

An Analytical Approach to Solving Motor Vibration Problems

Copyright material IEEE
Paper No. PCIC-99-20

William R. Finley
Senior Member
Large Motors & Pumps
Siemens Energy & Automation, Inc.
4620 Forest Ave.
Norwood, OH 45212

Mark M. Hodowanec
Member
Industrial Products Division
Siemens Energy & Automation, Inc.
4620 Forest Ave.
Norwood, OH 45212

Warren G. Holter
Industrial Products Division
Siemens Energy & Automation, Inc.
4620 Forest Ave.
Norwood, OH 45212

Abstract: Vibration problems in induction motors can be extremely frustrating and may lead to greatly reduced reliability. It is imperative, in all operations and manufacturing processes that down time is avoided or minimized. If a problem does occur the source of the problem is quickly identified and corrected. With proper knowledge and diagnostic procedures, it is normally possible to quickly pinpoint the cause of the vibration. All too often erroneous conclusions are reached as a consequence of not understanding the root cause of the vibration. This may result in trying to fix an incorrectly diagnosed problem, spending a significant amount of time and money in the process. By utilizing the proper data collection and analysis techniques, the true source of the vibration can be discovered. This includes, but is not limited to:

- Electrical imbalance
- Mechanical unbalance – motor, coupling, or driven equipment
- Mechanical effects – looseness, rubbing, bearings, etc.
- External effects - base, driven equipment, misalignment, etc.
- Resonance, critical speeds, reed critical etc.

Once the electrical and mechanical interactions in a motor are understood, and the influence external components have on the apparent motor vibration, identification of the offending component is usually straightforward. This paper provides an analytical approach for expeditiously understanding and solving these types of problems.

Index Terms: Induction Motors, Troubleshooting Vibrations, Cause of Vibration.

I. INTRODUCTION

Much has been written about vibration over the years. This includes many papers and books on vibration in general and a number of papers on vibration in induction motors in particular. This is an ongoing subject, continually extended by advances in analytical and diagnostic tools and methods. For this reason, and because this is an important and complex subject, it is worthwhile periodically to both present any new knowledge and experience as well as to review prior knowledge and concepts.

Vibration problems can occur at anytime in the installation or operation of a motor. When they occur it is normally critical that one reacts quickly to solve the problem. If not solved quickly, one could either expect long term damage to the motor or immediate failure, which would result in immediate loss of production. The loss of production is oftentimes the

most critical concern. To solve a vibration problem one must differentiate between cause and effect. For this to happen, one must first understand the root cause of the vibration. 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 support structure is just not stiff enough to minimize the displacement. In this paper the various sources of electrical and mechanical forces will be explained. Additionally, how the motor reacts or transmits this force and how this force can be amplified or minimized will be explained as well. When a vibration problem occurs it is important that one use a good systematic, analytical approach in resolving the problem. This includes performing the proper diagnostic tests. The process starts by listing all the possible causes for the particular identified frequency of vibration and any variations under different operating conditions. Then eliminate the incorrect causes one by one until all that remains is the true source of the problem, and now this can be efficiently eliminated.

II. SOURCES OF VIBRATION

There are many electrical and mechanical forces present in induction motors that can cause vibrations. Additionally, interaction of these various forces make identification of the root cause elusive. In subsequent sections, the major mechanisms are discussed. For a more comprehensive list of electrically and mechanically induced vibrations Table I should be referenced.

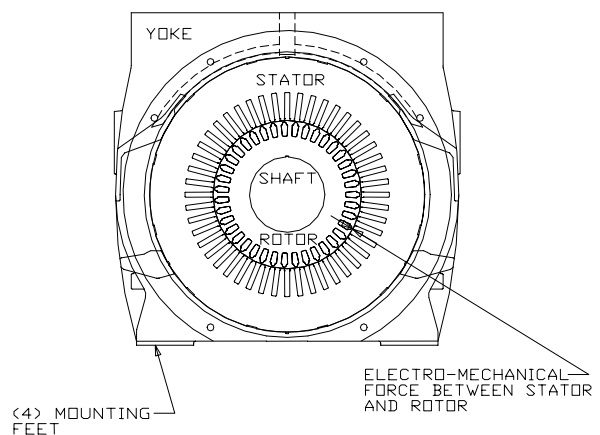


FIG. 1. Stator and Rotor

Twice Line Frequency Vibration:

There are many different forces and interactions as a result of the power source and the interactions between the stator and rotor as seen in Fig. 1. The power source is a sinusoidal voltage that varies from positive to negative peak voltage in each cycle. Many different problems either electrical or mechanical in nature can cause vibration at the same or similar frequencies. One must look closely to differentiate between the true sources of vibration.

A power supply produces an electromagnetic attracting force between the stator and rotor which is at a maximum when the magnetizing current flowing in the stator is at a maximum either positive or negative at that instant in time. As a result there will be 2 peak forces during each cycle of the voltage or current wave reducing to zero at the point in time when the current and fundamental flux wave pass through zero as demonstrated in Fig. 2. This will result in a frequency of vibration equal to 2 times the frequency of the power source (twice line frequency vibration). This particular vibration is extremely sensitive to the motor's foot flatness, frame and base stiffness and how consistent the air gap is between the stator and rotor, around the stator bore. It is also influenced by the eccentricity of the rotor.

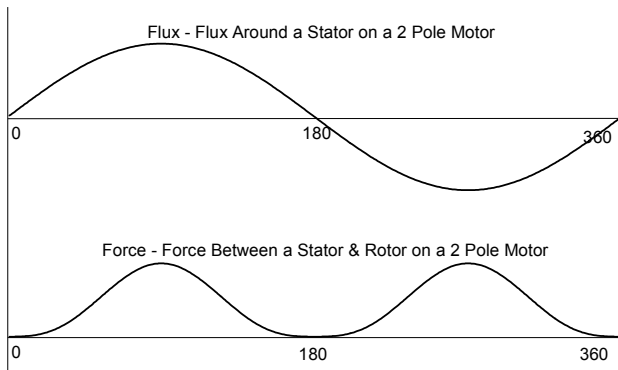


Fig. 2. One Period Flux Wave & Magnetic Force Wave

Some people are inaccurately under the premise that twice line frequency vibration varies with load. This misconception comes from the belief that twice line frequency vibration excitation is due to a magnetic field generated by the current in the stator coil which varies with load and creates a magnetic force which varies with the load current squared. In reality the ampere-turns of the stator and rotor tend to balance one another except for the excitation ampere-turns. To explain this to those not familiar with motor electrical theory, the excitation ampere-turns are created by the motor no load current. This establishes the magnetic field in the motor necessary to generate a back EMF approximately equal to the applied voltage. As load is applied to the motor, both stator and rotor currents increase together and balance one another, therefore, there are no significant changes in flux. This means that the basic magnetic forces are independent of load current and are nearly the same at no load or full load. Therefore the main component of twice line frequency

vibration which is created by an unbalanced magnetic pull due to air gap dissymmetry and does not change with load.

On 2 pole motors, the twice line frequency vibration level will appear to modulate over time due to it's close relationship with 2 times rotational vibration. Problems in a motor such as a rub, loose parts, a bent shaft extension or elliptical bearing journals can cause vibration at 2 times rotational frequency. Due to it's closeness in frequency to twice line frequency vibration the two levels will add when they are in phase and subtract when they are out of phase and then add again when they return to being in phase. This modulation will repeat at a frequency of 2 times the slip on 2 pole motors. Even at no-load, twice rotation vibration on 2 pole motors will vary from 7200 cpm (120Hz) due to slip. Since there is some slip on Induction motors, although small at no load, it may take 5 to 15 minutes to slip one rotation. For those of you not familiar with the term slip, there is a rotating field around the stator that the rotor is trying to stay in phase with, but the rotor will fall behind the stator field a certain number of revolutions per minute depending upon the load. The greater the load the greater the slip. 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, to measure vibration, many times it is necessary to perform what is referred to as a modulation test. In a modulation vibration test the motor is allowed to run for a period of typically 10 or 15 minutes, and vibration is recorded continuously to allow the maximum and minimum to be established.

Elliptical Stator due to Fundamental Flux:

As can be seen in Fig. 3, for 2-pole motors the electromechanical force will attempt to deflect the stator into an elliptical shape. The primary resistance to movement is the strength of the core back iron and the stiffness of the housing around the stator core, which is restraining the core's movement. On 4 pole motors the distance between the nodes is only 45 mechanical degrees, 1/2 that seen on 2 pole motors, thereby making the 4 pole stator core much stiffer to movement resulting in much lower twice line frequency vibration. Calculations on a typical 1000 HP two pole motor at 60 Hz show 120 Hz vibration at the stator core OD of about .12 inches per second, peak, while values for a four pole motor of the same size are only about .02 to .03 inches per second, one sixth to one quarter of this value. This twice line frequency vibration is transmitted through the motor frame to the bearing brackets where it is reduced somewhat in amplitude.

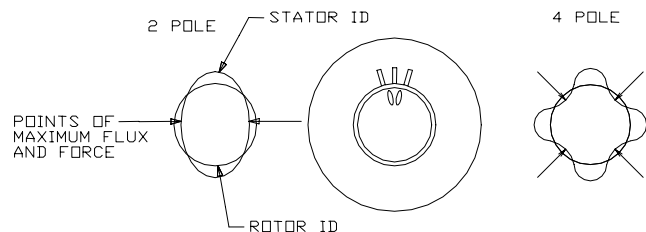


Fig. 3. Electromechanical Force on 2 & 4 Pole Motors

Non Symmetrical Air-gap:

Twice line frequency vibration levels can significantly increase when the air gap is not symmetrical between the stator and rotor as shown in Fig. 4.

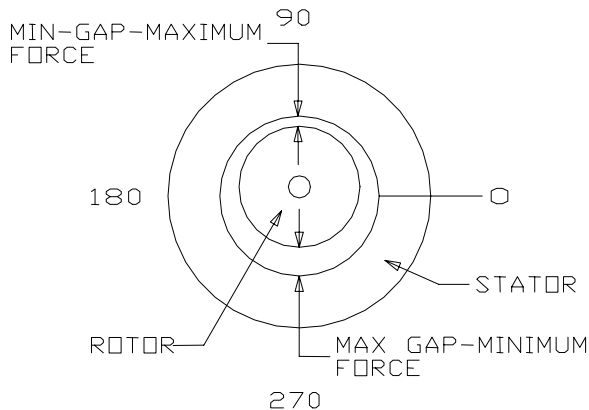


Fig. 4. Unsymmetrical Air Gap Around Rotor

This particular condition will result in the force being greater in the direction of the smaller air gap. That is, an unbalanced magnetic pull will exist in the direction of the minimum air gap.

Force $\approx B^2/d$
 Where B= Flux density
 And d= distance across air gap

Of interest here, not only is the stator pulled in one direction, but also the rotor is pulled in the opposite direction, to the side that has the minimum air gap. This causes higher shaft vibration, which is more detrimental to bearing life. Note that in Fig. 4 the rotor OD is concentric with the axis of rotation thereby causing the force to remain a maximum in the direction of minimum air gap.

One Times Line Frequency Vibration:

Although not nearly as prevalent as twice line frequency vibration, one times line frequency vibration can exist. Unbalanced magnetic pull may result in vibration at line frequency (one times line frequency) as well as the usual twice line frequency vibration. If the rotor or stator moves from side to side, the point of minimum air gap may move from one side of the motor to the other. When the frequency of this motion corresponds to the frequency of the traveling flux wave, the unbalanced magnetic pull will shift from side to side with the point of minimum gap, resulting in vibration at line frequency. This line frequency vibration is normally very small or non-existent, but if the stator or rotor system has a resonance at, or near, line frequency, the vibration may be large.

One Times Rotation Vibration - Electrical

Eccentric Rotor:

An eccentric rotor, which means the rotor core OD is not concentric with the bearing journals, creates a point of minimum air gap which rotates with the rotor at one times rotational frequency. Associated with this there will be a net balanced magnetic force acting at the point of minimum air gap, since the force acting at the minimum gap is greater than the force at the maximum gap, as illustrated in Figure 5. This net unbalance force will rotate at rotational frequency, with the minimum air gap, causing vibration at one time rotational frequency.

The flux causing the magnetic force is the fundamental flux wave, which rotates around the stator at the synchronous speed of the motor. The rotor attempts to keep up with the rotating flux wave of the stator, but the rotor slips behind the stator field as needed to create the necessary torque for the load. When the high point of the rotor (point of minimum air gap) aligns with the high point (maximum) of the stator flux, the force will be a maximum, and then it will decrease, becoming small under a point of minimum flux. Thus, an unbalance force is created which rotates at rotational speed and changes in magnitude with slip. The end result is a one times rotational speed vibration, which modulates in amplitude with slip. This condition occurs at no load or full load. At no load, the frequency approaches synchronous speed and could have a modulation period of 5 to 15 minutes. At full load the frequency of modulation in CPM will equal the slip in rpm times the number of poles. The slip is equal to the synchronous speed minus the full load speed, typically 1% of rated rpm. For example, a 2-pole motor with a full load speed of 3564 rpm at 60 Hz will have a slip of $3600-3564 = 36$ cycles per minute (1% slip) and will result in a modulation frequency of $2 \times 36 = 72$ cycles per minute.

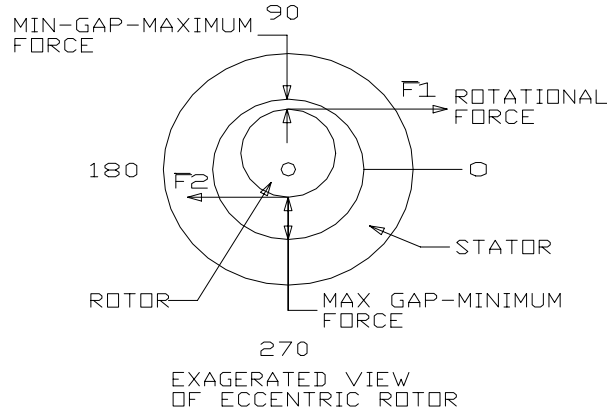


Fig. 5. Eccentric Rotor

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. 6. As a result the field in

the rotor around that particular 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.

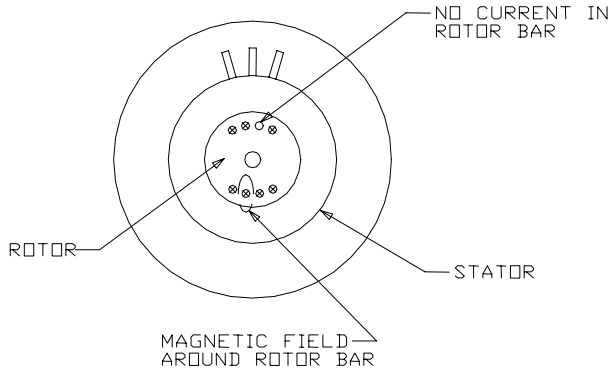


Fig. 6. Rotor with Broken Rotor Bar

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 can not be seen at no-load. But there is an additional phenomenon associated with this condition that 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. These radial and tangential forces which are applied to the stator teeth, as seen in Fig. 7, create vibration of the stator core and teeth.

This source of vibration is at a frequency which is much greater than frequencies normally measured during normal vibration tests. Due to the extremely high frequencies, even very low displacements can cause high velocities if the frequency range under test is opened up to include these frequencies. Though these levels and frequencies can be picked up on the motor frame and bearing housings, significant levels of vibration at these higher frequencies will not be seen between shaft and bearing housing where they could be damaging. For this reason vibration specification

requirements normally do not require that these frequencies be included in overall vibration.

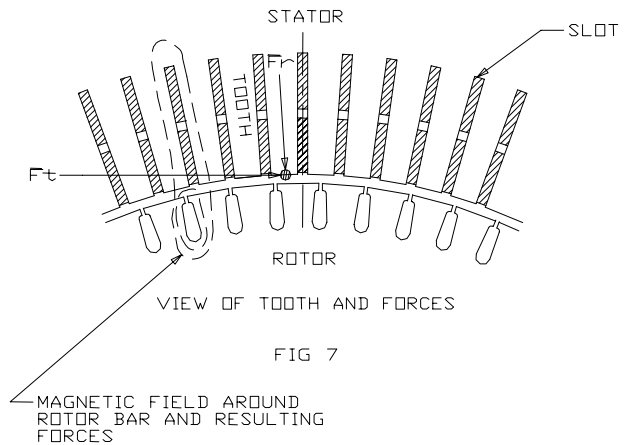


Fig. 7. Magnetic Field around Rotor Bar and Resulting Force on Stator Teeth

Since vibration at rotor bar passing frequency occurs at a high frequency, the vibration velocity level may be significant, but the effect on motor reliability is insignificant. Considering the stress that results in the motor as a consequence of the vibration makes this determination. For example, suppose a two pole motor exhibiting a vibration at 2800 Hz due to rotor bar passing frequency plus a 120 Hz side band:

Velocity, (IPS)	0.1	0.5
Displacement (mils)	0.011	.057
Stress in Stator Core Iron	30 psi	150 psi
Stress in Stator Tooth Iron	50 psi	250 psi

The typical fatigue strength of the core iron is 35,000 psi. Similar low stress levels can be calculated for all parts of the motor (including the stator windings). In addition, the typical minimum oil film thickness ranges from 1.0 mils to 1.5 mils. Since only a small displacement such as .011 to .057 mils as mentioned above could be seen, this vibration will not have an adverse affect on bearing performance.

The rotor slot and side band frequencies are in the frequency range normally related to noise rather than vibration performance, and are taken into account in noise predictions during motor design. In fact, these force components are the principal sources of high frequency noise in electrical machines, which has been for some time subject to noise regulations and limits. Experience has shown that where noise has been within normal or even high ranges, there has been no associated structural damage. The significance of these high frequency vibrations is distorted by taking measurements in velocity and then applying limits based on experience with lower frequency vibration.

Load Related Magnetic Force Frequencies and Mode Shapes

The frequencies of the load related magnetic forces applied to the stator teeth and core equal the passing frequency of the

rotor bars plus side bands at + or - 2f, 4f, 6f and 8f Hz, where f is the line frequency. A magnetic force is generated at the passing frequency of the rotor slot (FQR), which is motor speed in revolution per second times the number of rotor slots as shown in (3).

$$FQR = RPM * Nr / 60, \text{ Hz (3)}$$

where

$$Nr = \text{number of rotor slots}$$

For the typical two pole 3570 rpm motor with 45 rotor slots in the example above, FQR = 2680 Hz.

The side bands are created when the amplitude of this force is modulated at two times the frequency of the power source. On a 60 Hz system the 120 Hz modulation produces the side bands, giving excitation frequencies of FQR, FQR + 120, FQR - 120, FQR + 240, FQR - 240 Hz, etc.

The forces applied to the stator teeth are not evenly distributed to every tooth at any instant in time; they are applied with different magnitudes at different teeth, depending upon the relative rotor- and stator-tooth location. This results in force waves over the stator circumference. The mode shape of these magnetic force waves is a result of the difference between the number of rotor and stator slots as shown in (4).

$$M = (Ns - Nr) +/- KP \quad (4)$$

Where

- N_s = number of stator slots
- N_r = number of rotor slots
- P = number of poles
- K = all integers 0, 1, 2, 3, etc.

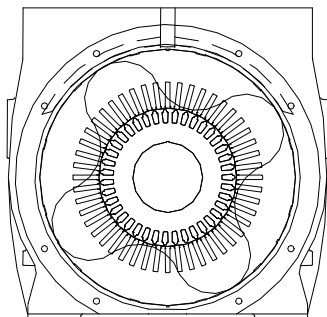
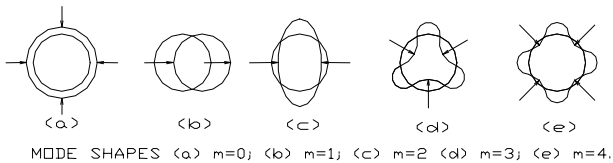


Fig. 8. Mode Shapes

Mode Shapes and Natural Frequencies of Core Vibration:

Under the applied magnetic forces the stator core is set into vibration in the same manner that a ring of steel would respond if struck. Depending upon the modal pattern and frequencies of the exciting force, as described above, the stator would vibrate in one or more of its flexural modes m of vibration, as shown in Figure 8. Each of the mode shapes has its associated natural frequency. The core may be somewhat influenced by the stator frame in actuality, but in analysis the frame is usually neglected, both due to complexity and because the effect on higher frequency modes is minimal.

To understand the resonant frequency of the core at a given mode of vibration, the core can be represented as a beam, which is simply supported on both ends and flexes between the ends due to forces applied on the beam. The length of the beam is equal to the circumferential length of the mean diameter of the stator core for one-half the mode wave length (see Fig. 9) [8].

$$L = \frac{\pi D_s}{2M}$$

If the resonant frequency of the core is close to the forcing frequency, a high level of vibration will result. The lower modes of vibration may produce resonant frequencies that are close to the primary forcing frequencies.

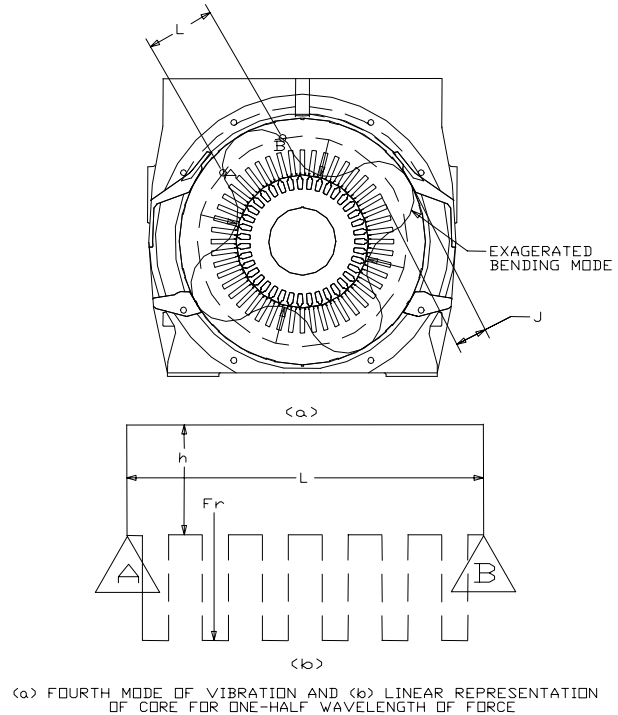
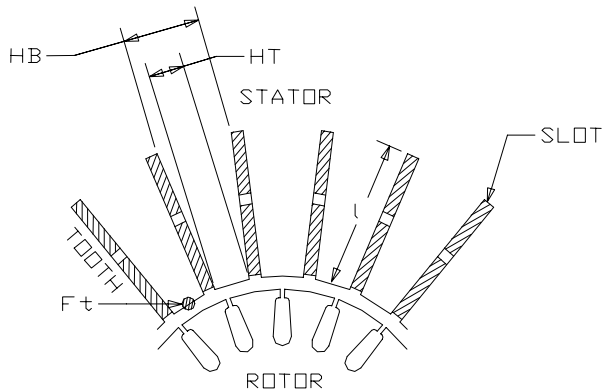


Fig.9. a) Fourth Mode of Vibration
b) Linear Representation of Core for one-half Wavelength of Force



VIEW OF TOOTH AND FORCES

Fig. 10. Stator Tooth Forces

The frequency of stator tooth resonance is also a concern. The tangential forces applied to the teeth can excite a resonant condition in the tooth. The tooth is a cantilever beam supported at the root by the core. The resonant frequency of the cantilever beam is a function of the beam length and width. A longer and narrower beam will produce a lower resonant frequency.

The force applied to each tooth produces displacement of the tooth and the core. The displacement will have a greater amplification the closer the forcing frequency is to the resonant frequency of the core or tooth (5):

$$\text{Amplification Factor} = \frac{1}{1 - (f/f_0)^2} \quad (5)$$

Where f is the line frequency and f_0 is the natural frequency.

This vibration is sometimes incorrectly associated with loose rotor bars, but there are reasons why loose rotor bars won't create rotor slot passing frequency vibration.

First, on most larger motors the centrifugal forces are so great that the only time there could possibly be rotor bar movement is while the rotor is accelerating. This in itself could be a serious problem since the number one cause of rotor bar to end connector failure is rotor bar movement as a result of multiple restarts of a high inertia load. But, the only increase in vibration at speed due to loose rotor bars would be due to a shift in the rotor cage causing a one times rotational mechanical unbalance.

Secondly, looking at any one rotor bar, the bar itself is never subject to a force at the rotor slot passing. The bars are rotating at rotational speed. There is an alternating field in the rotor, which has a frequency close to 0 cycles per minute at, no-load, then increases to a frequency equal to the slip frequency times the number of poles at full load. On a 2 pole motor typically 2 (poles) times 36 Rpm (typical slip) or 72 cycles per minute. To make this easier to understand consider one point or bar on the rotor of a 2 pole motor, and that this

point is rotating at 3564 Rpm. There is a field around this bar at a very low frequency. It is applying a force to the stator at varying magnitudes depending on the level of flux in the rotor at that instant in time. This flux pulsates each time it passes by a stator slot. Note that the force that the rotor sees is at the stator slot passing frequency and is modulating at twice the slip. This will produce vibration of the rotor bars at the stator slot passing frequency plus and minus side band frequencies in multiples of the (slip) x (poles).

One Times Rotation Vibration - Unbalance

Motor Unbalance:

Balancing is required on all types of rotating machinery, including motors, to obtain a smooth running machine. This is performed in the factory in a balance machine at a level of precision determined by the motor speed, size, and vibration requirements. The highest precision is required for two pole motors. Two pole and large four pole motors should be balanced at their operating speed in the balance machine. The assembled motors are then run in test to confirm that vibration requirements are met in operation.

Although they do not usually concern the user directly, a few salient factors affecting factory balance will be discussed here. These mainly apply to two pole motors.

Most medium to large motors are used for constant speed applications, although there has been a recent increase in the number and size used for variable speed applications on adjustable speed drives. Constant speed motors need only be precision balanced at one speed, their operating speed. Variable speed applications require that good rotor balance be maintained throughout the operating speed range, which typically may be from 40% to 100% of synchronous speed.

Rotor balance involves the entire rotor structure which is made up of a multitude of parts, including the shaft, rotor laminations, end heads, rotor bars, end connectors, retaining rings (where required) and fans. These many items must be controlled in design and manufacture to achieve stable precision balance.

Fundamental requirements for precision balance on any machine are:

- Parts must be precision manufactured for close concentricities and minimal unbalance individually.
- Looseness of parts, which can result in shifting during operation, causing a change in balance, must be avoided or minimized.
- Balance correction weights should be added at or near the points of unbalance.

For motors, rotor punchings must be precision manufactured with close concentricities of all features and have a shrink fit on the shaft that is maintained at all operating speeds and temperatures. The punchings 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 of

high strength material designed with proper interference fit. Rotor bars are shimmed and/or swaged so they are tight in the slots. There are other methods to assure tight rotor bars, such as heating the core and chilling the bars, but these methods are not common. End connectors should be induction brazed symmetrically to the bars, which helps eliminate variations in balance due to thermal change. The shaft and assembled rotor are precision machined and ground to concentricities well within .001 inch. The rotor is prebalanced without fans, then the fans are assembled and final balanced on the rotor. The fans are individually balanced before assembly on the rotor. For motors with a heavy external fan, two plane balance of the fan may be required.

Constant speed applications are usually satisfied with either a stiff shaft design, for smaller machines, or a flexible shaft design for larger motors. A "stiff shaft" design is one that operates below its first lateral critical speed, while a "flexible shaft" design operates above the first lateral critical speed [12]. When the rotor is precision designed and manufactured as described above, a two plane balance making weight corrections at the rotor ends, will usually suffice even for flexible rotors. Occasionally, however, a flexible rotor may require a three plane balance to limit vibration as the machine passes through its critical speed during runup or coastdown. This is accomplished by also making weight corrections at the rotor center plane as well as at the two ends.

Adjustable speed applications require a stiff shaft to prevent major balance changes with speed due to shaft deflection, such as may occur with a flexible shaft. In addition, however, the many other factors affecting balance in this complex structure, discussed above, must also be controlled to maintain good balance at varying speeds. In particular, any bar looseness will result in excessive change in balance with speed. This is prevented by rotor bar shimming and sometimes swaging as noted above. Shims around bars, such as used here allow the bars to be driven tightly into the slots without concern for having the laminations shear pieces of the bar off, causing bars to be loose. This design also prevents the bars from becoming loose over time in the field due to a similar phenomenon, which may occur during heating and cooling where the bars may not expand and contract at the same rate as the core.

During balancing and no load testing in the shop, the shaft extension keyway is completely filled with a crowned and contoured half key held in place by a machined sleeve to avoid any unbalance from this source. Load testing is carried out with the motor mounted on a massive, rigid base, accurately aligned to a dynamometer and coupled to the dyne with a precision balanced coupling and proper key.

Thermal Unbalance:

Thermal unbalance is a special form of unbalance. It is caused by uneven rotor heating, or uneven bending due to rotor heating. The proper solution is to determine the reason for uneven heating affecting shaft straightness, and fix the rotor. Before such major rework is performed, the severity of the thermal situation needs to be ascertained. All rotors will have some change in vibration in transitioning from a cold

state to a hot one. API 541, 3rd edition allows 0.6 mils change in shaft vibration (at rotational frequency, 1X), and, 0.05 inches per second change in housing vibration. However, if the application is one of continuous duty, and, vibration levels are not excessive during startup (i.e. motor cold), it is permissible to allow more change cold to hot without any damage to the motor. In these situations if the lowest vibration levels are desired at operating conditions, a hot trim balancing procedure can be performed. To perform this procedure, run the motor until all conditions thermally stabilize, and quickly perform a trim balance. If necessary, run the motor again after the initial trial weights have been installed and let the motor thermally stabilize before taking additional vibration measurements for final weight correction

Coupling Unbalance:

The coupling unbalance limit given in API 671 of 40W/N, when applied to a typical 1000 HP 3600 rpm 2 pole motor for example, gives a value equal to about one-third of the motor unbalance limit for one end.

Analysis shows this would be about the correct value to have minimal effect on motor vibration. Comparing this to AGMA coupling unbalance limits commonly used in the industry, it is comparable to a Class 11 balance which requires a balanced coupling. It is considerably better than a Class 9 balance (by a factor of 3) which is not a balanced coupling. AGMA Class 9 balance couplings are sometimes used for 2 pole motors, but do not meet API 671 and can give vibration problems with precision motors.

Use of a proper key and a balanced coupling leaves the machine alignment and mounting and the driven equipment balance as the remaining major factor in system vibration.

Oversize Coupling:

One consideration in coupling selection is coupling size. The coupling should be large enough to handle the application, including the required service factor, but should not be exceptionally large. Potential results of oversize couplings are:

- Increased motor vibration due to increased coupling unbalance and/or a change in the critical speed or rotor response due to increased weight. This is particularly true for flexible shaft machines.
- A greatly oversize coupling can result in greatly severe shaft bending, excessive vibration, and, heavy rubbing of seals, ultimately resulting in catastrophic shaft failure.

The predominant vibration frequency as a consequence of an oversized coupling would be at one times rotation, just like an unbalance condition. The concept of 'bigger is better' does not hold true here!

Driven Machine Unbalance:

Under normal circumstances, the unbalance of the driven machine should not significantly affect the motor vibration. However, if the unbalance is severe, or if a rigid coupling is

being used, then the unbalance of the driven machine may be transmitted to the motor.

Maintaining Balance in the Field:

When a finely balanced high speed motor is installed in the field, its balance must be maintained when the motor is mated to the remainder of the system. In addition to using a balanced coupling, the proper key must be used.

One way to achieve a proper key is to have the shaft keyway completely filled, with a full key through the hub of the coupling and the entire key outside the coupling crowned to match the shaft diameter. A second approach is to use a rectangular key of just the right length so that the part extending beyond the coupling hub toward the motor just replaced the unbalance of the extended open keyway. This length can be calculated if the coupling hub length and keyway dimensions are known.

An improper key can result in a significant system unbalance, which can cause the vibration to be above acceptable limits. For example, calculations for a typical 1000 HP, 2 pole 3600 rpm motor show that an error in key length of .125 inches will give an unbalance of .7 oz.-in. This is about equal to the residual unbalance limit for each end of the rotor of 4W/N given in API 541 for motors, and exceeds by a factor of 3 the residual unbalance tolerance of a typical one-half coupling of 40W/N given in API 671 for couplings.

A problem occasionally arises in the field when a flexible shaft machine with a high speed balance is sent to a service shop for repair. If the rotor is rebalanced in a slow speed balance machine at the service shop, then this usually results in unbalance at operating speed, and the machine will run rough when tested or reinstalled. The solution, of course, is to not rebalance unless absolutely required by the nature of the repair. If rebalance is absolutely required, than it should be done at the operating speed of the rotor, otherwise, a trim balance may need to be performed after the motor is reassembled.

Forcing Frequency Response Vibration

Weak Motor Base:

If the motor is sitting on a fabricated steel base, such as a slide base, then the possibility exists that the vibration which is measured at the motor is greatly influenced by a base which itself is vibrating. Ideally the base should be stiff enough to meet the "Massive Foundation" criteria defined by API 541 [1]. Essentially, this requires that support vibration near the motor feet to be less than 30% of the vibration measured at the motor bearing. To test for a weak base, measure and plot horizontal vibration at ground level, at bottom, middle, and top of the base, and at the motor bearing. Plotted, this information would look like Fig. 11, for a motor sitting on a weak base. In this particular example, had the motor been on a rigid base, the vibration at the bearing would

have been closer to .25 mils rather than the measured 2.50 mils.

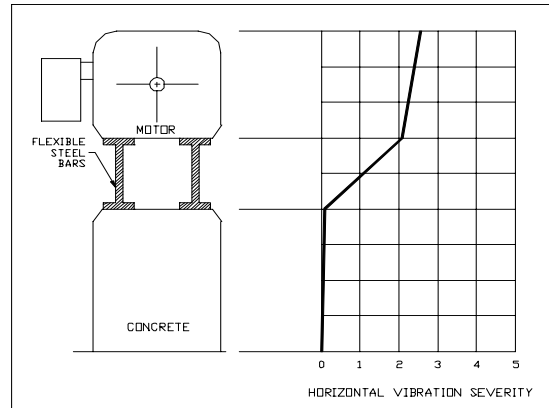


Fig. 11. Plot of Vibration (in mils) Vs. Base/Motor Position

A weak motor base usually results in high 1x vibration, usually in the horizontal direction as shown in Fig. 11. However, it may also result in high 2X (twice rotational frequency) or 2f (twice line frequency) vibration, which also is a common vibration frequency in motors. To determine the nature and source of this high 2x vibration requires vibration measurements be made at the motor feet in both the vertical and horizontal direction, taking phase as well as amplitude to determine a mode shape. The "rocking mode" of the motor observed in a particular case is illustrated in Fig. 12. The horizontal component δ_{HV} due to the rocking adds to the inherent δ_{HM} of the motor alone to give a high total at the bearing housing, as shown by the equivalency below.

$$\delta_H = \delta_{HM} + \delta_{HV}$$

where:

δ_H = Actual motor horizontal vibration measured in the field

δ_{HM} = Horizontal vibration of motor alone measured on a massive base in shop

$\delta_{HV} = \frac{D}{E} V_B$, calculated horizontal vibration component due to δ_{VB} , measured vertical vibration at each motor foot in the field.

The recommended repair for the weak motor base illustrated is that the support posts be tied together and heavily stiffened with the intent to meet the criteria for a "massive foundation." Even where resonance of the base is not a factor, heavy stiffening of a light support structure can greatly reduce vibration.

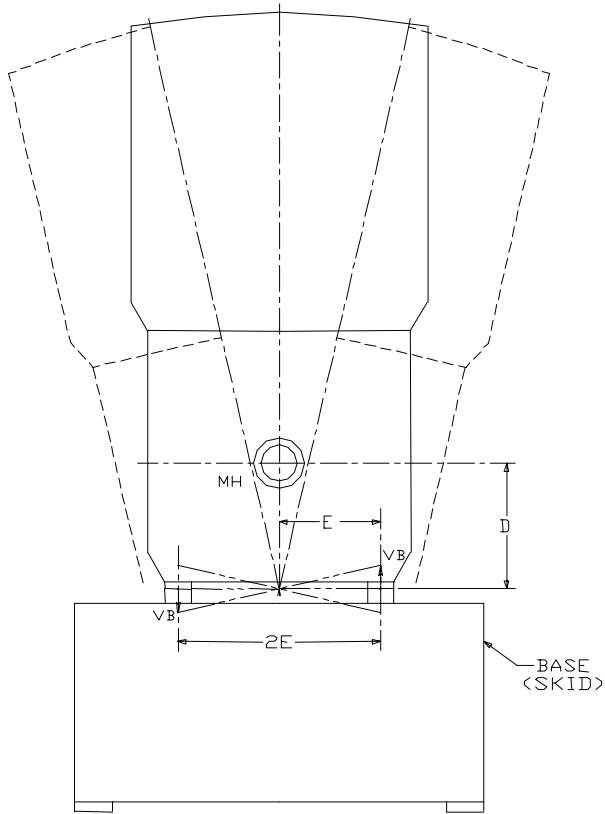


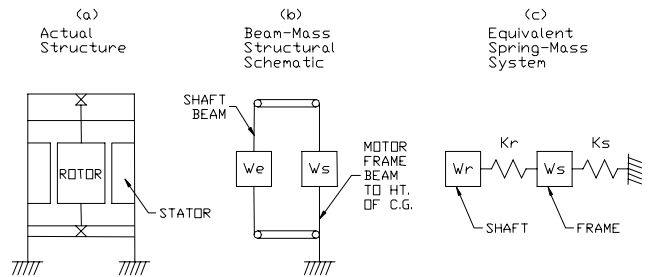
Fig. 12. Rocking Mode due to Weak Base

Reed Critical Base Issues:

A vertical motor's reed critical frequency is a function of its mass, distribution of mass, and base geometry. The reed critical should not be confused with the motor rotor's lateral critical speed. However, in large vertical motors, the rotor lateral critical speed may be a determining factor in the reed critical frequency, particularly of the motor alone. The effect of the rotor may be determined by considering it as a separate mass and including rotor shaft flexibility in the reed frequency calculation. That is, consider the motor as a two mass, two degrees of freedom system as shown in Figure 13, rather than a single degree of freedom system as described in NEMA MG 1-20.55. Figure 13 shows that the motor structure (a) is basically a two mass system which can be progressively simplified, first to a beam-mass structural schematic (b), then to an equivalent two mass, two spring system (c).

Where the lateral critical speed of the rotor is less than the reed frequency calculated as a single degree of freedom system, the true reed frequency will be lower than calculated. It will be approximately equal to the rotor lateral critical speed. However, when mounted on a flexible base in the field, the rotor shaft effect will be less and a single degree of freedom calculation is usually adequate. Just as in the case of a lateral critical, if the motor's operating speed (or any other frequency at which a forcing function is present) coincides with the reed critical, great amplification in the vibration amplitude will occur. Motor manufacturers routinely issue

reed critical data. This includes the reed critical that the motor alone would have if it were mounted on a rigid, seismic mass. In addition the motor manufacturer supplies the following information to aid in determining the system resonant frequency with the motor mounted on the user's base: Machine weight, center of gravity location, and static deflection. Bases found in typical installations are not as stiff, and correspondingly, the reed critical frequency will be lowered. If the reed critical drops into a frequency at which there is a forcing function present (most commonly the operational speed), the reed critical frequency will have to be changed. Usually, this is not difficult to do, and is most commonly accomplished by either changing the stiffness of the base, or by changing the weight of the base/motor. Where the reed critical drops below the operational speed to about 40% to 50% of running speed, this can result in subharmonic vibration at the system resonant speed in motors with sleeve guide bearings. This could be due to either oil whip effects or inadequate guide bearing oil film.



- W_R = rotor weight
- W_S = stator weight (motor weight – W_R)
- K_R = rotor shaft and bearing stiffness (lbs./in.)
- K_S = motor frame stiffness (lbs./in., considering bending, shear deflection, and flange bending).

Fig. 13. Structural Representation of Vertical motor for Reed Critical Frequency Calculation Including Rotor Shaft Flexibility

Resonant Base:

If the motor's operating speed (or any other frequency at which a forcing function is present) coincides with the base resonant frequency, great amplification in the vibration amplitude will occur. The only solution to this problem is to change the resonant frequency of the base. Usually, this is not difficult to do, and is most commonly accomplished by either changing the stiffness of the base, or by changing the weight of the base/motor.

Bearing Related Vibration:

Bearing related vibrations are common to all types of rotating equipment, including motors, and in themselves encompass extensive fields of technology. They will be dealt with briefly here.

Sleeve bearing machines may occasionally experience "Oil Whirl" vibration, which occurs at a frequency of approximately

45% of running speed. This may be quite large, particularly if there is a critical speed at or just below 45% of running speed, which is referred to as an "oil whip" condition. Other than basic bearing design considerations which will not be dealt with here, a common cause is high oil viscosity due to low oil temperature in flood lubricated motors operating in cold ambient conditions. Similar subharmonic vibration, but low in amplitude, may occur in ring lubricated bearings, probably due to marginal lubrication. Other causes of vibration are journal out of roundness or bearing misalignment.

Anti-friction bearings have four identifiable rotational defect frequencies for which formulas for calculation or tabulations of values are given in the literature. These defect frequencies are for the inner race, outer race, ball (or roller) spin, and cage fundamental train. Much research has proven that no absolute answer can be given to allowable amplitudes at bearing defect frequencies. Therefore, the most important thing to look for indicating significant bearing wear is the presence of a number of bearing defect frequency harmonics, particularly if they are surrounded by sidebands independent of amplitude [14]. Tracking of vibration should be carried out starting at installation, observing these indicators to predict remaining bearing life.

III. IDENTIFICATION OF CAUSE OF VIBRATION PROBLEM

Now that the causes of vibration are understood it is time to establish a systematic approach to solve any problem that may arise.

Vibration Data Gathering/Analysis:

Many of the details of rotor dynamics, vibration data gathering, and analysis have not been presented in detail in this paper. For additional information references [1] and [2] can be reviewed.

Now one must keep in mind that all of the electrical sources of vibration and the mechanical sources of vibration are not necessarily at the same phase angle or exactly the same frequency. To make matters worse, the electrical vibration may modulate, and when superimposed on the mechanically induced vibration may result in an overall vibration signature that is unsteady in amplitude and phase. Through proper data collection, testing, and analysis, it is possible to identify the root cause of the vibration.

Vibration Units:

Vibration can be measured in units of displacement (peak to peak, mils), units of velocity (zero to peak, inches per second), or units of acceleration (zero to peak, g's). Acceleration emphasizes high frequencies, displacement emphasizes low frequencies, and velocity gives equal emphasis to all frequencies. This relationship is better illustrated in Fig. 14. In this figure the vibration level is constant at .08 inches per second throughout the entire frequency range, with corresponding vibration levels shown in acceleration (in g's) and displacement (in mils). It is possible to convert from one unit of measurement to another at

discrete frequencies of the vibration. To do so on an overall vibration measurement, complete knowledge of the entire spectral data is required (i.e. amplitude for each frequency band, for all the lines of resolution).

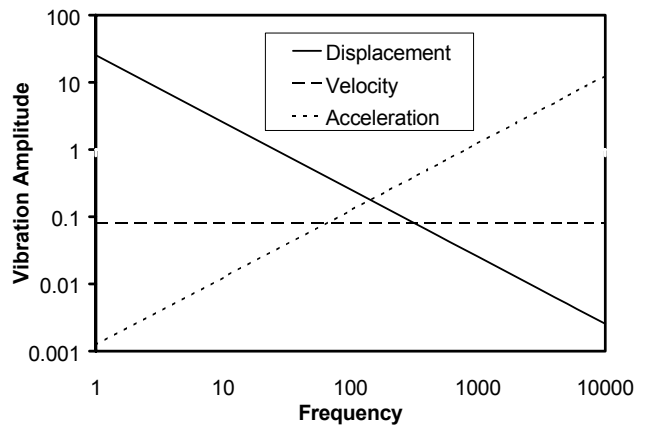


Fig. 14. Comparison of vibration amplitudes Expressed in acceleration, velocity, and displacement

Today, the most common units are displacement for shaft vibration measurement, and velocity for housing vibration measurement. The use of these units is further reflected in most current standards such as API and NEMA.

Direction of Measurement:

Measurements should be made in three planes (vertical, horizontal, and axial) on both bearing housings, as shown in Fig. 15.

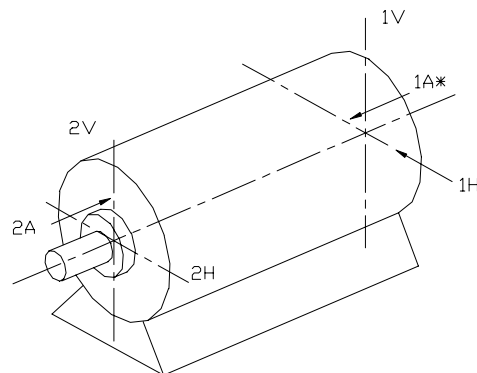


Fig. 15. Vibration Measurement Positions

Shaft Vibration vs. Housing Vibration:

The determination of obtaining shaft vibration data vs. housing vibration data is dependent upon the type of problem being experienced. Oftentimes it is advantageous to have both shaft and housing vibration data. If the problem originates in the rotor (unbalance or oil whirl for instance), then shaft vibration data is preferable. If the problem originates in the housings or motor frame (twice line frequency vibration for instance), then housing vibration data is preferable. Housing

vibration is generally obtained with magnetically mounted accelerometers. Shaft vibration can be obtained one of two ways: shaft stick or proximity probe. There is an important distinction between the two methods of obtaining shaft vibration data: the proximity probe will give vibration information of the shaft relative to the housings, whereas measurements obtained with a shaft stick yield vibration information with an absolute (i.e. inertial) reference. Housing vibration data is always obtained in terms of an absolute reference. If the motor has proximity probes then they should be used. If it does not, then proximity probes may be carefully set up with magnetic mounts. In this case it is important to have the tip of the proximity probe on a ground, uninterrupted surface. Even with this precaution taken, the electrical runout will be higher than in a motor specifically manufactured for use with proximity probes.

Modulation vs. Snapshot:

A snapshot refers to obtaining spectral vibration data at an instant in time. Details of amplitude vs. frequency is readily available in this format. A modulation refers to collecting vibration data for a period of time (typically ten or fifteen minutes), so that the variation in vibration as a function of time can be analyzed. Typically, the following frequencies are tracked when taking a modulation: 1/2X, 1X, 2X, and 1f, 2f, and overall vibration levels (i.e. unfiltered), where X corresponds to rotational frequency and f, line frequency. Additionally, the phase information should be tracked when taking the modulation, especially for the one times rotational frequency. This will make the identification and subsequent correction of various vibration problems possible.

It is sometimes desired to separate twice line frequency and twice rotational frequency vibration. Different methods are required to do this at no load and full load. Under full load the difference in frequency is large enough so that the separate components can each be measured directly with most vibration analyzers. However, at no load, the frequencies are so close together that this can not be done, even using the zoom mode on a high resolution analyzer, so that an indirect method is required. This can be accomplished by measuring the 2 x RPM value at reduced voltage (25%) where the 2 x line component is negligible, and then subtracting this from the peak 2 x component in the modulation test which is the sum of 2 x line and 2 x RPM components. This is usually only possible at a motor manufacturer's facility or at a motor service shop.

Troubleshooting Procedure:

If a vibration problem occurs there are various tests that should be performed. But first, the following maintenance items should be checked.

Maintenance Items

- Check for loose bolts – mounting or other loose parts
- Keep motor clear of dirt or debris
- Check for proper cooling and inlet temperatures or obstructions such as rags, lint or other enclosures
- Check Bearing and Stator Temperatures
- Lubricate as recommended

Check proper oil levels

Check vibration periodically and record

The affected frequencies and other vibration characteristics are listed in Table I.

- Are all bolts tight? Has soft foot been eliminated?
- Is hot alignment good? If it's not possible to verify hot alignment, has cold alignment been verified (with appropriate thermal compensation for cold to hot)?
- Is any part, box top cover, piping vibrating excessively (i.e. are any parts attached to motor in resonance)?
- Is the foundation or frame the motor is mounted to vibrating more than 25% of motor vibration (i.e. is the motor base weak or resonant)?
- Is there any looseness of any parts on motor or shaft?
- Integrity of fans and couplings – have any fan blades eroded/broken off, are any coupling bolts loose/missing, is coupling lubrication satisfactory?

If all of the above items check out satisfactorily, and vibration remains high, a thorough vibration analysis shall be required.

Essentially, there are only two steps in diagnosing a problem:

- Obtain vibration data – not always clear cut because of noise, sidebands, combination of signals, modulation, etc.
- Determine what conditions increase, decrease, or have no effect on vibration through different test conditions to help isolate root cause.

Ideally, vibration measurements should be obtained with the motor operating under the following conditions:

- Loaded, Coupled, Full Voltage, All Conditions Stabilized (i.e. normal operating conditions):
 - First measurement to be obtained.
 - Represents state of machine in actual operation.
 - May indicate which test should be taken next.
- Unloaded, Coupled, Full Voltage:
 - Removes load related vibration, while everything else remains the same.
 - Not always possible to get to zero load, but some reduced load is usually possible.
- Unloaded, Uncoupled, Full Voltage:
 - Removes all effects of coupling and driven machine.
 - Isolates motor/base system.
- Unloaded, Uncoupled, Reduced Voltage (25% if possible):
 - Effect of magnetic pullover forces minimized (most effective use is in comparison to vibration at full voltage,
 - 25% usually only possible at motor service shop or motor manufacturers facility. If motor is a Y-Δ connected motor, then Y connection is effectively 57% voltage as compared to Δ connection at the same terminal voltage. A comparison of vibration under both connections will reveal voltage sensitivity of motor.
- Unloaded, Uncoupled, Coast Down:
 - Will make any resonance/critical speed problem apparent for entire motor/base/driven equipment system.

- Observation of vibration change when the motor power is cut will give information similar to reduced voltage operation as illustrated by Figure 16.

Both frequency domain and time domain data should be acquired. During coastdown a cascade (waterfall) plot will yield frequency domain data (vibration spectral data) vs. speed which can be very helpful. A Bodé plot will yield amplitude and phase vs. speed. It is understood that 25% voltage is not readily available in the field. Reduced voltage vibration measurement is one of the most powerful methods available to separate electrically induced vibration (which manifests itself at twice line frequency) from mechanically induced vibration (manifesting itself at twice rotation speed) in two pole motors. Therefore, the method was included.

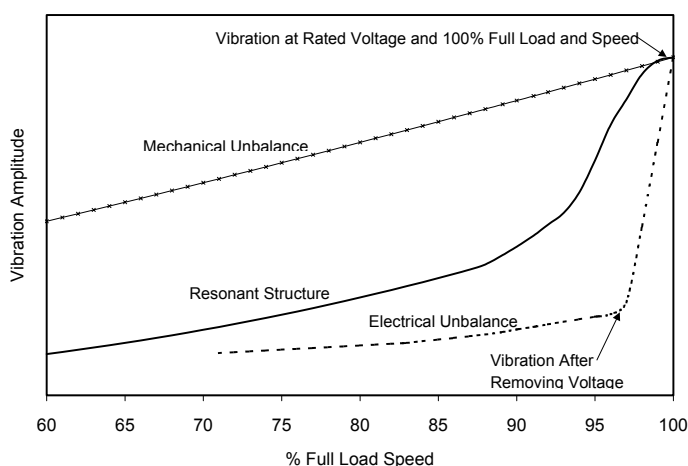


FIG. 16. Vibration After Power is Removed

Vibration Limits:

Many publications of 'vibration limits' exist. Table II lists various industry vibration limits. Both current revisions, as well as older revisions of these standards are listed, as these older revisions are commonly referenced. Furthermore, these motor vibration limits are applicable to a motor mounted on a seismic mass, and either uncoupled, or coupled to a piece of equipment in such a way that any vibration influence from the driven equipment is totally eliminated.

As a motor ages, the vibration levels may slowly increase. There may be a multitude of reasons of why the levels may increase over time:

- degradation of the bearings (sleeve bearings)
- loosening of rotor bars
- accumulation of debris in the oil guards, between rotor and stator, etc.
- changes in mounting conditions: deterioration of grouted base, changes in alignment/soft foot, etc
- Loosening of things mounted to the motor

Obviously, if conditions are identified which increase the motor's vibration level, they should be corrected. If for

whatever reason it is not feasible to rectify the identified condition or identify the offending condition, the level of vibration needs to be compared to what the motor can safely tolerate. The appropriate vibration limits for a particular application are dependent upon several factors such as motor speed, size, design, and lastly, criticality of the process. In the end, allowable motor vibration limits depend greatly upon what the user is willing to tolerate, tempered with knowledge of what the motor can safely tolerate. In the absence of any other information, Table III can serve as a guide for alarm limits. Trip limits can be safely set at 10% above the alarm limits

TABLE III
MOTOR VIBRATION ALARM LIMITS

Speed	3600	1800	1200	900
Housing IPS	0.2	0.2	0.2	0.2
Shaft mils	3.0	3.4	3.9	4.5

The factor limiting the vibration limits at these levels is the motor bearings. Generally, sleeve bearings (as compared to anti friction bearing motors) are more restrictive in terms of vibration limits. Sleeve bearing motors can operated continually at one-half their diametrical bearing clearance, without any damage. They can operate at slightly higher levels for short periods of time as well, but these higher limits must be established with the motor manufacturers.

If the motor is sitting on a weak base, higher housing vibration limits and shaft vibration limits (if measured by shaft stick, and not by a proximity probe) can be tolerated. Effectively, the vibration measured at the motor feet can be subtracted from the vibration measured at the bearing. Refer to Fig. 11, and the section on forcing frequency response vibration for further explanation.

IV. CONCLUSION

Vibration problems can vary from a mere nuisance to an indication of imminent motor failure. With solid knowledge of motor fundamentals and vibration analysis, it is possible to identify the root cause of the problem, and more significantly correct, or ascertain the impact of increased vibration on motor reliability and longevity.

V. REFERENCES

[1] API Standard 541 Third Edition, Form-Wound Squirrel Cage Motors - 250 Horsepower and Larger Washington, D.C., 1995
 [2] NEMA Standards Publication No. MG 1-1993 Rosslyn, VA, 1996

- [3] Sommers, Ernest W., "Vibration in Two Pole Induction Motors Related to Slip Frequency" Transaction, AIEE, (April 1955), pp. 69-72.
- [4] Brozek, B., "120 HERTZ Vibrations in Induction Motors, Their Cause and Prevention," IEEE, Catalog #71C35-IGA, Paper PLI-7, 1-6 (1971).
- [5] Robinson, R.C., "Line Frequency Magnetic Vibration of A-C Machines", Trans. AIEE, Power Apparatus and Systems, Vol. 81, pp 675-679, 1962-1963.
- [6] Alger, P.L., "Induction Machines," Second Edition, Gordon and Breach Science Publishers, New York, 1970.
- [7] Finley, W.R., and Burke, R.R., "Troubleshooting Motor Problems," IAS, 1993, IEEE Transactions of Industry Applications, 1994, vol. 30, no. 5.
- [8] Finley, W.R., "Noise in Induction Motors – Causes and Treatments," IAS, 1991, IEEE Transactions of Industry Applications, 1991, Volume 27, Number 6.
- [9] Eis, R.O., "Electric Motor Vibration – Cause, Prevention, and Cure," IEEE Transactions on Industry Applications, Vol. 1A-11, No. 3, May/June 1975.
- [10] Baumgardner, J., "Vibration in Squirrel-Cage Induction Motors," Proceedings of the 18th Turbomachinery Symposium, College Station, TX, pp 179-183, October 1989.
- [11] Costello, M.J., "Understanding the Vibration Forces in Induction Motors," Proceedings of the 19th Turbomachinery Symposium, Dallas, TX, pp 67-76, September 1990.
- [12] Finley, W.R. and Holter, W.G., "Retrofitting Pipelines with Induction Motors," IEEE Transactions on Industry Applications, Vol. 34, No. 5, September/October 1998.
- [13] Hodowanec, M.M., and Bezesky, D.M., "Field Motor Testing: Procedures which Limit Amount of Risk Involved, IEEE IAS PCIC Conference Records, pp 79-90.
- [14] Berry, J.E., "How to Track Rolling Element Bearing Health with Vibration Signature Tracking," Sound and Vibration, pp 24-35, November, 1991.
- [15] Taylor, A.G., "Computer-assisted Diagnosis of Instability Problems: Four Case Histories," Orbit Publication, Bentley Nevada, Vol. 8, No. 1, pp 5-13. February 1987.

William R. Finley received his BS Degree in Electrical Engineering from the University of Cincinnati, Cincinnati, OH.

Presently, as Manager of Engineering for Siemens Energy & Automation, Bill is responsible for Above NEMA Induction Motor designs. Over the many years in the business he has worked in various Engineering design and management positions, including Electrical

and Mechanical Design, Product Development, Quotation and Computer Systems. He is a Senior member of IEEE and has previously published 10 technical articles. He is currently active in over 10 NEMA and IEC working groups and Sub-committees. He is Chairman of the Large Machine Group and International Standardization Group of NEMA.

Mark M. Hodowanec received a B.S. and M.S. degree in mechanical engineering from the University of Akron, Akron, OH. Currently, he is the Manager of Mechanical Engineering for ANEMA induction motors built in the U.S. at Siemens Energy and Automation, Inc., Cincinnati, OH. For the past nine years he has worked in a variety of engineering positions including design, product development, order processing, shop testing, and field support. In addition to his ANEMA motor experience, Mr. Hodowanec has worked on a wide assortment of induction motors such as NEMA, submersible, and MSHA motors. He is the author of several published technical articles.

Warren G. Holter received B.S. degrees in mechanical engineering and electrical engineering from the University of Minnesota, Minneapolis, MN.

Prior to his retirement in 1995, he had been a Senior Product Engineer with Siemens Energy & Automation, Inc., Cincinnati, OH, where he handled the mechanical engineering for the design, quotation, order processing, shop testing and field follow up for induction motors ranging from 200 to 10000 horsepower. He is currently continuing this work as a Consultant for Siemens. His prior experience included positions as Chief Mechanical Engineer and Manager of Engineering for large induction, synchronous, and D.C. motors and generators. He is the holder of three patents, including a patent for vibration reduction in electric motors.

TABLE I
ELECTRIC MOTOR DIAGNOSTIC CHART

CAUSE	FREQUENCY OF VIBRATION	PHASE ANGLE	AMPLITUDE RESPONSE	POWER CUT	COMMENTS
Misalignment: ① Bearing	Primarily 2 x Some 1 x Radial High at DE and Axial	Phase angle can be erratic.	Steady.	Drops slowly with speed.	① 2 x can dominate during coast-down. ② 2 x is more prevalent with higher misalignment.
Misalignment: ② Coupling	Primarily 1 x Some 2 x Radial High at DE and Axial	Drive 180° out Phase with NDE.	Steady.	Level drops slowly with speed.	① Parallel causes radial forces and angular causes axial. ② Load dependent.
Rub - ① Seal/or bearing	1/4x, 1/3x, 1/2x or 10-20x can be seen Primarily 2 x Some 1 x. Radial.	Erratic.	Erratic depending upon severity.	Disappears suddenly at some lower speed.	① Full rubs tend to be 10 to 20x higher. ② Bearing misalignment can give rub symptoms.
② Rotor	1/4x, 1/3x, 1/2x, & 1x with slip freq. side bands. Radial.	Erratic.	High.		① Severe pounding.
Looseness: ① Bearing (non- rotating)	2 x 3 x may be seen Radial	Steady.	Fluctuates	Disappear at Some lower speed	① Bearing seat looseness. ② Looseness at bearing split.
② Rotor Core (rotating)	1-10x with 1, 2, & 3 predominant. Radial	① Can exist relative to type of looseness ② General core loose gives erratic symptom.	Erratic, high amplitude	① Drops with speed. ② Can disappear suddenly.	① End plates loose. ② Core ID loose.
③ Pedestals (non-rotating)	1-10x with 2 & 3 predominant Radial & Axial	Steady.	Fluctuates.	Disappears at some lower speed.	
④ External Fans	1 & 3 x Radial & Axial – OE (fan end)	N/A	Fluctuates.	① Drops with speed. ② Can disappear suddenly.	
Unbalance Rotor	1x rotor speed. Radial	① NDE & DE in phase. ② Couple gives out of phase condition	Steady.	Level drops slowly.	Rotor has unbalance - can be due to thermal problems.
Unbalance of External Fan	① 1X Radial high at NDE (fan end). ② 1X Axial with high at fan end.	① Couple DE 180° out of phase with EO.	Steady.	Level drops slowly.	
Coupling Unbalance	1 x Radial & higher on drive end		Steady	Level drops slowly	Unbalance due to coupling or key
Bent Shaft Extension	2 x Primarily 1 x may be seen Axial	EO 180° out of phase with DE.	Steady.	Level drops slowly.	DE runout should give higher 2x axial at that end. Normal runout on core - 1-2 mil.
Eccentric Air Gap	Strong 120 Hz Radial	N/A	Steady	Immediately drops	Difference between max. and min. air gap divided by ave. should be less than 10%.
Soft Foot Eccentric rotor.	1x Primarily Some 60 & 120 Hz Radial	Unsteady.	Modulates in amplitude with slip	Immediately drops	① Eccentricity limit 1-2 mil. ② Slip beat changes with speed/load.
Loose stator core.	120 Hz. Axial & radial	Frame & bearing brackets in phase at 120 Hz Unsteady	Steady	Immediately drops	① Look for relative motion of core with respect to housing.
Rotor Bow (Thermal Bow)	1x Primary Some 120 Hz may be seen May have Modulators on 1X & 2X vib. - Radial		① Changes with temperature. ② Time or load related. ③ Varies at Freq. slip x poles	Some drop but high level would come down with speed.	① Heat related. ② Examine rotor stack for uneven stack tightness or looseness. ③ Shorted Rotor Iron ④ Check bar looseness.

TABLE I
ELECTRIC MOTOR DIAGNOSTIC CHART

CAUSE	FREQUENCY	PHASE ANGLE	AMPLITUDE OF VIBRATION	POWER CUT	COMMENTS
Broken rotor bars	1x and modulates at slip x # poles May have high stator slot frequencies On slower speed Motors	Dependent upon where broken bars are located.	STRONG BEAT POSSIBLE. - Varies @ Freq. Slip x poles - Amplitude increased with load	Immediately drops	① Sparking in the air gap may be seen. ② Long term variation in stator slot frequencies can be indicator of bar problems. ③ Broken bars cause holes in magnetic field. ④ Large current fluctuations. ⑤ Current analysis shows slip frequency side bands.
Loose bars.	① 1 x Possible balance effect with thermal sensitivity. Radial ② Stator slot freq. plus sidebands @ ±(# Poles*Slip)	1. 1 x vibration will be steady 2. Stator slot freq. will modulate causing a fluctuation in phase angle on overall vibration	Steady	① Stator slot freq. will immediately disappear. ② Imbalance effect can suddenly disappear at some lower speed.	Excessive looseness can cause balance problems in high speed motors.
Interphase fault	60 & 120 Hz Radial	N/A	Steady and possible beat.	Immediately disappears.	
Ground fault	60 Hz & 120 Hz slot freq. - Radial	N/A	Steady and possible beat.	Immediately disappears.	
Unbalanced Line Voltages	120 Hz Radial	N/A	Steady 120 Hz & Possible beat.	Immediately disappears.	
Electrical Noise Vibration	(RPM x # of Rotor slots)/60 +/-120, 240, etc. - Radial	Due to modulation overall vibration will fluctuate	Steady	Immediately disappears	Increases with increasing load.
System Resonance	1 x RPM or other forcing frequency One plane – usually Horizontal	Varies with load and Speed	Varies	Disappears rapidly.	Foundation may need stiffening- may involve other factors
Strain	1 x RPM		Steady		Caused by casing or foundation distortion from attached structure (piping).
Poorly shaped Journal	2x Rotational Usual	Erratic	Steady	May disappear at lower speed	May act like a rub.
Oil Film Instability (Oil Whirl)	Approx. (.43-.48)*rotational	Unstable	Steady		
Anti-Friction Bearing Problems	Various Frequencies dependent on bearing design	Unstable	Steady		Four basic frequencies.
Resonant Parts	At forcing Frequency or Multiples	N/A	Steady♣	Drops rapidly	May be adjacent parts
Top Cover Fit	120 Hz. Radial	N/A	Steady.	Disappears immediately.	① Magnification of 120 Hz electrical. ② Top cover rests on basic core support.

TABLE II

Industry Housing Vibration Limits

	NEMA - 1993	NEMA - Old			API 541 3rd Ed.	API 541 2nd Ed.			IEEE 841
	2, 4, 6 Pole	2 Pole	4 Pole	6 Pole	2, 4, 6 Pole	2 Pole	4 Pole	6 Pole	2, 4, 6 Pole
Unfiltered (Overall)	.12 IPS	1 Mil	2 Mils	2.5 Mils	.1 IPS	.8 Mils	1.5 Mils	1.5 Mils	.08 IPS
Filtered - 1X	.12 IPS				.1 IPS	.5 Mils	1 Mil	1 Mil	
Filtered - 2X	.12 IPS				.1 IPS				.05 IPS
Filtered - 2f					.1 IPS				.05 IPS

Industry Shaft Vibration Limits

	NEMA - 1993			API 541 3rd Ed.	API 541 2nd Ed.		
	2 Pole	4 Pole	6 Pole	2, 4, 6 Pole	2 Pole	4 Pole	6 Pole
Unfiltered (Overall)	1.0 Mils	2.0 Mils	2.5 Mils	1.5 Mils	2.0 Mils	2.5 Mils	3.0 Mils
Filtered - 1X				1.2 Mils	1.5 Mils	2.0 Mils	2.5 Mils
Filtered - 2X				0.5 Mils	1.0 Mils	1.5 Mils	1.7 Mils
Filtered - 2f				0.5 Mils			