An Analytical Approach to Solving Motor Vibration Problems

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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 downtime 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, the following: 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—Cause of vibration, induction motors, troubleshooting vibrations.

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.
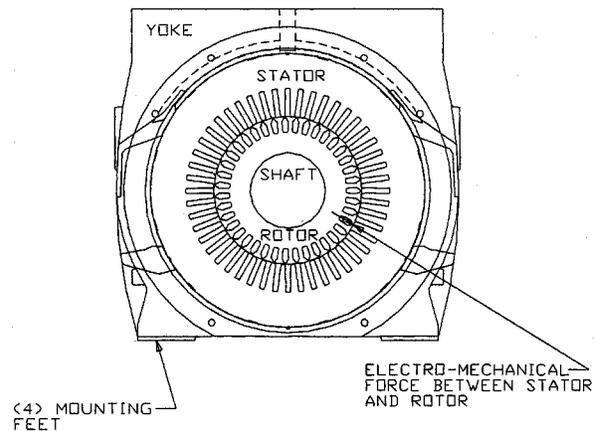
Fig. 1. Stator and rotor.

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, from where does the force come? 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.

1) 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

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	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	Unsteady	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.

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 at a point on the stator 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 two 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× 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.

TABLE I (*Continued*)
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.

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 does not change with load.

On two-pole motors, the twice-line-frequency vibration level will appear to modulate over time due to its close relationship
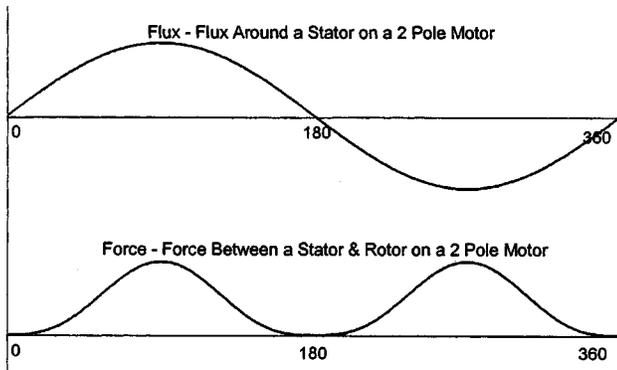
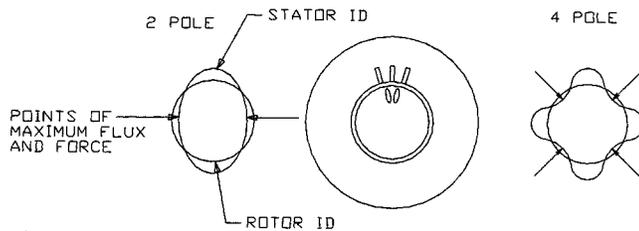Fig. 2.   One period flux wave and magnetic force wave.

Fig. 3.   Electromechanical force on two- and four-pole motors.

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 its 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 two-pole motors. Even at no load, twice rotation vibration on two-pole motors will vary from 7200 cycles/min (120 Hz) due to slip. Since there is some slip on induction motors, although small at no load, it may take 5–15 min 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 min, and vibration is recorded continuously to allow the maximum and minimum to be established.

2) Elliptical Stator Due to Fundamental Flux: As can be seen in Fig. 3, for two-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 four-pole motors the distance between the nodes is only 45 mechanical degrees, one-half that seen on two-pole motors, thereby making the four-pole stator core much stiffer to movement, resulting in much lower twice-line-frequency vibration. Calculations on a
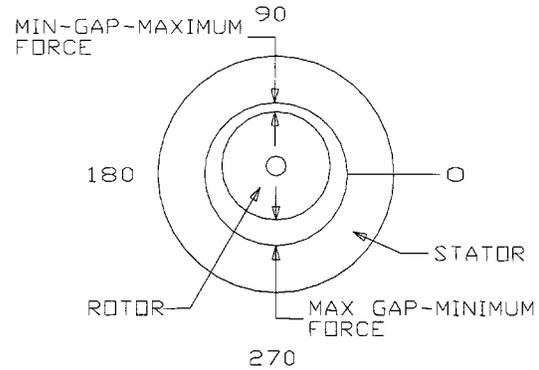

Fig. 4.   Unsymmetrical air gap around rotor.

typical 1000-hp two-pole motor at 60 Hz show 120 Hz vibration at the stator core OD of about .12 in/ s, peak, while values for a four-pole motor of the same size are only about .02–.03 in/s, 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.

3) Nonsymmetrical 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.

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

$$\text{force} \approx B^2/d \tag{1}$$

where B is the flux density, and d is the distance across the 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.

4) 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 nonexistent, but if the stator or rotor system has a resonance at, or near, line frequency, the vibration may be large.

A. One-Times-Rotation Vibration—Electrical

1) 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 unbalanced magnetic force acting at the point of
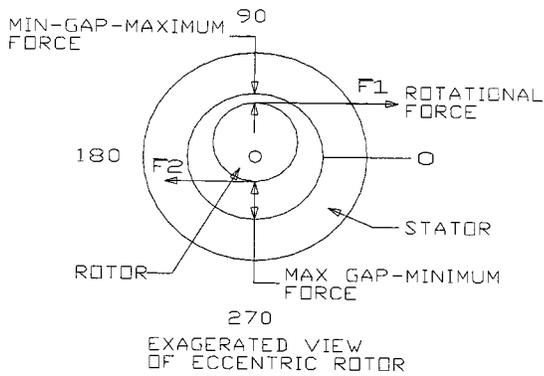
Fig. 5.   Eccentric rotor.

Fig. 6.   Rotor with broken rotor bar.

minimum air gap, since the force acting at the minimum gap is greater than the force at the maximum gap, as illustrated in Fig. 5. This net unbalance force will rotate at rotational frequency, with the minimum air gap, causing vibration at one times 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 5–15 min. At full load, the frequency of modulation in cycles per minute will equal the slip in revolutions per minute times the number of poles. The slip is equal to the synchronous speed minus the full-load speed, typically 1% of rated revolutions per minute. For example, a two-pole motor with a full-load speed of 3564 r/min at 60 Hz will have a slip of 3600-3564 = 36 cycles/min (1% slip) and will result in a modulation frequency of $2*36 = 72$ cycles/min.

2) 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.

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. However, 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 accel-
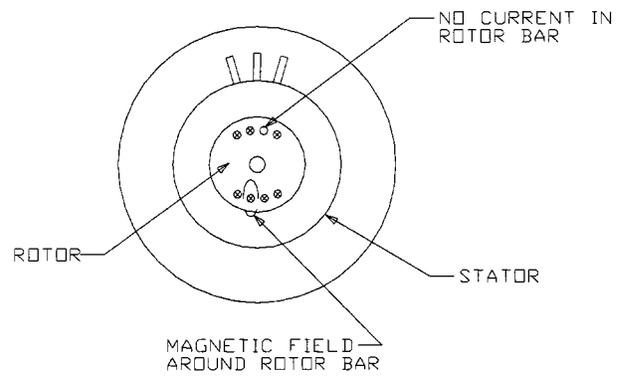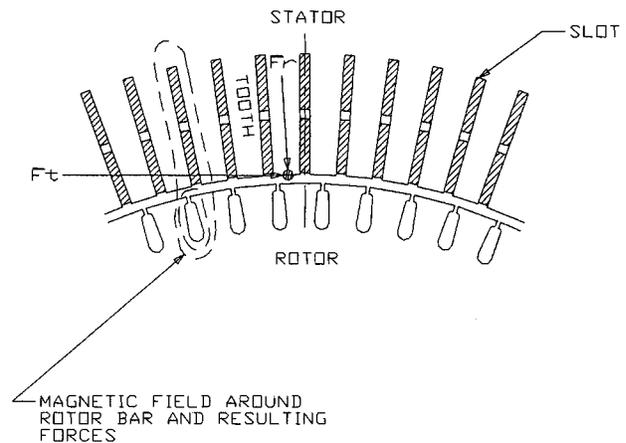

Fig. 7.   Magnetic field around rotor bar and resulting force on stator teeth.

erations 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.

3) 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. Although 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.

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, a two-pole motor exhibiting a vibration at 2800 Hz may give the following:

velocity, (IPS)	0.1	0.5
displacement (mils)	0.011	.057
stress in stator core iron	30 lb/in^2	150 lb/in^2
stress in stator tooth iron	50 lb/in^2	250 lb/in^2.

The typical fatigue strength of the core iron is 35 000 lb/in^2. 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 to 1.5 mils. Since only a small displacement such as .011–.057 mils as mentioned above could be seen, this vibration will not have an adverse affect on bearing performance.

The rotor slot and sideband 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.

B. 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 sidebands 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

$$\text{FQR} = \text{r/min} * \text{Nr}/60 \text{ Hz} \qquad (2)$$

where Nr is the number of rotor slots.

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

The sidebands are created when the amplitude of this force is modulated at 2× the frequency of the power source. On a 60-Hz system, the 120-Hz modulation produces the sidebands, 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
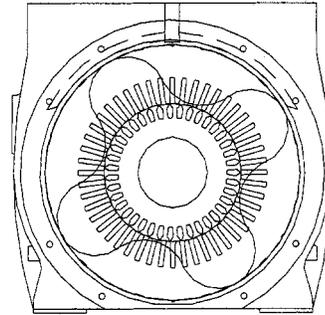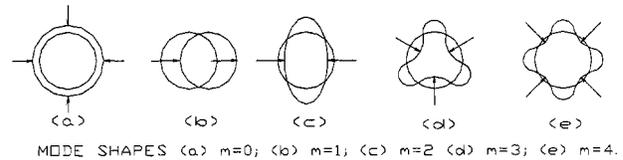

MODE SHAPES (a) m=0; (b) m=1; (c) m=2 (d) m=3; (e) m=4.

Fig. 8.   Mode shapes.

force waves is a result of the difference between the number of rotor and stator slots as shown in

$$M = (N_s - N_r) +/- KP \qquad (3)$$

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.

1) 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 Fig. 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].

$$L = \frac{\Pi D_s}{2M} \qquad (4)$$

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.

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
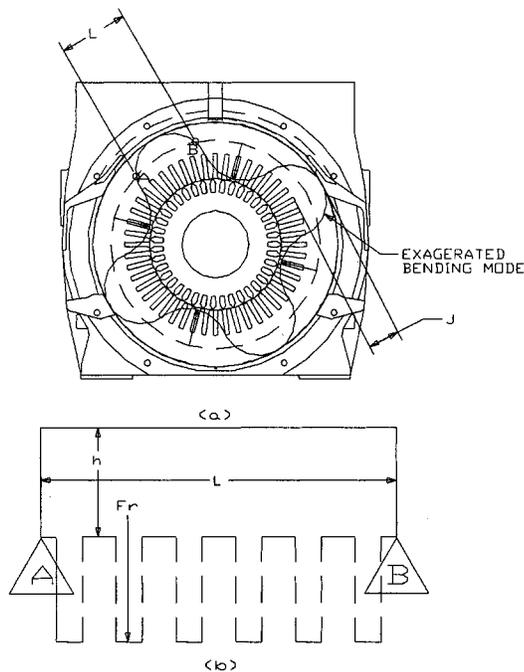
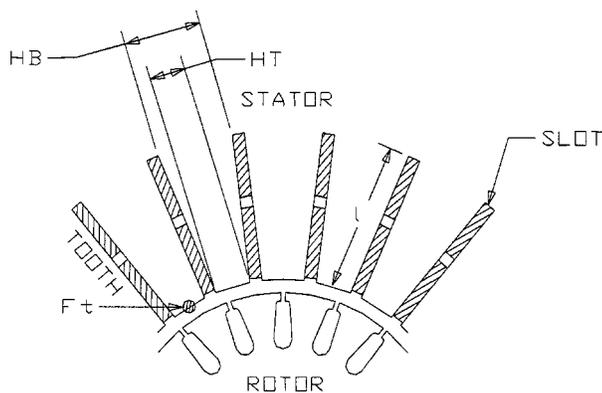Fig. 9. (a) Fourth mode of vibration. (b) Linear representation of core for one-half wavelength of force.

Fig. 10. Stator tooth forces.

beam is a function of the beam length and width. A longer and narrower beam will produce a lower resonant frequency (see Fig. 10).

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 (3)

$$\text{amplification factor} = \frac{1}{1 - (f/f_0)^2} \qquad (5)$$

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 will not 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. However, 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/min at no load, then increases to a frequency equal to the slip frequency times the number of pole pairs at full load. On a two-pole motor this is typically 1 (pole pair) $\times$ 36 r/min (typical slip) or 36 cycles/min. To make this easier to understand, consider one point or bar on the rotor of a two-pole motor, and that this point is rotating at 3564 r/min. 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 sideband frequencies in multiples of the (slip) $\times$ (poles).

C. One-Times-Rotation Vibration—Unbalance

1) 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 as follows.

- 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 in. 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.

2) 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, $1\times$), and, 0.05 in/s change in housing vibration. However, if the application is one of continuous duty, and, vibration levels are not excessive during startup (i.e., motor is 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

3) Coupling Unbalance: The coupling unbalance limit given in API 671 of 40W/N, when applied to a typical 1000-hp 3600-r/min two-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 two-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.

4) 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 the following.

• One is an 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.

5) 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.

6) 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 two-pole 3600-r/min motor show that an error in key length of .125 in will give an unbalance of 0.7 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, then 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.

D. Forcing-Frequency-Response Vibration

1) 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 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.

A weak motor base usually results in high $1\times$ vibration, usually in the horizontal direction as shown in Fig. 11. However, it may also result in high $2\times$ (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 $2\times$ 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
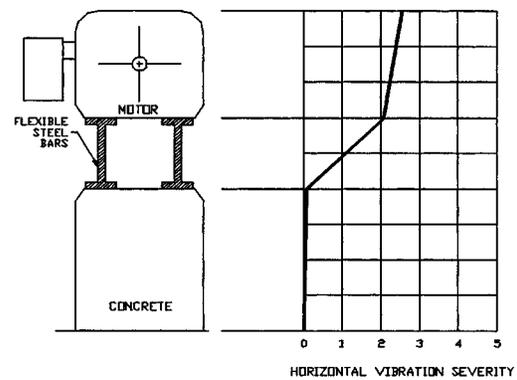

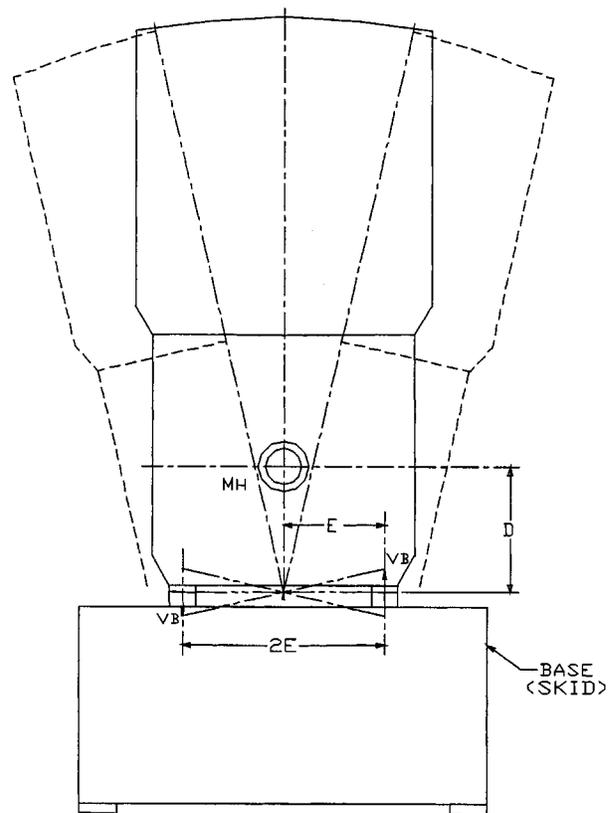Fig. 11.   Plot of vibration (in mils) versus base/motor position.

Fig. 12.   Rocking mode due to weak base.

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 following equivalency:

$$\delta_H = \delta_{HM} + \delta_{HV} \tag{6}$$

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;

δ_{HV} $= (D/E)VB$ 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."
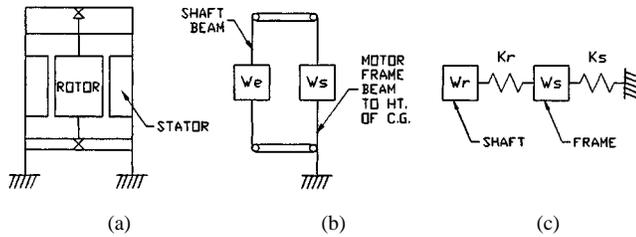
Fig. 13.   Structural representation of vertical motor for reed critical frequency calculation including rotor shaft flexibility (W_R: rotor weight; W_S: stator weight (motor weight-WR); K_R: motor shaft and bearing stiffness (lb/in); K_S: motor frame stiffness (lb/in, considering bending, shear deflection, and flange bending).

Even where resonance of the base is not a factor, heavy stiffening of a light support structure can greatly reduce vibration.

2) 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 Fig. 13, rather than a single-degree-of-freedom system as described in NEMA MG 1–20.55. Fig. 13 shows that the motor structure [Fig. 13(a)] is basically a two-mass system which can be progressively simplified, first to a beam-mass structural schematic [Fig. 13(b)], then to an equivalent two-mass two-spring system [Fig. 13(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%–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.

3) 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.

4) 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.

Antifriction 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.

1) 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, [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.
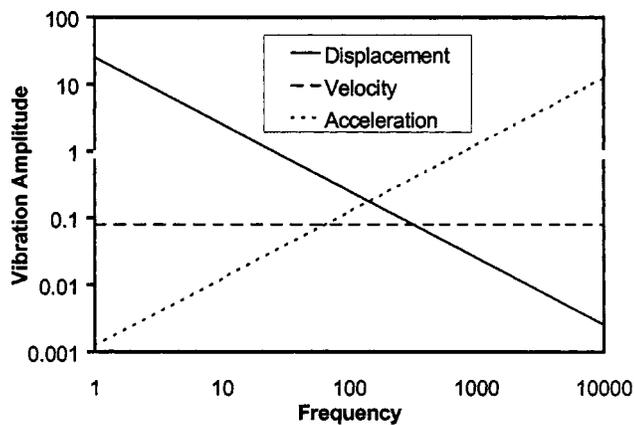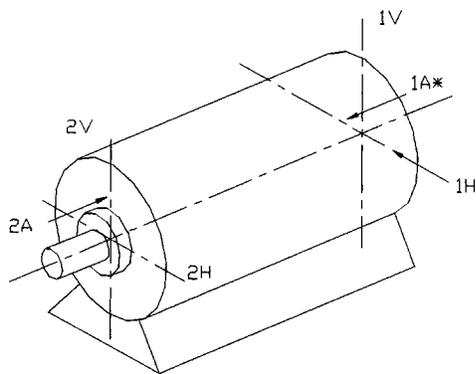
Fig. 14.   Comparison of vibration amplitudes.

Fig. 15.   Vibration measurement positions.

2) 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 in/s 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).

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.

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

4) Shaft Vibration versus Housing Vibration: The determination of obtaining shaft vibration data versus 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.

5) Modulation versus Snapshot: A snapshot refers to obtaining spectral vibration data at an instant in time. Details of amplitude versus frequency are readily available in this format. A modulation refers to collecting vibration data for a period of time (typically, 10 or 15 min), so that the variation in vibration as a function of time can be analyzed. Typically, the following frequencies are tracked when taking a modulation: $1/2\times$, $1\times$, $2\times$, and $1f$, $2f$, and overall vibration levels (i.e., unfiltered), where $\times$ 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 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 cannot 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 $2 \times$ revolutions per minute value at reduced voltage (25%) where the $2 \times$ line component is negligible, and then subtracting this from the peak $2 \times$ component in the modulation test which is the sum of $2 \times$ line and $2 \times$ revolutions per minute components. This is usually only possible at a motor manufacturer's facility or at a motor service shop.

6) 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 is 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, or 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?
- As for the integrity of fans and couplings, have any fan blades eroded/broken off, are any coupling bolts loose/missing, and 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.

1) Obtain vibration data. This not always clear cut because of noise, sidebands, combination of signals, modulation, etc.
2) 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, coastdown:
  - 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 Fig. 16.

Both frequency-domain and time-domain data should be acquired. During coastdown a cascade (waterfall) plot will yield
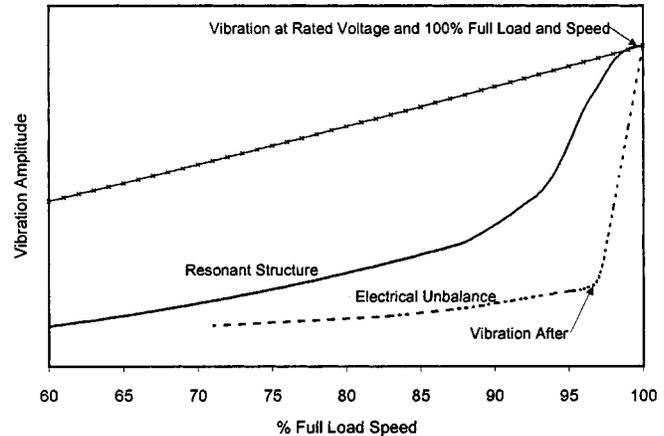

Fig. 16. Vibration after power is removed.

frequency-domain data (vibration spectral data) vs. speed which can be very helpful. A Bodé plot will yield amplitude and phase versus 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.

7) 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, including the following:

- degradation of the bearings (sleeve bearings);
- loosening of rotor bars;
- accumulation of debree 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 II
INDUSTRY HOUSING VIBRATION LIMITS AND INDUSTRY SHAFT VIBRATION LIMITS

Industry Housing Vibration Limits

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

Industry Shaft Vibration Limits

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

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 antifriction bearing motors) are more restrictive in terms of vibration limits. Sleeve bearing motors can be 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 Section II-D 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.

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