Vibration Matters: Reliability of Electric Motor Driven Pumps and Compressors

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Abstract: Reliable rotating equipment systems—such as pumps and compressors—are critical to achieving plant production goals and ensuring the financial success of organizations. Unexpected vibration behavior in rotating equipment can lead to production delays and introduce uncertainty into plant planning and operations.

Achieving high reliability and availability to meet production and growth targets—and ultimately ensuring financial predictability—begins with proper equipment specification, procurement, and the selection of appropriate components that keep the rotating systems running efficiently.

In this paper, we will explore the causes of vibration in electric motor-driven systems and present real-world field issues. We will also discuss how these problems can be avoided through the correct selection of motors and couplings used to drive pumps and compressors.

Index Terms — Induction motor, vibration, misalignment, ambient temperature

I. INTRODUCTION

Rotating machinery, such as compressors, pumps, blowers, motors, and other related equipment, forms the backbone for processing and transporting petroleum and chemical products. Induction motors are commonly used to drive rotating equipment, given their high reliability, efficiency, and simplicity.

While many factors may cause failures in these motors, among the most complicated causes are bearing failure and high vibration. High vibration on the driven equipment leads to operational challenges as it could reduce a plant's process capacities or bring about an unplanned shutdown of plant operations if the vibration is severe.

If a drive train is not properly designed for a site's ambient and process conditions, excessive vibration could happen, which would delay the whole start process. Overall vibration is an important issue which should not be ignored as reduces the reliability and life of equipment. Excessive vibration introduces dynamic loads that can cause component fatigue, shorten bearing life, and loosen fasteners. If the vibration problems happen in the field, then it is very critical to resolve as soon as possible.

To solve a vibration problem, one must differentiate between cause and effect by identifying the vibration's root cause. In other words: where does the force come from? Is the vibratory force the cause of the high levels of vibration or is there a resonance that amplifies the vibratory response. Perhaps the vibration is caused due to a misalignment between the drive train and the driven equipment.

In this paper, we discuss the real-life case study where vibration instability in the motor gearbox pump application is caused due to site ambient temperature swing along with improper coupling type leading to excessive drive train misalignment.

II. SOURCES OF VIBRATION IN INDUCTION MOTORS

A. Mechanical Excitations

1) **Mechanical Unbalance**

API 684 [1] describes balancing as "A procedure for adjusting the radial mass distribution of a rotor so that the mass centerline (principal inertial axis) approaches or coincides with the rotor's rotational axis, thus reducing the lateral vibration of the rotor due to unbalanced inertia forces and forces on the bearings, at once-per-revolution frequency (1X)." API 684 emphasizes that achieving a balanced condition for a rotating assembly is a fundamental element of maximizing machinery reliability**.**

An excess mass on one side of the rotor will cause unbalance. Rotor balance involves the entire rotor structure, which is made up of many parts. These include the shaft, rotor laminations, end heads, rotor bars, end connectors, retaining rings (where required) and fans.

Unbalance in the motor rotor can occur due to following reasons:

1) Tolerances in fabrication, machining, and assembly.
2) Variations within materials, such as voids, porosity, density, and finishes.
3) Nonsymmetrical structure during operation, such as shifting of parts due to rotational stress, aerodynamic forces and temperature changes.

Balancing problems can be minimized by precision manufacturing technology. For motors, rotor punching must be precision manufactured with close concentricity of all features and have a shrink fit on the shaft that is maintained at all operating speeds and temperatures.

The punching must be stacked square with the bore, uniformly pressed and clamped in position when shrunk on the shaft to prevent movement with speed change. When end connectors require retaining rings, the rings are made of high-strength material and designed with proper interference fittings. Rotor bars are shimmed and/or swaged, so they are tight in the slots.

Other less common methods to assure tight rotor bars include heating the core and chilling the bars, although these methods are not common. End connectors should be induction brazed symmetrically to the bars so they can help eliminate variations in balance due to thermal change. Ideally, the shaft and assembled rotor are precision machined and ground to concentricity well within .001 inch.

In induction motors, the rotor core is usually the source and location of the unbalance as most rotor shafts are machined to within very precise tolerances. The API 541 [2] balance requirements are of particular interest with respect to the vibration limits at operating speed. It requires residual unbalance not to exceed 4W/N ounce-inches at each journal, where W is one half the weight of the rotor and N is the maximum operating speed of the machine.

In SI units, this permitted unbalance level is 6350W/N gram-millimeters where W is the weight per journal in kilograms and N is the maximum operating speed. This permitted unbalance level corresponds to about G 0.70 in the ISO 1940-1 [3] system. Balance is more critical and more difficult to perform on two pole motors.

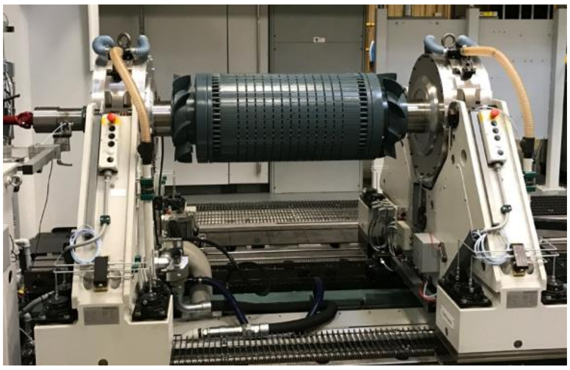


Figure1: Rotor in Balancing Machine

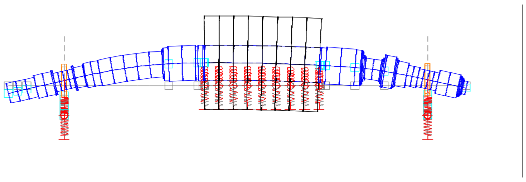


Figure 2: Bending Critical Speed of rotor

2) **Critical Speed**

A rigid rotor is defined as a rotor which operates far below their first pin-pin critical speed [1]. First pin-pin critical speed of rotor can be calculated based on bending stiffness of rotor only, not considering the bearing or support stiffness and is independent of bearing type used. However, the first critical speed of a rotor-bearing system changes when a rotor is mounted on fluid film bearings. Critical speed of a rigid rotor-soft bearing system (as shown in figure 2) can be predicted by damped critical speed analysis. This includes oil film bearing stiffness and damping coefficients besides shaft and support stiffness coefficients.

The combined stiffness of a rotor-bearing system is a function the shaft and the two bearings stiffness as given by [1].

$$\frac{1}{k_{system}} = \frac{1}{k_{shaft}} + \frac{1}{k_{bearing}} \qquad (1)$$

Equation 1 indicates that the combined stiffness of a rotor-bearing system will be less than the stiffness of the most flexible element which is generally fluid-film bearings in the case of sleeve bearing motors.

Fluid film stiffness and damping in plain cylindrical journal bearings are anisotropic (having a different stiffness and damping coefficients in different directions) in nature where normally the horizontal spring is softer (weaker) than vertical spring.

Due to this fact, a stiff shaft/soft bearing system will pass through two resonances also known as rigid body mode of vibration [1] before first bending mode occurs. API 684 defines critical speed as a shaft rotational speed that corresponds to a non-critically damped (AF>2.5) rotor system resonance frequency.

To prevent catastrophic equipment failure and increase reliability of mechanical components, continuous operation near or at critical speed should be avoided at all costs. API specification requires at least a 15% separation margin between critical speed and operation speed.

3) **Thermal Unbalance**

Thermal unbalance is a specific form of rotor unbalance caused by uneven rotor heating or differential bending due to thermal gradients. All rotors exhibit some change in vibration when transitioning from a cold to a hot operating state. API 541 permits up to 0.6 mil change in shaft vibration at the rotational frequency (1×) and 0.05 in/s change in housing vibration.

For continuous-duty applications, if the vibration levels during startup (i.e., with the motor cold) are not excessive, a greater cold-to-hot vibration change may be acceptable without risk of motor damage.

When the lowest possible vibration levels are desired at steady-state operating conditions, a hot trim balance procedure can be performed. To conduct this procedure, the motor should be run until thermal conditions stabilize, after which a trim balance should be performed promptly.

If required, the motor can be operated again after installing the initial trial weights and allowed to reach thermal stability before acquiring additional vibration measurements and applying final unbalance corrections per plane. Variation in bearing clearance affects the stiffness and damping characteristics of the lubricating oil film, thereby influencing rotor dynamics and vibration response.

In practice, bearing-clearance variation arises from the cumulative manufacturing tolerances of the shaft, bearing babbitt, bearing housings, and stator housing. Improper bearing clearance may also lead to oil-whirl–related vibrations, with larger clearances generally being more susceptible to oil whirl and increased lateral rotor vibration.

4) **Oil whirl**

In rare instances [5], induction motors with sleeve-bearing configurations are susceptible to large-amplitude lateral vibrations caused by a self-excited instability known as oil whirl.

Oil whirl is independent of rotor unbalance or misalignment and arises from hydrodynamic forces generated within the lubricating oil film. During oil whirl, the rotor orbits within the bearing clearance at a non-synchronous frequency typically slightly less than one-half of the rotor's rotational speed, and in the same rotational direction. If not controlled, this non-synchronous, self-excited motion can grow without bound, potentially leading to catastrophic bearing failure and major equipment damage.

At the onset of oil whirl, rotor behavior differs from classical critical-speed resonance. In critical-speed resonance, vibration amplitude increases as the rotor approaches its critical speed and decreases after passing through it. In contrast, during the inception of non-synchronous whirl, the vibration amplitude continually increases at a frequency near one-half of rotor speed and does not decay.

Lightly loaded bearings or excessively large bearing clearances are the most common causes of oil-whirl instability in induction motors. Several design charts based on theoretical and experimental studies have been developed to predict the onset of oil whirl [6].

These charts relate operating, geometric, and design parameters for various rotor-stiffness values and can be used to evaluate bearing stability once key geometric parameters are established. If the operating speed exceeds the predicted instability threshold, geometric parameters—such as bearing length, diameter, or clearance—can be adjusted to increase bearing stability and mitigate the risk of oil whirl.

B. Electromagnetic Excitations

1) Unbalance Magnetic Pull

An uneven air gap can cause electromagnetic excitations due to unbalanced magnetic pull, producing higher magnetic forces in the direction of minimum air gap. Errors in parts specification or loose manufacturing tolerances in stator and rotor components may cause an uneven air gap in the motor due to one or more of the following factors:

1) Out of round stator bore
2) Out of round rotor core
3) Out of round bearings housings and frame
4) Bent rotor shaft
5) Tolerance stack-up of mechanical components
6) Thermal bow of the rotor

Magnetic forces increase with the increasing air gap eccentricity. Lateral vibration of rotor and rotodynamic is strongly influenced by these magnetic forces and is discussed in later section of this paper. To control rotor vibration within API 541 limits, the air gap eccentricity of manufactured motor must be less than a maximum of 10% of designed air gap.

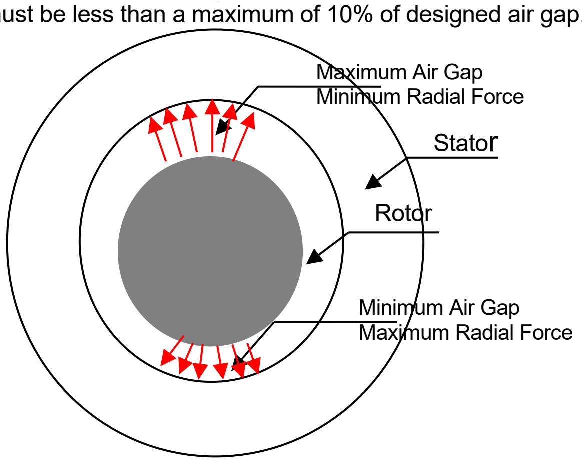


Figure 3. Schematic of Uneven Air Gap

2) Twice-line vibration (electrical vibration)

Twice line frequency vibration can also be significant part of the overall vibration in induction motors. For machines at speeds up to 1200 RPM, for example, the filtered and unfiltered vibration limit is 1.6 mils peak-to-peak displacement and 0.1 inches per second true peak velocity for rated speeds above 1200 RPM.

The source of this vibration is dependent on various parameters within the motor. The power source is a sinusoidal voltage that varies from positive to negative peak voltage in each cycle. The power supply applied to the stator produces a rotating magnetic field developing an attractive electromagnetic force between the stator and rotor.

This force reaches its maximum magnitude when the magnetizing current flowing in the stator is at a maximum, either positive or negative at that instant in time. As a result, two peak forces exist during each cycle of the voltage or current wave, which reduces stator and rotor to zero at the point in time when the current and fundamental flux wave pass through zero.

In turn, this will cause a vibration frequency equal to two times the frequency of the power source (twice-line frequency vibration). This vibration is extremely sensitive to the motor's foot flatness, frame and base stiffness, and the consistency of the air gap between the stator and rotor. It can also be influenced by the eccentricity of the rotor.

API 541 4th Edition requires the motor feet to fall within 0.005 inch of a common horizontal plane. Additionally, it limits the foot flatness to 0.0005 inch per foot and requires that different mounting planes be parallel to each other within 0.002 inch per foot.

One Period Flux Wave & Magnetic Force Wave

The basic forces are independent of load current and are nearly the same at both no load and full load. This is due to the main component of twice line frequency vibration, created by an unbalanced magnetic pull caused by air gap dissymmetry, does not change with load.

For 2-pole motors, the twice-line frequency vibration level will appear to modulate over time due to its close relationship with two times rotational vibration. Motors with problems such as a rub, loose parts, a bent shaft extension or elliptical bearing journals, can cause vibration at 2 times rotational frequency.

Due to its closeness in frequency to twice-line frequency vibration, the two levels will add together when they are in phase and subtract when they are out of phase. This modulation will repeat at a frequency of 2 times the slip on 2-pole motors.

Slip occurs in induction motors due to the rotor trying to stay in phase with the rotating field around the stator. The rotor falls behind the stator field by a certain number of revolutions per minute (slip speed) depending upon the load.

Even at no-load, twice-rotation vibration on 2-pole motors will vary from 7200 cpm (120Hz) due to slip. Since there are some slip-on induction motors, although small at no load, it may take 5 to 15 minutes to slip one rotation.

A larger load will produce a greater slip speed. Slip is typically 1% of rated speed at full load and decreases to near 0% slip at no-load.

Since vibration levels are not constant over time, API requires a modulation test be performed when measuring vibration. In a vibration modulation test the motor is allowed to run for a period of 15 minutes, and vibration is recorded continuously to allow the maximum and minimum to be established.

Other standards require only a vibration snapshot, which may not reveal the peak vibration over a period. In general, the methods used to reduce this level of vibration are the responsibility of the motor manufacturer.

Frame stiffness, flux densities, and isolation of the stator from the bearing housings will all influence this vibration level, but only foot flatness and parallelism are defined by API standards. The remaining design parameters are left to the motor manufacturer.

Good foot flatness has the added benefit of consistent results when the motor is placed in different locations. Although the design methods can vary, achieving lower levels of vibration is the primary objective.

Broken Rotor Bar

If a broken rotor bar or open braze joint exists, no current will flow in the rotor bar as shown in Fig. 5 [6]. As a result, the field in the rotor around that bar will not exist. Therefore, the force applied to that side of the rotor would be different from that on the other side of the rotor again creating an unbalanced magnetic force that rotates at one times rotational speed and modulates at a frequency equal to slip frequency times the number of poles.

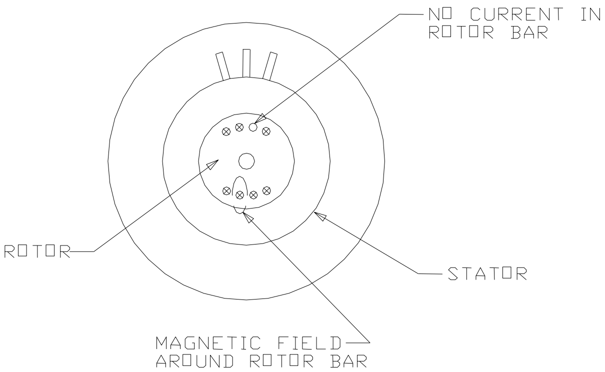


Fig. 4. Rotor with Broken Rotor Bar

If one of the rotor bars has a different resistivity, a similar phenomenon (as in the case of a broken rotor bar) can exist. It should be noted that this is one of the few conditions that cannot be seen at no-load.

But an additional phenomenon associated with this condition can be seen at no load after the motor is heated to full load temperature by any method that creates rotor current. These methods would include, coupled full load test, dual frequency heat run, multiple accelerations or heating by locking rotor and applying voltage.

In addition, broken rotor bars or a variation in bar resistivity will cause a variation in heating around the rotor. This in turn can bow the rotor, creating an eccentric rotor, causing basic rotor unbalance and a greater unbalanced magnetic pull, thereby creating a high one times and some minimal twice-line frequency vibration.

Rotor Bar Passing Frequency Vibration

High frequency, load-related magnetic vibration at or near rotor slot passing frequency is generated in the motor stator when current is induced into the rotor bars under load. The magnitude of this vibration varies with load, increasing as load increases. The electrical current in the bars creates a magnetic field around the bars that applies an attracting force to the stator teeth. For additional detail magnetic vibration, including stator teeth and stator core [6].

C. Vibrations due to coupled systems

1) Misalignment

The motor should be coupled to the driven equipment and aligned such that the vibration levels do not exceed the limits specified for the coupled assembly. The coupling must not be considered a vibration-damping device; instead, it should be aligned in accordance with the coupling manufacturer's specifications.

Proper alignment in both cold and hot conditions reduces shaft and bearing stress and minimizes vibration. Field experience and rotordynamic simulations indicate that a certain degree of misalignment has minimal influence on rotor vibration.

However, once misalignment exceeds a threshold, vibration amplitudes increase significantly with further misalignment. This behavior has been observed in the field, where drive-train vibration changes with ambient temperature due to thermally induced variations in alignment caused by differing thermal growth rates of mechanical drive components.

2) Base Excitation

If the motor is installed on a fabricated steel base, such as steel rails, the vibration measured at the motor may be significantly influenced by vibration originating from the base itself. Ideally, the base should be sufficiently stiff to meet the "massive foundation" criteria defined in API 541 [2]. This criterion requires that support-structure vibration near the motor feet be less than 30% of the vibration measured at the motor bearings.

Figure 5. Motor on concrete foundation

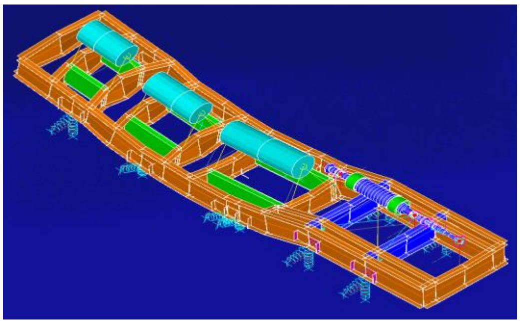


Figure 6. Drive Train on Steel Foundation

A weak or flexible motor base typically results in elevated 1× vibration, most commonly in the horizontal direction. However, it may also lead to high 2× (twice-rotational frequency) or 2f (twice-line frequency) vibration, both of which are common in induction motors.

Determining the nature and source of elevated 2× vibration requires vibration measurements at the motor feet in both the vertical and horizontal directions, including amplitude and phase, to identify the associated mode shape [7].

Pulsating air-gap torques invariable-frequency-drive-powered induction motors may cause not only torsional vibrations in the drive train but also lateral vibrations when the motor is mounted on a soft or flexible foundation.

3) Torsional Vibration

When the motor is coupled to the driven equipment such as compressor, pumps or extruders, with or without gearbox, its rotational inertias also get coupled resulting in a new system which has torsional natural frequency. Torsional vibration mode causes back and forth angular twisting of the rotor and its components and hence the name. Torsional natural frequency is a strong function of the ratios of the inertias of the individual components and magnitude of the coupling stiffness.

Issues related to torsional vibration are more common in variable frequency driven 4-lobe, 3-lobe pumps or 2-stroke, 4-stroke and 6-stroke reciprocating torque. These applications generate pulsating torques of various orders besides fundamental driving torque.

Such pulsating torques along with the variable speed can produce large spectrum of excitation torques which could line up with torsional frequency and excite torsional vibrations in the drivetrain.

Torsional vibration problems could be avoided by performing torsional analysis at the design stage of the drive train components. Rotational inertia of the rotor, torsional stiffness of the components are required to build the system model.

API 684 provides well-documented guidance for the building the models and various excitations to be considered for stress analysis. It is highly recommended to perform the torsional measurements during commissioning and compare the measurements with predicted models.

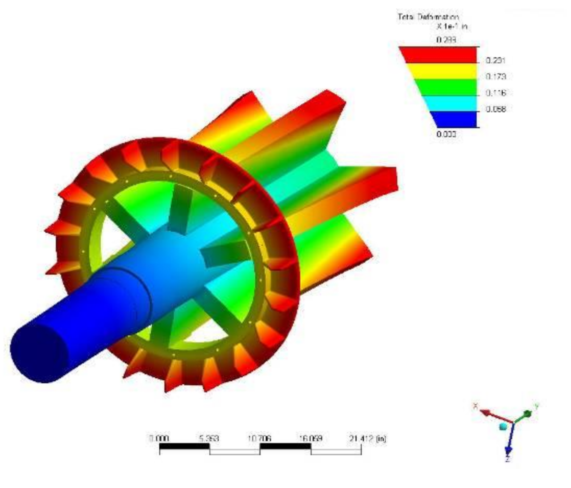


Figure 7. Twisting of shaft due to torque

III. HOW AMBIENT TEMPERATURE CHANGES AFFECT VIBRATIONS IN DRIVE TRAIN SYSTEMS

This next example will illustrate how a drive train consisting of motor-gearbox and pump vibration can change from acceptable levels to unacceptable levels because of completely normal ambient temperature changes. It is not unusual for motors and driven equipment to be used and installed outdoors. No matter what geographical location the product is located, it will always experience ambient temperature swings. It is easier to understand that the motor shaft height will grow from cold to hot due to heat from inside the motor, and that the thermal growth of the motor and driven equipment may not be the same. However, it may be more difficult to know how ambient temperature swings can drive this phenomenon. The following example will demonstrate the effect of ambient temperature swing on drive train vibration.

For example, the motor shaft height on a 580 frame is 14.5 inches from the foundation to the center of the shaft as shown in Figure 7. As the changes in shaft height are very small (0.001 inches of displacement), it is important to first understand that the shaft height has a significant tolerance. This tolerance is typically compensated during installation by placing shims under the motor feet. This is unimportant for the purposes of this discussion, but, if necessary, the relative shaft height can easily be adjusted during installation using shims. Once installed, the shaft height will be fixed at approximately 14.5 inches, plus or minus a small shift in height depending on temperature. When the equipment gets hot, the shaft height will grow . A rule of thumb commonly used in the industry is : 1 mil per inch for every 100 degrees centigrade of temperature change.

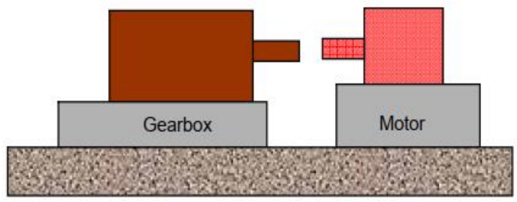


Figure 7. Thermal growth of Drive Train components

The stator winding inside of the motor will normally have a temperature rise of 80–105°C. On a totally enclosed fin-cooled machine, the bearing housings could rise as much as the stator winding rise. However, in this example, an open machine was used, and it was determined that the housing rise would be only 50% of the stator rise. For a Class B rise (80°C), the housing would then be 40° C. Using the above rule of thumb, the shaft height rise would be:

$$1 \text{ mil/in} \times 14.5 \text{ in} \times 40/100 = 5.8 \text{ mil}$$

If the driven equipment expands to the same center line as the motor, there is no problem. However, if they expand at different rates, misalignment and excessive vibration could occur.

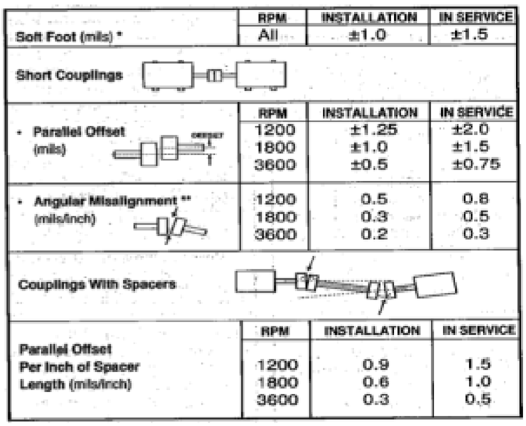
The normal variation between shaft centerlines cannot exceed 1.0 –2.5 mils total indicator run-out (TIR). This depends on the rotational speed, unless a special coupling is used, which can allow for this excessive offset. It is important to note that the offset of shaft centerlines is equal to one-half the TIR.

If the equipment driven expands at a much different rate, significant vibration could occur. The driven equipment may even be located on a different foundation and have a totally different shaft height, which can affect its sensitivity to thermal change.

See the drive train example in Figure 7. To calculate the vertical expansion from cold to hot, one would need to know the thermal change, the expansion in the driven equipment and foundations, and any other mounting medium that could affect the total shaft height.

If the coupling contains a spacer or spool piece extension or if a flexible coupling is used (which many times also includes a spacer), the misalignment allowance can be increased.

Figure 8 provides an estimation on the length of spacer required per inch of shaft offset.



	RPM	INSTALLATION	IN SERVICE
Soft Foot (mils) *	All	±1.0	±1.5
Short Couplings			
	RPM	INSTALLATION	IN SERVICE
• Parallel Offset (mils)	1200 1800 3600	±1.25 ±1.0 ±0.5	±2.0 ±1.5 ±0.75
• Angular Misalignment ** (mils/inch)	1200 1800 3600	0.5 0.3 0.2	0.8 0.5 0.3
Couplings With Spacers			
	RPM	INSTALLATION	IN SERVICE
Parallel Offset Per Inch of Spacer Length (mils/inch)	1200 1800 3600	0.9 0.6 0.3	1.5 1.0 0.5

* "Soft foot" describes the condition where the four mounting feet are not all in the same plane. Measured in mils (1 mil = .001 inches)

** To find angular misalignment in mils/inch of coupling diameter, measure widest opening in mils, then subtract narrowest opening in mils, and divide by diameter of coupling in inches.

Note: Up and down motion of driving and driven shafts with temperature may be in either direction.

Figure 8. Suggested Alignment Tolerances

If a spacer is required after the product is installed, it may be difficult or often impossible to move the motor away from the driven equipment. However, special couplings are available today that can be flexible in very short distances between shaft ends. An example of this coupling will be discussed later in this example.

Once the change from cold to hot for both the load and driven equipment is known, adjustments can be made to the cold alignment to compensate for the hot alignment. Keep in mind that this compensation only applies to the vertical shaft height change, and that either the motor or load shaft centers could also move horizontally.

This is common on gear boxes where the distance between gears will tend to grow with increasing temperature. Torque applied to the shaft can also cause the equipment centers to move apart. This movement is harder to measure and normally not done. However, it is common to measure the hot alignment at standstill to ensure the calculations are accurate.

The analysis so far has been straightforward, but it has not yet considered the effects of large ambient temperature swings. In this example, the hot alignment was perfect under a full-load condition, but the ambient changed a total of 50°C -10°C to +40°C.

In this case, the driven load was a water-cooled gear box and the motor was air-cooled. The cooling water temperature remained constant with ambient temperature. And for the sake of calculation, it can be assumed that the gear box temperature also remained constant. At the same time, the air-cooled motor was directly affected by the ambient air temperature, and as a result, the motor temperature rose 50°C with a calculated offset as follows:

1 mil/in. x 14.5 in. x 50/100 = 7.25 mils off-set

The installed coupling, which is common and good for many applications, is shown in Figure 9. The coupling's literature also stated that it was capable of handling 10 mils of misalignment. Although this coupling can take the force of misalignment, it was not known how this force would affect the performance of the drive train; therefore, the drive train had to be evaluated.

The force per mil of misalignment was measured to understand the effect on vibration, and it was discovered to be relatively linear at approximately 100 lbs. per mil of misalignment. In this case, the 7.25 mils of misalignment resulted in over 700 lbs. of radial force on the motor drive end bearing.

When the force was in the same vertical direction as the rotor weight (vertical down), the force started to approach the bearing loading limit. The vibration on the drive end, however, did not significantly change. The vibration on the non-drive end increased slightly due to the bearing being off loaded, but the vibration was still not excessive.

Now assume the misalignment was in the opposite direction. In this case the hot alignment was perfect at the maximum ambient temperature of 40°C and then the ambient temperature dropped 50°C. Since the gear box temperature did not change, the bearing was now being unloaded by 100 lbs. for every mil of misalignment.

If the bearings were antifriction bearings instead of sleeve bearings, the unloading would eventually lead to a bearing loading below the minimum level required to minimize skidding. Skidding would eventually lead to premature bearing failure. Sleeve bearings, however, require a certain level of loading to remain stable and maintain the oil film stiffness and damping which is required to minimize rotor vibration [9].

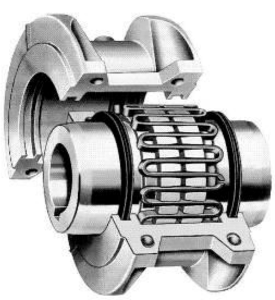


Figure 9 : Coupling [4]

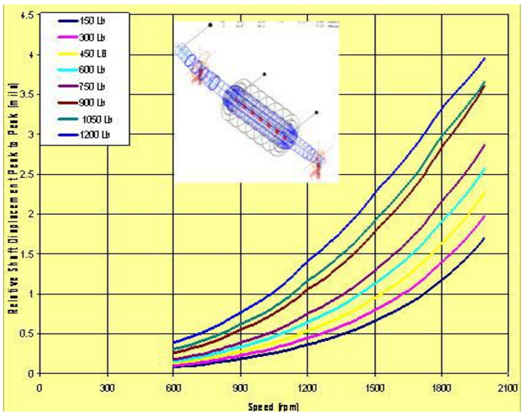


Figure 10. Vibration because of Unloading of Bearings

In an extreme case, the shaft could be lifted off its bearing losing the oil film stiffness and become totally dependent on the coupling and bearing load for any resistance to movement of the rotor. Since this set up would normally have minimal stiffness or resistance to movement, any unbalance of the motor rotor or coupling would go virtually unchecked and could vibrate out of control.

Other factors, such as magnetic pull, can also add or subtract from the force being applied to the bearing. In this case, where the rotor weight on each bearing was 1,200 lbs. based on a total rotor weight of 2,400 lbs., it would lighten the load to a point where the vibration would become excessive.

The result of unloading the bearing and the corresponding vibration was modeled, and the results of this analysis are shown in Figure 10. This phenomenon was also verified in the field under test. The vibration started at less than 1 mil at

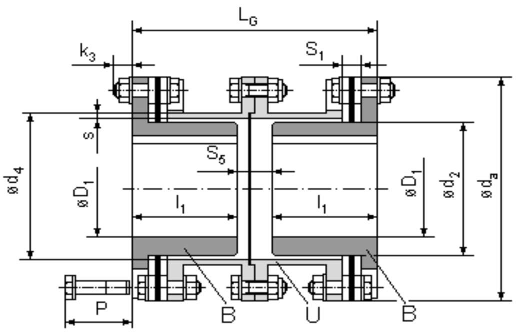


Figure 11: Flexible Disc Pack Coupling [4]

1800 RPM, increased to 2 mils vibration at 6 mils misalignment, and then climbed to 3-4 mils vibration as the misalignment approached 12 mils. There were more than 50 of these systems in the field, and they all performed differently. The quality of the alignment was inconsistent between units, and the coupling unbalance was sometimes in phase with motor unbalance and sometimes out of phase, where it would decrease the overall vibration. At times, the magnetic unbalance inside the motor would lift the rotor and off-load the bearing even further. Other applications would pull the rotor down, loading the bearings even further. As a result, none of the sets were performing in the same way and only a few were drastically different. Figure 11 shows a coupling that can handle significant misalignment. Normally, the flange for the disc pack would be at the end of the shaft.

Figure 12. Installed Drive Train

In this case, however, there was no room for a spool piece between the flanges which was needed for flexibility. In the picture the design has a flexible disc pack on the left and a flexible disc pack on the right which can be located against the motor housing and the driven equipment housing.

Since this application was already installed in the field and the shaft ends were very close together, a special coupling was required. In this case, the flanges were able to be turned away from the end shaft and placed closed to the motor and the driven equipment .

The spool piece in the center was split to provide all the required flexibility as shown in figure 12. This coupling now applied less than 10 lbs./mil of offsets compared to 100 lbs. per mil on the original coupling. Vibration remained unchanged throughout all future misalignment conditions.

Now all 50 applications in the field performed virtually the same and could have minimum issues when the radial loading on one bearing does not exceed 10% of the rotor weight. In the original drive train with the original rigid coupling, 10% would have amounted to only 120 lbs. or 1 mil offset maximum. Within that limit the vibration increases and bearing life reduction would not be significant. Any greater force would require the use of a flexible coupling to guarantee a long life the vibration requirement of 1.5 mils as defined in API 541 [1] .

In conclusion, these types of applications will have minimal problems when the radial loading on one bearing does not exceed 10% of the rotor weight. In the original drive train with the original rigid coupling, 10% would have amounted to only 120 lbs. or 1 mil offset maximum. Within that limit the vibration increases and bearing life reduction would not be significant. Any greater force would require the use of a flexible coupling to guarantee a long life.

VI. REFERENCES

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VII . VITA

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