GUIDELINES FOR PREVENTING TORSIONAL VIBRATION PROBLEMS IN RECIPROCATING MACHINERY

Presented by

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1 Introduction

There are numerous books [1, 2] and technical papers [3, 4] on the subject of torsional vibration, so the phenomenon should be well understood and easily controlled. However, numerous torsional vibration problems continue to occur in reciprocating and rotating machinery. One reason for this is the mating of equipment traditionally used in non-reciprocating applications (such as variable speed motors) with reciprocating compressors. Other reasons may include lack of monitoring engine or compressor performance, as well as improper application and maintenance of viscous dampers and elastomeric couplings.

The purpose of this short course 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 recommended items that should be considered in the initial design stage, analysis stage, and after the system is in service is provided to help attain maximum reliability. The need for torsional vibration measurements during commissioning to verify acceptability of critical applications is also discussed.

Ten different case histories are presented where failures were linked to torsional vibration. In general, the solutions to these problems were based on practical considerations that could be retrofitted in the field. Of particular interest are the failures that could not have been predicted if only an "ideal" operating condition was analyzed. The results of these investigations emphasize the need for more comprehensive torsional analyses in the design stage of critical systems. Advanced software for performing steady-state and time-transient torsional analyses should have the following capabilities:

- Undamped and Damped Torsional Natural Frequencies and Mode Shapes
- Interference/Campbell Diagram
- Forced Response: Dynamic Torque, Shear Stress, Torsional Oscillation, and Heat Dissipation
- Nonlinear Stiffness Couplings
- Viscous Dampers
- Standard Gearboxes and Epicyclic Gears
- Variable Frequency Drives
- Multiple Compressor Load Steps and Speeds
- Combined Response of Multiple Orders or Excitation Sources
- Determining and Comparing Results to Allowable Levels
- "Non-Ideal" Operating Conditions: Engine Misfire and Compressor Valve Failures
- Transient Events: Synchronous Motor Start-up, Electrical Fault, and Loaded Shutdown of Reciprocating Compressor
- Rainflow, Fatigue Damage, and Allowable Number of Events

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2 Torsional Modeling of Reciprocating Machinery

The first step in analytically determining the torsional response is to calculate the torsional natural frequencies of the system. To do this requires the stiffness and mass inertia of the shaft and components being analyzed (referred to as the mass-elastic data). For cylindrical shafts, 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 mass-elastic data is not provided by the manufacturer.

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 many 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 a 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 off the main connecting rod for the compressor cylinder for a total of three connecting rods per throw.

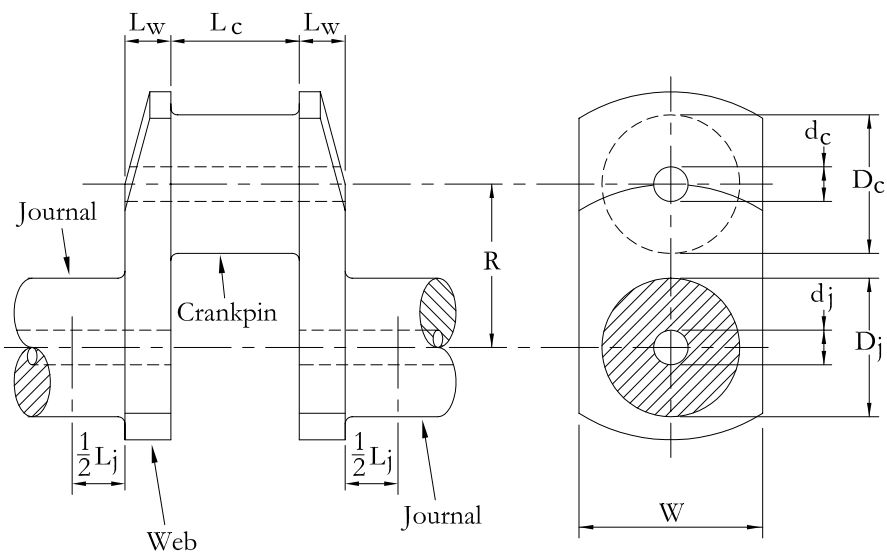

Figure 1: Typical Crankshaft Throw

2.1 Torsional Stiffness

Many equations are given in Ker Wilson [1] and BICERA [2] for calculating the torsional stiffness of a crankshaft. The basic dimensions of the journals, webs, and crankpins are needed, as well as the shear modulus of the shaft material. BICERA also developed curves based on test data for various types of crankshafts. These can be more accurate, but also more complex and are not discussed here.

<u>Carter's Formula</u>

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

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Ker Wilson's Formula

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

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. In the case history presented in Section 3.3, Carter's formula had the best correlation with measured test data. On the other hand, Ker Wilson's formula provided good correlation with field data from the case discussed in Section 8.6.

When conducting a torsional analysis, Ker Wilson has suggested using the average of his 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 1/3 rule should also be applied.

Equations (1) and (2) were developed before finite element analysis (FEA) was readily available. A finite element program such as ANSYS 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 fillet radii and oil holes. These models are from journal centers, which is the same as the distance between throw centers. One end was rigidly fixed and a moment was uniformly applied across the other end. The calculated torsional stiffness is equal to the moment divided by the angle of twist at the free end. It is interesting to note that for both cases, the calculated torsional stiffness using ANSYS fell between the values from the Carter and Ker Wilson formulas.

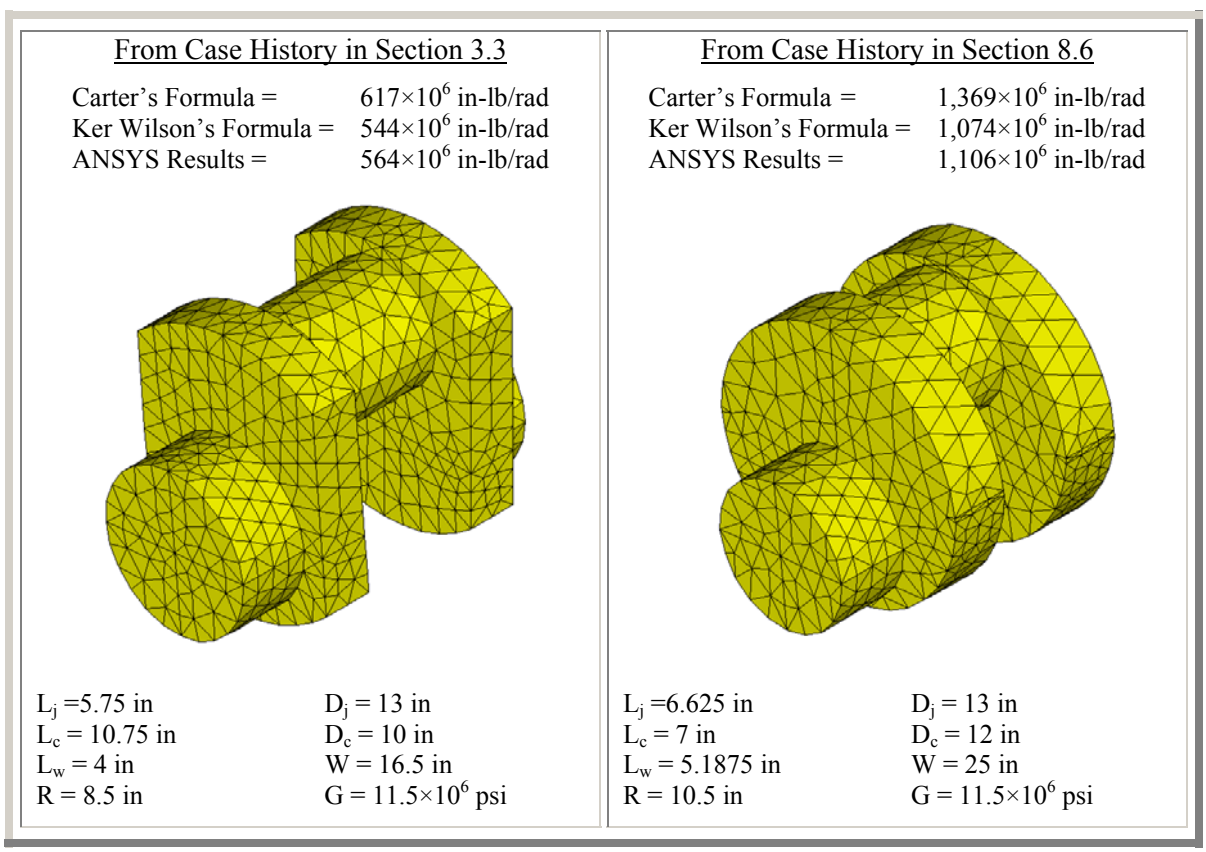

Figure 2: Finite Element Models Used to Calculate Torsional Stiffness of Crankshaft Sections

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2.2 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 [34].

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 counter-weights that may be bolted to a web should also be included. There are numerous 3D CAD packages that could perform this calculation.

The connecting rod is generally heavier at the crankpin 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	
Throw Radius	2.25	in	
Connecting Rod Weight	96	lb	
	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	377	380
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	2,428	2,436

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3 Viscous Dampers

Viscous dampers (Houdaille type) are often used in reciprocating engines to help limit torsional vibration and crankshaft stresses [5, 6, 7]. 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.

3.1 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 3). An untuned damper does not contain an internal torsional spring.

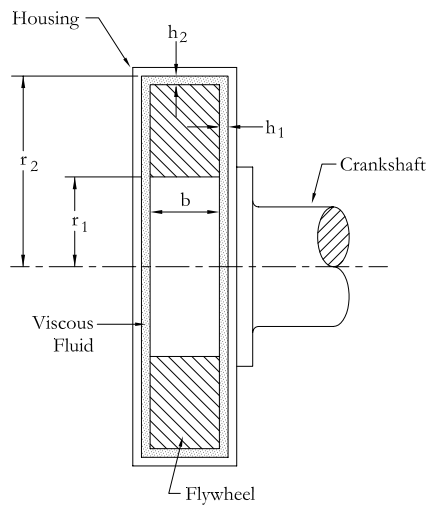

Figure 3: Untuned Damper

The shearing motion of the fluid between the flywheel and housing surfaces dissipates 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 [7], the damping constant is

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

According to Den Hartog [8], the optimum damping for maximum energy dissipation is

$$c_{opt} = I_d\omega \tag{5}$$

The equivalent damper inertia is

$$I_{eqv} = \frac{I_d}{1 + (I_d\omega / c)^2} + I_h \tag{6}$$

For optimum damping, equation 5 is substituted into equation 6 so that the equivalent damper inertia is equal to half of the flywheel inertia plus the housing inertia.

$$I_{eqv} = \tfrac{1}{2} I_d + I_h \tag{7}$$

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Engine manufacturers may provide a lumped inertia value for a damper. As shown in equation 7 for optimum damping, the equivalent damper inertia is equal to half of the flywheel plus the housing. 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 [9, 10] 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. In some situations, it may be beneficial to use two dampers to provide additional damping.

3.2 Service Life

Viscous dampers have a limited service life and require periodic checks and maintenance. However, dampers located inside an engine case can be easily overlooked and forgotten.

Continuous heat absorption reduces the fluid viscosity over time. Simpson Industries (formerly a division of Holset) manufactures dampers and recommends changing them when the fluid viscosity has reduced 50% and the efficiency is approximately 80%. According to Simpson Industries this occurs after approximately 25,000 to 30,000 hours of service [11]. Superior recommends replacement of dampers every 24,000 to 35,000 hours depending on the engine model [12].

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 become solid and turn black. This could cause the damper flywheel to seize to the housing and would no longer provide damping. Burnt or discolored paint could be a sign of overheating.

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3.3 Case History – Broken Engine Crankshaft Linked to Failed Engine Damper

Equipment	16-cylinder natural gas engine (four-stroke) driving a centrifugal compressor through a speed increaser (gear ratio = 10:1). Rated 3,800 HP at 500 RPM Operating Speed Range = 430 – 500 RPM
Problem	Engine crankshaft failure.
Cause	Seized engine damper due to improper maintenance.
Solution	Installed and properly maintained damper.

A crankshaft failure consisting of a crack at a 45° angle occurred, which is typical of a torsional failure. Upon inspection, it was determined that the damper located on the non-drive end of the engine crankshaft was seized. The silicon fluid was solidified and black from heat and age. It was in effect a solid flywheel without any damping properties.

After the engine was overhauled and the damper was replaced, field tests were performed to determine if there was a dangerous resonance within the operating speed range. To measure the torsional vibration response, a torsiograph (see Section 10.1) was mounted on the engine crankshaft at the damper end. Torsional measurements were also taken at the flywheel and bull gear with frequency modulation non-contacting probes (see Section 10.2). Torsional natural frequencies were measured at 15.5 Hz (first mode) and 35 Hz (third mode). These natural frequencies were intersected by various engine excitation orders within the speed range.

A steady-state torsional analysis was subsequently performed on the engine/compressor system to evaluate the crankshaft stresses without a functioning damper. The torsional stiffness of the crankshaft was calculated using Carter's Formula [2], which correlated well with the measured natural frequencies from the field data.

The forced response was calculated by considering the dynamic torque produced by the engine and evaluating the properties of the engine damper. The actual pressure-time curves from each cylinder were measured and combined with the inertial effects of the reciprocating weights. Rough engine operation caused increased torque modulation at certain orders; therefore damaging torsional vibration could occur if the engine was not well tuned.

From the field data and analytical work, it was determined that the crankshaft failure was caused by the 4.5× engine order exciting the third torsional natural frequency. The alternating shear stress in the crankshaft was highest on resonance near 460 RPM and was excessive without a properly functioning engine damper. The third torsional natural frequency was primarily a crankshaft mode, resulting in maximum twisting at the failure location of the crankshaft. With the new, properly functioning damper, the torsional vibration was controlled and the system operated satisfactorily.

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4 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 2 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 2: 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).
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.

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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 [13].
- **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 [14]. 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 trade-off 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 allowable levels. This 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.

4.1 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% from catalog values, which should be considered in the analysis. Figure 4 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 [15].

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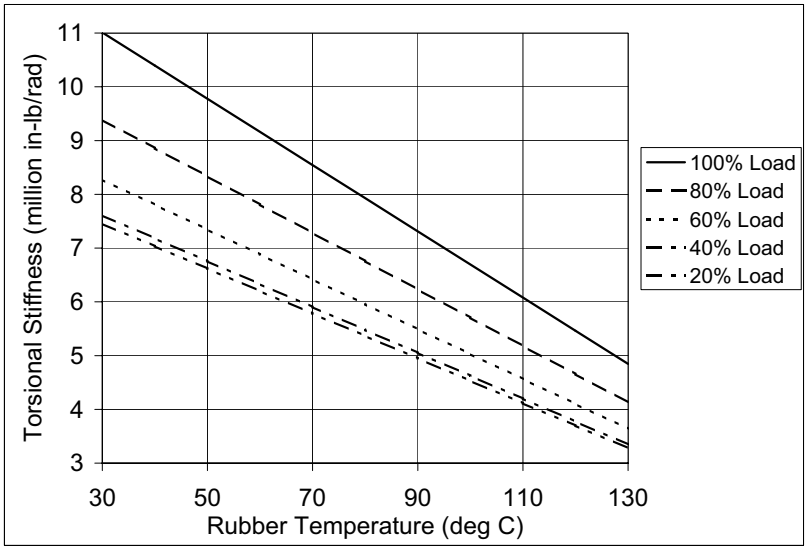

Figure 4: Variable torsional stiffness of coupling with rubber blocks in compression

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.

4.2 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 is typically 20% to 30% 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).

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 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. The flywheel can usually be integrated with the coupling hub or added as internal flywheels ("donuts") inside some compressor frames. Rotating counterweights can also add inertia similar to a flywheel.

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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.

4.3 Case History – Coupling Failure in Variable Speed Reciprocating Compressor System

Equipment	6,000 HP variable speed induction motor driving a single-stage, six-throw reciprocating compressor through a compressed rubber block coupling. Speed range = 300-1,200 RPM
Problem	Pre-mature failures of rubber blocks in coupling.
Cause	High dynamic torque in coupling.
Solution	Replaced soft coupling with flexible disc coupling, but had to significantly restrict the speed range.

The system was originally designed and installed with a coupling using rubber blocks in compression, which has a variable torsional stiffness depending on the transmitted torque. The operating speed range of the unit was an ambitious 300-1,200 RPM, although several speeds were blocked out (using VFD dead-bands) to avoid continuous operation at a torsional resonance.

After installation of the unit, several coupling failures occurred. These failures involved overheating of the rubber blocks, indicating excessive dynamic torque and corresponding heat load. The specific speed(s) at which damage occurred was not known; however, plant personnel observed excessive heat and smoke from the coupling as the unit approached the maximum speed of 1,200 RPM.

A field test was performed to measure the actual torsional natural frequencies of the system. The torsional oscillations at the free-end of the motor shaft were measured with a torsiograph. The alternating torque/shear stress in the compressor shaft near the coupling hub was measured using a strain gage telemetry system. The tests showed that the first torsional natural frequency was 68-70 Hz, while the second was near 133 Hz (Figure 5). Since the coupling has a progressive torsional stiffness, the torsional natural frequencies varied somewhat depending on load.

The third compressor order was significant, particularly near the maximum speed as the 3× compressor harmonic approached the first torsional natural frequency. The strong 3× is inherent in this six-throw compressor since the three throw pairs have 120 degree phasing. The high third order response contributed to the over-heating problem. In addition, numerous loaded shutdowns also produced high dynamic torque in the coupling. As a temporary solution, the maximum speed was limited to 1,000 RPM due to the high 3× harmonic. Also, operation between 610-770 RPM was prohibited to avoid the 6× harmonic exciting the first TNF.

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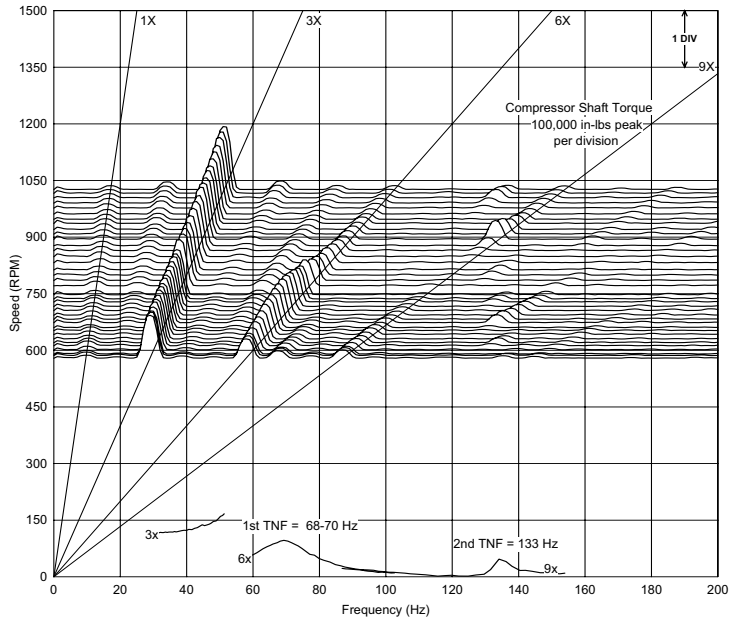

Figure 5: Waterfall plot with original rubber-in-compression coupling

A standard flexible disc coupling was subsequently installed in the unit and the torsional measurements were repeated. With the stiffer steel coupling, the first torsional natural frequency increased to 94 Hz and the second to 150 Hz (Figure 6).

The 3rd, 6th and 9th harmonics were the major orders for the six-throw compressor, but the 5th was also significant. Of concern were the resonances where the 5×, 6× and 9× harmonics intersected the first torsional natural frequency near 1,135 RPM, 945 RPM and 630 RPM, respectively. The amplitudes on resonance were higher since the system no longer had a damped coupling. Because the stresses near resonance were well above the allowable levels, dead-bands were programmed into the VFD from 610-650 RPM and 900-1,000 RPM. Under full load operation with all pockets closed, the unit tripped on high horsepower at 1,120 RPM. Therefore, the maximum rated speed could not be tested under full load. The maximum speed was restricted to 1,100 RPM.

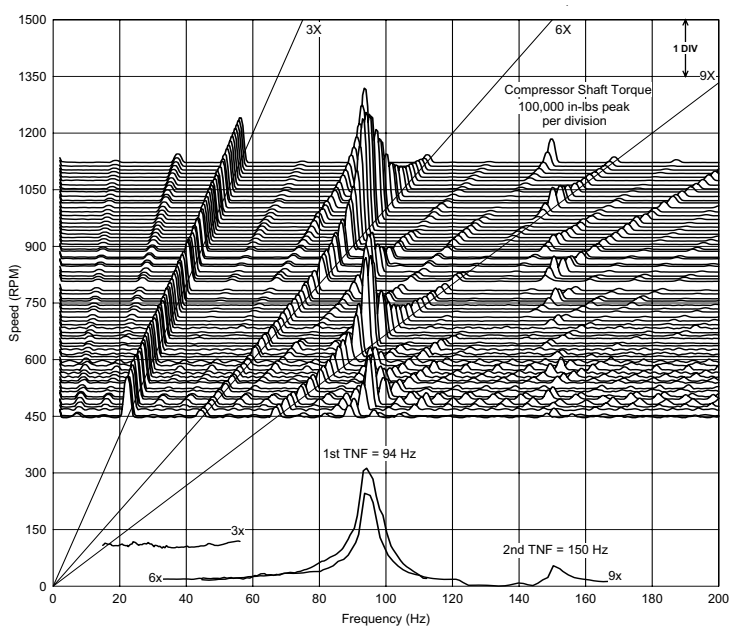

Figure 6: Waterfall plot with new flexible disc coupling

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In the analysis stage, larger separation margins would have been needed to allow for possible modeling inaccuracies. This would mean eliminating even more of the speed range, possibly offsetting the benefit and cost of installing a variable frequency drive.

This example demonstrates how variable speed reciprocating compressors warrant extra caution in the design phase. Multiple harmonics from the VFD and compressor almost guarantee torsional resonances within a wide speed range. Using a rubber coupling was good from the standpoint of trying to add damping to the system. However, the dynamic torque was still too high and the limited coupling life became a maintenance problem.

The solution of using a flexible disc (rigid) type coupling reduced maintenance and downtime, but a large portion of the speed range was forfeited. The torsional measurements had to be used to determine the exact speed-bands allowable for operation.

4.4 Case History – Oil Pump Failure

Equipment	12-cylinder gas engine driving a three stage, six-throw compressor through a flexible disc coupling. Rated 2,600 HP at 1,000 RPM
Problem	Compressor oil pump failures
Cause	Excessive torsional oscillation at oil pump end due to excitation of second torsional natural frequency.
Solution	Installed a rubber-in-shear coupling

A case history given in the EDI seminar manual [18] and also presented at the 2000 GMRC [19] shows how compressor oil pump failures were eliminated by detuning the system and adding damping using a rubber coupling. Torsional vibration measurements showed that the second torsional natural frequency of the original system was near 103 Hz and was excited by the 6× compressor harmonic near maximum speed (1,000 RPM). The overall torsional oscillations at the oil pump end of the compressor reached 1.25 degrees p-p at maximum speed, with a predominant response of 0.8 degrees p-p at 6× (100 Hz).

In technical bulletin #66, Ariel Corporation published torsional oscillation limits for the auxiliary end of compressor crankshafts based on their experience with JGC and JGD frames. These results were also presented at the 2001 GMRC [20]. Based on Ariel's criteria, the maximum allowable torsional oscillation at 100 Hz would be 0.36 degrees p-p. It is important to note that the compressor in this case history was not an Ariel; however, the measured torsional oscillations did exceed their allowable. The resulting oil pump failures would seem to further validate Ariel's criteria.

A rubber-in-shear coupling was subsequently installed in place of the original flexible disc coupling. This detuned the torsional resonance and added damping to the system. The oscillation at the oil pump end of the compressor was reduced to 0.1 degrees p-p at 6× running speed and the oil pump failures ceased.

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5 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 constructions:

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

5.1 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% to 40% 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 [21] and Frei [22]. 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 (FEA) 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 converted to an equivalent cylindrical shaft diameter over the motor core length.

Figure 7 shows a finite element model of an induction motor with six welded spider arms (webs) that was created with ANSYS. Table 3 summarizes some of the motor core dimensions and lists the resulting equivalent diameters based on the finite element analysis, and the API [21] and Frei [22] methods. The equivalent diameter determined from the finite element analysis was 7% larger than the base shaft diameter, which corresponded to a 33% 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.

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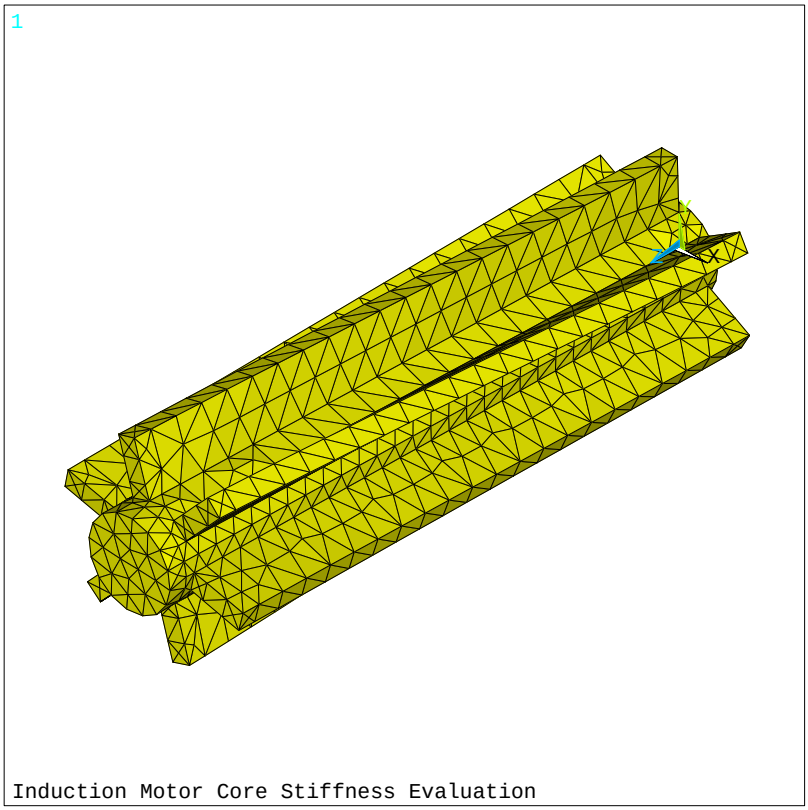

Figure 7: Finite element model of induction motor core

Table 3: Equivalent Shaft Diameter for Six Arm Motor Core

Core Length (in.)	64
Length of Arm Above Shaft (in.)	6
Arm Thickness (in.)	2
Base Shaft Diameter (in.)	9.5
Equivalent Diameter (in.) Using FEA	10.2
Equivalent Diameter (in.) Using API [21]	10.1
Equivalent Diameter (in.) Using Frei [22]	9.6 – 10.4

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5.2 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 at 1,200 RPM Speed Range = 720-1,200 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 720-1,200 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 8). The first TNF was within 1.5% of the calculated frequency, and the second TNF was within 3.5%, 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 un-intensified crankshaft stresses of 10,000 psi 0-peak. Since the coupling hub was keyed to the crankshaft, the intensified stresses at the base of the keyway were over 30,000 psi 0-peak. This stress level was well above the endurance limit of the shaft material and was determined to be the cause of the failure.

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 7.

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 motor model.

To detune the system, the original compressor flywheel was replaced with a smaller internal flywheel. This reduced the inertia of the compressor and increased the second TNF to 130 Hz (Figure 9). This 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.

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.

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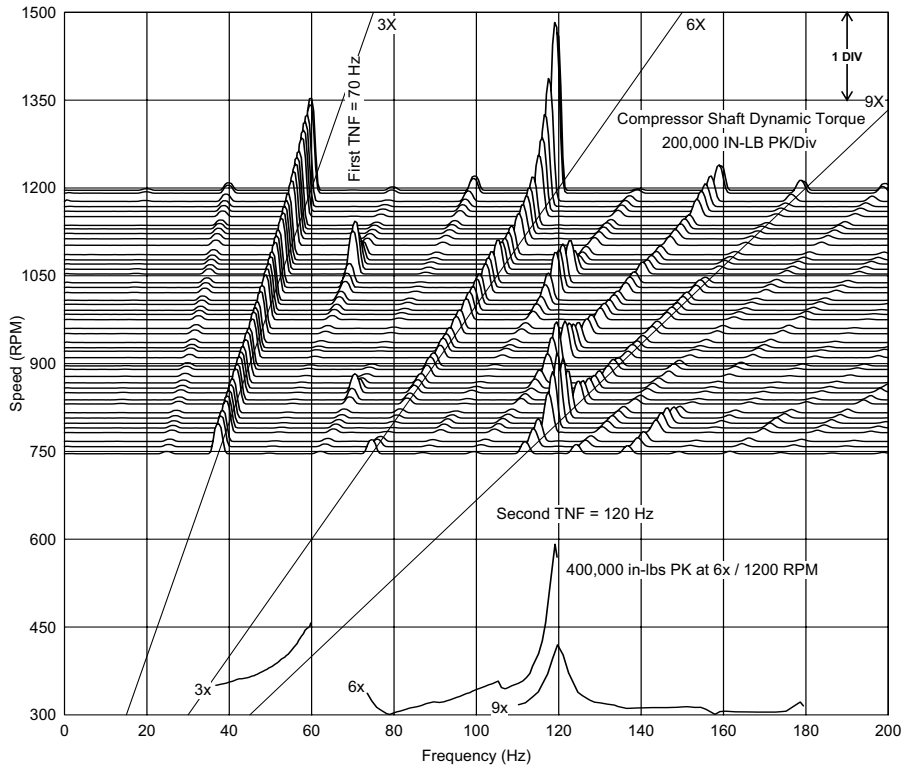

Figure 8: Waterfall plot of compressor torque – original system

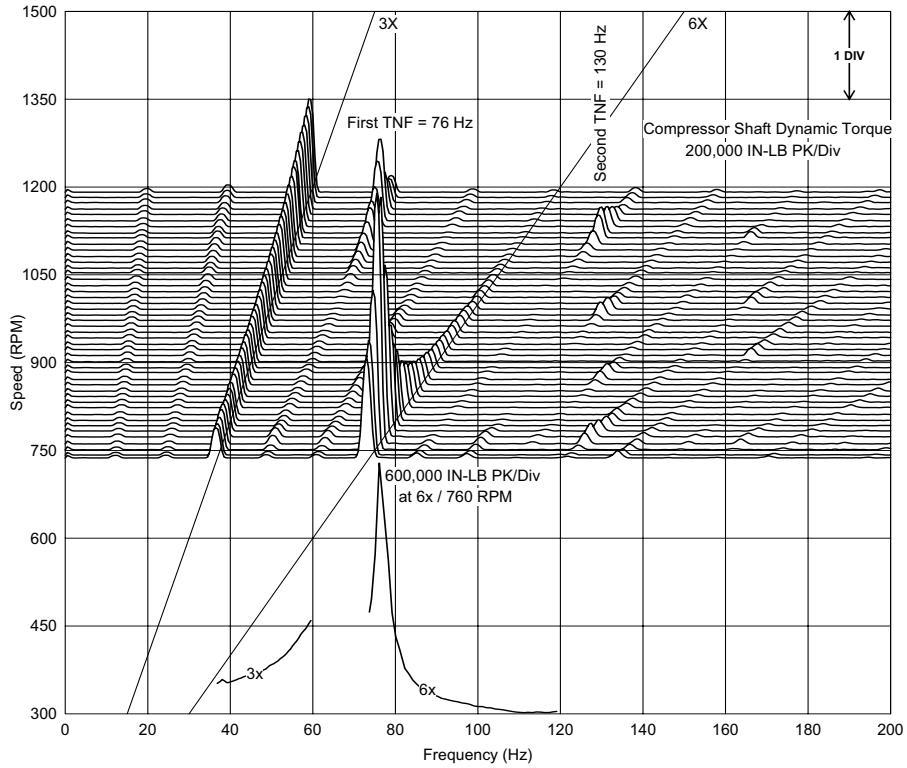

Figure 9: Waterfall plot of compressor torque – modified system

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6 Shaft Properties

The shaft size, material strength and stress concentration factors all have significant effects on the long-term reliability of a system. For example, shear stress in a shaft is inversely proportional to the diameter cubed, so a 10% increase in diameter results in a 25% reduction in stress. However, for obvious practical and economical reasons, shaft size is minimized. Similarly, 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. When the dynamic stress in a shaft section approaches the endurance limit of the shaft material, a fatigue failure can occur. Therefore, it is important to use good engineering practices when designing shafts and selecting materials.

6.1 Shaft Sizing

In many cases, shafting is designed based only on the mean transmitted torque, which is a function of the rated horsepower and speed of the unit [23]. A guideline for sizing a shaft in reciprocating service is to limit the mean stress level (excluding stress concentration) to 3,000 psi or less. Based on this value of mean stress, the minimum shaft diameter is determined by

$$D \geq \sqrt[3]{\frac{107P}{N}} \tag{8}$$

where D is the shaft diameter (inches), P is the rated power (HP), and N is the rated speed (RPM).

Experience has shown that many motors and centrifugal pump shafts are not adequately sized for use with reciprocating equipment, which can produce substantially higher torque modulation than rotating equipment. Also, crankshafts are often constructed with higher strength material and have lower stress concentration factors, since there are no welds and sometimes no keyways. A good "rule-of-thumb" for motors driving reciprocating compressors (or engine-driven generators) 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

While the shaft may be conservatively designed from a mean stress standpoint, dynamic torque must also be considered. In applications involving reciprocating equipment, the dynamic torque can be significant, particularly if torsional resonance or operation near resonance occurs. Many reciprocating compressor trains can have significant dynamic torque even in the absence of resonance. For example, a system with a two-throw horizontally opposed compressor can have a large response at 2× running speed. In either case, the reliability of the shaft now becomes a fatigue problem, not a static stress problem.

6.2 Endurance Limits

To evaluate the reliability of the unit from a torsional standpoint, the torsional shaft stresses must be compared to the endurance limit in shear for each shaft material. Shear endurance limits are not readily available for many steels. A method for determining the endurance limit is provided by Shigley [24] and ASME [25]. These are similar methods that convert the ultimate tensile strength (UTS) to an endurance limit and include fatigue factors such as surface, size, and reliability. Mean stress is subsequently considered using a Goodman diagram. The Goodman diagram de-rates the endurance limit with increased mean stress (higher mean stress results in lower endurance limit).

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% (0-pk) of the UTS. Therefore, if the combined intensified shaft shear stress (total stress amplitude including any stress concentration factor) exceeds

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this value, a fatigue failure could occur. In this case, a fatigue analysis would be required to determine the number of cycles before failure.

Figure 10 is a plot of the endurance limit (with a safety factor of 1) versus ultimate tensile strength assuming typical fatigue modifying factors and various mean stress levels. Higher grade steel improves the capability of withstanding dynamic stress. Therefore, the choice of steel should be based on the stresses due to the dynamic as well as the transmitted torque.

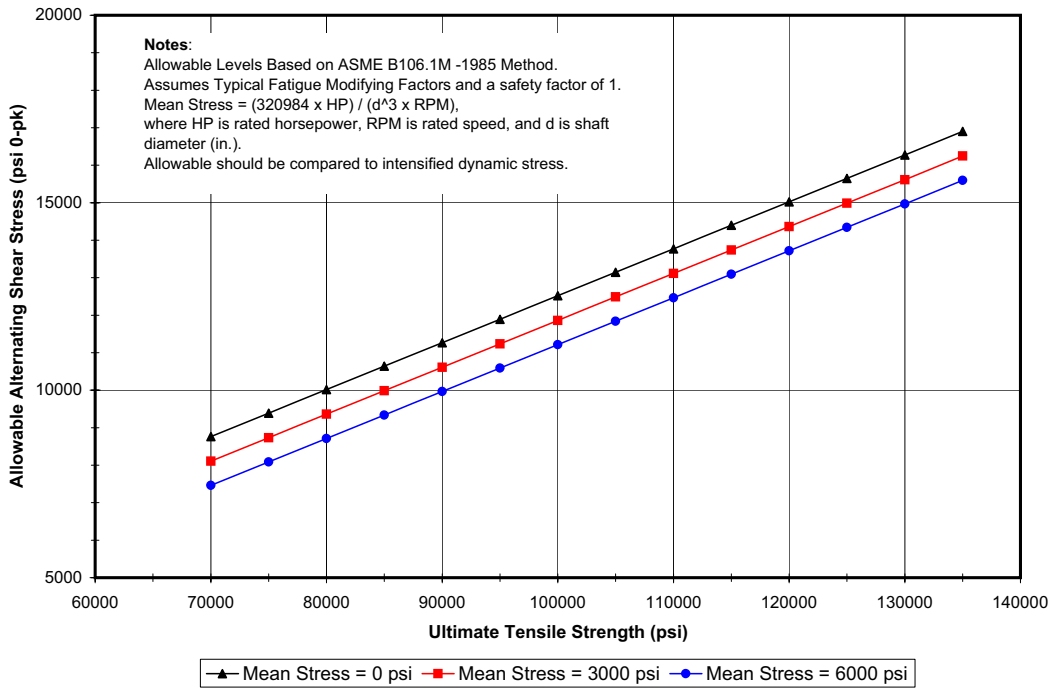

Figure 10: Allowable shear stress versus ultimate tensile strength

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% of the UTS. This safety factor is intended to compensate for possible variations in material properties, as well as mass-elastic data and toque excitation data. Several of these variables are discussed later.

The endurance limit for a specific shaft material is often provided by the equipment manufacturer. If the allowable limit appears to differ dramatically from other criteria, it may be because some limits are to be compared to un-intensified shear stress or applied to only a single harmonic of vibration. Any supplied criteria should be clarified before evaluating the calculated stress levels. The Shigley and ASME methods are to be compared to the intensified, overall shear stress.

For example, one engine manufacturer gives a design limit of 3,045 psi 0-peak based on a crankshaft with an UTS of 120,000 psi. This allowable is to be compared to un-intensified stress levels due to a single harmonic. Using the ASME method, the design allowable would be approximately 7,200 psi 0-peak, but is to be compared to an overall, intensified stress. Considering a typical SCF of 2 for the crankshaft, the ASME limit could be de-rated to 3,600 psi 0-peak for an overall un-intensified stress.

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6.3 Stress Concentration Factors

When designing shafts, stress concentration factors (SCF) are used to account for stress risers caused by geometric discontinuities such as shaft steps, keyways, welds, shrink fits, etc. SCF's should always be included in a torsional analysis. A common reference for determining SCF's is Peterson's Stress Concentration Factors [26]. Note that the SCF depends on the type of stress: shear or bending. Peterson provides SCF's for both cases; so care should be taken to use the proper value.

For stepped shafts and keyways, the fillet radius and shaft diameter must be known to determine the SCF. The worst-case SCF occurs with a square cut 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 (heat shrink or hydraulic) should be considered. However, the shrink must be sufficient to prevent slippage and galling of the shaft. If a keyway is required, use a fillet radius of at least 2% of the shaft diameter as described in USAS B17.1 – Keys and Keyseats [27]. This will limit the SCF at the base of the keyway to 3.

Specific SCF details for reciprocating equipment can be found in Ker Wilson [1], BICERA [2], and Lloyd's Registry [28]. The typical SCF for a crankshaft ranges from 1.5 to 2 in torsion (excluding any keyways). Therefore, using a stress concentration factor of 2 is reasonable for crankshafts when the fillet radii are unavailable and no keyways are present.

6.4 Case History – Torsional Failure, But Not a Torsional Resonance

Equipment	Variable speed motor driving a two-throw, single-stage, natural gas reciprocating compressor through a flexible disc coupling. Rated 1,000 HP at 1,200 RPM Speed Range = 600-1,200 RPM
Problem	Torsional failure of the compressor crankshaft that initiated at the keyway for the coupling.
Cause	High shear stress at upper end of speed range due to strong second order compressor excitation and stress concentration. However, system was not operating on or near resonance.
Solution	Increased compressor drive stub diameter from 5.5″ to 6.75″.

The crankshaft experienced a torsional fatigue failure that initiated in the vicinity of the coupling keyway. Fretting had also been observed between the flywheel and crankshaft near the keyway, which may have resulted in an increased stress concentration factor. The SCF in this area was estimated to be at least 3.2.

A strain gage telemetry system was installed on the crankshaft to measure the torsional response of the system. A torsiograph was also installed at the free end of the motor. The first TNF was identified at 98 Hz, and the second TNF near 234 Hz. Neither resonance was significantly excited within the operating speed range of 600-1,200 RPM. However, significant dynamic torque and alternating shear stress were measured in the 5.5″ diameter compressor crankshaft, particularly at 2× running speed (Figure 11). Since this was a two-throw, horizontally opposed compressor, a significant 2× excitation was inherent due to the acceleration/deceleration of each piston. The combined intensified stress amplitude was in excess of 12,300 psi 0-peak (based on an SCF of 3.2). The endurance limit for this shaft was estimated to be 15,000 psi 0-peak using the ASME method. Therefore, a fatigue failure could occur depending on operating conditions and the actual keyway SCF.

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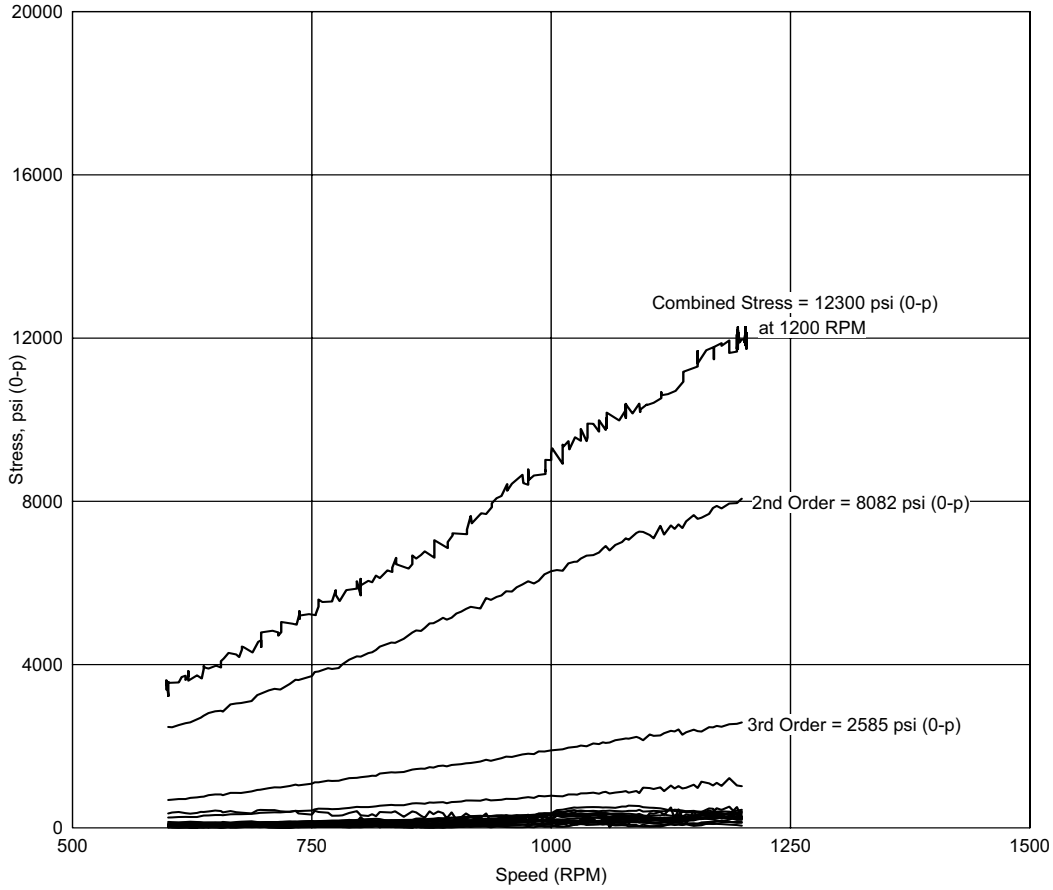

Figure 11: Compressor shaft stress based on an SCF of 3.2

A torsional analysis was subsequently performed to evaluate modifications to reduce the shaft stresses. Since the stresses were not significantly influenced by a torsional resonance within the speed range, reducing the stress by tuning the system was not feasible. Instead, the stresses had to be lowered by adding damping (e.g., rubber coupling), minimizing the SCF, and/or increasing the shaft diameter. The SCF could easily be reduced from approximately 3.2 to a maximum of 2 (a 38% reduction) by eliminating the keyway and preventing fretting. Increasing the shaft diameter from 5.5″ to 6.75″ diameter would decrease the stress approximately 46%.

Installing a soft coupling was not desirable to the user company since the coupling could require maintenance. The compressor drive shaft diameter was increased to 6.75″, but a keyway was still used. Based on the torsional analysis, the maximum combined intensified stress was reduced to approximately 8,800 psi 0-peak, which corresponded to a safety factor of 1.7. Although the safety factor was less than the normally recommended value of 2, no additional crankshaft failures have been reported since the shaft was modified.

This case history illustrates how high alternating shear stress can occur even though there are adequate separation margins from resonance. The original shaft diameter was sufficient for transmitting the mean torque, but was undersized when considering the high level of dynamic torque. The problem was made worse by the stress concentration factor associated with the keyway and shaft fretting.

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7 Special Considerations for 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" will occur producing an audible noise. Torsional vibration can also couple into lateral vibration due to variation in torque at the gear mesh [29]. 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.

7.1 Dynamic Torque at Gear Mesh

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% to 40% 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 (ESD's) 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.

7.2 Case History – Gearbox and Coupling Failures Caused by High Dynamic Torque

Equipment	Steam turbine driving a three-throw hydrogen reciprocating compressor through a two-stage speed reducer (gear ratio 14.5:1). Rubber-in-compression coupling between gearbox and compressor (LS coupling). Compressor Rated 3,200 HP at 327 RPM Compressor Speed Range = 245-327 RPM
Problem	Gear failures; high-speed and low-speed coupling failures.
Cause	High dynamic torque from compressor caused misalignment of gear teeth.
Solution	Detuned first torsional natural frequency by changing the low-speed coupling to a helical-spring coupling and increasing the flywheel size. Replaced gear journal bearings with rolling element type bearings with tight clearances. Replaced original gears with new carburized and ground gears with higher service rating.

The gearbox experienced high vibration since installation at the chemical plant. In addition to vibration alarms and shutdowns on the gearbox, noticeable contact pattern was found on the no-load side of the teeth, indicating a possible back loading condition. Eventually, failures occurred on the high-speed coupling, low-speed coupling (rubber blocks), and the gear teeth.

Field tests were performed to measure the torsional vibration responses. A torsiograph was mounted directly to the blind-end of the low speed bull gear to determine the amount of torsional oscillation. Gearbox shaft vibrations were simultaneously recorded from the permanently installed proximity probes as the compressor speed was varied over the speed range. When loaded, there was a strong response near 8 Hz (2× excitation at 245 RPM) and was evident in the torsional and lateral vibration readings due to coupling through the gearbox.

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The torsional response at 8 Hz was not evident when the unit was tested unloaded. This may be due to the variable stiffness coupling which had rubber blocks in compression. The torsional stiffness of the coupling varied with transmitted torque. Under the no load condition, the coupling stiffness was softer which lowered the first torsional natural frequency.

Operating near the torsional critical speed could result in gear backlash and is not recommended. As a short-term fix, the minimum compressor speed was raised to 300 RPM to provide an adequate separation margin. To develop a long-term solution, lateral and torsional analyses were performed.

The speed reducer was configured as a double reduction parallel shaft gearbox with double helical meshes. The intermediate pinion was down loaded while the LS bull gear was up loaded. The tangential and separating forces acting on the gears were calculated for various loads, as shown in Table 4. The separating, tangential and weight forces were vectorally summed to compute the resultant. The convention for the resultant angle is 0° for up and 180° for down.

Table 4: Forces Acting on the Inboard End of the LS Gear

Load Case	Separating Force (lb)	Tangential Force (lb)	Resultant Force (lb)	Resultant Angle
0%	0	0	7757	180°
10%	528	1270	6509	175°
25%	1320	3174	4769	164°
50%	2641	6348	2993	118°
75%	3961	9523	4337	66°
100%	5282	12697	7232	47°

The analysis showed that the mean torque at 75% to 100% load lifted the inboard end of the LS gear shaft up in the bearing. However, the large torque modulation from the compressor (more than 50%) reduced the tangential force acting in the upward direction to less than the shaft weight. The LS gear shaft was lifted then dropped once per revolution causing high vertical vibration. This lateral vibration can cause significant dynamic misalignment of the gear teeth, resulting in overloading and rapid gear wear. To reduce the dynamic misalignment, the gear mesh direction could have been changed. However, this was not practical for an existing unit, so the shaft journal bearings were replaced with rolling element type bearings with tighter clearances. The failed gears were replaced with new carburized and ground gears having a higher service rating.

Modifications to the LS coupling and flywheel were considered that would detune the first torsional natural frequency and help isolate the compressor dynamic torque from the gearbox. Using a stiffer coupling was not sufficient because the dynamic torques generated by the compressor were still being transmitted through the gearbox, resulting in high stresses and coupling torques. The torsional analysis also showed that the rubber blocks would generate heat and deteriorate rapidly due to the high transmitted and vibratory torque. Another problem encountered with the resilient coupling was that the stiffness varied with torque. Since the dynamic torque through the coupling was significant (in excess of 50%) and changed through the crankshaft rotation, the coupling stiffness was nonlinear and would vary significantly during a revolution. This shifted the torsional resonance, especially the first mode, making it difficult to tune.

In order to avoid these uncertainties, a steel-spring coupling was selected. The advantage was that it provided a low and constant torsional stiffness. However, the disadvantage was that it did not provide additional damping to the system. In addition to changing the LS coupling, the compressor flywheel size was increased to further lower the first torsional natural frequency. The increased inertia also helped to reduce the dynamic torque being transmitted into the gearbox. After installation of the new coupling, gears and flywheel, the failures subsided.

8 Torque-Effort Considerations

Reciprocating compressors and engines produce unsteady torque. This torque variation can be much higher than in rotating equipment and flywheels are often used to smooth the torque. The amplitudes and frequencies of the torque excitation should be considered to avoid coincidence with torsional natural frequencies, which could potentially cause problems.

8.1 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 throw centerline is equal to the moment imposed on the crankshaft. At BDC and TDC, the throw is inline 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).

The inertial forces are caused by the reciprocating mass of the connecting rod, cross-head and piston. The rotating inertia of the crankshaft must be considered in the mass-elastic model, but does not cause any torque variation. Mehta, Farr, and DeWitt [35] give equations of motion for a slider-crank mechanism in terms of displacement, velocity, and acceleration. By multiplying the mass and acceleration of the reciprocating parts, the inertial forces at the crankpin can be determined. At bottom and top dead center, the displacement is maximum and the velocity is zero. The inertia forces will vary with the speed squared.

The gas force is equal to the differential pressure across the piston times the cross-sectional area of the bore. The stroke, or travel of the piston, is equal to twice the crank throw radius. 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. The torque can then be determined versus crank angle. Any distortion in the pressure waveform will affect the dynamic torque and torsional response [36]. 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.

Once the inertia and gas forces have been determined, they must be correctly added together for each cylinder. The throws must then be properly phased for the entire machine as shown in Figure 12. A Fourier analysis can then be performed on these curves to represent the complex wave as a series of sinusoidal curves at various harmonics. At each harmonic, the amplitude and phase can be calculated, or the values can be presented as coefficients of sine and cosine functions. 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 when examining the overall torque output from the machine. Figure 13 shows an example of the computed amplitudes from a Fourier analysis of the torque curves in Figure 12.

Compressor and engine manufacturers will often provide this 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 [37] and Lloyd's Shipping Register.

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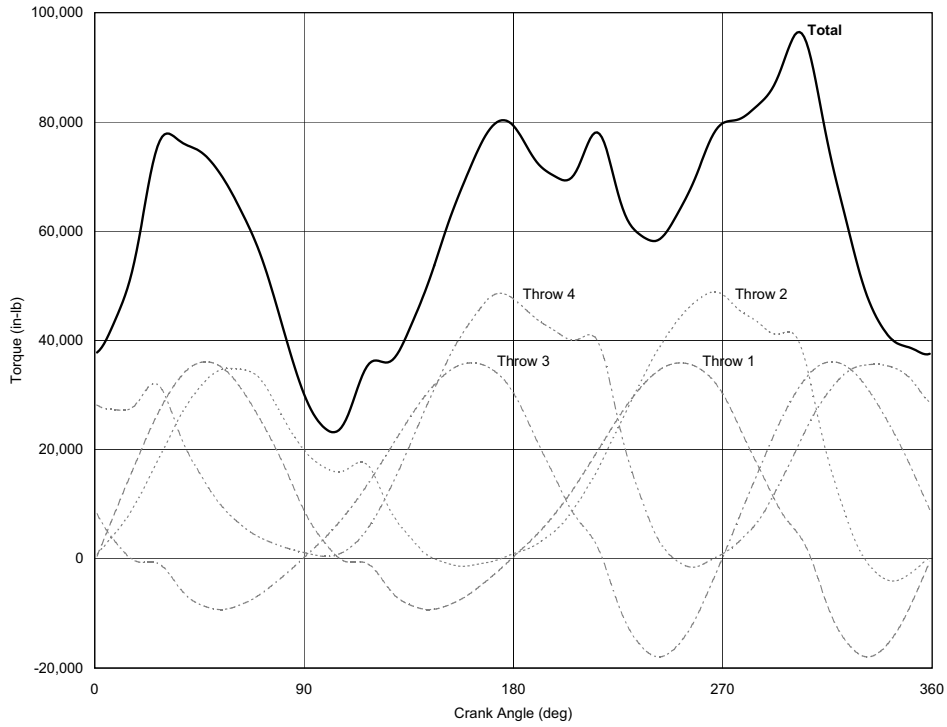

Figure 12: Compressor Torque Variation Versus Crank Angle

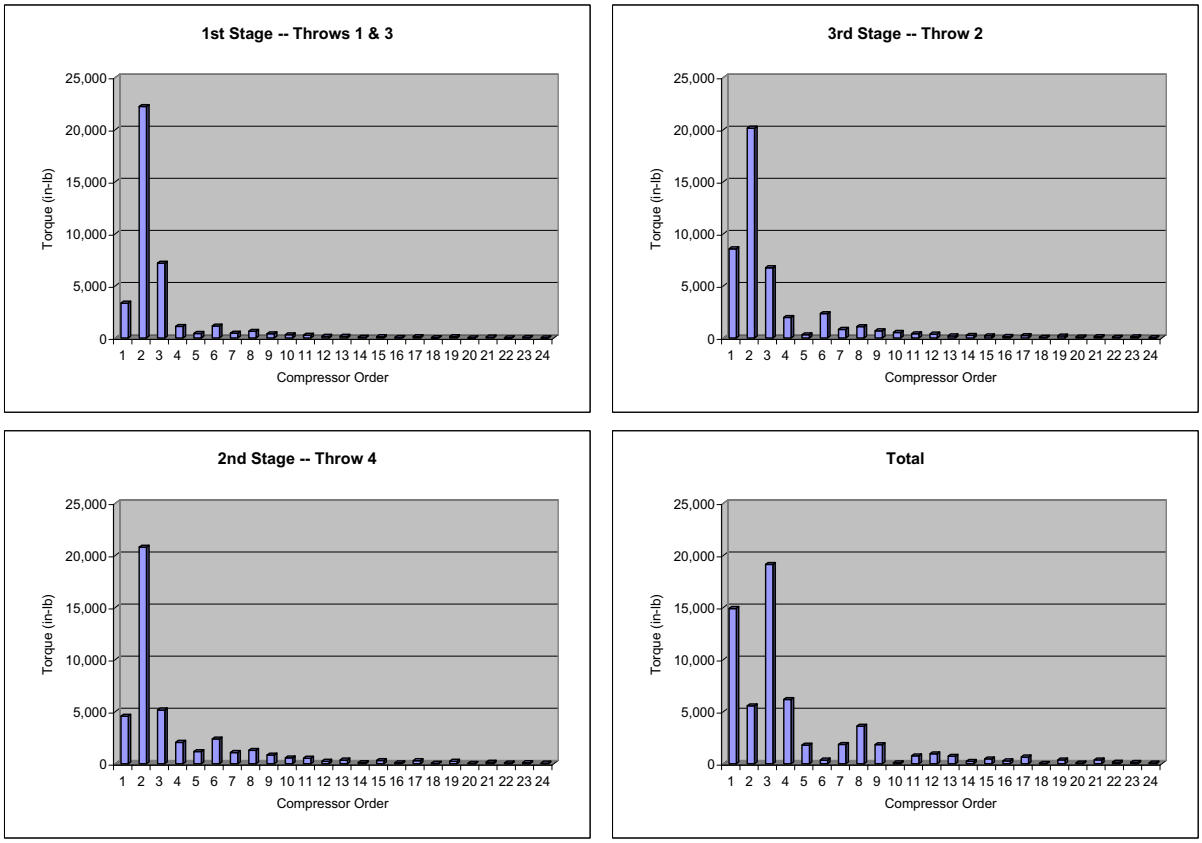

Figure 13: Fourier Analysis of Compressor Torque Excitation

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8.2 Compressors

For compressors, suction valves allow the gas to enter when the piston approaches BDC and the cylinder pressure is lower than 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 in the discharge line, causing the valves to open. Single-acting compressors use only one side of the piston while double-acting cylinder use both the crank and head ends. Unloaders and pockets can also have a major effect on 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 14 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. Situations where torsional vibration problems can occur in compressors:

- **Valve Failure** – A compressor valve failure can be analyzed 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 and used in the analysis.
- **Load Steps** – All load steps must be considered in the analysis, such as unloaders, pockets, etc., as these load steps can significantly affect the harmonic content and influence the torsional responses. The maximum horsepower case will not necessarily correspond to the maximum torsional excitation at all harmonics. The full range of operating conditions (pressures, flows, gas mole weights, etc.) must also be considered.

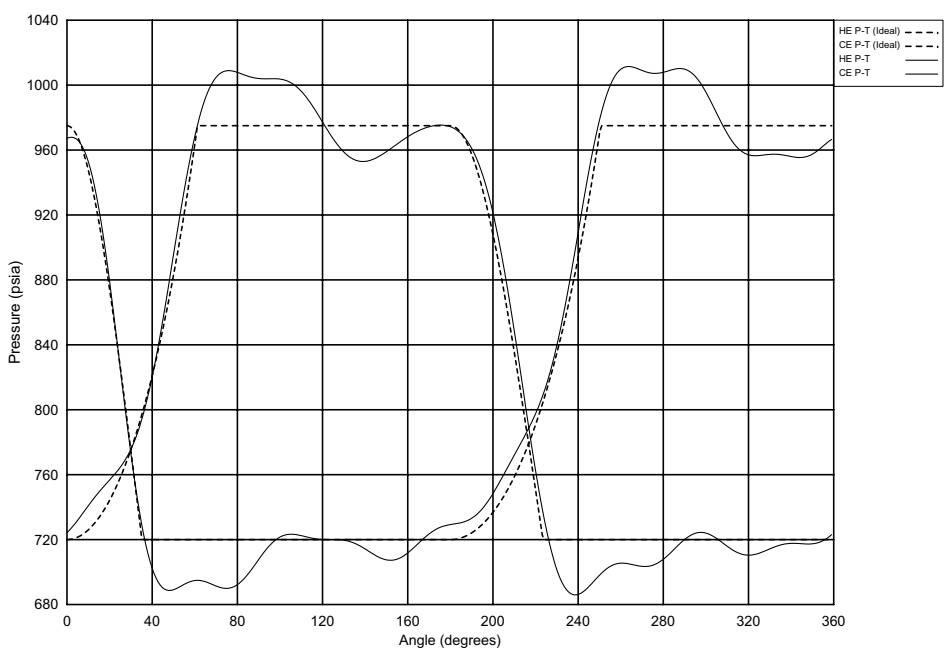

Figure 14: Comparison of actual versus predicted pressure-angle of compressor cylinder

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8.3 Engines

Combustion 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.

Some engines have a choice of firing orders, which can change the strong harmonics. In critical systems, the best firing order could be chosen to reduce the torsional response. Identical P-T cards are normally used for every cylinder, although in reality, the cards 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 15 shows large variation in peak cylinder pressures for an engine with multiple performance problems. Poorly maintained machines will tend to operate at non-ideal conditions that can cause high torsional vibration. Situations where torsional vibration problems can occur 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.
- **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.
- **Leaks** – Peak pressure can also be affected by leaks in fuel valves or compression pressure.

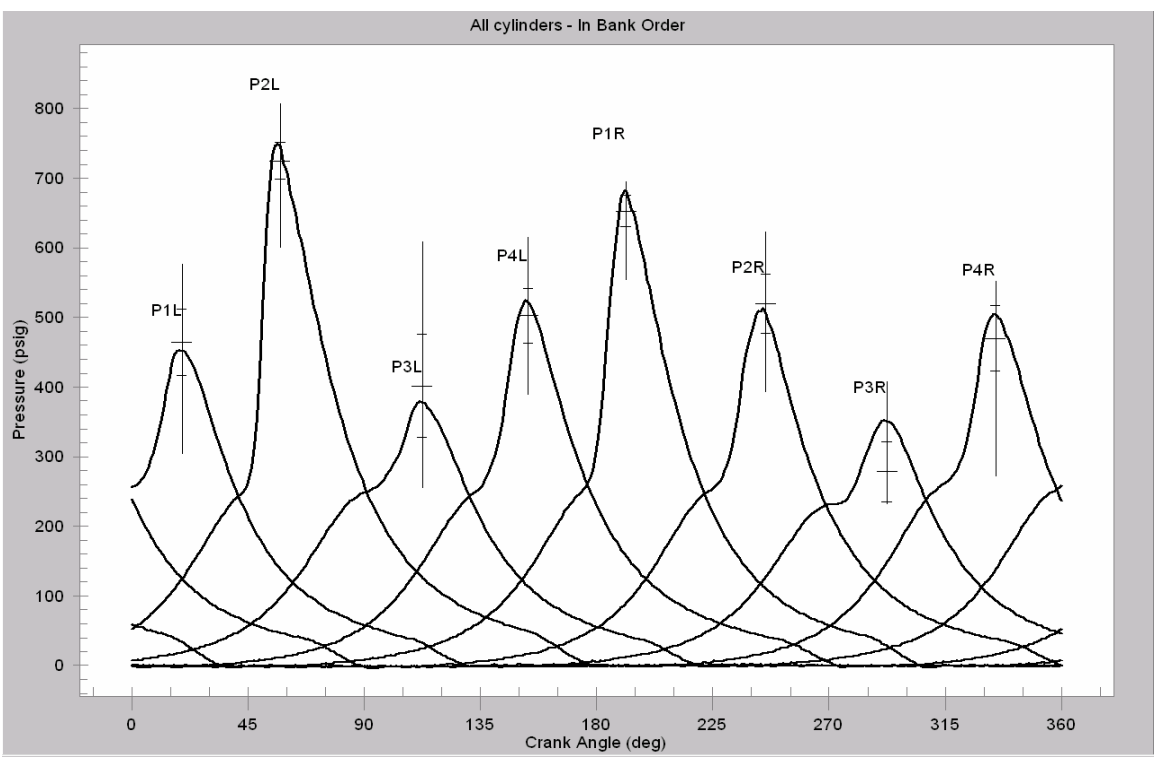

Figure 15: P-T parade for engine with performance problems

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8.4 Case History – Variable Operating Conditions Causes Failure of Rubber Coupling

Equipment	Natural gas engine driving a six-throw, two-stage reciprocating compressor through a rubber-in-shear coupling. Rated 3,335 HP at 1,000 RPM Speed Range = 750-1,000 RPM
Problem	Rubber element failures ranged from radial cracks to complete melting.
Cause	Coincidence of first torsional natural frequency with 1× at minimum running speed. Abnormal operating conditions (valve failures) resulted in increased 1× excitation.
Solution	Increased minimum speed to allow adequate separation margin. Recommended new scrubber to prevent slugging compressor valves.

A natural gas compressor station experienced several failures of a rubber-in-shear coupling. Typically, the units were operated at 1,000 RPM, but would operate at speeds as low as 750 RPM depending on the pipeline conditions. The compressors were normally operated double-acting (both ends loaded), although the first stage cylinders were equipped with head-end unloaders.

A strain gage telemetry system was installed on the compressor shaft near the coupling. The measurements showed the first TNF to be near 760 CPM or 12.7 Hz (Figure 16), which is within the operating speed range. However, with normal loading conditions the measured dynamic torque was acceptable.

Plant personnel reported that valve failures were common, so an abnormal operating condition was tested with cylinders 1 and 5 single-acting. Data obtained during the speed run revealed high vibratory torque at speeds below about 850 RPM (Figure 16). The amplitudes were approximately two times above the allowable vibratory torque of the coupling, resulting in excessive heat load and eventual failure of the rubber elements.

This case history shows that cylinder loading, valve failures, etc., can significantly change the torsional response of a system. Also, this demonstrates why the first torsional natural frequency should not be located within or near the operating speed range when using a soft coupling.

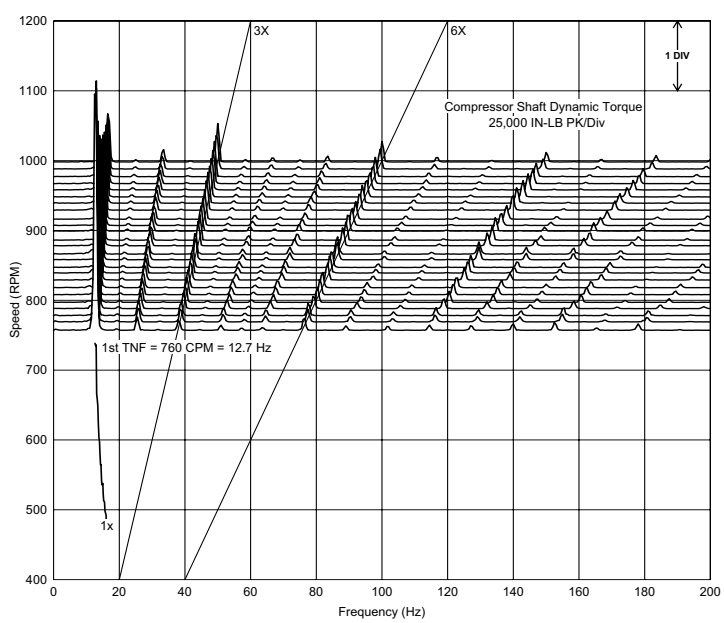

Figure 16: Waterfall plot of compressor shaft torque

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8.5 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 at 895 RPM
Problem	Motor shaft failure and coupling damage.
Cause	Coincidence of first TNF with 4× running speed. Acoustical resonance at 4× 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 17. Note that this is a photo of a section that was cut out of the failed motor shaft. The cracks occurred at a 45° angle to the shaft axis, which is indicative of torsional fatigue. The cracks reportedly initiated from poor welds at the shaft-spider interface.

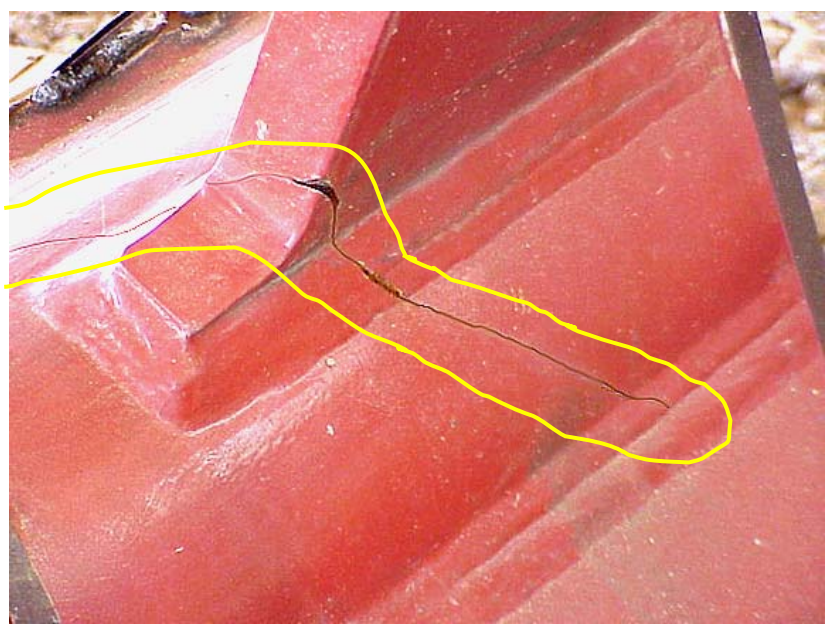

Figure 17: Failed motor shaft

After the motor shaft was replaced, the unit was 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.

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 18), which was coincident with 4× running speed.

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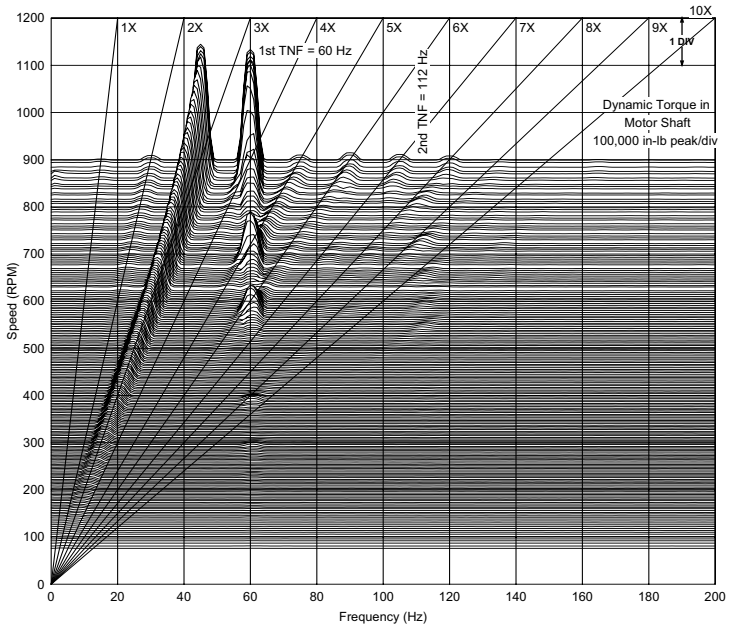

Figure 18: Waterfall plot of compressor shaft torque during coastdown – original system

The maximum overall dynamic torque in the shaft was 750,000 in-lbs peak and occurred during operation at load step 3 (Figure 19). This was approximately 2.5 times the full load torque and corresponded to a stress of 18,100 psi peak (based on an SCF of 2). At load step 3, cylinders 1 and 2 are double-acting, while 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.

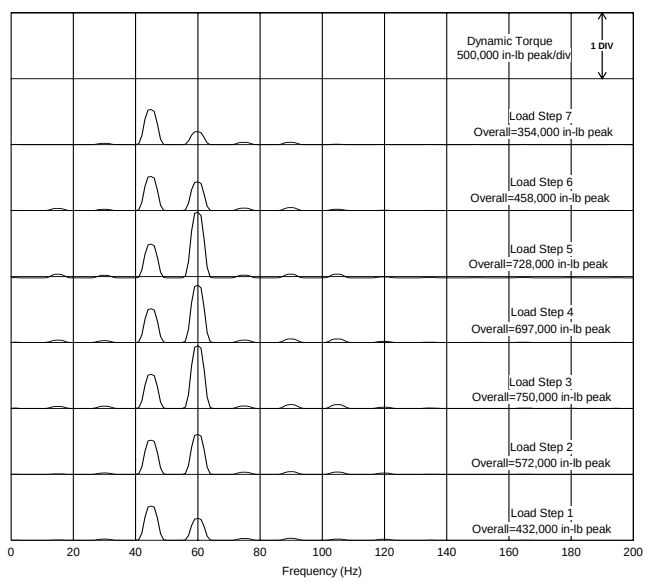

Figure 19: Frequency spectra at each load step – original system

A torsional analysis had been performed by another company prior to installation of the unit. The original analysis predicted the first TNF to be near 70 Hz, which is approximately a 17% error versus the actual frequency of 60 Hz. A second analysis was then performed to evaluate modifications to detune the system. Using a multi-mass motor model and including the effects of the core stiffening, the first TNF correlated with 60 Hz.

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To detune the first torsional natural frequency from the 4th order excitation, an internal flywheel (five donuts) was recommended between cylinders 4 and 5. Follow-up testing verified the first TNF was reduced to 55 Hz with the new flywheels (Figure 20), which provided an adequate separation margin from the 3rd and 4th compressor harmonics.

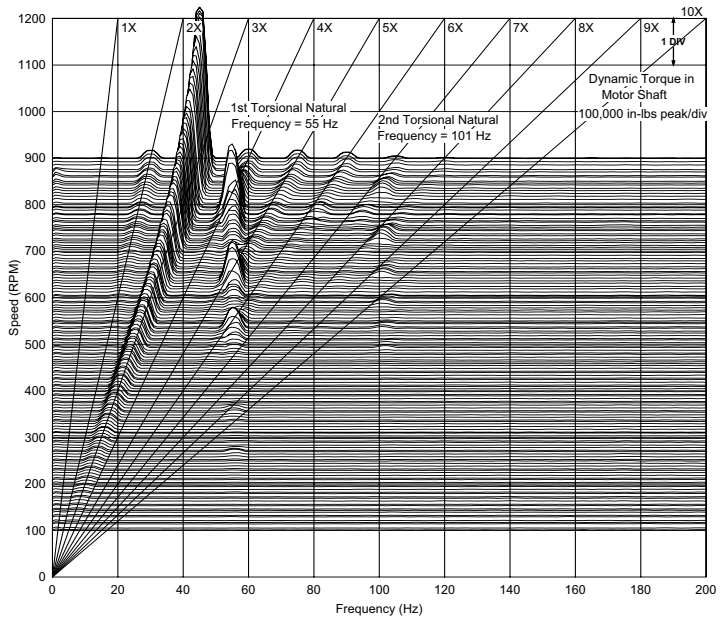

Figure 20: Waterfall plot of compressor shaft torque during coastdown – modified system

A significant discrepancy was noted between the calculated and measured 4th order response amplitudes at certain load steps. The calculations using an ideal P-V card for the torque-effort calculations, meaning performance degrading phenomena such as gas pulsation and valve losses were not considered, were as much as 17 times lower than those measured at load step 3. Therefore, actual P-T cards were acquired from each cylinder and used in the torque-effort calculations.

The existing and proposed (compressor with internal flywheels) systems were re-analyzed on the computer using the measured P-T card data. The correlation between the measured and calculated torsional amplitudes of the existing system was greatly improved. The P-T data revealed significant 4th order pulsation, which was later determined to be 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.

This case history shows that:

- 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 (see Section 5).
- 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.
- 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.

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8.6 Case History – Incorrect counter-weights combined with rough operating conditions leads to crankshaft failure of engine

Equipment	Diesel engine driving a synchronous generator. Rated 6 MW at 450 RPM
Problem	Engine crankshaft failure.
Cause	Coincidence of first torsional natural frequency with 4× running speed. Poor engine maintenance and rough operation resulted in increased 4× excitation.
Solution	Installed new counter-weights in engine to balance and detune torsional. Recommended engine monitoring program to improve engine performance.

Within two years, two crankshaft failures occurred in a diesel engine driving a 6 MW synchronous generator at 450 RPM. Both failures were at journal 8 (between throws 7 and 8) and consisted of a 45° crack, indicating a torsional vibration problem (Figure 21).

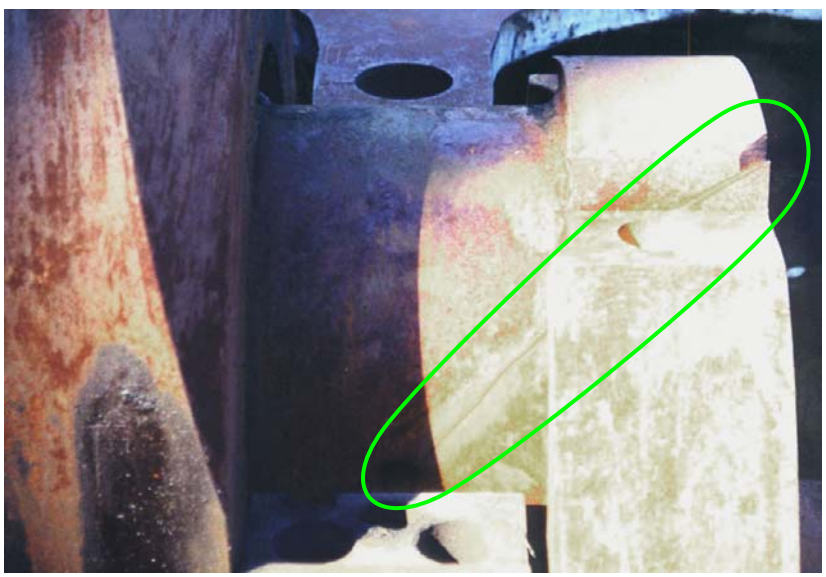

Figure 21: Failed engine crankshaft

The engine, originally certified for nuclear service as an emergency back-up generator, was purchased used but had only a few operating hours by a power plant. After the first crankshaft failure, there was concern that the flywheel that came with the engine was mismatched with the generator, which had been taken from another site. Therefore, the larger flywheel from the original set that the generator came from was used with the new crankshaft. Unfortunately, changing the size of the flywheel did not correct the problem, as a second crankshaft failure occurred.

Torsional vibration testing was performed after the second new crankshaft was installed. Torsional oscillation at the non-drive end of the engine was measured with a torsiograph mounted to the crankshaft. The engine speed was varied with the generator unloaded and the major engine orders were tracked (Figure 22). The first torsional natural frequency was determined to be 29.6 Hz, which corresponded to a 1% separation margin from the fourth engine order. Using the half power point method, the first mode was found to have an amplification factor of 36. The amplification factor was high because, unlike many engines, this unit did not have a viscous damper.

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The four-cycle engine has 16 cylinders. When operating smoothly, the engine will theoretically have low excitation at the fourth order. However, during rough operation (poor combustion, misfire, etc.), the fourth order could be much higher. This was the case as the engine was converted to heavy fuel and bank-to-bank imbalance was occurring. Variation in temperatures among the cylinders was noted indicating uneven loading.

Further investigation found that this particular engine was configured with four large rotating counter-weights. Reciprocating machinery produces unbalanced shaking forces and moments at 1× and 2× running speed. Rotating counter-weights are often mounted on the crankshaft webs to offset the unbalanced forces at 1× running speed. Depending on the type of service, this type of engine could be fitted with as many as 16 counter-weights (for marine application). In addition to balancing, the inertia of the counter-weights has a flywheel effect, which can affect the torsional natural frequencies of the system.

To detune the first torsional natural frequency away from the running speed of the engine, a torsional analysis was performed. Ker Wilson's formula [1] for crankshaft stiffness was found to give good correlation to measured data. The first calculated mode shape showed that the engine flywheel was actually at a node and that using a larger flywheel would have little influence on the system. This is why changing the flywheel after the first crankshaft failure did not make a difference. Instead of changing the flywheel, it was recommended that four counter-weights be added to the crankshaft. This allowed better balance of the unit and added inertia to detune the first torsional natural frequency.

In addition to performing the natural frequency calculations, pressure data measured from an engine cylinder was input into the torsional program to calculate the forced response. By simulating smooth and rough operation, it was possible to see how the fourth engine order could be increased. Since shaft strains could not be easily measured inside the engine at the failure location, the computer analysis was used to predict shaft stresses levels.

With eight counter-weights installed, the unit was re-tested by varying the speed unloaded (Figure 23). The first torsional natural frequency was lowered to 28.1 Hz (half way between the 3.5× and 4× engine orders). With the resonance safely removed from the engine order, torsional vibration readings were taken at various load steps until the generator was fully loaded. The torsional amplitudes were found to increase linearly with load.

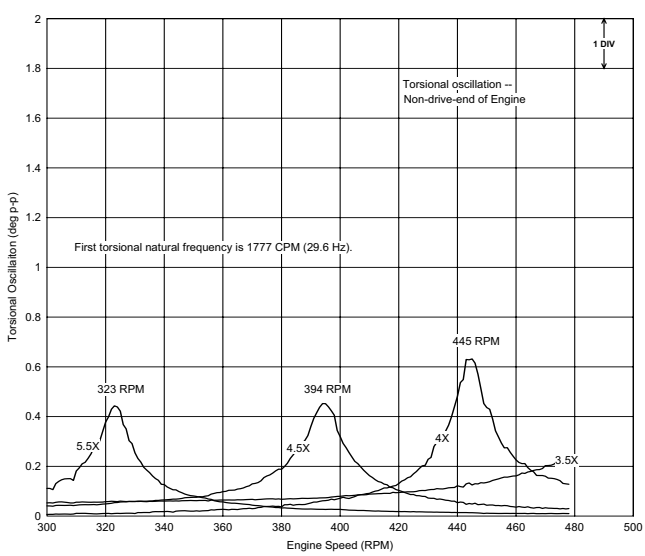

Figure 22: Order track plot of original system

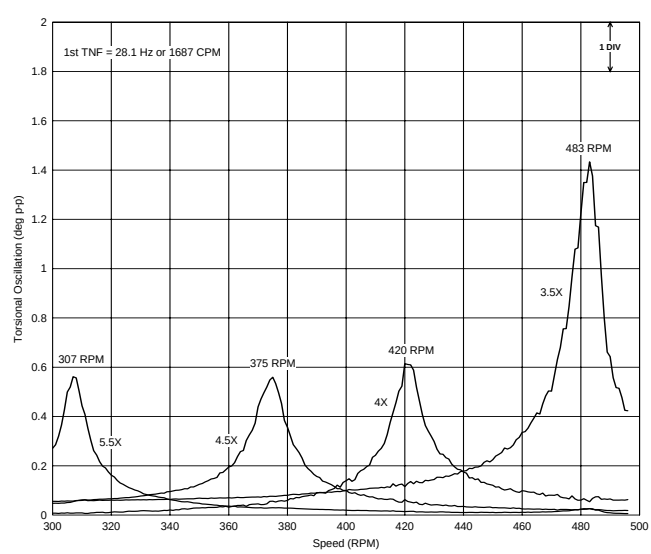

Figure 23: Order track plot with 8 counter-weights

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The measured data were input into the torsional program and it was verified that the crankshaft stresses were below allowable levels, particularly at the previous failure location. DEMA recommended levels for engines have been given as 7,000 psi for overall combined stress and 5,000 psi for individual orders [30].

This case history shows that:

- Continuous operation at a torsional resonance should be avoided. Rotating counter-weights are typically used for balancing shaking forces produced by reciprocating equipment; however, they can also influence torsional natural frequencies. Counter-weights should be looked at from both standpoints during the design phase.
- Relying on an engine to run perfectly all the time is unrealistic and can lead to failures. If only ideal P-T cards were considered and all cylinders were assumed to behave identically, the fourth engine order would not be predicted to be a problem. However, experience indicates that rough operation commonly causes the forth order to be fairly significant. In many instances, particularly in remote areas with limited operation and maintenance skills, equipment may not be properly operated and/or maintained; therefore, "rough" operating conditions should always be included in a torsional analysis.
- A regular engine performance monitoring program will help minimize problems due to poor engine operation.

9 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, such as acoustical resonances that cause increased pulsation and corresponding 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% to 35%. 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% 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% separation margin between a calculated torsional natural frequency and any significant excitation order. In many cases, a minimum 15% 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.

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9.1 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 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 6,000 HP 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 pound 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. A strain gage telemetry system on the motor shaft between the flywheel and core measured shear stresses and torques.

A synchronous motor produces mean and pulsating torque 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 torsional natural frequency of the system 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 time-domain data, the first torsional natural frequency 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 24 shows the first 14 compressor orders as diagonal lines. The first torsional natural frequency can be seen as 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 non-linearity sensitive to centrifugal force, such as looseness in the system.

The flanged connection between the compressor crankshaft and motor shaft at the flywheel was checked and the bolt torques increased. The unit was re-tested 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% in the torques [31].

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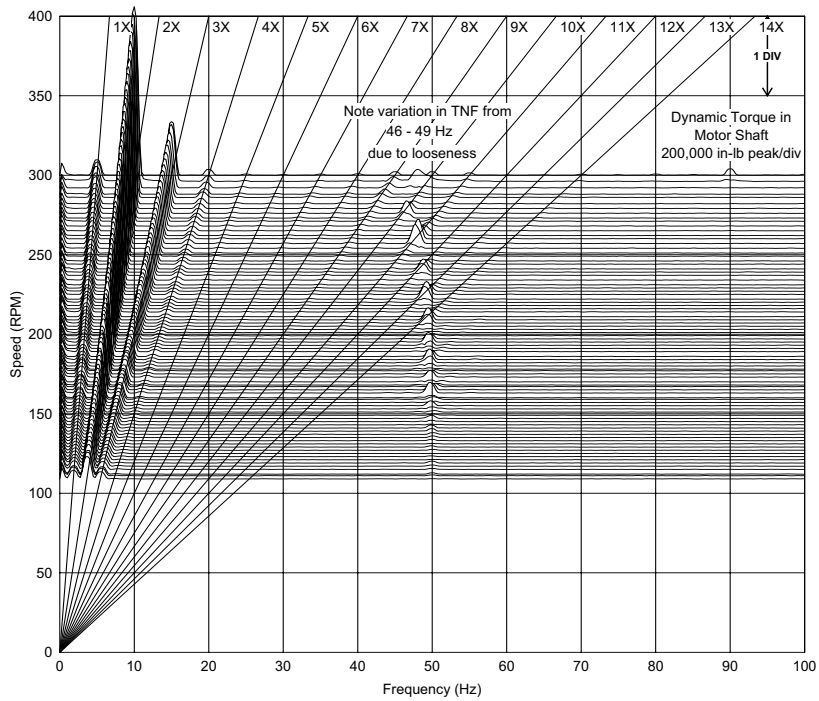
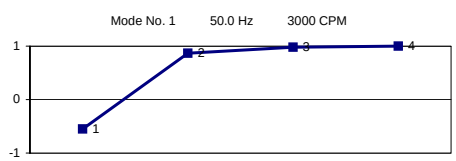


Figure 24: Waterfall plot of motor shaft torque during unloaded coastdown – unit as-found

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 torsional natural frequency (Figure 25).

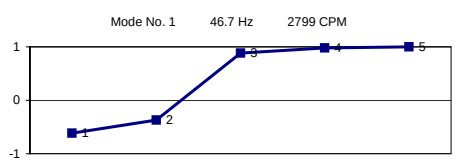
Normalized Model

Station	WR^2 in-lb-s^2	Stiffness million in-lb/rad	Description
1	59,490		Motor
		2,276	
2	34,701		Flywheel
		2,281	
3	1,286		Cylinder
		5,902	
4	1,317		Cylinder



Loose Motor Poles

Station	WR^2 in-lb-s^2	Stiffness million in-lb/rad	Description
1	46,000		Motor Poles
		10,000	
2	13,490		Rest of Motor
		2,276	
3	34,701		Flywheel
		2,281	
4	1,286		Cylinder
		5,902	
5	1,317		Cylinder



Note: Poles approximately 77% of total motor WR^2.

Ring Added to Flywheel

Station	WR^2 in-lb-s^2	Stiffness million in-lb/rad	Description
1	59,490		Motor
		2,276	
2	42,155		Flywheel
		2,281	
3	1,286		Cylinder
		5,902	
4	1,317		Cylinder

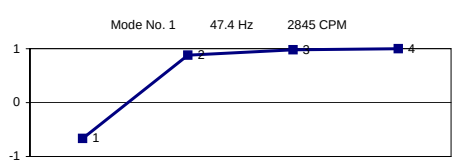


Figure 25: Torsional natural frequency calculations

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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×. However, higher orders can be greatly amplified by torsional resonances in stiff systems without couplings. The half power point method was used to determine the amplification factor (AF) of approximately 80, indicating little damping in this system at this frequency. While this observed AF value is high, similar AF's (100-110) have been documented by others [32, 33].

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 9th and 10th compressor orders at synchronous speed. The system was re-tested 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 26). All of the compressor load steps were tested and the modified system was found to be acceptable.

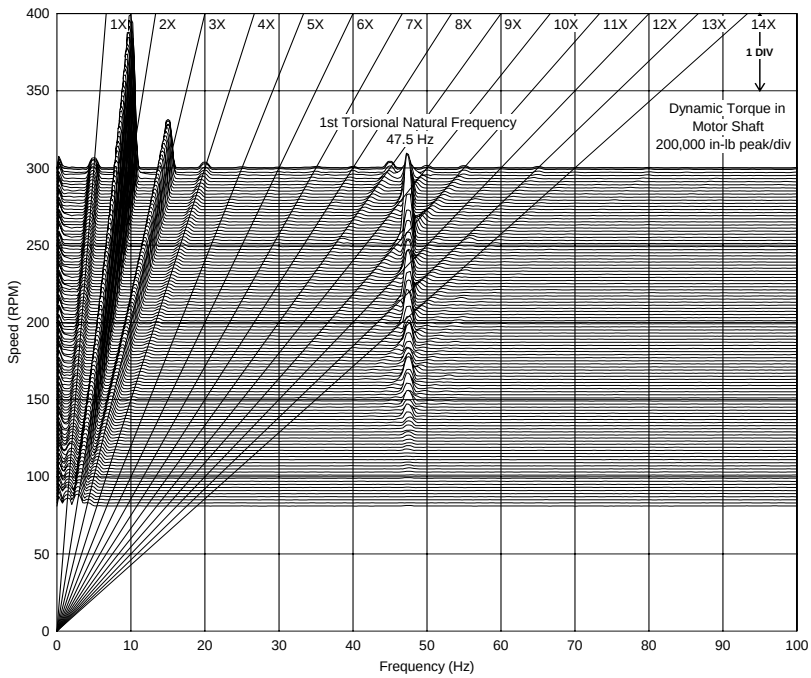

Figure 26: Waterfall plot of motor shaft torque during unloaded coastdown – modified system

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10 Torsional Measurements

Performing a computer torsional analysis is always recommended for new systems and should result in an acceptable design if current standards are used. However, there are still situations where torsional measurements may be required.

- **Previous torsional failures** – Torsional vibration is often "silent" and not detected by shaft proximity probes or accelerometers. Coupling chatter and compressor oil pump failures can be indicators, but many times torsional problems are not detected until a shaft failure occurs. Torsional measurements can be used to help diagnose the cause.
- **Critical applications** – A system that poses unusually high risks to life, other machinery, or plant processes. This may be 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.
- **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 re-staging and/or changing operating conditions, should be re-analyzed or tested.
- **New unit for municipality** – Many municipalities have specifications that require torsional vibration testing of new units by a profession engineer.

10.1 Test Conditions

For compressor systems, torsional measurements should be obtained at multiple conditions, including start-up, unloaded operation, all 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 the preferred way of plotting. 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. All data should be simultaneously stored using a multi-channel recorder. The following devices can be used to measure torsional vibration.

10.2 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 - 1,000 Hz. The device must be mounted on a free end of the shaft (Figure 27), 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.

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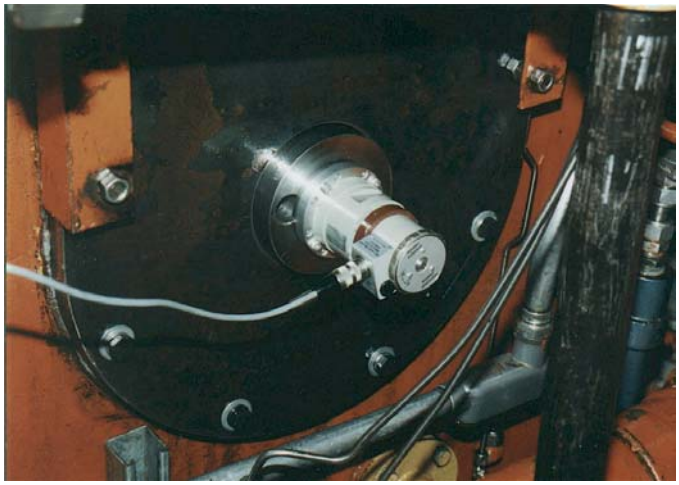

Figure 27: Typical torsiograph installation

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.

10.3 Strain Gage Telemetry System

A strain gage telemetry system can be used to evaluate actual transmitted and dynamic torque in a shaft. A full bridge arrangement with 4-gages can measure torsional strain while negating strains due to bending, axial and temperature effects (Figure 28). The voltage signal proportional to strain can be converted to torque or shear stress. The measured stress can then be compared to allowable levels.

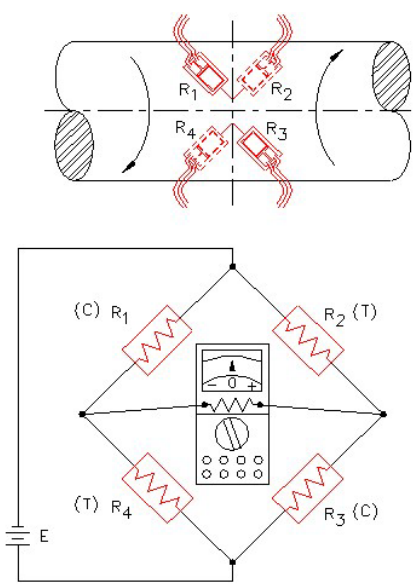

Figure 28: Full bridge arrangement with 4-gages

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If possible, the gages should 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 (Figure 29). Installation of the gages and telemetry system normally requires several hours with the unit down.

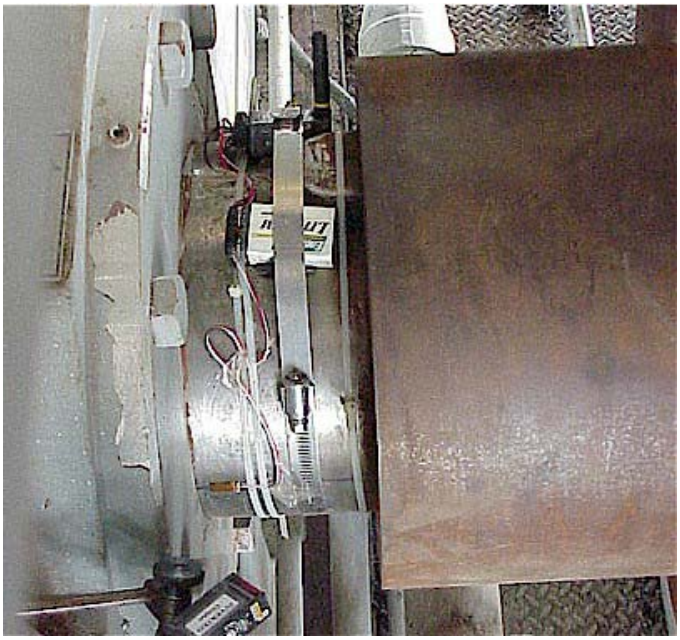

Figure 29: Strain gage telemetry system installed between gearbox and coupling hub

10.4 Frequency Modulation

The frequency modulation system uses proximity probes or a magnetic pick-up 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.

10.5 Laser Vibrometer

A laser vibrometer can also be used to measure angular displacement in degrees. This technology has recently been improved. The laser is non-contacting and can be "pointed" at the rotating surface of interest. No downtime is normally required. However, the laser vibrometer may not be capable of accurately evaluating transient conditions, such as start-ups, speed sweeps, or shutdowns. This limits its ability to measure natural frequencies of a system, as speed changes are normally required to track through a resonance. Many of the original laser vibrometers had poor frequency response (could not measure below about 10 Hz) and displacement dynamic range (could not measure less than about 0.1 degree), which limited its usefulness, particularly in low speed reciprocating equipment. However, the newer models have a frequency range down to 0.5 Hz and a dynamic range of 0.01 to 12 degrees.

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11 Guidelines for Torsionally Reliable Reciprocating Systems

A summary of recommendations has been grouped into three stages: Initial Design, Torsional Analysis, and Operation/Maintenance.

11.1 Initial Design

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% of the shaft diameter to minimize the stress concentration factor (SCF) per USAS B17.1 – Keys and Keyseats.
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. Care must be taken when using couplings with rubber blocks in compression. This type of coupling has nonlinear stiffness which varies with torque. This makes it difficult to tune a natural frequency between compressor orders due to varying torque during each rotation. The stiffness of all rubber couplings can vary with temperature and age, which could also affect the torsional natural frequencies over time.
7. Fabrication details, such as the motor pole bolt torque, should consider loads due to centrifugal force and torsional vibration.
8. 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.
9. 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.

11.2 Torsional Analysis

10. A steady-state torsional analysis should be performed that includes 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.
11. If required, a time-transient analysis for synchronous motor start-up, electrical faults or to predict peak torques during loaded shutdowns should be performed. This may also include fatigue damage calculations.
12. Identify any ambiguous or incomplete information. Verify the manufacturer's inertia and stiffness calculations when possible. Consider that the torsional stiffness of couplings could

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differ by as much as ±35% from catalog values. Determine if the coupling stiffness includes the 1/3 shaft penetration effect.

13. Check that dynamic torque and heat dissipation limits are not exceeded if using a rubber coupling. Consider that the rubber stiffness can vary with temperature. Continuous running at a resonant condition could easily overheat and damage the coupling before the elevated temperature decreases the stiffness and moves the natural frequency.
14. If a problem is found with a system as designed, possible modifications to consider include: changing the coupling, flywheel (external or internal), or damper.
15. Engine dampers are designed to protect the crankshaft and not necessarily the attached equipment. A torsionally soft coupling may be required to isolate sensitive equipment, such as gearboxes from engine excitation and to add more damping to the system.
16. A reciprocating compressor driven by a VFD motor over a large speed range will have multiple excitation frequencies that virtually guarantee torsional resonances within the operating speed range. Therefore, adding a rubber coupling in shear or locking out speeds to avoid resonances is usually required. In some cases, it may be possible to install a damper on the auxiliary end of the compressor. The trade-off is additional maintenance of the coupling to protect the shafts and/or auxiliary equipment.
17. In a system with a VFD and rubber coupling, the ramp rate or frequency at which the VFD controls the speed of the unit should not be coincident with a torsional natural frequency.
18. Constant speed units should have sufficient separation margins between torsional natural frequencies and significant excitation sources such as compressor orders, engine orders and electrical frequencies.
19. In systems with a gearbox, the dynamic torque at the gear mesh should be limited to 30% to 40% of the rated torque. To prevent backlash, the dynamic torque should not exceed the transmitted torque (even when unloaded). The gear manufacturer should review and approve the results.
20. Excitation orders that should be theoretically low can still cause failures due to bank-to-bank imbalance, misfire, failed valves, or acoustic resonances. In addition to design operating conditions, these "non-ideal" operating conditions should also be considered in the torsional analysis.
21. 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.
22. If a flywheel is at a node for a particular mode, a larger flywheel will not lower the torsional natural frequency further. Other methods would have to be used to detune torsional resonances. It may be possible to install an internal flywheel on reciprocating compressors or add rotating counter-weights to an engine.
23. Amplification factors (AF) for motor-driven reciprocating compressors can vary from 25 to 50. However, short, stiff systems with a flanged connection (no coupling) could have much lower damping (AF≈ 100), greatly amplifying higher orders typically not of concern.
24. Motors with long cores should be modeled as multiple masses and springs to more accurately predict the torsional natural frequencies. Synchronous motor cores with a wheel design could be modeled satisfactorily as a single inertia.
25. Finite element analysis should be used to more accurately calculate the torsional stiffness of a spider constructed motor core, crankshafts, etc.
26. The torque-effort (torque modulation) of a compressor varies depending on the number of throws. For example, a two-throw compressor can generate significantly higher 1× and 2× harmonics than a six-throw compressor. This can result in high shaft stress, even in the absence of a resonance. Simply detuning the system may not be sufficient.

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27. Calculated pressure pulsation from an acoustic analysis could be included into harmonic torque coefficients for torsional analysis. High valve losses or acoustic resonances associated with the cylinder gas passage and bottle nozzles can create higher dynamic torque. If torsional problems occur after start-up, actual P-V cards should be obtained and used in the analysis.

11.3 Operation/Maintenance

28. Avoid continuous operation at a torsional resonance.
29. For critical applications, testing may be necessary during commissioning to verify the torsional natural frequencies and stresses.
30. Existing systems with a resonance could be retrofitted with a different coupling or flywheel to detune a natural frequency.
31. 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.
32. The silicon fluid in viscous dampers can overheat or degrade with time. Therefore, regular maintenance is required (every 24,000-35,000 hours). It can be easy to overlook dampers when hidden inside an engine case. An inoperative damper could cause a crankshaft failure.
33. A regular engine and compressor performance monitoring program could identify rough operating conditions and help prevent failures.
34. Changing the loading sequence could help reduce torsional vibration in a single-stage reciprocating compressor.
35. 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.

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13 Nomenclature

b	width of damper flywheel
c	damping constant (in-lb-s/rad)
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
G	shear modulus (psi)
h_1, h_2	damper internal clearances
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)
I_{rot}	rotational inertia of crankshaft throw
K_t	torsional stiffness
L_c	crankpin length
L_j	crankshaft journal length
L_w	web thickness
M_{recip}	reciprocating mass
N	rated speed (RPM)
P	rated power (HP)
R	throw radius
r_1	inside radius of damper flywheel
r_2	outside radius of damper flywheel
SCF	stress concentration factor
TNF	torsional natural frequency
UTS	ultimate tensile strength (psi)
W	web width
μ	viscosity
ω	frequency (rad/sec)

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