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Retrofitting pipelines with induction motors

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Abstract

This paper includes a discussion of design considerations, test and application of a large stiff shaft two pole motor designed for fixed or adjustable frequency drive (AFD) of pumps for a pipeline retrofit. A rotor-dynamics analysis of the rotor bearing system is presented for the as-built design and for alternate bearing designs. Test data is presented for complete shop tests with the motor and drive system. The project consisted of 22 motors plus 2 spares rated 3000 HP at 1.15 service factor and 3600 RPM at 60 Hz. These motors, rated 4000 volts are for use at variable speeds and torque from 23.5 to 60 Hz. A second more recent project is also briefly described in the paper consisting of 18 motors and one spare rated 4000 HP at 1.15 service factor and 3600 RPM at 60 Hz. These 4000V motors are for use at variable speeds and torque from 24 to 66 Hz.

Index Terms: Rigid shaft motor, AFD motor, pipeline motor, high speed motor, sleeve bearing.

Introduction

There has been much discussion in recent years in regard to retrofitting pipelines with induction motors driven by adjustable frequency drives (AFD). These motors will typically replace gas turbines which have been in service for many years. The motor drive combination has certain advantages over the old gas turbines. These advantages include improved efficiency, lower maintenance, lower noise, and less pollution.

In the past, induction motor manufacturers were not building large 2 pole motors with stiff shaft designs which operate below the first lateral critical speed of the rotor system [1]. If they had, the cost would have been excessive. Normally, standard flexible shaft motors were used [2] except for a few recent cases where stiff shaft designs have been applied [3]. For variable speed applications of flexible shaft motors, various speed ranges were blocked out as needed. The following discussion will explain why flexible shaft designs are no longer necessary for variable speed applications.

When considering the retrofit, special consideration must be given to the installation, the drive design, and the motor. This paper will discuss all the design considerations that had to be evaluated for all the possible alternatives, and issues that showed up later which needed to be addressed. The retrofit of 22 motor/drive installations has been completed and is performing well. The second 18 motor/drive project consisting of some new and some retrofit has been installed and also reported to be running well.

Motor Design

Due to the various retrofits not being identical, special design constraints were placed on the motor which may not have been necessary on new installations. Other requirements were due to the location, environment and application.

Typical considerations common for pipeline application:

- Rating in the range of 2000 HP to 4000 HP at 3600 RPM
- Weather Protected Type II outdoor motor
- API 541 3rd edition
- Class I Group D Division 2
- CSA Approval for Canada
- Underwater High-Pot

The following are special operating conditions due to varying applications at different locations. Some of the motors are intended to operate without adjustable frequency drives requiring the motor to be suitable for both across the line start and AFD applications. In addition various locations had different pump designs requiring across the line starting under different load conditions and different reduced voltages. Typical load and motor torque are shown in Fig. 1 for 65% voltage start.

Special Considerations

- Adjustable Frequency Drive (variable torque) from 1400 to 3600 RPM or higher
- Stiff rotor design for low vibration on AFD at all speeds
- 90 C Rise by Resistance @ 1.15 Service Factor on AFD @ all speeds
- Suitable for across the line start and constant speed application
- Low vibration levels @ all speeds
- 600% locked rotor current due to weak power system
- 65% Voltage start with 53% load @ full speed
- 70% Voltage start with 60% Load at full speed
- 88 dBA noise @ 3 feet per IEEE 85
- 95.3% Efficiency
- Special fan design and venting to provide adequate cooling and still meet efficiency
- Special 5500 Volt insulation system to handle voltage swing coming from 4000 volt power system (AFD)

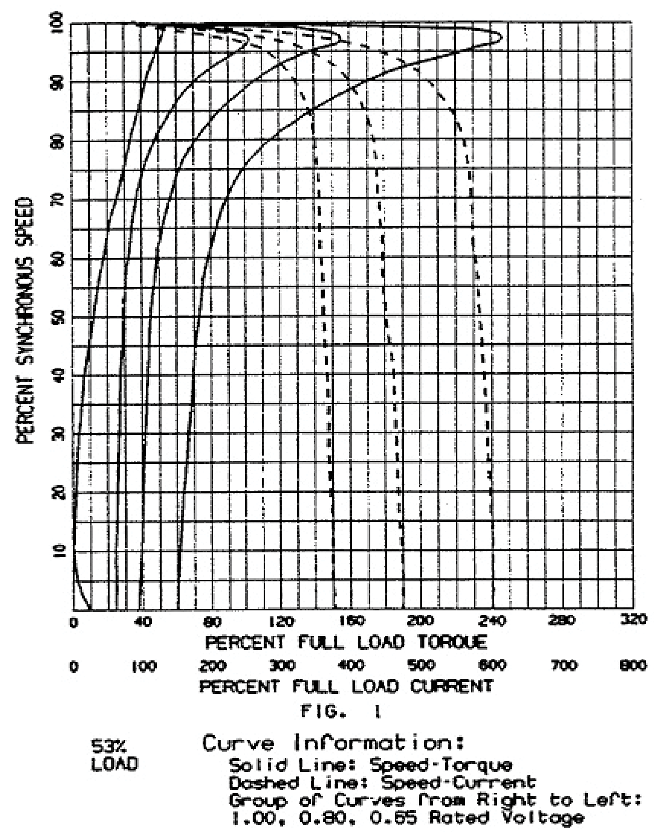

Figure 1 Speed torque curves of all applications

Electrical Design

Many of these design constraints are in direct opposition to one another. Making the motor suitable for across the line reduced voltage start is usually accomplished by either increasing the locked rotor current or by going to a high resistance rotor or narrower rotor bar.

It should be noted that the higher locked rotor torque and pull up torque which is required for reduced voltage across the line starting is directly proportional to the I2R loss of the rotor at start and up to 70 or 80% speed.

A high resistance rotor bar or narrower smaller bar, though it increases accelerating torque, also increases rotor bar losses, decreasing efficiency and increasing temperature rise. This is of particular concern when high frequency harmonic currents exist such as would be expected on an AFD.

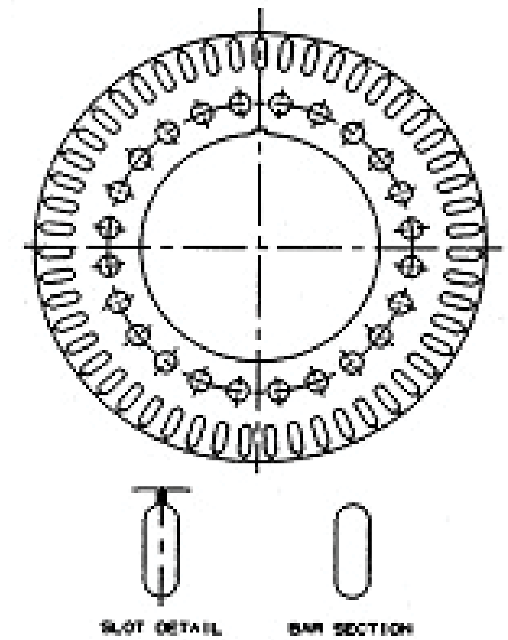

Figure 2 Rotor lamination/ axial vent holes

While recognizing the negative effects of going to a narrow high resistance rotor bar it was still necessary to do so to achieve the starting performance needed for this application. It was not possible to achieve the high starting torque by increasing the starting current due to the weak power system. The rotor bar chosen was CDA 220 copper alloy which has 2.3 times the resistance and losses of the standard copper bar rotor. Keeping in mind that the rotor current only flows in the upper .5 inches of the bar at "0" speed and upper 1.3 inches at 80% speed, it was necessary to keep the bar narrow through the upper 1.3 inches away from the air gap. Beyond this depth, the bar might be made wider, but in this case there was no space. Since the current will flow in the entire rotor bar at full speed, the bar area should be maximized to minimize rotor losses (Fig. 2).

Due to the Hall Effect which forces high frequency currents to the outer surface of conductors, the high harmonic frequency current will also only flow in the top of the rotor bar which now has been designed narrow with high resistance to achieve high starting torque. This will tend to cause the motor to run hotter while running on the AFD assuming there is significant harmonic content. Figures 3 through 6 below at various frequencies show there to be minimal harmonic content. One other option was to make the slot (air space) above the bar deeper creating higher reactance for the high frequency currents, minimizing the harmonic current. This was done to a small extent by making this section .065 inches deep. Deeper would have been better but it reduced the fundamental current too far thereby reducing starting torque. There were also some mechanical constraints which minimized how deep the rotor bar could be made. Since the motor is to operate at various speeds the laminations must be designed so as to minimize deflection or expansion of the rotor core at different speeds and temperatures which could cause a change in balance. Though this is important on all motors it is even more critical when the motor is to be operated throughout a wide speed range. Most 2 pole motors are balanced and run at 3600 RPM, so that lower speed balance would not be as critical.

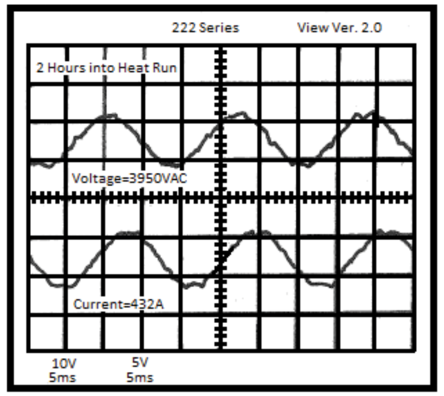

Figure 3 Voltage and Current Waveform (60 Hz)

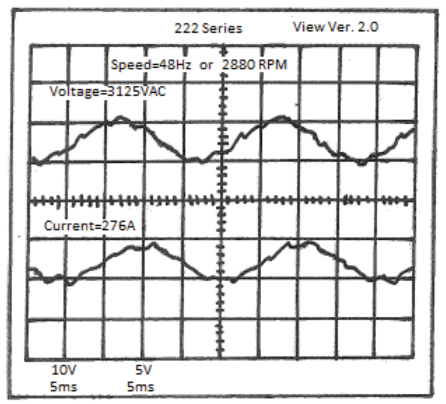

Figure 4 Voltage and Current Waveform (48 Hz)

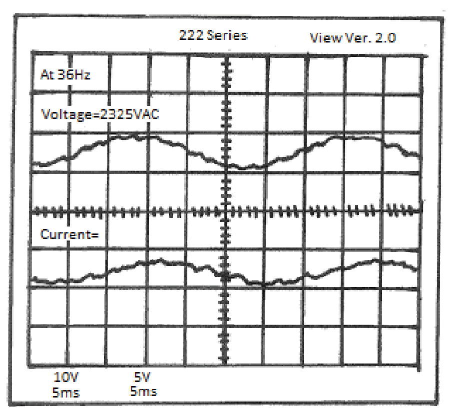

Figure 5 Voltage and Current Waveform (36 Hz)

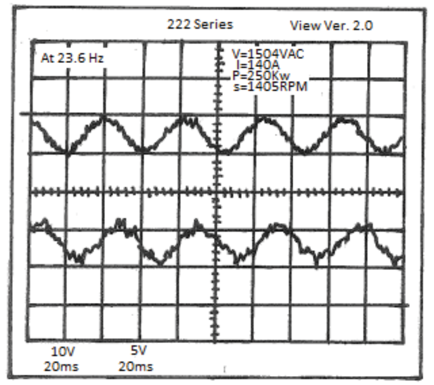

Figure 6 Voltage and Current Waveform (23.6 Hz)

In addition, the low noise and high efficiency requirements dictate a reduction in the fan diameter which will reduce cooling and increase the temperature rise. This must be done with extreme caution when operating on an AFD which may cause the motor to run hotter. A special fan design was required here.

Obviously, it would be better to design around a more specific application to better custom fit the motor, but this is not always possible, requiring a balancing act by the designer. The listing in Table I is a summary of the test results of the motor and drive running in the motor factory coupled to a dynamometer simulating the load conditions of the field. The results show that the drive harmonics were very minimal causing only minimal increase in temperature rise. Vibration also remained low throughout the entire speed range.

TABLE I Test Results Loaded Heat Run & Vibration

Frequency	60 Hz Sine-wave	60 Hz AFD	48 Hz AFD	36 Hz AFD	23.4 Hz AFD
Horse Power	3485	3500	1790	760	370
Rise by Resistance	58.8	61.4	31.1	N/A	N/A
Rise by RTD	83.6	87.3	46.8	N/A	22.3
Voltage	4040	4120			
Vibration Hsg. Inches/Sec Vertical Horizontal Axial	ODE / DE .038 / .100 .080 / .063 .085 / .096	ODE / DE .007/ .069 .071 / .066 .038 / .061	ODE / DE .035 / .037 .058 / .054 .083 / .039	ODE / DE .017 / .027 .054 / .052 .045 / .025	ODE / DE .014 / .018 .111 / .098 .012 / .012
Shaft Vibration Mils P-P 315 Deg 45 Deg	 .816 / .360 .615 / .337	 .791 / .405 .579 / .320	 .565 / .353 .358 / .263	 .427 / .435 .417 / .311	 .591 / .618 .469 / .404
Figure	N/A	Fig. 3	Fig. 4	Fig. 5	Fig. 6

Nameplate Rating

Rated HP	3000	Amperes	376
Service Factor	1.15	Insul. Class	F
Rated RPM	3574	Temp. Rise	90 °C
Voltage	4000	Type	WPII
Freq./Ph. Hz	60/3	Frame	808S

Tested Efficiency 95.5 (sine wave)

Tested Noise Level at one meter at no load in a free field (sine wave) = 84.5 dbA.

Noise Levels (per IEEE Std. 85-1973): A walk around sound test was done under load on the AFD (93.7) dBA and on a sine wave (93.3 dBA). No significant change was observed. Levels were higher than no-load due ambient noise, load machine noise, and reverberation.

With motor coupled and hot, motor was operated on AFD from 24 to 60 Hz and no unusual noise or vibration was observed.

Mechanical Design

The determining consideration in the mechanical design of the motor was the requirement for a "stiff shaft" design which requires that the motor operate below its first lateral critical speed. This requirement is unusual for a two pole motor of a rating this large which would normally be of a flexible shaft design with the first critical speed below operating speed. To achieve a stiff shaft design, special design features were employed consisting of a large diameter rotor and shaft (Fig. 7) with a shorter core length and distance between bearing centers than for the normal flexible rotor design and a corresponding large diameter, short frame structure. This was accomplished by going to the next larger standard frame diameter and creating a new, short frame length. In addition to a larger diameter, this provided a larger air gap and more rugged frame and bearing bracket structures.

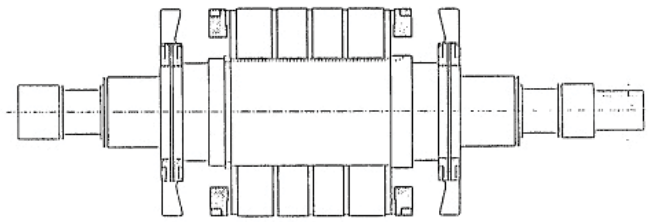

Figure 7 Rotor Assembly

The criteria for a "stiff shaft " design, also referred to as a "stiff rotor" design, as opposed to a "flexible shaft" or "flexible rotor" design, has been discussed at some length in prior papers in the literature [1], [2], [3]. Ref [1] suggests a criteria where the rigid bearing lateral "critical" is 130% or more of the maximum operating speed of the machine. The design for the 3000 HP motor discussed here, based on rotor dynamic analysis including bearing oil film properties and bearing support properties, has a rigid bearing critical well above 130% of maximum running speed. Liberal margins were allowed to assure good vibration performance over the machine operating range based on experience and familiarity with the rotor dynamics software. A conservative design was achieved while yet observing all other normal performance and structural design margins and criteria. The design of the 4000 HP motor was similar except slightly longer with a larger shaft. It had similar design margins and similar vibration performance.

Bearing Selection and Design: Cylindrical bore 4.5 inch sleeve bearings were employed having two oil rings and provisions for flood lubrication. Flood lubrication is required at two pole speeds for this bearing bore diameter. The bearings are near standard self aligning bearings except for minor modifications to the external dimensions to achieve interchangeability with a tilting pad bearing which was offered as an alternate design.

In the initial project stages, a choice of either a standard cylindrical bore sleeve bearing or a tilting pad bearing was offered. General preference was for the cylindrical sleeve bearing which is a rugged and proven design, although analysis showed that the tilting pad bearing might offer an advantage for adjustable speed operation. It was concluded that either design would provide satisfactory vibration performance with low vibration throughout the operating range as was later proven out by test. However, to allow an option for change at any time, a tilting pad bearing was also designed which is completely interchangeable with the sleeve bearing. Complete rotor dynamics and performance analyses were run with this bearing as well as the sleeve bearing, and one set of tilting pad bearings was built and tested. The tilting pad bearing is a four pad design with load between pads except utilizing only the bottom two pads and employing two oil rings along with the normal flood lubrication. The oil rings will provide lubrication during coast-down in the event of loss of flood lubrication. Where coast down time is short, a four pad bearing without oil rings could be supplied and would have sufficient lubrication for coast down without oil rings.

Rotor Dynamics: Rotor dynamics analyses were carried out using the ROTBRG rotor bearing system program which is described in some detail in [4]. In this program, the rotor is discretized as a number of beam elements with specified length, diameter, shear and inertia properties. These individual elements are then assembled by the finite element method to form the rotor system mass, damping and stiffness matrices. Bearing properties which are speed dependent, and several foundation levels beyond the bearing can be included. After the system matrices are formed, either a forced response or natural frequency analysis can be performed.

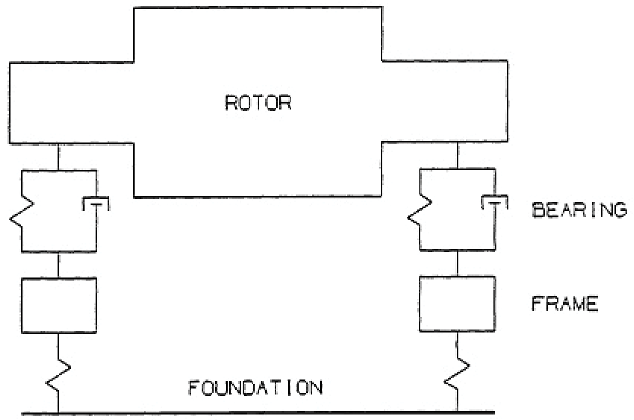

Figure 8 Schematic of Rotor Bearing System

Damped synchronous response to unbalance analyses were run for each bearing type using unbalance distributions to excite the first two modes of the rotor, the cylindrical bouncing and conical rocking modes respectively (Fig. 9). This was accomplished by locating unbalance equal to four times the allowable residual

unbalance, 4UB, as defined below on the shaft at each end of the rotor, first "in phase" for the cylindrical mode, then "out-of-phase" for the conical mode. Stiffness and mass values for one foundation level beyond the bearings were included representing the motor bearing support structure (Fig. 8). No damping was included for the foundation structure since this is negligible with the only significant damping coming from the bearings. The unbalance response plots approximate the vibration amplitude vs. speed curves for actual motor coast-down tests run with the same unbalance values and locations used in analysis. Experience has shown that this is the best tool for predicting motor vibration performance. Undamped calculations were made to obtain the rigid bearing critical speed (6500 RPM) which is independent of the bearing type used. Undamped calculations were also made to obtain rotor mode shapes (Fig. 9).

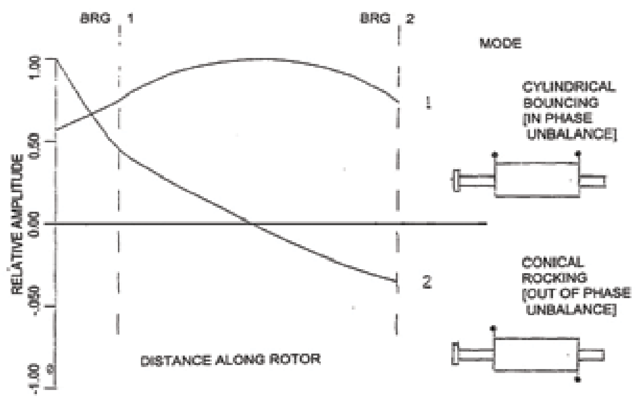

Figure 9 Rotor Mode Shape at Critical Speed

The first mode unbalance response plot for the sleeve bearing, shown in Fig.10, shows resonance peaks at 5000 RPM and 3200 RPM. Although the latter speed, corresponding to horizontal resonance, is within the operating speed range, it is very highly damped having an amplification factor of 1.37. This gave smooth and stable operation even when operating at the resonance speed as shown by test. The unbalance response orbit plot in Fig. 11 shows the maximum vibration amplitude independent of machine axis. Calculated shaft vibration is absolute in all cases.

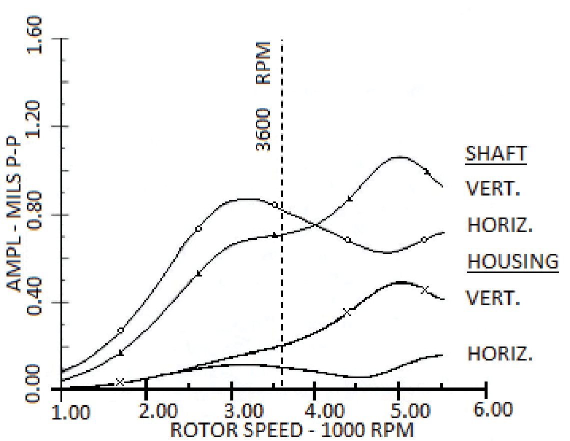

Figure 10 Calculated Unbalance Response - Sleeve Bearings

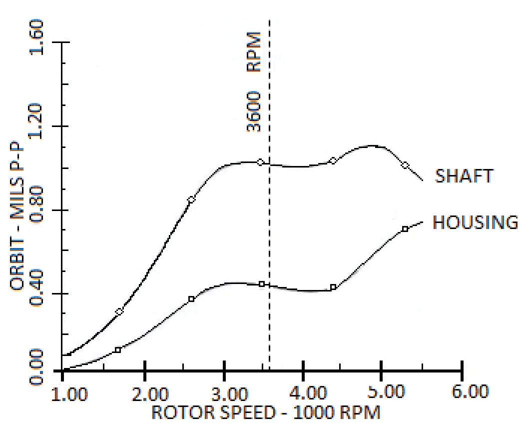

Figure 11 Calculated Unbalance Response Orbital Plot - Sleeve Bearings

The effect of damping on rotating system performance has been recognized and included in the API requirements for "critical speed" for pumps and centrifugal compressors which state in effect that a resonance peak is not considered a critical speed when the amplification factor is below 2.5 and therefore does not require the specified separation margin from operating speed. This criteria is not specifically stated in API 541 for motors, but the third edition provides for machines which do not comply with the separation margin when a well-damped resonance can be demonstrated by an unbalance response coast down test with deliberate unbalance added equal to 4UB. For this test for two pole 60 Hz motors the allowable limit for shaft vibration from 0 to 120% of rated speed is 2.7 mils. This requires customer approval, which was given for this 3000 HP motor except with an agreed upon shaft vibration limit of 1.5 mils from 1400 to 3600 RPM and a limit of 2.0 mils from 3600 to 4320 RPM (120% speed).

The calculated first mode unbalance response plot for the tilting pad bearing is shown in Fig. 12. This shows a single response peak at 4800 RPM which is well above operating speed and approximately corresponds to the vertical response peaks for the sleeve bearing, although showing slightly lower speed and less damping. The single response peak is achieved by using a four pad configuration with load between pads. This gives equal bearing stiffness in the horizontal and vertical directions. Since the horizontal and vertical foundation stiffnesses are also equal, the two response curves are superimposed, giving a circular orbit.

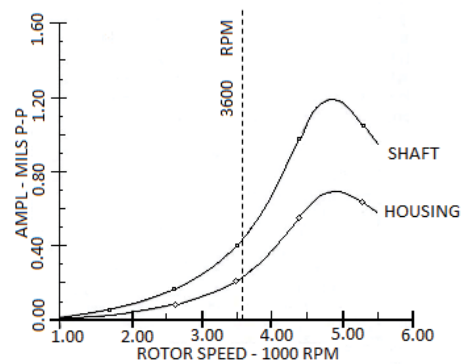

Figure 12 Calculated Unbalance Response - Tilting Pad Bearings

Vibration Tests

Figures 13 and 14 below show vibration vs. speed curves obtained during motor coast-down from overspeed for an unbalance response test with deliberate rotor unbalance. Rotor unbalance weights equal to four times the allowable residual unbalance (4UB) per API 541 were applied to each end of the rotor, where UB = 4 W/N ounce inches, W is the journal loading or one-half the rotor weight and N is the maximum normal operating speed in RPM, in this case 3600 RPM. The motor was brought to 120% speed (4320 RPM) and allowed to coast down while measuring vibration on the shaft and housing. Shaft vibration was measured with four proximeter probes located at 45° on either side of vertical on both ends measuring shaft to housing relative vibration. Housing vibration was also recorded at both ends in the

vertical, horizontal, and axial directions. Data is shown for one end of the motor only, which were the highest readings, but otherwise are typical of both ends.

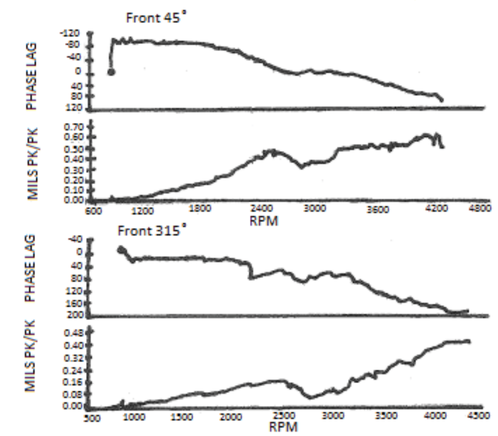

Figure 13 Coastdown Test - Shaft Relative Vibration Runout Compensated for in Phase Unbalance

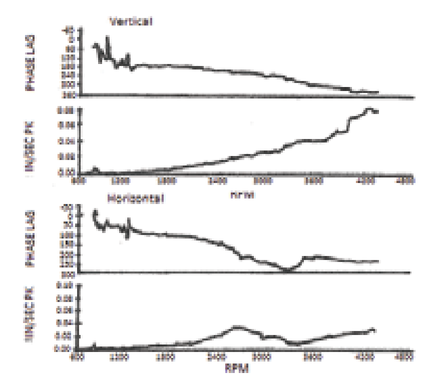

Figure 14 Coastdown Test - Housing Vibration for in Phase Unbalance

Shaft vibration amplitudes during the coast-down tests were all within .6 mils, well within the limit of 1.5 and 2.0 mils given above. Since shaft probes measure shaft to housing relative vibration, these measured values should be comparable to the difference between calculated shaft and housing values. Referring to figures 10 and 11, this difference is about .6 to .7 mils or below, showing test values about equal to calculated values.

Complete vibration readings as tabulated above in Table I were obtained at load and temperature for 60 Hz sine wave operation and under AFD operation over the range from maximum frequency of 60 Hz to minimum frequency of 23.4 Hz. In comparison with limits given in API 541 third edition, shaft readings were well within the limit of 1.5 mils, and housing readings essentially met the limit of .1 inches per second over the speed range. Vibration versus frequency readings were also obtained at no load under sine wave power for this sleeve bearing motor and for another unit tested with the tilting pad bearings. Shaft vibration readings from these tests are plotted in Fig.15, along with shaft readings for the loaded motor from the tabulated data above. Three observations of interest can be drawn from these curves: (1) Vibration is very low with both types of bearings over the entire speed range. (2) The sleeve bearing motor has only slightly higher vibration under load than at no load, indicating little or no thermal unbalance. (3) Vibration is somewhat lower for the machine with tilting pad bearings, particularly at lower frequencies, as is predicted by analysis.

Measured vibration for the 4000 HP motor under coupled load test at 60 Hz showed maximum values of .09 in/sec on the housing and .75 mils p-p on the shaft. Readings at reduced frequencies on AFD were equal to or lower than the 60 Hz values.

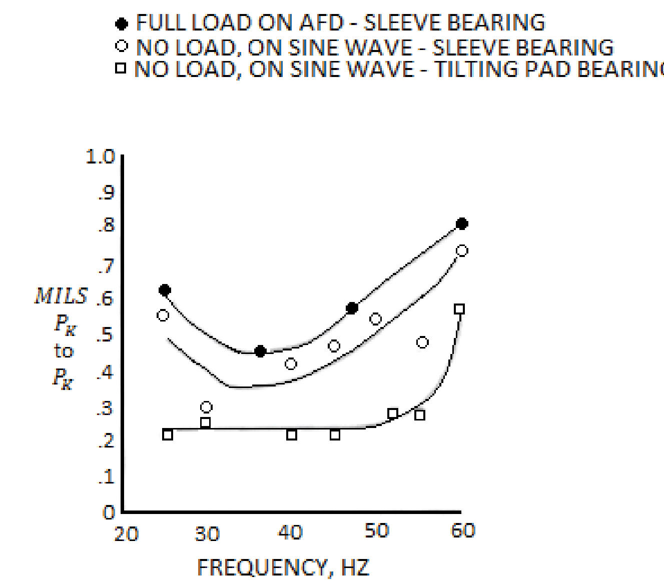

Figure 15 Measured Shaft Vibration

Discussion

Although the electrical design of these AFD motors is complex because of the many demanding and sometimes conflicting requirements, the most special aspect of these motors is the mechanical design relating to the rotor dynamics and vibration performance. The many factors which can affect vibration under variable speed are discussed below including rotor balance, motor bearings, motor structure, and electromagnetics.

Rotor Balance

Rotor balance involves the entire rotor structure which is made up of a multitude of parts including the shaft, rotor laminations, rotor bars, end connectors, retaining rings and fans (Fig. 7). These many items must be controlled in design and manufacture in compliance with the electrical design to achieve stable, precision balance. A stiff shaft is required to prevent major balance changes with speed due to shaft deflection such as may occur with a flexible shaft. Rotor punchings must be precision manufactured with close concentricities and must have a shrink fit on the shaft which is maintained at all speeds and operating temperature conditions. With the large shaft and limited rotor outside diameter, this affects rotor bar size and shape, ventilation holes, and lamination material (see Fig. 2).

The end connectors require retaining rings of high strength material with proper interference fit. Rotor laminations are stacked square with the core, uniformly pressed, and clamped in position when shrunk on the shaft to prevent movement with speed change. Rotor bars are shimmed and then swaged so that they are tight in the slots. End connectors are 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 a thousandth of an inch. The rotor is prebalanced without fans and then the fans are assembled and final balanced on the rotor. All balancing of two pole rotors is done in a high speed balance machine at maximum continuous speed which is 3600 RPM for this motor.

Tests were made on the first few rotors in the high speed balance machine to observe any change in balance with speed. It was found that any bar looseness resulted in excessive change in balance with speed and that this could be corrected either by swaging or by tighter shimming of bars as shown by Fig. 16. Final rotors were made with both tighter shims and swaged bars. As it was, slight change in balance with speed still occurred, but was not excessive. There are many factors affecting balance in this non-solid, complex structure which must be controlled in addition to furnishing a stiff shaft to obtain good balance for variable speed machines. It should be noted that shims around the rotor bars such as used here allow the bars to be driven tightly into the slot without concern of having the laminations shear pieces of the bar off causing bars to be loose, particularly opposite the driven ends. This design also prevents the bars from becoming loose over time due to a similar phenomenon which could occur during the heating and cooling cycle where the bars may not expand and contract at the same rate as the core.

Thermal unbalance was not a problem for these machines, as discussed previously, which can be attributed to a stiff shaft and short rotor core along with other good practice listed above. Also, coupling unbalance in test was not a problem. This was due to rotor balancing with a half key completely filling the keyway and use of a stepped key completely filling the keyway in test, along with a precision balanced spacer coupling.

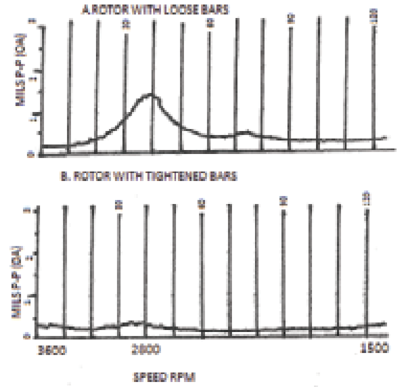

Figure 16 Rotor Coastdown in Balance Machine before and after Tightening Bars

Bearings

Factors in bearing analysis and selection relating to motor vibration are discussed in detail above. In the final analysis the sleeve bearing was selected based on rotor dynamic analysis and experience, knowing it would give low and stable vibration along with high reliability. Even though a resonance occurs within the operating speed range, it is indeed highly damped and not a concern. Tests confirmed the performance showing that the bearing is a little more highly damped than calculated. Tilting pad bearings would also give satisfactory service based on both analysis and tests and have been used successfully in other applications, but not with the extent of experience possessed for sleeve bearings.

Motor Structure

A possible contributor to excess vibration is structural resonance. This is particularly true under variable speed where a design has previously been applied at constant speed. Resonances which are passed through in starting or stopping of constant speed machines may occur in the operating range of a variable speed machine. Also, when a new frame length is designed, slight changes in resonance frequencies may occur. In this design three such occurrences were experienced. An axial resonance in the bearing brackets near 7200 RPM was observed which was corrected by stiffening the bracket design. The bottom plate also showed resonance at higher frequencies which was controlled by stiffening. A problem also arose in the field where the auxiliary conduit boxes showed high vibration. This was corrected by increasing the size of the box support. Resonances of secondary structural members or components not only affect the individual part, but cause increased vibration of the total motor structure. Testing of the motor and drive together in the factory is needed to check vibration at all speeds. Aside from these problems, the generally rugged frame and bearing bracket structures contributed to successful vibration performance.

Electromagnetics

Significant electromagnetic factors in these machines are a large air gap, moderate flux densities, two circuit stator winding, uniform air gap, low rotor runout and generally concentric and symmetrical magnetic parts and assemblies. These all contribute to minimizing magnetic unbalance forces and excitations and corresponding vibration.

Conclusion

In conclusion, proven, reliable designs with low vibration can be offered for large 2 pole motors which are to operate on AFDs through a wide speed range. These motors can be built using the more standard sleeve bearings when properly designed. These motors will operate well, giving low vibration over the entire operating speed range as required and provide generally reduced vibration levels below the 60 Hz speed. Therefore, this confirms the successful application of the combined stiff-shaft sleeve bearing design for adjustable speed drives.

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