuring performance and/or efficiency, a mechanical calibration may be necessary.

By applying a known unbalance in the bridge circuit, the telemetry system can be calibrated with the shunt resistor. With some telemetry systems, the shunt resistor may be built-in. While at rest, the zero and span can be adjusted on the receiver to obtain a scale factor. Note that the shunt calibration does not take into account possible deviation from the ideal strain/torque relationship due to variation in gage factors and exact gage placement. Therefore, a second mechanical calibration could be performed where a known moment was applied directly to the shaft.

Example Calibration of Motor

Mean and dynamic torque values were measured during a shop test of back-to-back motors. These motors have a "drive through" design for starting, helping, generating power with a Frame 9 gas turbine-NGL compressor train. The need for very accurate torque measurements required mechanical calibration of the strain gage telemetry system. To accomplish this, a small beam was bolted to the compressor end of the motor and supported at the end by a stand and hydraulic jack (Figure 38). The purpose of this beam was to prevent rotation and to provide a way to level the longer beam mounted on the other end. Next, the longer beam was bolted to the turbine end of the motor (Figure 39). The drive-through design of the motor allowed the beams to be attached to both ends.

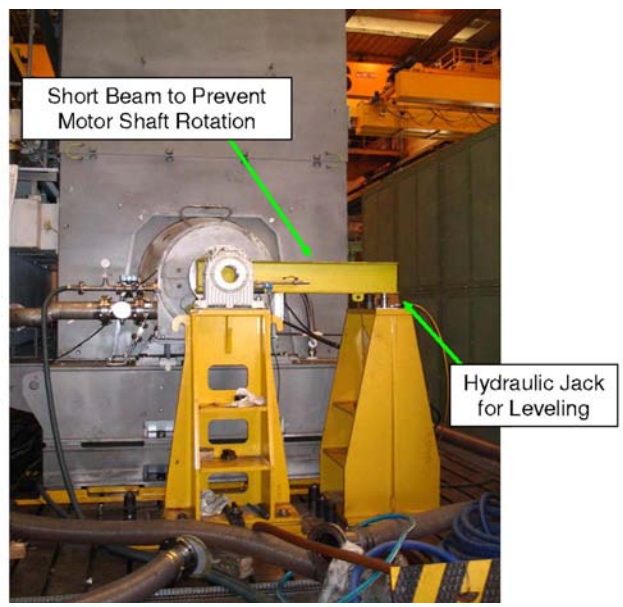


Figure 38. Beam on Compressor End of Motor.

Figure 39. Beam of Turbine End of Motor.

Before calibration was started, jacking oil was activated for both motors. This elevates the motor shafts in the bearings onto an oil film, reducing the effect of friction. The hydraulic jack on the shorter beam was adjusted to level the longer beam on the turbine end of the motor, and to ensure that the shafts were freely rotating within the bearings on the oil film. The telemetry system receiver was zeroed to eliminate the moment effect due to the overhung weight of the beam.

Three weights were available for the test. A calibrated electronic scale was used to verify each weight. The weight of the tackle used to attach the weights to the beam was also included in the total weight. Each weight ranged between 869 lbs and 999 lbs (394 to 453 kg). The weights were loaded onto the beam one at a time.

After a weight was attached to the beam, it was releveled using the hydraulic jack on the other end. This was to ensure that the measured distance from the hanging weight to the center of the motor shaft would be the correct moment arm length for the torque calculation. The output voltage from the receiver was then noted and recorded in a spreadsheet.

With all three weights hanging from the end of the longer beam (Figure 40), a torque of 23,600 ft-lb (32 kN-m) was applied. For reference, the torque with 60,346 hp (45 MW) load at 3000 rpm is 105,600 ft-lb (143.2 kN-m). Therefore, the maximum applied calibration moment was approximately 22 percent of the rated torque.

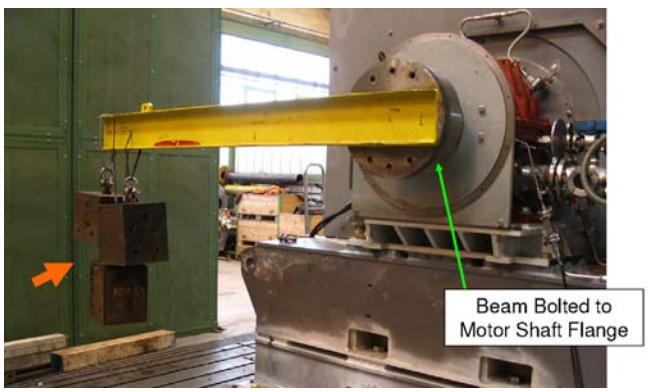


Figure 40. Weights Attached to Beam.

Weights were then removed one at a time in the reverse order. These points were also noted in the spreadsheet, since hysteresis can cause a small variation in the indicated torque values. This part of the calibration was used to check that the strain gage system returned to the zero point with all of the weights removed.

All points were plotted on a graph with output voltage along the X-axis and applied torque along the Y-axis as shown in Figure 41. Linear regression was applied to the data points. The goodness of fit (R^2) was computed, which is a value that can range between 0.0 and 1.0 without units. The R^2 value for this example was 1.0000, which indicates the best fit of the line to the data points. The calculated slope is the scale factor of torque per voltage. When compared with the shunt calibration performed previously, these two scale factors agreed within 0.6 percent of each other.

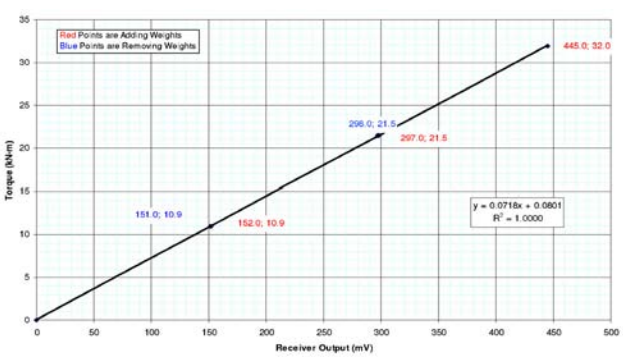


Figure 41. Plot of Strain Gage Calibration.

Rotary Shaft Encoder

A rotary encoder can be used to measure angular position of a shaft. The device is normally attached to the free-end of a rotating shaft. There are mechanical and optical encoders available both of which generate pulses that can be processed by a data acquisition system to calculate speed, torsional oscillation, etc. The pulses from the encoder signal can be processed via a Hilbert transform to measure torsional vibration (Randall, 1990).

Setup and use of the encoder is typically simple and straightforward; however, the shaft must have proper accommodations for the encoder, such as a tapped hole in the shaft end. Normally an encoder would be mounted at the free-end of a shaft using an adaptor.

An alternate way to install an encoder is shown in Figure 42. In this case, a jack shaft from the front of the engine was driving a cooling fan, which prevented installing the encoder directly to the engine crankshaft. Therefore, a special mounting bracket was designed with a spring to apply a constant force to the encoder wheel to prevent slippage relative to the jack shaft. The measurements were adjusted by the diameter ratio of the jack shaft and the rider wheel.

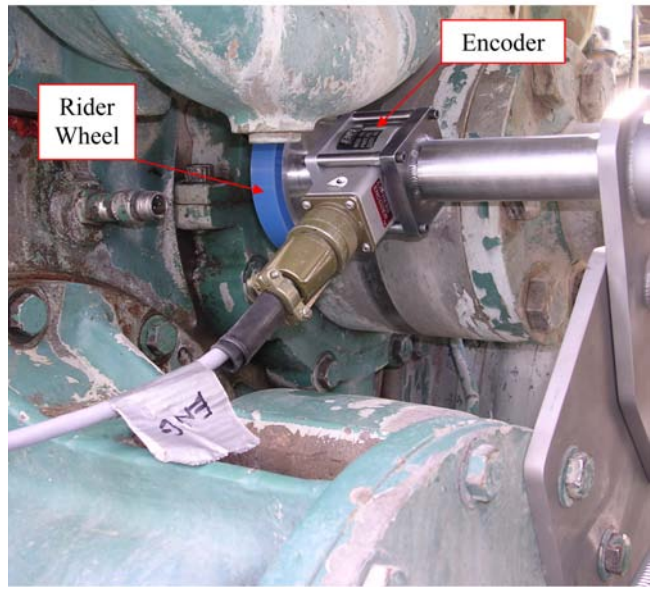


Figure 42. Encoder with Rider Wheel.

Laser Vibrometer

A laser vibrometer can be used to measure angular displacement in degrees. The laser is noncontacting and can be "pointed" at the rotating surface of interest. No downtime is normally required for installation. However, the laser vibrometer may not be capable of accurately evaluating transient conditions, such as start-ups, speed sweeps, or shutdowns. This could limit its ability to measure natural frequencies of a system, if speed changes occur too quickly. Newer laser vibrometers have a frequency range down to 0.5 Hz and a dynamic range of 0.01 to 12 degrees.

Example of Laser Measurement on Engine

A laser was used to measure torsional vibration on the front end of a gas engine (Feese and Smith, 2009). The installation can be seen in Figure 43. Two red dots can be seen from the laser beams on the reflective tape wrapped around the jack shaft. The laser is powered by a separate computer module, which processes the signal, applies filters, and produces a voltage proportional to torsional oscillation in degrees. For comparison, the torsional vibration was also measured with the encoder shown in Figure 42.

Figure 43. Laser Aimed at Reflective Tape on Engine Shaft.

During the test, the engine speed was varied slowly from minimum to maximum operating speed (860 to 1,000 rpm). The torsional oscillation in degrees zero-peak (0-pk) was plotted using a waterfall plot format where frequency spectra were taken every 3 rpm. Diagonal lines superimposed on the plot indicate harmonics or engine orders. A four-stroke engine will produce half orders in addition to the multiples of running speed due to the firing cycles. Peaks in the waterfall plots can indicate torsional natural frequencies of the system.

Figure 44 shows the data obtained using the laser. Above the waterfall plot is a slice plot that tracks individual engine orders. The 3× order was of interest because there was a torsional natural frequency of the system near 58 Hz. This harmonic was approaching the torsional natural frequency, but the engine did not reach a high enough speed to peak through this resonance.

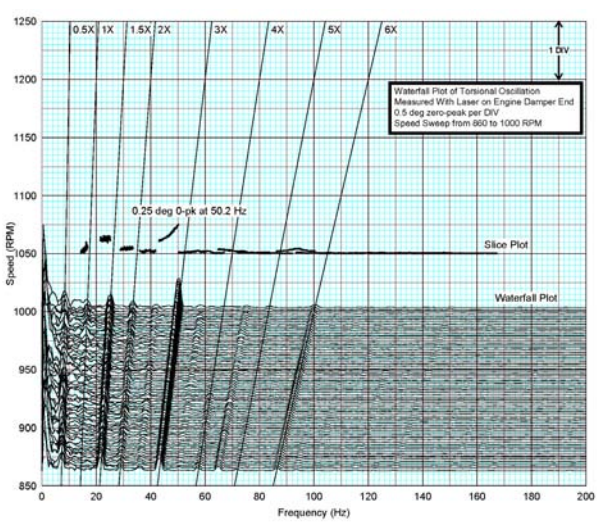


Figure 44. Waterfall Plot from Data Taken with Laser.

For comparison, Figure 45 shows the data from the shaft encoder that was located near the laser. The laser and encoder measurements were taken simultaneously during the test. At the order of concern (3×) the values were very similar, although the measurement techniques were totally different.

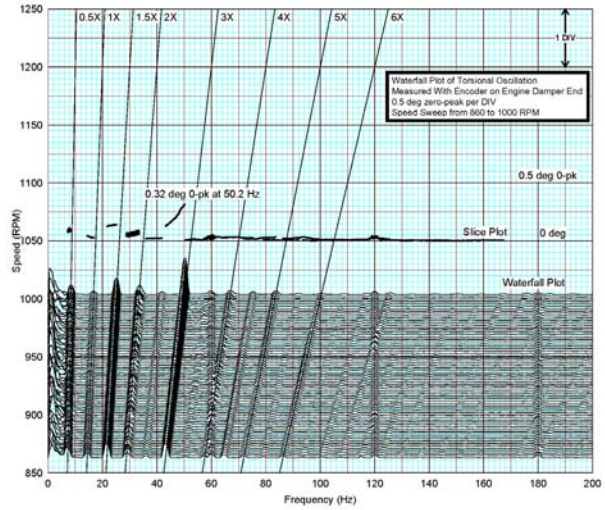


Figure 45. Waterfall Plot from Data Taken with Shaft Encoder.

There appeared to be less low-frequency, nonsynchronous noise (below 5 Hz) from the encoder versus the laser. This may have been due to vibration of the tripod holding the laser. The encoder produced electrical noise at multiples of line frequency (60 Hz, 120 Hz, and 180 Hz), which is not real torsional vibration. This noise may have been due to electrical interference or a ground loop.

It was concluded that the laser provided accurate data during the test and compared favorably with the shaft encoder. The laser was easier to setup and did not require a special adaptor to be fabricated.

Frequency Modulation

The frequency modulation system uses proximity probes or magnetic pickups to measure the pulse rate or gear tooth passing frequency of a train component. Assuming the gear teeth are equally spaced and the lateral vibration is low, variations in time between tooth passing will indicate torsional vibration. The signal can be demodulated and converted to angular velocity or integrated to angular displacement. Two probes, 180 degrees apart, are preferred to cancel the effects of lateral vibration.

Inductive Torque Measurement

One of the newest methods for measuring torque is a contactless inductive sensor that measures the magnetic permeability variation of a shaft material. The concept is based on the anisotropic magnetostrictive effect in ferromagnetic shaft surfaces (Fraunhofer, 2007). The magnetic permeability of the shaft material differs in the tensile versus compressive directions and is proportional to the torsional stress. The sensor is able to measure the permeability variation at the surface over a wide torque range. The sensor does require a narrow air gap (similar to a proximity probe), and typically has a frequency response range of 0 to 200 Hz.

Torsiograph

A torsiograph is an instrument that rotates with the shaft and is used to measure angular velocity (deg/sec) or displacement (degrees). For example, an HBM torsiograph operates on the seismometer principle, with a mass retained by springs whose relative motion compared to the stator is converted into an electrical signal by inductive proximity detectors. The frequency range is approximately 3 to 1,000 Hz. The device must be mounted on a free end of the shaft (Figure 46), preferably near an anti-node or point

of maximum torsional oscillation for best results. While the instrument is easy to install, it is sensitive to lateral vibration and will require that the shaft end be true and drilled and tapped such that the torsiograph is centered on the shaft. Downtime of the compressor system will be required for installation.

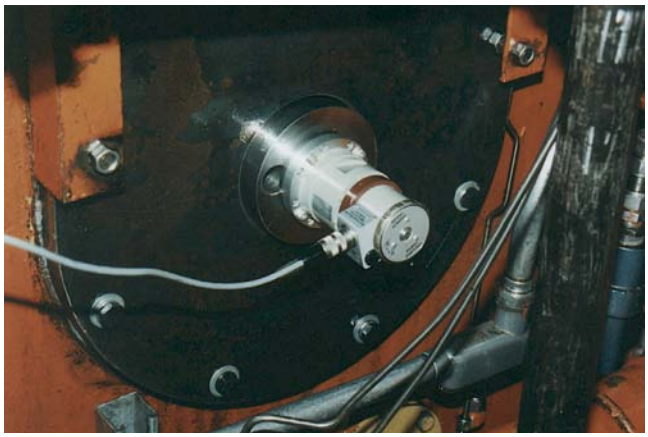


Figure 46. Typical Torsiograph Installation.

The amount of oscillation may not be an indicator of shaft stresses. For example, high oscillation can occur in a system with a soft coupling, but the stresses may be low. If only torsional oscillations are measured, the torsional analysis should be normalized based on the measurements to evaluate the stresses in the system.

HBM torsiographs are no longer manufactured. Some specifications particularly from water districts and municipalities may still call for a "torsiograph test"; however, today shaft encoders or torsional lasers are more commonly used to measure torsional vibration instead of a torsiograph.

Electrical Measurements

It may be necessary to measure voltage and current to help diagnose the torsional vibration problems with systems involving electric motors. For example, VFDs can produce torque ripple, which could excite torsional natural frequencies of the system (Feese and Maxfield, 2008). Electrical measurements can be used to determine electrical power and the amount of torque fluctuation potentially being applied to the motor by the drive.

The stator voltages can be measured with voltage dividers, which have a large frequency response well in excess of 10,000 Hz. Figure 47 shows the three stator voltage probes (one for each phase) inside a motor cabinet. The output signal from the voltage probes requires impedance matching (1 MW) and in some cases amplification.

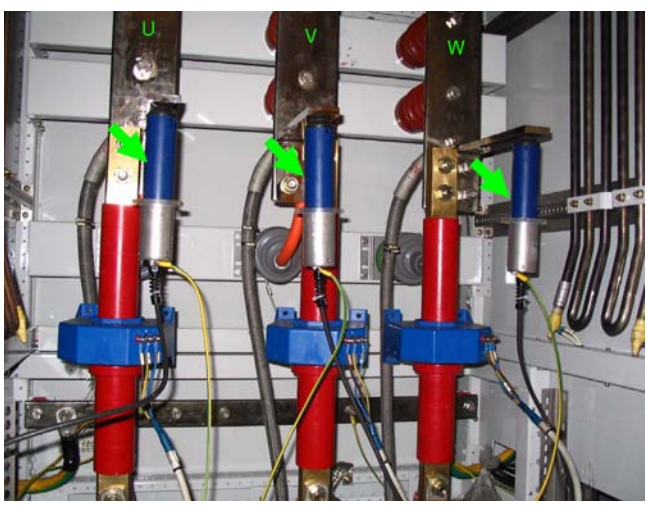


Figure 47. Voltage Probes.

Stator currents can be measured using a flexible head that mounts around the phase cables. The current probes are based on the Rogowski principle. Figure 48 shows the installation of the stator current probes around each of the three phases in the motor junction box.

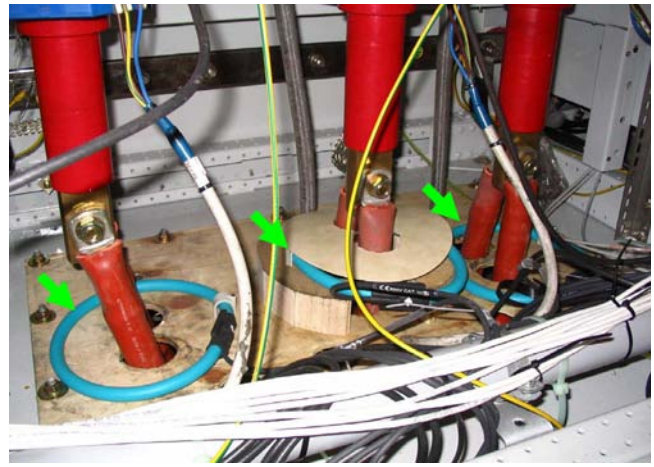


Figure 48. Current Probes.

Reciprocating compressors can cause current pulsation in motors that experience high torsional oscillation. The National Electrical Manufacturers Association (NEMA) specification states that current pulsations should not exceed 66 percent. Figure 49 shows an example where the current pulsations in a synchronous motor were 101 percent when the two-throw compressor was fully loaded.

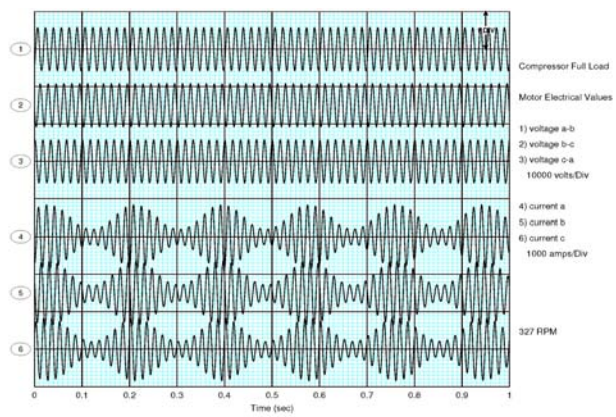


Figure 49. Example of Voltage and Current Measurements.

Adding inertia to the system with a larger flywheel and/or motor rotor will help to smooth out these current pulsations. High torsional vibration can also cause failures of motor pole bolts, wire connections on the rotor, exciters, diode wheels, and cooling fan blades.

Test Conditions

In addition to the instrumentation, an important consideration during torsional testing is the operating conditions of the machinery being tests. For compressor systems, torsional measurements should be obtained at multiple conditions, including start-up, unloaded operation, multiple load steps (e.g., unloaders or pockets), and during loaded and unloaded shutdowns.

During start-up and shutdown, many transient events can happen quickly, so time-domain data obtained with a high sampling rate is required. Coastdown time can be maximized by removing the compressor valves, thus obtaining a higher resolution waterfall plot. For variable speed machines, data should be acquired during a slow, smooth speed run so that waterfall plots and/or order tracks can be obtained. Cylinder pressure-time cards are also useful to correlate back to torsional excitation and can help identify acoustic

resonances and/or valve problems. When possible, all data should be simultaneously obtained and digitally recorded.

As already shown in this paper, waterfall plots of frequency spectra are good for displaying multiple harmonics over a speed range. Order tracking can also be used to separate responses at the various harmonics. Peaks often indicate resonant conditions. Phase angles can be determined from a once-per-revolution tach signal and used to confirm a phase shift through a suspected resonance. For four-stroke engines, a phase signal is needed every other crank revolution (once per firing order). This may be obtained from a high temperature pressure transducer installed in the cylinder or a current probe located around an unshielded ignition wire.

CONCLUSIONS AND RECOMMENDATIONS

Many times torsional vibration problems can be avoided in the design stage with careful specifications. Improved reliability can also be achieved in the field through proper operation and maintenance programs. Recommendations are summarized as follows.

Design and Specification

1. Shaft material should be high-strength steel, with an ultimate tensile strength (UTS) of 95,000 psi or greater.

2. If welds are required on the shaft (e.g., induction motor spiders), a weldable shaft material should be used. Proper welding procedures and material compatibility must be considered when welding to a shaft to minimize the stress concentration factor.

3. Avoid keyways, particularly for motor driven reciprocating compressor systems. Use interference/hydraulic fits instead, particularly at the coupling. If keyways are required, use a fillet radius of at least 2 percent of the shaft diameter to minimize the stress concentration factor per USAS B17.1, "Keys and Keyseats" (1973).

4. In a motor driven reciprocating compressor system, make the minimum diameter of the motor shaft between the core and coupling (including the drive-end bearing journal) at least equal to the compressor crankshaft diameter.

5. When selecting couplings, use an appropriate service factor. For rigid type couplings, a service factor of at least 3 is required. For rubber-in-shear couplings, use a multiplier of 1.5 to 2 times the catalog rated torque value of the coupling. The coupling manufacturer should always review the torsional analysis results and confirm the coupling selection.

6. Fabrication details, such as the motor pole bolt torque, should consider loads due to centrifugal force and torsional vibration.

7. When possible, gearboxes should be designed with the tangential force acting downward on the LS bull gear. In reciprocating systems with high torque variation, an uploaded gear could "bounce" causing dynamic misalignment of the mesh.

8. The speed range of a compressor will have a limit due to torsional resonance, as well as other acoustical/mechanical issues. For additional flow control, consider using evenly distributed pockets or unloaders.

9. A complete torsional analysis should be performed. This may include calculation of: torsional natural frequencies, mode shapes, interference diagram, shaft shear stress, coupling vibratory torque, gear mesh torque, torsional oscillation (particularly at the auxiliary end of a compressor with a lube oil pump or damper end of engine), heat dissipation (for systems with dampers or rubber couplings), and damping effects. The analysis results should be compared to allowable separation margins, material endurance limits, coupling or gearbox torque, motor current pulsation, etc. All load cases should be considered over the full speed range. If required, a time-transient analysis for synchronous motor start-up, electrical faults or loaded shutdowns to predict peak torques and fatigue damage should also be performed.

10. If a problem is predicted with a system as initially designed, possible modifications to consider include: changing the coupling, flywheel (external or internal), or damper in order to achieve acceptable separation margins.

Operation and Maintenance

11. Avoid continuous operation at torsional resonances (if possible).

12. For critical applications, testing may be necessary during commissioning to verify the torsional natural frequencies and stresses.

13. Existing systems with a "dangerous" resonance could be retrofitted with a different coupling or flywheel to detune a natural frequency.

14. Plan regular maintenance for a soft coupling to replace rubber elements. When replacing coupling elements, be sure that they are the correct durometer and material.

15. The silicon fluid in viscous dampers can overheat or degrade with time. Therefore, regular sampling and maintenance is required. It can be easy to overlook dampers when hidden inside an engine case. However, an inoperative damper could cause a crankshaft failure.

16. Excitation orders that should be theoretically low can still cause failures due to bank-to-bank imbalance, misfire, failed valves, or acoustic resonances. Therefore, a regular engine and compressor performance monitoring program could identify rough operating conditions and help prevent failures.

17. To reduce torque excitation, it may be possible to optimize the loading sequence of the compressor cylinders or to select an alternate firing order for some engines. For example, changing the loading sequence could help reduce torsional vibration in a single-stage reciprocating compressor.

18. When possible, avoid full-load shutdowns for reciprocating compressor systems with torque sensitive equipment, such as rubber couplings in shear and quill/torque shafts with limited fatigue life.

NOMENCLATURE

A	= Piston area
AF	= Amplification factor
b	= Width of damper flywheel
BDC	= Bottom dead center
c	= Damping constant (in-lb-s/rad)
CE	= Crank end of cylinder
cpm	= Frequency in cycles per second
c_{opt}	= Optimum damping (in-lb-s/rad)
D	= Outside diameter of shaft (in)
d_c	= Crankpin inside diameter
D_c	= Crankpin outside diameter
d_j	= Journal inside diameter
D_j	= Journal outside diameter
FEA	= Finite element analysis
F_i	= Force due to inertia
G	= Shear modulus (psi)
$\lvert G(i\omega)\rvert$	= Magnification factor
HE	= Head end of cylinder
h_1, h_2	= Internal clearances in damper
I_d	= Damper flywheel inertia (in-lb-s^2)
I_{eqv}	= Equivalent inertia (in-lb-s^2)
I_h	= Damper housing inertia (in-lb-s^2)

ips = Vibration measured in terms of velocity (inches per second)
I_{rot} = Rotational inertia of crankshaft throw
K_t = Torsional stiffness
L = Connecting rod length
L_c = Crankpin length
L_j = Crankshaft journal length
L_w = Web thickness
M_{recip} = Reciprocating mass
N = Rated speed (rpm)
n = Connecting rod to crank ratio
ODC = Outer dead center
P = Rated power (hp)
p = Gas pressure
p-p = Peak-to-peak
P-T = Pressure versus time (or crankshaft angle)
P-V = Pressure versus volume
Q = Quality factor
R = Throw radius
R^2 = Goodness of fit
rpm = Speed in revolutions per minute
r_1 = Inside radius of damper flywheel
r_2 = Outside radius of damper flywheel
SCF = Stress concentration factor
TDC = Top dead center
T_g = Instantaneous torque due to gas forces only
TNF = Torsional natural frequency
UTS = Ultimate tensile strength (psi)
VFD = Variable frequency drive
W = Web width
w = Total reciprocating weight (lb)
0-p = Zero-peak
θ = Crank angle (degree)
m = Fluid viscosity
ω = Frequency in radians per second
ω_d = Damped natural frequency
ω_n = Undamped natural frequency
ζ = Damping ratio

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ACKNOWLEDGEMENTS

The authors would like to thank our clients for the opportunity to assist in analyzing and diagnosing many of these torsional vibration problems. They would also like to thank the EDI staff for their assistance, especially Donald Smith for his review prior to final publication. The authors appreciate Bruce Bayless of Valero and member of the Turbomachinery Advisory Board for sponsoring this tutorial.