UNDERSTANDING THE VIBRATION FORCES IN INDUCTION MOTORS

by

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ABSTRACT

Squirrel cage induction motors have been used extensively in industry for over 50 years. While it appears that vibration problems are more pronounced nowadays, certain basic construction features have always existed and have created considerable difficulty from the initial stages of motor development. As induction motor theory has never changed, the electromechanical forcing functions have always existed and created vibration problems. In fact, some of the most complete and best references on vibration in induction motors were written 30 to 40 years ago. It appears, however, that only recently has the induction motor been critically reviewed by mechanical engineers and rotating machinery specialists. Motors are now being treated for what they are—extremely complex rotating machines having not only the associated mechanical forces, but electromagnetic and electromechanical forces as well. The basic operating principles of motors are discussed as well as the lateral vibration forcing functions encountered when troubleshooting motor vibration problems. All motors described herein are squirrel cage, polyphase, 60Hz design.

INTRODUCTION

The present trend in industry is towards long term reliability on all major equipment. In order to accomplish this, more and more motors are being outfitted for vibration and temperature monitoring systems. While proximity probes have been in service to measure vibration for over 20 years on turbines and compressors, most motor manufacturers have not used them until the last five to seven years. It was only five years ago when a manufacturer stated he knew a motor's mechanical performance was acceptable when he could stand a nickel on end on the bearing housing.

There is no doubt that the induction motor has evolved considerably over the past 20 years; however, this evolution was dictated primarily from an electrical standpoint. Insulation materials were developed which allowed manufacturers to build larger horsepower machines, and run them at progressively higher and higher temperatures. As an aftermath of government legal actions in the 1950s, the "White Sale" eliminated price fixing between the manufacturers. This brought competition and effectively lowered motor prices drastically. During the 1960s and 1970s, material improvements and new manufacturing procedures resulted in significantly more efficient machines. Motor base prices continued to drop and even now are lower than they were 15 years ago. During this period, the mechanical aspect of the motor became altered significantly. Motor frames were reduced in physical size, weight, and structural strength for a given horsepower. However, they were still to contain the same forces as their larger and more robust predecessors.

As a result of recent problems, the need for equipment reliability, more knowledge in rotordynamics and more stringent user specifications, motor manufacturers are presently being forced to evaluate their product's mechanical performance.

INDUCTION MOTOR OPERATING PRINCIPLES

To understand the operation of an induction motor, it is important to become familiar with its major components. A cutaway of a typical large motor is shown in Figure 1. A squirrel cage induction motor consists of the following major components:

- *Stator*—The stator consists of an electrical winding and a cylindrical laminated steel core in which the winding coils are inserted. After insertion, the coils are connected in a manner to produce alternate pole polarity, the number of which dictates the speed of the motor.
- *Rotor*—The rotor is made up of a shaft, and a cylindrical laminated steel core in which the rotor winding is inserted. In a squirrel cage design, the rotor winding consists of nonmagnetic bars which are inserted through slots in the core. The bar ends connect to end rings which short circuit the bars. The bars and end rings together make up the rotor "squirrel cage."
- *Frame*—The frame of the machine is either a fabricated or cast structure in which the stator is inserted. This frame must be strong enough to withstand mechanical and electromechanical forces along with providing air passages employed to cool the motor.
- Enclosure—Various enclosures can be specified such as DP (drip proof), WPI (weather protected I), WPII (weather protected II), TEWAC (totally enclosed, water-air-cooled), etc. These enclosures are either integral with or are installed on top, bottom, or sides of the frame.

The basic theory of the induction motor is actually very simple. As an alternating polyphase voltage is applied to the ends of the stator windings, currents flowing in coil groups produce a multipole alternating magnetic field which rotates around the

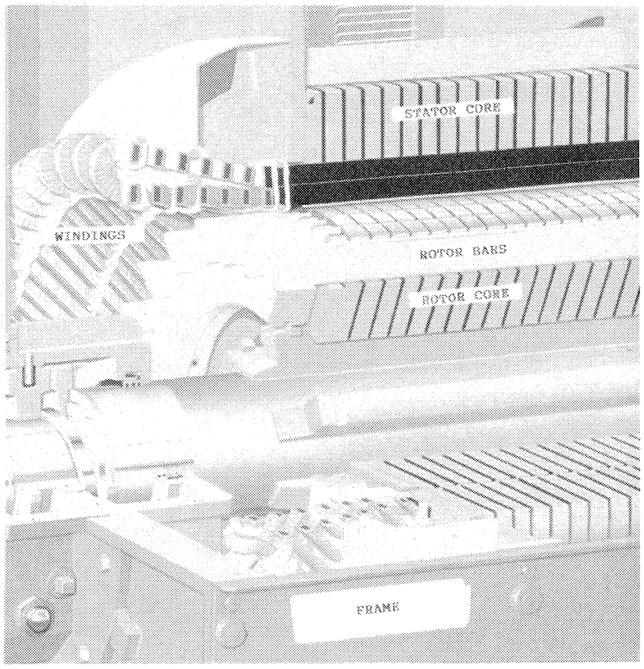

Figure 1. Cutaway of a Typical Large Squirrel Cage Induction Motor Outlining the Various Major Components.

stator ID. The number of alternate polarity magnetic poles set up by the winding connections dictate the speed of the rotating magnetic field. The motor synchronous speed is as follows:

$$\text{motor synchronous speed} = \frac{120 \times \text{frequency of applied voltage (Hz)}}{\text{number of poles}}$$

Currents in the rotor cage are induced across the air gap of the motor in a manner similar to those induced in the secondary of a transformer. It is different from the transformer, however, since the secondary or rotor winding rotates physically, trying to gain synchronism with the stator winding rotating field [1]. The rotor cannot achieve synchronism because of torque or load on its shaft. The amount of speed by which the rotor lags the stator synchronous field is called "slip." The amount of slip and motor current are higher as the motor torque load is increased. As the field rotates around the stator, the reactionary tangential force, which is a function of load torque, core loss, and friction and windage losses, slips behind the stator magnetic field. Under no load conditions, slip is low because torque needs only to overcome the core and friction and windage losses; however, a slight current is induced in the rotor cage. Because of this, some slip is present even at no load, therefore, synchronous speed can never be reached.

The torque to accelerate an inertial load to an operating speed, will at any speed, be equal to the difference between the motor torque developed and the load torque. The motor torque is a function of the applied voltage during starting conditions, and the load torque usually varies with speed. The rate of acceleration is proportional to this torque difference, and the motor will hang up at a low speed value if the torque differential reaches zero. This may also occur if the motor design is such that its speed characteristic curve has parasitic torques or cusps.

MOTOR VIBRATION

Vibration analysis and rotordynamics has become a science in itself, generally studied and reviewed by mechanical engineers or rotating machinery specialists. It is not uncommon for turbine and compressor rotors to successfully operate at speeds in excess of 12,000 rpm. Therefore, most rotating machinery specialists cannot understand how motors, having maximum speeds of 3600 rpm, can exhibit vibration problems which are so difficult to diagnose. While motor speeds are relatively low compared with turbomachinery, the dynamics associated with them can be extremely complex due to additional forcing functions present which are not found in a mechanical machine. Add to this a rotor which is a laminated steel cylindrical core held axially under compression and shrunk on a shaft, as well as a rotor cage which is inherently "loose," and the difficulty of motor vibration diagnostics increases dramatically.

It is true that two pole (3600 rpm) motors behave the worst when discussing mechanical performance; however, lower speed motors operate on the same principles. Electromagnetic forcing functions are generated at all motor rotor speeds; however, they are more pronounced for two pole machines due to the greater force between the magnetic poles [2]. In addition, two pole motors have greater centrifugal forces and normally operate much closer to the first critical speed.

Does the addition of electromagnetic forces really complicate motor vibration diagnostics? Not in itself; however, when they are combined with the mechanical forces seen on all rotating machines, analysis becomes difficult. The fact that the electromagnetic forcing frequencies may be very close to running speed, or its multiples, makes it easy to understand why mechanical and electrical engineers alike have such difficulty in motor vibration diagnostics.

To understand vibration in motors, the first thing to remember is that it is also a mechanical machine having all the forcing functions as any rotating mechanical machine. The shaft must be straight, the rotor must be balanced, bearings must be adequate, etc. Electrically, various inherent electromagnetic forces exist which cannot be eliminated. Problems will occur when either the mechanical, electromagnetic, or electromechanical forces become excessive, which can occur due to a number of reasons.

As this presentation is directed towards the mechanical engineer, the purely mechanical dynamics of a rotating system will not be discussed. The electromagnetic and electromechanical aspects will be the point of concentration. Electromagnetic forces are those which are purely magnetic, created by the rotating magnetic field. Electromechanical forces are those forces most commonly generated as a result of an electromagnetic force and a mechanical force, such as unbalance or a bent shaft, acting in cooperation with one another. It can also result from an electromagnetic force and electrical dissymmetry (broken bars or cracked end rings) also working in concert.

Electromagnetic Forces

The two main electromagnetic forces in an induction motor occur at 60 Hz and 120 Hz. The frequency of the main air gap magnetic flux wave is 60 Hz; however, it is actually a torsional function on the rotor. Any dissymmetry in the magnetic circuit will produce a lateral force whose frequency is at 60 Hz. This component is generally very small and normally not a concern.

Inherent 120 Hz Force—The existence of a 120 Hz force can result primarily from two sources. It is the result of the inherent magnetic attraction between the rotor and stator acting on a single point on the stator core each time it comes under the influence of a rotating magnetic field pole.

In one cycle of voltage, a magnetic field pole will pass this stationary point twice in one rotation of the magnetic field for two pole motors, four times in one rotation of the magnetic field for four pole motors, six times for six pole motors, etc. The speed of rotation of the magnetic field poles is exactly 60 Hz for two pole motors, exactly 30 Hz for four pole motors, and exactly 20 Hz for six pole motors. As a result, the frequency of the force of attraction between the rotor and stator is 60 Hz times two for two pole motors, 30 Hz times four for four pole motors, and 20 Hz times six for six pole motors. This 120 Hz force can, therefore, be defined as being a function of the speed of the rotating magnetic field times the number of magnetic field poles. Of course, as the number of poles dictate the speed of the rotating magnetic field, this force as defined above, must always have a frequency of 120 Hz and, therefore, is independent of the number of magnetic field poles.

While it is demonstrated prviously that an inherent 120 Hz force is present on all induction motors, the amplitude of this force is typically more pronounced on two pole motors. This is due to the much greater distance between the poles on a two pole motor (180 degrees) as opposed to on slower speed machines, 90 degrees for four pole motors, 60 degrees for six pole motors, etc.

120 Hz force due to air gap dissymmetry—The second source for 120 Hz vibration forces, while not inherent in the motor, generally always exists due to a point of minimum air gap being present in the motor. Ideally, the rotor should be perfectly concentric with the stator bore. Practically, however, due to manufacturing and assembly tolerances, this situation is impossible to achieve. Hopefully, the maximum deviation in the air gap will not be greater than five percent from the average (especially on higher speed motors).

If it is assumed that the rotor is perfectly concentric with the stator, the net effect of the magnetic flux forces in the air gap is entirely balanced between magnetic field pole pairs (north and south pole). If, however, a point of minimum air gap exists in the motor, and a magnetic pole lines up with this point, it creates an area of maximum flux density and therefore the magnetic forces between the pole pairs are unbalanced. This unbalance creates a magnetic pull force occurring each time a magnetic field pole passes the point of maximum flux density. It can, therefore, be stated that this 120 Hz force due to air gap dissymmetry is also defined as a function of the speed of the rotating magnetic field times the number of poles. As was the case for the inherent 120 Hz force, the frequency of the air gap dissymmetry force is independent of the number of poles and occurs at exactly 120 Hz [3]. The generation of the 120 Hz force due to air gap dissymmetry is demonstrated in Figure 2.

The electromagnetic forces discussed previously are dependent entirely on voltage and the rotating magnetic flux wave. This means that they exist whether the machine is running at no load or full load. It is possible, however, that an increase in temperature resulting from a full load run, can alter the air gap mechanically, thereby increasing the air gap dissymmetry and with it the 120 Hz forces from the no load runs. It is for this reason that both a full load and no load test are valuable when diagnosing motor vibration problems.

Electromechanical Forces

Electromechanical forces are present on all motors to some extent and are directly related to the motor slip speed. The forces can be generated by a number of either electromagnetic or mechanical dissymmetries, which creates an unbalance magnetic pull force with a frequency of modulation. The two most common modulating unbalance magnetic pull forces occur at a frequency of 1) the number of poles times the slip speed, and 2) one times the slip speed.

To produce a force having a frequency of the number of poles times the slip speed, a revolving point of minimum air gap must

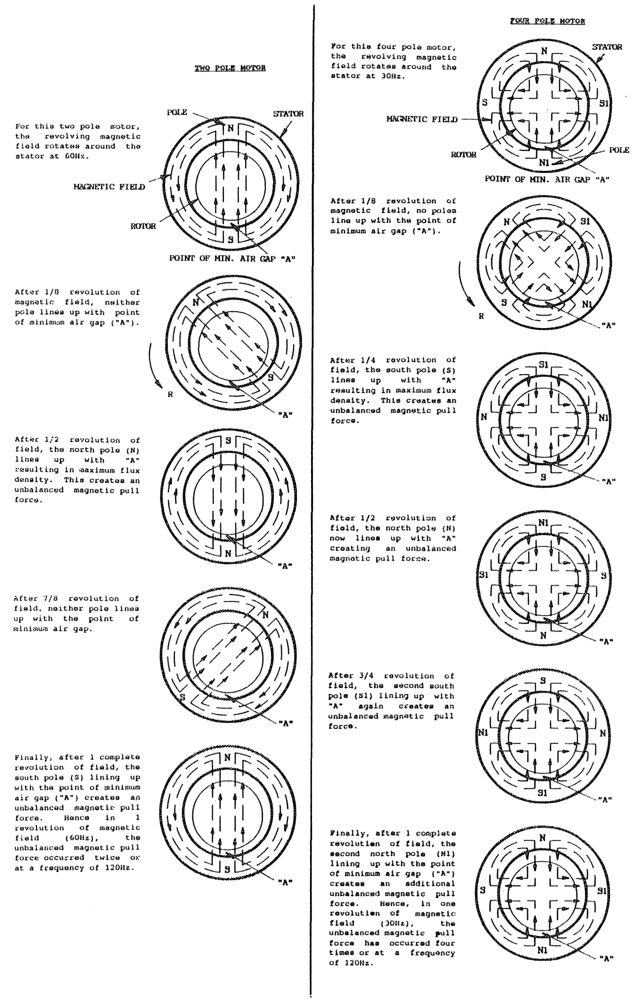

Figure 2. Generation of 120 Hz Electromagnetic Force Due to Air Gap Dissymmetry.

be present in the machine. Due to the reasons discussed for electromagnetic forces, this minimum air gap is present on all machines; however, its deviation from the average is what determines the amplitude of the resulting modulating force. In addition, besides resulting from normal manufacturing, this can result from much more severe problems such as a bent shaft, broken rotor bars, excessive unbalance, etc. This revolving point of minimum air gap will, as shown in Figure 3, comes under the influence of maximum flux density (magnetic pole) twice in one slip cycle for a two pole motor and four times in one slip cycle for a four pole machine. This produces an unbalanced magnetic pull force modulating, pulsating, or beating at a frequency of the number of poles times the slip speed [2].

Although not as prevalent, the second most common electromechanical forces has a frequency of one times the slip speed. For this to occur, two dissymmetries must occur simultaneously between the rotor and stator. An example would be of a rotor which was not adequately centered radially in the stator and which also exhibits excessive unbalance. Assuming this example is of a two pole motor, when a magnetic pole lines up with the point of minimum air gap, the mechanical unbalance is 180 degrees from this point and therefore, the unbalance magnetic pull will tend to "balance" the rotor [2]. The resulting force will, therefore, be negligible during one half cycle of slip. During the other half cycle, the magnetic pull will line up with the unba-

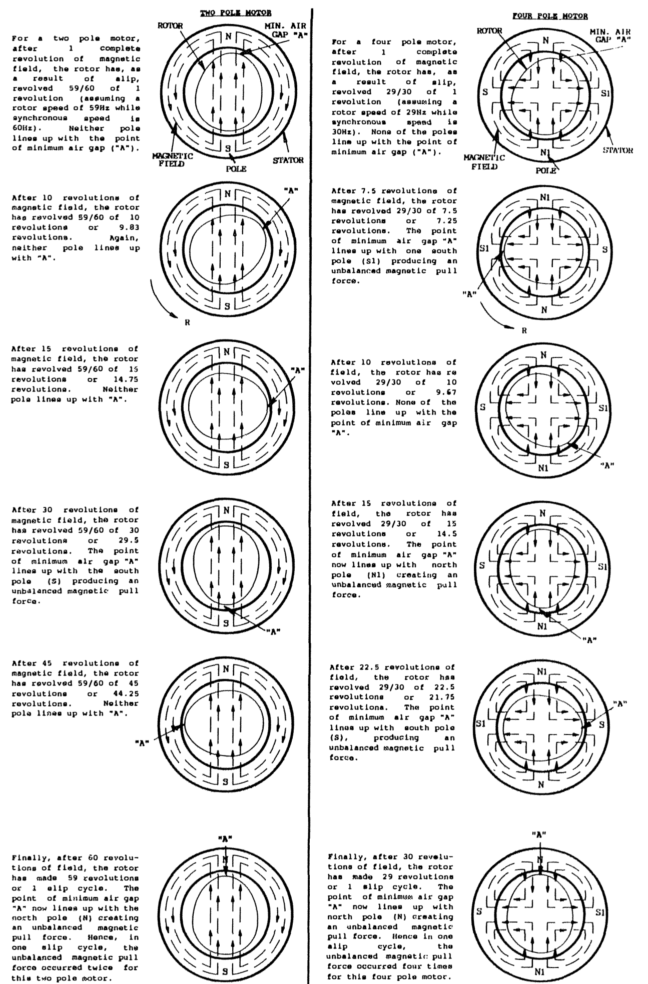

Figure 3. Demonstration of Electromechanical Force with a Frequency of the Number of Poles Times Slip Speed.

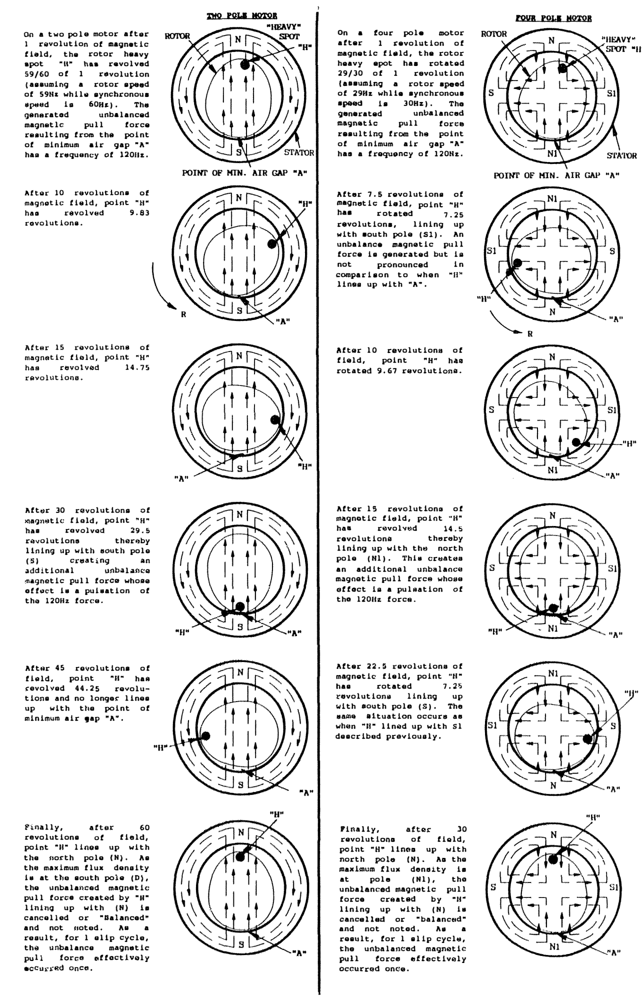

Figure 4. Demonstration of Electromechanical Force Having a Frequency One Times the Slip Speed.

lance amplifying the unbalanced magnetic pull force. Therefore, the resulting modulating force will occur once in one cycle of slip. This situation is demonstrated in Figure 4.

While it is also possible for this to occur on slower speed motors, a one times slip frequency force is much more difficult to produce. As seen in Figure 4, for the four pole motor, a pronounced force occurs when the unbalance force lines up with the unbalanced magnetic pull force created by the point of minimum air gap. In addition, a force is created each time the unbalance force comes under the influence of a magnetic pole except when the pole is 180 degrees from the point of minimum air gap. This, in effect, also creates an unbalanced magnetic pull force; but, it is not as great as when the unbalance lines up directly with the pole. Because of this, the unbalanced magnetic pull force will have a tendency to modulate at one times slip frequency. As this forcing function has yet to be seen on slower speed machines, it should not be of a concern.

Either of the electromechanical forces previously described above would be superimposed on any of the vibration components whether it be the unfiltered or filtered values. The result would be that particular vibration component modulating at either one times slip speed or the number of poles times slip speed. It is important to remember that a slip frequency related unbalance magnetic pull force is always present; however, it should not be excessive. The amplitude of modulation should not be above 20 to 25 percent. If it is, a more severe dissymmetry is present, most likely due to sloppy machining or excessive tolerances during motor assembly. Critical problems such as broken rotor bars, end ring cracks or bent shafts exhibit the same modulation characteristics; therefore, all excessive modulation must be investigated.

A common misconception is that these unbalance magnetic pull forces exist only when the machine is operating at full load or close to full load. This is probably due to the fact that one can hear an audible beat at full load while not hearing one at no load. The unbalance magnetic pull force will not exist at no load if the dissymmetry is load related such as a broken rotor bar or a thermal bow; however, if it is from sloppy manufacturing techniques, the forces exist even at no load.

Since the unbalance magnetic pull is dependent upon slip, and as described in the basic theory section of this paper, slip is present even at no load, it makes sense that these modulation forces can also exist at no load. At no load, however, the slip and, therefore, the forcing frequency is very low. Typical no load slip frequencies are between 0.001 Hz and 0.003 Hz (0.06 rpm to 0.18 rpm), whereas, full load slip values typically range between 0.25 Hz to 0.5 Hz (15 rpm to 30 rpm). As a result, during no load operation the time for the revolving point of minimum air gap to come under the influence of maximum flux density will be approximately five to seventeen minutes. However, during full load operation, this time is only between two and four seconds! It is for this reason, extremely low frequency, that a beat cannot be heard at no load.

It is, therefore, possible to ascertain the slip frequency modulated electromechanical forces during a no load run by monitoring a vibration component continuously for up to approximately 15 minutes. In addition, the no load slip speed can be determined by aiming a strobe light, set at exactly 60 Hz for two pole motors, exactly 30 Hz for four pole motors, etc., at the shaft end keyway. One can then see the keyway start to lag behind the strobe very slowly. The time required for the keyway to make one complete revolution is the no load slip speed. An example of using this technique is demonstrated in Case History Number 3.

For modulation forces to be eliminated, a perfectly symmetrical machine, both electrically and mechanically would have to be manufactured. As this is not possible, the motor should be made as symmetrical as tolerances allow.

The vibration troubleshooting chart presented in Table 1 should be beneficial when motor vibration problems occur. It covers most of the common occurrences in a simplified way; however, because of the multitude of electrical and mechanical factors which may precipitate a vibration problem, it is often necessary to perform a basic analysis based on fundamental concepts, most of which are discussed herein. Since not all combinations of electrical and mechanical problems can ever be accounted for, it is very important to understand the nature, origin, and behavior of the forces discussed here.

Table 1. Vibration Symptoms and Causes for Squirrel Cage Induction Motors.

Frequency Component	Cause	Comment
Fractions of Rotational Frequency	MECHANICAL	
	a) oil whirl	Frequency commonly falls around 40 to 45% of the rotational frequency
	b) poor lateral bearing support	Amplitude greater in the horizontal direction than the vertical. Subharmonic resonance may cause vibration at 1/2, 1/3, 1/4, etc., of rotational frequency.
1 Times Rotational Frequency	MECHANICAL	
	a) Unbalance	Most common cause of vibration. Amplitude steady. Responds readily to balance weights.
	b) Bent shaft	Addition of large amount of balance weights have little effect. May also create excessive slip related modulation.
	c) Improper balancing techniques	Can overly flex shaft as in the instance of adding balance weights to ends of rotor for unbalance in center of rotor.
	d) Critical speed is near operating speed	Difficult to balance using normal techniques. Vibration peaks near rotational speed during coastdown.
	e) Frame or enclosure resonance	Vibration is insensitive to normal balancing methods.
	f) Excessive bearing clearances/or excessive lateral clearances	Shaft vibration is often greater in the horizontal direction than the vertical direction.
	g) Rotor rub	Possible unpredictable vibration amplitude and phase angle. Local overheating effects can cause thermal bow.
	h) Misalignment	Can differentiate from unbalance due to the fact that misalignment often appears as increased vibration also at 2x, 3x, and higher multiples of rotational frequency. Effect is proportional to the stiffness of the coupling radially and the degree of misalignment.
	i) Defective anti-friction bearings	Can also produce multiples of rotational frequency and frequencies associated with the number of balls and ball train rotation May also appear as multiples of rotational frequency often at 6 to 50 times rotational Vibration is unsteady as bearing deteriorates.
	j) Coupling unbalance	Adjacent bearing vibration is highest. Difficult to make balance correction on rotor body.
	k) Loose rotor laminations on shaft	Erratic and unpredictable response to balance corrections. See also ELECTROMECHANICAL a) and MECHANICAL c) for multiples of rotational frequency.
	l) Eccentric journals	Vibration cannot be reduced by normal balance methods below a certain value. Frequency may also be 2 or more times rotational.
1 Times Rotational Frequency (cont'd)	m) Poor bearing to bearing housing fit	Large rotational vibration amplitude not proportional to balance corrections. May also result in multiples of rotational frequency.
	n) Non-uniform anchoring of rotor bars	Results in differential expansion within or among rotor bars. Cage and end ring displacement or stressing results in unbalance. Could lead to bar breakage. Can sometimes be corrected by reswaging.
	o) Non-uniform rotor core clamping	Mechanically forced shaft bow either at assembly or due to thermal effects as the rotor heats up.
	p) Soft feet	Twisting of frame or base, poor clamping, improper shimming, etc. Often produces magnetic dissymmetry and excessive 120Hz vibration.
	ELECTROMECHANICAL	
	a) Excessive rotor core eccentricity with respect to journal	Punching eccentricity, machining error and/or loose rotor laminations on shaft. See also MECHANICAL c) for multiples of rotational frequency and MECHANICAL k) above.
	b) Defective rotor bars or end rings	Broken rotor bars or connections to the end rings. Shows up as number of poles times slip frequency with modulation of twice line frequency component.
	c) Thermal bowing of rotor	Can result from improperly machined rotor surface and smeared rotor laminations, broken rotor bars, etc. Will also cause slip frequency related modulation.
Multiples of Rotational Frequency	MECHANICAL	
	a) Out of round journals	Vibration cannot be reduced by normal balance methods below a certain value. Frequency may also be 2 or more times rotational.
	b) Inadequate oil film thickness	Load region of bearing is starved of lubricant, is too hot, wrong grade of oil, etc.
	c) Out of round rotor	Usually produces twice rotational frequency vibration. May also produce slip related modulation of twice line frequency. See also ELECTROMECHANICAL a) and MECHANICAL k) for 1 times rotational frequency.
	d) Mechanical looseness of bearing support structure	Vibration high at support feet and foundations.
Multiples of Rotational Frequency	ELECTROMECHANICAL	
	a) Non-sinusoidal or multiple rotational frequency excitation forces	Example: out of round rotor in addition to the rotor being eccentric with respect to the journals. Modulations of vibration related to slip frequency.
2 Times Line Frequency	MECHANICAL	
	a) Stator bore not round	Sensitive to stator winding connections and voltage.
	b) Rotor not centered in the stator core.	Sensitive to stator winding connections and voltage.
	ELECTROMECHANICAL	
	a) Inherent magnetic forces compounded due to winding circuit or phase imbalance.	Causes local stator heating. Vibration compounds as motor heats up.
	b) Shorted stator turns	Unit fails in a short period of time.

CONCLUSIONS

While squirrel cage induction motors have been used widespread in industry for over 50 years, it appears that the recent trend in long term reliability on major machines has now been instituted on motors. As a result, users are evaluating motors in the same manner they evaluate higher speed compressors' and turbines' mechanical performance. In the purchasing stage, it is important to recognize that stringent specifications alone do not guarantee a well running and reliable machine. A thorough understanding of induction motors must be known and employed when making up job specifications. The following aspects of motor procurement should be followed in order to assure the user of a well running and reliable motor:

- Write detailed motor specifications which rely on standards already published, if applicable to the motor application. Keep in mind that vibration and manufacturing tolerances do not have to be overly stringent for a good operating machine. Also, remember that a motor is a "completely different animal" from a turbine or compressor and, therefore, not all requirements can be interchanged.
- Submit the job specifications to motor vendors for comments. Qualify various motor manufacturers to ascertain their manufacturing, quality control and testing capabilities.
- Contact other users of identical motors to learn of their experiences. Judgement has to be made here since a particular machine can be either the "greatest" or a "piece of junk," depending upon who is talking. It helps to have a specific list of questions available and make certain that all failures are fully explained to determine responsibility.
- Evaluate the motor bids from a technical and an economic standpoint. Know the various manufacturers' major design features and drawbacks.
- A comprehensive design review should be made as soon as the initial electrical and mechanical designs are finalized. If possible, crosscheck the motor starting characteristics and the lateral critical speed analysis. Get satisfactory explanations on significant deviations between results.
- Develop a complete and comprehensive shop inspection and witness test plan. It is most important to utilize qualified inspectors knowledgeable in motor manufacturing and construction. Witness agents must be knowledgeable in motor design, and electrical and mechanical testing. It is easy to understand that more than one person may be required to satisfactorily complete the inspection and witness testing.

In order to understand the vibration forces within a motor, it is important that the basic operating principles of motor theory be known. Vibration forces in motors can be of three types: mechanical, electromagnetic, or electromechanical. As this presentation is geared towards the mechanical engineer, only the latter two cases were reviewed. Electromagnetic vibration consists of 60 Hz (line frequency) and 120 Hz (twice line frequency) forces. Electromechanical forces consist of a unbalanced magnetic pull force working in combination with an electromagnetic force so that the resulting vibration is modulating at a frequency in relation to slip speed.

Lastly, five actual case histories of motors exhibiting various vibration problems are presented. Each case is detailed in terms of the troubleshooting and corrections necessary which resulted in well running machine.

FIVE CASE HISTORIES

- Rotor bar breakage on a compressor driver
- Rotor Thermal bow due to smeared laminations
- Stator core 120 Hz vibration transmitted to shaft and bearing housings
- Demonstration of non linear damping of oil film
- Demonstration of vibration modulation at one times slip speed

Case History 1

Three identical 1250 hp, 6900 Volt, 3600 rpm, induction motors were placed in service in 1978 at a waste water concentrator plant for a utility company. The units were driving vapor compressors having a connected load inertia of four times the listed "allowable NEMA WK^2 to accelerate without injurious temperature rise." Over the next five years, numerous rotor failures occurred on all three motors, with at least two failures per unit. Each time, the rotor was repaired and placed back into service.

Vibration measurements were subsequently recorded on one of the repaired motors after it had been in service for two months. Maximum unfiltered vibration levels on the bearing housing modulated between 1.6 and 3.4 mils. The frequency of the modulation was determined to be twice slip frequency with a very strong twice line frequency vibration component of 0.30 ips which was also modulating at twice slip frequency (Figure 5). A higher harmonic of bar passing frequency was also noted in the unfiltered value.

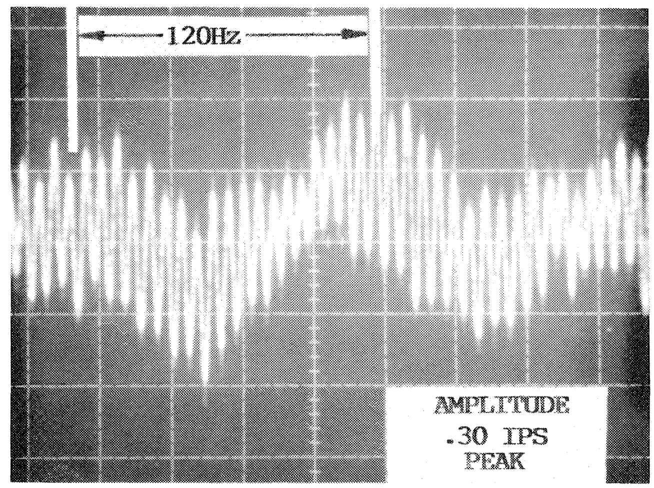

Figure 5. Opposite Drive End Bearing Housing Horizontal Vibration as Measured by Accelerometer.

The following month, the motor was shut down and transported to a service shop for inspection and repair. When the rotor was pulled from the stator, numerous cracks were noted in the end ring and rotor bar to end ring joints (Figures 6 and 7). Additionally, the rotor cage was extremely loose, such that the service shop started to swage the bars at an attempt to tighten the rotor cage (Figure 8). Reportedly, this had been the exact condition of all the rotors which had previously failed. Work was stopped when it was decided that an analysis and assessment of this problem should be performed.

A review of the rotor design was made, and it was discovered that to compensate for the loose cage, the manufacturer utilized a balance/support ring assembly. This assembly was bolted to the end ring and the ID was then shrunk to the shaft, thereby containing the radial looseness of the rotor cage. A cross section of the rotor construction is shown is Figure 9. Restraining a loose item is acceptable on a mechanical machine, however, on an electrical machine, certain electrical and thermal parameters were neglected.

Each time a motor is started, a large current flows in the rotor bars and end ring. The heat generated during this start is almost

Figure 6. View of Crack in Rotor End Ring after Motor was Disassembled at Service Shop.

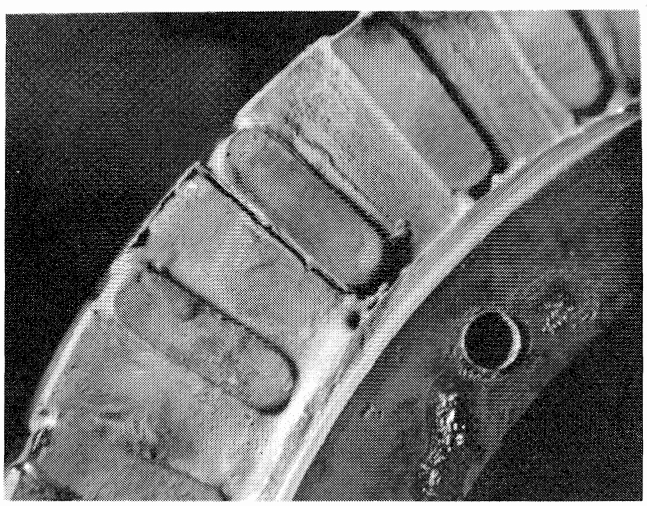

Figure 7. Crack in Rotor Bar to End Ring Braze.

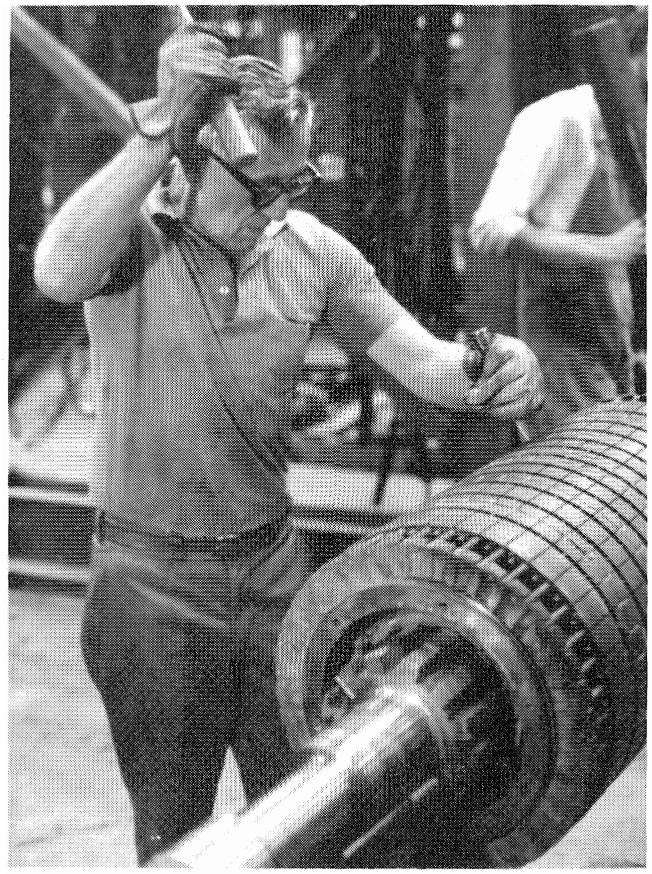

Figure 8. Service Shop Worker Attempting to Tighten Rotor Cage by Swaging Rotor Bars. Swaging is the act of mechanically displacing bar material in a manner such that it becomes tighter in the slot.

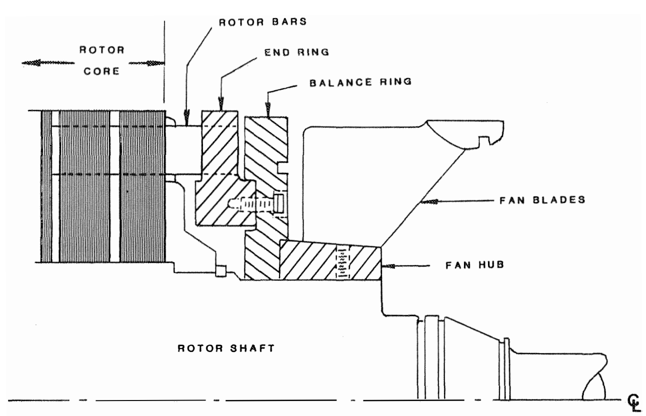

Figure 9. Original Rotor Cross Section Showing Balance Ring/ Support Ring Assembly.

equal to the amount of energy imparted to the rotating system. This heat, of course, is greater when accelerating high inertia loads such as this. The temperature rise of the bars and end ring results in an axial growth of the bars pushing the end ring assemblies away from the rotor core. Stresses are created due to centrifugal forces acting on the end ring and bars along with temperature gradients on the bars due to skin effect. These factors create bending stresses at the bar protrusion section and at the brazed bar joints to the end rings. For these reasons, it is important to allow an unrestrained and predictable axial growth of the rotor bars. The end ring balance/support ring assembly the manufacturer utilized on this machine may have avoided a balance problem; however, it did not allow for the design considerations just mentioned.

Once this problem was identified, the motor manufacturer designed and assembled new rotors utilizing a tight cage, larger end rings for the high inertia starting, and a balance ring which was allowed to slide on the shaft. Problems have not been reported on these machines since.

Case History 2

During acceptance testing of a 1500 hp, 3600 rpm motor, no load coupled vibration levels on a dynamometer stand were very low. Maximum unfiltered levels of 0.80 mils on the shaft and 0.25 mils on the bearing housing were measured. When the motor was placed under load, the corresponding vibration levels gradually increased over a four hour period and stabilized at unfiltered levels of 2.5 mils on the shaft and 0.75 mils on the bearing housing. Filtered rotational speed vibration levels showed approximately the same increase in shaft vibration and a corresponding phase angle change of almost 180 degrees.

Even though the measurements exhibited classic signs of a rotor thermal instability (Bow), the manufacturer stated that the vibration acceptance criteria of 2.0 mils had not been guaranteed under loaded conditions. They felt that since the motor vibration was below 2.0 mils under no load, the motor was acceptable. During initial discussions, they stated that this condition was the result of vibration influence from their dynomometer; however, this was disproved from a review of the vibration spectra which clearly confirmed original suspicions of a thermal bow.

The rotor was disassembled from the stator and inspected very thoroughly. As discussed earlier, the rotor core is made up of thousands of cylindrical laminated steel sheets held in compression axially and shrunk onto the shaft. Each of these laminated sheets are insulated from one another in order to limit rotor surface eddy current losses and thereby reducing stray load losses. Since the laminated core is placed on the shaft, it is effectively shorted at the core ID. Local currents cannot flow unless a short occurs at a different radial location, most often at the core OD. One method of lamination shorting at the core OD can come from excessive burrs touching one another or piercing the coreplate of an adjacent lamination.

An inspection of this rotor's core OD, as viewed under a magnifying glass, showed areas of the rotor with "smeared" laminations over approximately 30 percent of the core surface. In these areas, localized eddy currents were circulated, thereby increasing the rotor surface temperatures nonuniformly and resulting in a thermal bow. Since the surface losses increase with slip, the rotor did not bow until the machine was loaded, when the surface eddy currents were highest. The smearing of laminations was caused by a dull lathe cutting tool, which was subsequently corrected. The entire rotor OD was then turned down to a lesser diameter and the motor then reassembled. When the vibration was again measured with the machine under load, levels did not increase by more than 10 percent over the test duration, while the phase angles did not change more than 10 degrees over a four hour test.

The vibration data shown in Figure 10 was recorded both before and after the repair procedures.

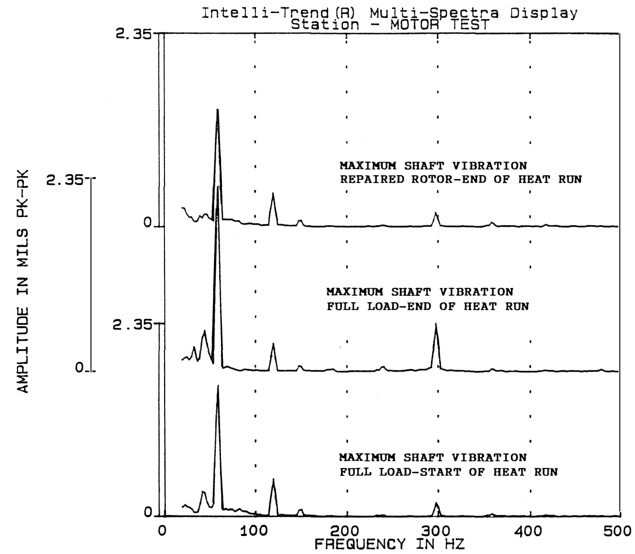

Figure 10. Vibration Data Recorded Both Before and After Repairs.

Case History 3

A 600 hp, 2300 Volt, NEMA 5000 frame motor was experiencing very high vibration, modulating at two times the slip frequency on both bearing housings and all shaft probes. Additionally, the twice line frequency vibration component was predominant and was also modulating at two times slip frequency.

It is important to note that twice line frequency vibration is present on all induction motors no matter how many poles. On a two pole motor such as this, it inherently results from the rotating magnetic field passing a single point of the stator twice in one voltage cycle (sine wave). The resulting stator and rotor attraction forces are independent of the voltage polarity, therefore, the magnetic force occurs at exactly twice line frequency. As the stator core is the primary forcing function, the decision was made to isolate it from the bearing housing.

The frame construction was a fairly standard cast frame which is line bored to accept the stator core (Figure 11). In order to maintain close air gap tolerances, ribs on the frame ID are bored concentric with the end bell rabbit fits. In this machine, there were six ribs into which the stator core and winding were shrunk. As these ribs extend along the entire axial length of the frame, the core vibration easily transmits to the end of the frame which supports the bearing housing. To isolate the core vibration, notches were machined into the ribs as shown.

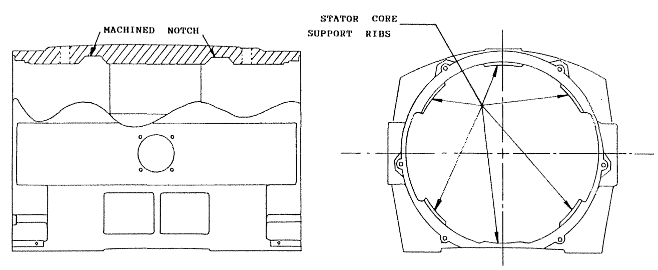

Figure 11. Cost 500 Frame Showing the Ribs in Which the Stator Core is Placed. To limit the transmission of the 120 Hz vibration from the stator core, each of the six ribs were notched as shown in the cross section. This tends to isolate the core from the end frame to which the bearing housings are mounted.

When the motor was assembled, the twice line frequency vibration was lower; however, the excessive modulation was still present. Only when another frame was modified was the cause identified and corrected which resulted in reduced vibration levels. It was discovered that in addition to the 120 Hz vibration problem, the original frame was improperly bored so that it was deflecting excessively in one direction simulating an out of round stator. The maximum shaft vibration is shown in Figure 12 both before and after the modifications to the frame.

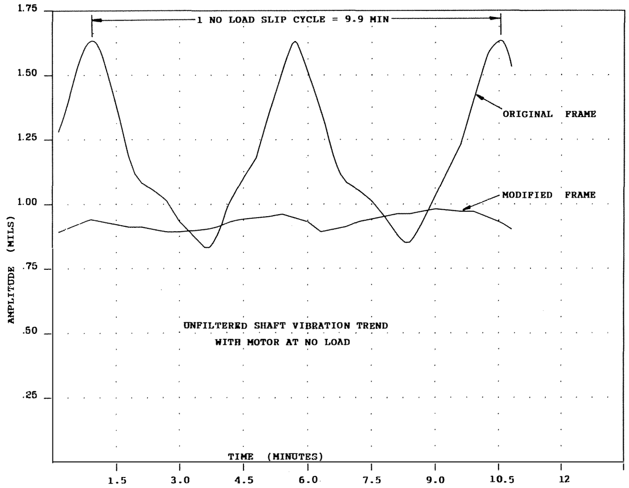

Figure 12. Plot Showing Instantaneous Peak to Peak Shaft Vibration Levels Modulating at 2 × the Slip Speed During a No Load Slip Cycle.

Case History 4

A two pole, 1250 hp motor had undergone vibration testing at the motor manufacturers plant without any problems. Unfiltered vibration levels reached a maximum of 1.13 mils on the shaft and 0.50 mils on the bearing housing during the full load dynamometer testing. Predominant vibration was noted to be at running speed with no excessive 120 Hz or slip frequency effects noted. The motor was very stable throughout the load test as amplitudes, both filtered and unfiltered, and phase angles did not change significantly.

Contract requirements stated that an unbalance response test for the determination of the first lateral critical speed was to be performed for this rigid shaft rotor. (The calculated first lateral critical speed was 4250 rpm) To do this, the motor was run at an overspeed up to 4200RPM and allowed to coastdown freely while plotting the filtered shaft vibration and corresponding phase angle. The critical speed would show an increase or peak at a given speed as well as a phase angle change. During the first coastdown, shaft vibration reached a maximum of only 1.20 mils. As this value was fairly low, an unbalance was placed on each end of the rotor in phase with one another to excite the first mode.

The residual unbalance (RU) tolerance of the rotor was 19 gm-in. The second coastdown was performed with an unbalance of 51 gm-in placed on each end of the rotor. This time, the maximum amplitude was 1.4 mils. The decision was made to add 95 gm-in per end (6 times RU). The machine was then being brought up to overspeed and a sudden increase in shaft vibration up to 5.0 mils occurred at 4150RPM. The occurrence was extremely rapid and the amplitude and phase angle characteristics did not indicate a true critical speed. Upon bearing disassembly, there was a "polish" on the top bearing halves due to their being contacted by the shaft journals. The maximum probe vibration during each of the three coastdowns performed is shown in Figure 13.

Once the bearings were "scotchbrited" and reassembled, the unbalance response tests were repeated but only up to adding the unbalance weights of 51 gm-in. All commercial obligations had been satisfied as there were no critical speeds within 20 percent of operating speed, and nothing in the customer specifications stated that the addition of five times the residual unbalance tolerance could not cause a vibration increase. The motor was accepted following an explanation from the manufacturer stating that the 98 gm-in unbalance resulted in a nonlinear behavior of

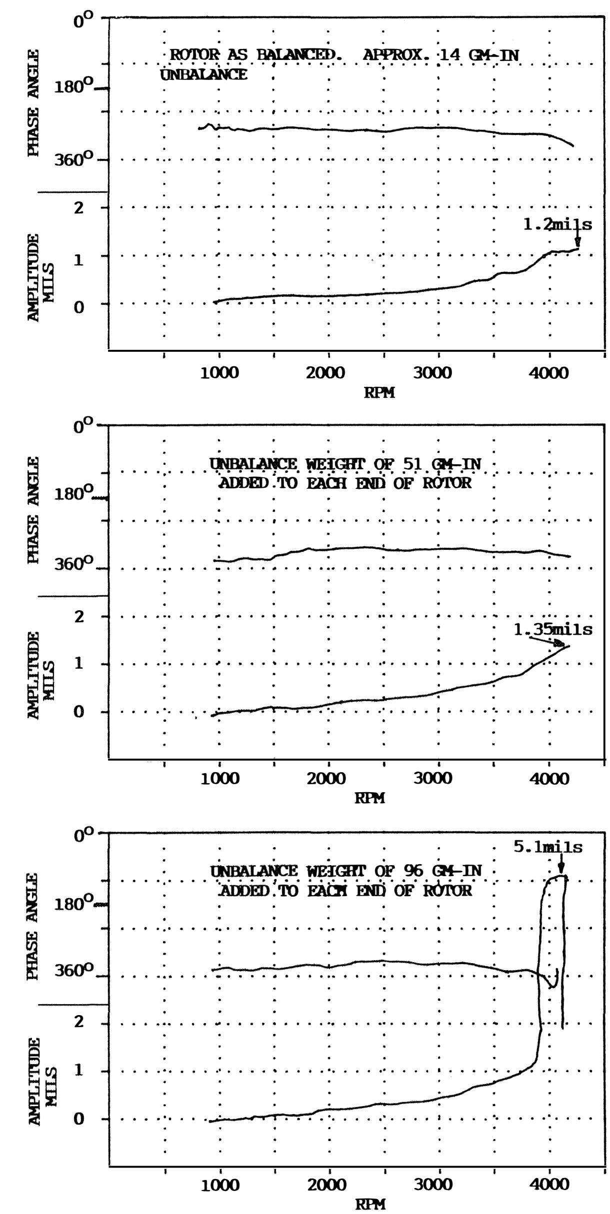

Figure 13. Rotor Unbalance Response Curves with Rotor in the "As Balanced" and Unbalanced Conditions of 51 gm-in and 96 gm-in.

the oil film perimeters. The unbalance caused the shaft journal displacements to exceed the linear portion of the oil film and permitted the shaft to contact the bearing.

This motor was subsequently installed and has been in service for approximately two years without any known problems.

Case History 5

Three 700 hp, 3567 rpm motors were purchased as the drivers for feedwater pumps. As the first motor was undergoing acceptance testing, vibration measurements with the machine under no load were well below the user specified levels of 0.10 ips on the bearing housings (filtered, all frequencies) and 2.0 mils on the shaft (unfiltered).

The motor was briefly loaded for instrumentation verification by use of a dynamometer and immediately an audible "beat" was noted whose frequency corresponded directly to one times slip frequency. Shaft vibration spectra was recorded as shown in Figure 14. It is quite common for a beat or vibration modulation of twice slip frequency to occur on two pole motors, and as long as this is not excessive, should not be a concern. Due to machining and assembly tolerances, always present is a point of minimum air gap in the motor. This will come under the influence of maximum flux density twice in one slip cycle, producing the unbalanced magnetic pull force whose frequency is twice slip speed. When this is superimposed onto other vibration frequency components, the result is a modulation of that component at twice slip frequency. As the motor exhibited a modulation of once slip frequency, an unusual situation was occurring on this machine.

What cannot be shown in Figure 14 is the 100 percent modulation of all the vibration components. Zooming into the 60 Hz

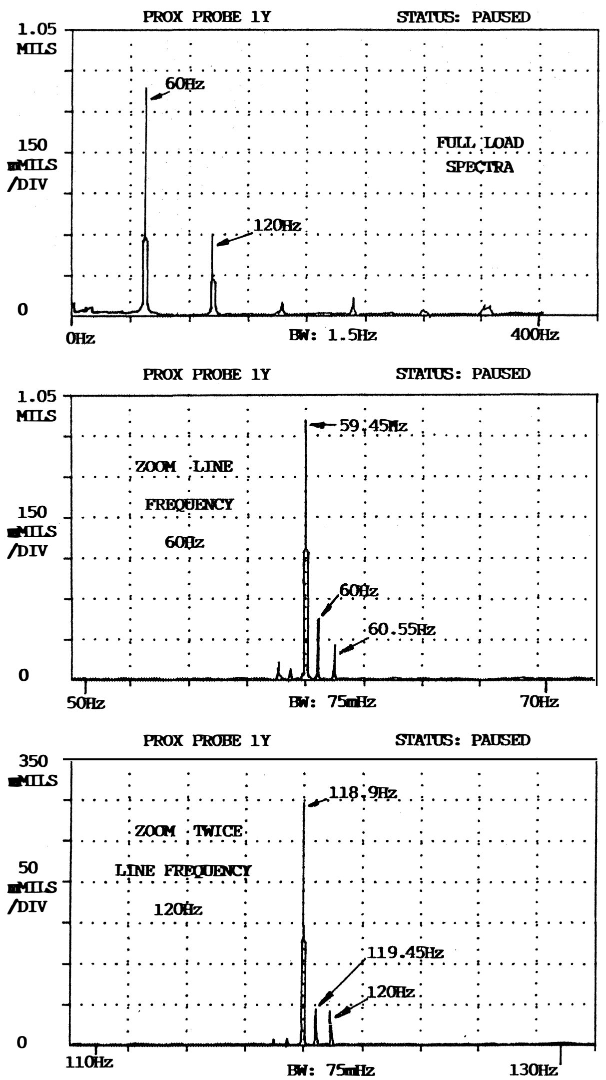

Figure 14. Full Load Shaft Vibration Spectra from 0-400 Hz, 50-70 Hz, and 110-130 Hz. The latter two plots differentiate the running speed from the line frequency components and multiples thereof.

component revealed that one and two lines slip speed sidebands were prevalent. Overall vibration levels at this time were not excessive; therefore, electrical performance testing was completed during which the vibration was not monitored. When the machine was again run under no load, the overall vibration levels had increased significantly and were now above the aforementioned specified values. Because of this, the motor was disassembled and inspected thoroughly.

As was stated previously, it is normal for vibration modulation to occur at twice slip frequency as a result of manufacturing tolerances of the machine. In this instance, there is a force whose frequency is one times slip frequency. This can occur if there is a multiple dissymmetry, such as when a mechanical dissymmetry and magnetic dissymmetry occur simultaneously between the stator and rotor. An example of this would be a rotor not adequately centered in the stator and also exhibiting excessive unbalance. For instance, when a magnetic pole lines up with a point of minimum air gap, the mechanical unbalance is 180 degrees from this point; therefore, the magnetic pull will tend to equalize the mechanical unbalance and the resulting force will be negligible for one half cycle of slip. During the other half cycle, the magnetic pull will be in phase with the mechanical unbalance and the resulting force will, therefore, occur once in one cycle of slip.

On this motor, the significant vibration increase from one no load test to the next pointed to a mechanical shift from its original residual unbalance condition. An inspection of the rotor revealed significant end ring cross section variations over its entire circumference. This leads to nonuniform resistivity in the end ring, producing a variation in the induced current circuit and, hence, air gap flux dissymmetry. Since these two problems, excessive unbalance and magnetic variation, were occurring simultaneously, the resulting electromagnetic force occurred once per cycle of slip. Actions for correction of this problem included machining of the end ring eliminating the variation and rebalancing the rotor. The motor was subsequently tested, and vibration levels were low, thermally stable and repeatable. Modulation at one times slip speed was still present, however, to a much lesser degree (down to 35 percent instead of the original 100 percent).

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