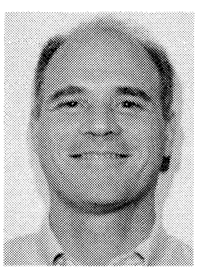
HIGH SPEED ELECTRIC DRIVE APPLICATIONS—AN OVERVIEW

by
John C. Rama
Commercial Marketing Manager
Robicon
New Kensington, Pennsylvania
and
Miles Griggs
Senior Systems Engineer
Siemens Energy & Automation
Alpharetta, Georgia

John C. Rama is Commercial Marketing Manager for Medium Voltage Drives with Robicon, in New Kensington, Pennsylvania. Before this position, he spent seven years at Ansaldo Ross Hill as Manager of Applications Engineering, and North American Product Manager for Ansaldo Motors. Mr. Rama spent 17 years in the large motor and drives industry, beginning his career with assignments in applications engineering at Louis Allis and Toshiba, before joining Ansaldo Ross Hill.

Mr. Rama received a B.S.M.E. degree, graduating with honors, from Milwaukee School of Engineering (1980). He has authored several articles and technical papers and has made presentations to the IEEE, EPRI, EASA, and the Vibration Institute, concerning large motors and electric drive systems.



Miles Griggs is a Senior Systems Engineer with Siemens Energy and Automation. His current duties involve business development support for the gas transmission marketing group within Siemens Oil & Gas Systems Business Unit. Before this position, he was Manager of Project Engineering with Ansaldo Ross Hill for six years. He has been involved in project management and system design, since joining that organization in 1988. Those projects have included blower, fan, pump, and compressor variable frequency applications for power and pipelines.

Mr. Griggs received his degree from the University of Houston (1988). Before that, he completed four years of U.S. Army duty. He has also authored a paper delivered to the IEEE on a high speed motor application for the gas pipeline industry.

ABSTRACT

Worldwide, current technology has opened a wide range of new applications for electric motors, many of which have traditionally employed mechanical (turbine) drivers. While the use of electric motors for these high speed applications offers tremendous potential for improved efficiency and productivity at reduced cost, it also presents new engineering challenges. The reasons will be presented why high speed electric drivers—both induction and synchronous—have become viable alternatives to traditional mechanical drivers. The technologies and operating issues involved in such applications will also be reviewed.

INTRODUCTION

The Basics

Since 1889, alternating current electric motors have proven to be reliable and efficient power conversion devices. Their operation and maintenance are routine and uncomplicated. Electric motor dependability and adaptability are such that approximately 60 percent of worldwide electric power generation is consumed by AC electric motors. Furthermore, the utilization and importance of AC electric motors stand to increase with the coming of the 21st century. The advent of reliable, cost effective adjustable speed drives has transformed the applications for these simple machines.

Adjustable frequency drives, which regulate motor speed by controlling input power (voltage and frequency), have vastly enlarged the envelope of AC motor applications and capabilities. This can be done without adding gears, clutches, or other equipment to the drivetrain. AC motors can now be employed in applications that have traditionally utilized mechanical drivers, such as gas turbines and combustion engines. These new applications offer many benefits, but also new engineering challenges. While 3600 rpm, two-pole motors have occasionally received consideration from the turbomachinery industry, new high-speed machines driven at frequencies well above 60 Hz require much of the industry's acquired mechanical sophistication.

OPPORTUNITY AND RATIONALE

Many of the first "high-speed" electric drive systems were installed in the power generation industry, primarily on boiler feedwater pumps, utilizing synchronous motors (with LCI drives) not dissimilar to the turbine-generators that are commonplace. Since the deregulation of the pipeline industry in the U.S. in 1990, many electric driven stations have been built. More recently, high-speed electric machines have found their way into applications within refining operations, replacing steam turbines.

There are three areas where electric drivers offer advantages over mechanical/combustion alternatives:

- Environmental impact
- Economics
- Operations

Environmental Impact

The elimination of local emissions is an obvious positive change. Throughout the world, there is a growing emphasis on controlling emissions, and increasing governmental oversight and regulation.

Here in the U.S., for example, sources having annual NO_x/VOC emissions greater than 25 tons, are required to report their emissions to their state environmental agency [1]. This is the approximate level generated at a compressor station with just 200 hp of reciprocating engines [2, 3]. A 7000 hp gas turbine, without benefit of BACT or RACT compliance options, is estimated to emit 150 tons/yr of NO_x. In ozone nonattainment defined areas, lowest achievable emission rate (LAER) technology may be required, of which the electric motor driven system is the prime existing example.

EPA limitations on emissions can make new installation licensing a burdensome exercise, even an impossibility in some areas. Yet saleable emissions credits may be available for operators adopting a strategy of integrating electric motor drive systems (LAER) into overall retrofit and expansion planning. Hence, the incorporation of electric drivers as part of the mix offers flexibility and benefits to an entire operation.

In addition to air emissions, the other environmental issue that favors electric drivers is noise. This is of special importance to pipeline operators who often operate in areas of high population density. U.S. regulations (and those in many other countries) and local ordinances define firm limits on the sound levels that are measured at each commercial/industrial property line. The typical electric drive system is 7-10 dB(A) quieter than any of the mechanical driver alternatives.

Economics

Electric motors hold an economic advantage over mechanical drivers in several important respects. While each of these issues could be the subject of an entire study, they will be quickly reviewed:

- Four to five percent improvement in system thermal efficiency, compared with single-cycle gas turbines and steam turbines (even greater advantage compared to reciprocating engines)
- Capacity is unaffected by external air temperature
- Automated electrical control allows unmanned operations
- Significantly lower annual maintenance cost
- Reduced initial equipment and installation cost
- Ability to consider strategic fuel source options

The efficiency of a typical natural gas driver might be 30 percent or less. Taking into account the efficiency of the average natural gas electric utility generating station, electricity transmission losses, and the losses of a typical electric motor and related controls, leaves the total efficiency of the electric drive system four to five percent greater than that of the typical mechanical alternative [4].

While gas turbine and engine capabilities vary inversely with air temperature (making operation more costly and complex during warm weather conditions), the electric drive system is not impacted by ambient temperature changes and can operate at constant capacity.

Along with capacity upgrades to existing operations, modernization is bringing automation and smart control systems to optimize efficiency, throughput, and cost conservation. The accuracy and reliability of current design medium voltage speed controllers can be an aid to such automated control of remote operations.

Annual maintenance costs for electric driven pipeline stations (where large numbers are available for comparative analysis) have proven to be substantially less than those at stations with turbine or engine drivers (Table 1).

The initial cost of an electric and adjustable speed controller, even one of the "new" high-speed electric machines, is well below the cost of a turbine driver (approximately half the system capital cost). Extrapolated life cycle costs for a typical 5000 hp pipeline

Table 1. Comparison of Annual Maintenance Cost.

Comparison Of Annual Maintenance Cost At Pipeline Compressor Stations With Alternate Drive Systems [3]	
Gas Engine Driven	\$65 - \$100 / hp / year
Gas Turbine Driven	\$30 - \$ 40 / hp / year
Steam Turbine Driven	\$30 - \$ 40 / hp / year
Electric Motor Driven	\$7 - \$ 10 / hp / year

compressor driver, assuming operation for 50 percent of the year, show turbine driver costs to be more than nine times greater than electric driver costs, and reciprocating drivers have costs 16 times greater [5].

Because of favorable seasonal and daily load factors that make pipeline operations ideal for electric utility load shaping, these operations will become favorite targets for electric utility marketing efforts. Favorable rates are already available to most operations and the coming deregulation of the power generation supply will provide even greater opportunities to negotiate favorable electric "fuel" costs, without consuming the product to be delivered.

Operating Issues

While electric drivers are nontraditional and electric power in the past was viewed as the "competition" by gas companies, there are currently more than 170 installations of medium voltage "high-speed" electric drivers, representing more than 2 1/4 million hp of load throughout the world. These range up to 55,000 hp and up to 20,000 rpm. Some of these installations have been operating for more than 15 years (APPENDIX A). In many cases, the improved availability, reduced maintenance, and increased operating speed range of electric drivers has won over the critics in nearly every installation where they have been applied.

An updated comparison of various drive systems as compiled by a major gas/energy company is shown in Table 2.

Table 2. Comparison of Drive Systems.

Parameter	Reciprocating Engine	Turbine Driver	Electric Driver
Total Installed Cost	1.00	0.65	0.52(1)
Space Requirement	1.00	0.75	0.65
Maintenance	1.00	0.60	0.15
Spare Parts	1.00	0.60	0.40
Environmental Cost	1.00	0.80	0.10
Availability	96%	98%	99%
Fuel Efficiency	31%-40%	25%-40%(2)	35%-45%(3)

Notes: (1) Inclusive of substation costs. (2) Simple cycle, includes turndown factor. (3) Includes power plant conversion and transmission losses.

It is important to note that while most of these applications are driving centrifugal loads with a classic quadratic torque *vs* speed characteristic, the density of compressor gases may vary significantly at a given installation, dependent on process, temperature, product mix, etc. Therefore, the operating envelope of the drive system must be broadened to cover all of the expected operating conditions. As will be seen in some of the case examples that follow, failure to recognize or communicate the full range of operating requirements has led to some difficult field problems. With some designs, this is more critical than with others, but as with any system, a genuine understanding of the requirements is essential to success.

EQUIPMENT DESIGN

The development of high-speed electric machine technology has been more evolution than revolution. Electric motors and generators have seen more than a century of use in industrial applications. Adjustable speed controls date back more than 70 years. The first turborotor technology was applied for synchronous power generation in the 1930s. Large high speed synchronous motors were applied with power electronic based adjustable speed control beginning about 1980. High power, high speed, induction motor applications became part of the mix in 1990.

The development of adjustable speed controllers has evolved with the advent of new, improved, and lower cost power switching devices, from transistors to thyristors to GTOs and IGBTs. The evolution of switching devices has allowed higher power applications with low harmonics and high efficiency. The continued development and increasing competition have reduced costs and improved reliability. Drives have actually become simpler, because the new devices allow new topologies with fewer components, much in the way large scale integrated circuit chips have improved size, reliability, and costs of computers and electronic appliances.

High-speed electric motors became possible with the evolution of the adjustable frequency controls. Nevertheless, the design of such motors has borrowed heavily from techniques developed for other turbomachinery applications. Using the methods proven by compressor and turbine designers, motor engineers have been able to model and analyze the rotational dynamics of higher-speed rotors. The cylindrical rotor of a high-speed electric motor is a simpler mechanical element than that of any compressor or turbine, without the significant aerodynamic complications. Nonetheless, the rotor of an electric motor is a very large rotating mass. Yet, for a given power rating motor size, it is inversely proportional to the speed.

Utilizing the experience gained from turbine-generators, motor manufacturers have time-tested techniques for rotor construction, cooling, and balancing. As in a parallel world, the electric problems of eddy-current reduction have been resolved by employing techniques similar to those applied to resolution of bearing oil film breakdown problems.

Stators and insulation systems do not differ from those of standard motors. Rotors are cylindrical, fabricated from a one-piece forging, or assembled from steel laminations. Shaft/rotor metallurgy and heat treatment must be carefully chosen to minimize any residual stresses that could contribute heat or load dependent deformation (rotor bowing). The use of laminations or shallow circumferential grooves in the rotor periphery breakup skin effect eddy currents. Stainless steel retaining rings are generally used to contain the end ring, or end winding connections, against outward movement due to centripetal forces. The size/speed combinations are currently limited by the strengths of the materials.

The approximate size (horsepower) and speed combinations available are shown in Figure 1. Not all suppliers will manufacture all combinations shown on this chart. Because of differences in construction and materials, manufacturers have varying limitations.

Most of the motor designs utilize a base and pedestal construction that gives the greatest flexibility in selecting bearings and support stiffness. Bearings can be hydrodynamic (cylindrical, lobed, or tilting pad type), or magnetic. The use of magnetic bearings on a high-speed motor, combined with magnetic bearings and gas seals on a compressor, offers complete elimination of any lubrication system for the drivetrain. Such a system reduces cost, complexity, parts count, and the environmental costs inherent with handling of lubricants.

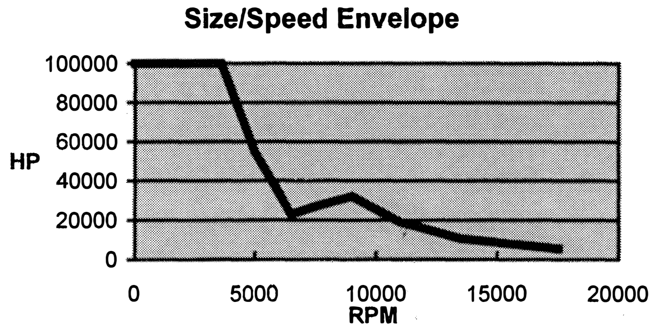


Figure 1. Size/Speed Envelope.

Virtually all these machines are blower-cooled to provide constant airflow and constant cooling over the entire speed range. The concurrent elimination of shaft or rotor mounted fans further simplifies the rotordynamics. In Division I hazardous areas, ducted air from a nonhazardous source can be supplied to cool and pressurize the machine.

Rotordynamics analysis and support system modelling are used to avoid operation at a critical speed. Typical charts of such design steps are shown in Figures 2, 3, and 4. The example shown is a motor operating between its second and third lateral critical speed. For a complete description of the analysis and design issues of this particular example, refer to the literature [6].

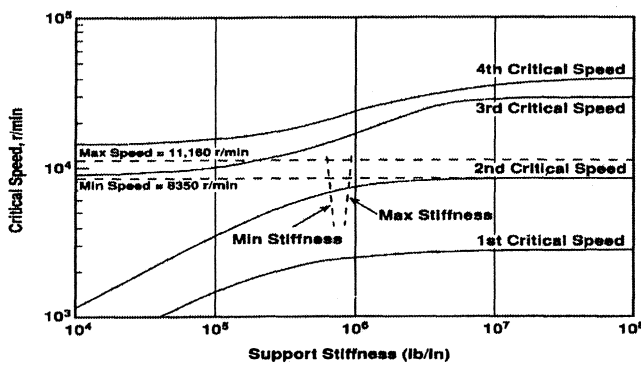


Figure 2. Lateral Analysis Undamped Critical Speed Map.

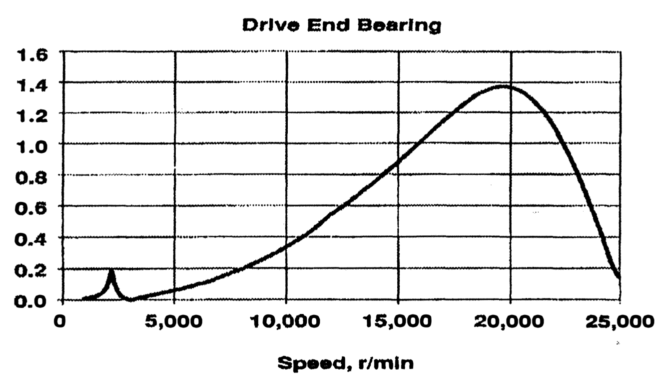


Figure 3. Calculated Rotor Unbalance Response—First Mode.

A torsional analysis of the system would normally be performed, but it is important to realize that many of the current adjustable speed drive designs produce significant torsional excitations. In the example shown in Figure 5, the excitation frequencies include integer multiples of the 6× electrical frequency from the six pulse inverter, in addition to the normal 1× running speed, and 1× and 2× electrical excitations.

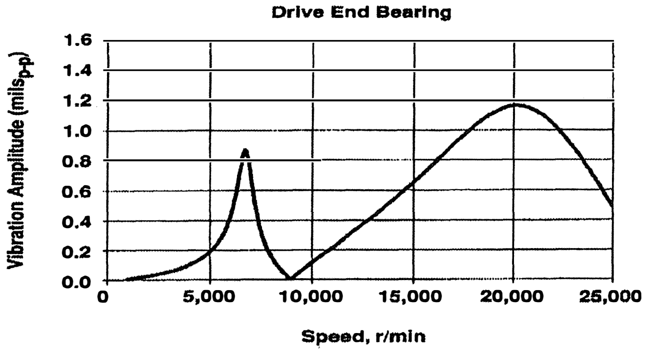


Figure 4. Calculated Rotor Unbalance Response—Second Mode.

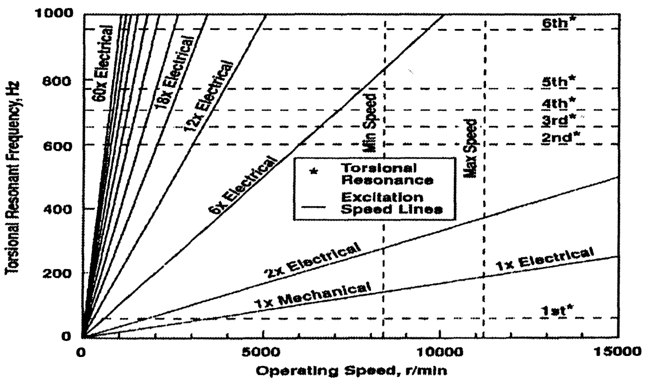
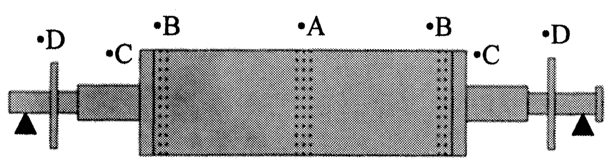


Figure 5. Torsional Resonance Interference Diagram.

Because machines like the one shown in Figure 5 are subject to multiple vibration modes, the balance is critical. Balancing must be performed in multiple planes to minimize unbalance in any of the possible modes. A sketch of a typical unit is shown in Figure 6.



- Seven Balance Planes
  - Central Plane (Low Speed Balancing) - A
  - Two Lateral Planes (Low Speed Balancing) - B
  - Two Internal Planes (High Speed Balancing) - C
  - Two External Planes (High Speed Balancing) - D
- Low-Speed Balance, High-Speed Balance + Trim Balance Over Speed Spectrum - Check Balance Hot and Cold

Figure 6. Seven Balance Planes.

Following are some case examples to illustrate the reasons for employing high-speed electric drive systems and some of the possible problems. As with any new application, mistakes have occurred and lessons have been learned.

Case Example 1

A refinery in California had to revise some processes to produce the new reformulated gasoline required by recent legislation [6]. Unfortunately, the revised process required additional steam, which was not available without adding a new package boiler. The addition of a new package boiler would entail a lengthy application process with no guarantee of approval, not to mention major expenditures. As an alternative, a project to replace some existing steam turbines with electric motor drive systems, combined with other steam saving efforts, freed enough steam to meet the needs of the reformulation project, in a shorter time and at lower cost.

The motors used for this application were 3500 hp, 11,160 rpm, directly replacing the existing steam turbines, mounted on the top deck of a three-story structure. A more conservative approach using conventional two or four pole motors with speed increasing gear assemblies would have required extensive modifications to the structure and foundation, and rework of large diameter hydrogen piping. Hence, the rationale for employing high-speed electric systems was the avoided cost and longer schedule of the more expensive alternatives.

These motors are used in the example rotordynamics curves shown in Figures 3 and 4. Again, a more detailed description can be found in the literature [6].

During testing, the motors experienced oil leaks from the bearing housings. The bearings were tilting-pad type, custom designed for the application. When oil flow was reduced to minimize the leakage, subsynchronous vibration was observed. Cooperation between the user, bearing manufacturer, and motor manufacturer resulted in new tilt pads, machined to tighter tolerances; rerouting of the oil flow more directly to the bearing pads to improve development of the oil film; and by adding an air purge to the outer seal on each bearing.

One of the most important requirements of an electric drive system is the ability to ride through any input voltage disturbance. Without this capability, the driver is subject to trips and shutdowns due to very normal, momentary line disturbances. The user of these systems specified very comprehensive testing, including tests of the "ride-through" capability, to minimize installation and startup time at the refinery, and to provide maximum confidence in the equipment once installed. During this testing, it was realized that these very high speed motors were designed with very low magnetic flux levels, to minimize iron core losses at the high operating frequencies. Unfortunately, that feature of the design also allowed these motors to produce potentially harmful voltages during the voltage sags. A saturating reactor was added in parallel to the motor and output filter. The reactor has lower saturation characteristics than the motor, limiting the possible self-excitation voltages to acceptable levels.

Because the adjustable speed drive topology is vulnerable to very fast load swings, a compressor surge event could cause a drive trip and drive component failure. A collaborative effort between the drive system supplier, the user, and the manufacturer of the surge detection system, produced a coordinated scheme. This scheme actively responds, upon detection of imminent surge, to move operation away from the surge line, while the adjustable speed controls temporarily assume a self-protective operating mode.

To date, these systems have been in operation more than two years with excellent reliability and performance. The potential problems were identified during the design and testing stage and resolved through joint action of the engineering staffs of all parties involved.

Case Example 2

An order for two induction motor drive systems for gas transmission service [7, 8] is reflected in Case Example 2. The systems were the first commercial units provided by the motor manufacturer for service at high speed. The rating of each motor is 7000 hp at 5500 rpm. The customer, to assure themselves of satisfactory design before shipment, required full-load, full-speed testing of the machines. This was made possible by connecting the two machines back-to-back, using one as a motor and the other as generator, with the variable frequency drives providing the appropriate loading.

The back-to-back tests quickly indicated a problem: motor shaft vibration levels measured at the bearings were low (less than 1.0

mils peak-to-peak) at the start of a full-load run, but climbed steadily to very high levels (greater than 4.0 mils peak-to-peak) within a few hours. The vibration primarily consisted of the 1× component. Sensitivity checks were made to determine what might have the strongest influence on the vibration. Lube oil temperature was varied. Lube oil flow was varied. Machine loading was varied. While all these had some effect, the most influential effect was that of loading. This led to the conclusion that the rotors had thermal sensitivity, since temperature rise is most closely related to loading in electric machines. If the thermal behavior of the machine was repeatable, then there was a possibility of finding a balance condition for the rotor that gave acceptable vibration performance in hot and cold conditions. After three balance runs, the same type of behavior was present, with the vibration vector having changed to a new nondetermined direction. New balancing strategies were devised and used, with no improvement. The thermal sensitivity was repeatable, but it could not be deterministically countered with balance weight placement.

After more than 20 balancing runs, it was decided to try to reduce the thermal sensitivity of the rotors directly. The means available to them were to increase the heat rejection at the rotor surface with more effective cooling, and to reduce one of the loss mechanisms on the rotor surface eddy currents. The solution chosen, circumferential grooving, served both purposes at the same time. This technique has been employed successfully for years in large turbine-driven generators. At the same time, new forgings were ordered for the machines, specifying metallurgical composition and heat treatment that would improve the thermal consistency of the rotor material. This was done in the interest of schedule, in case grooving was unsuccessful in bringing the vibration down to reasonable levels.

Grooving proved successful and an immediate improvement in the vibration performance was achieved after the modification. Vibration reduced to approximately 1.3 mils peak-to-peak, after achieving full machine temperature. The machine performance was accepted, and the units were shipped to site, where they performed satisfactorily for three winters of peaking service.

In their third year of service, one of the units began to show elevated vibrational characteristics. Attempts were made by a third party to improve the behavior with balancing techniques, with no improvement. The motor manufacturer offered one of the new forgings as a replacement for the poorly performing rotor. During the next winter season, the station was called into service under high station loads. Because the motors were under torque limitations via current limit controls in the variable frequency drives, the motors were unable to reach the speed demanded by the unit controller. The drive manufacturer was called to site to examine what was perceived to be a problem with the units. In an effort to enable operations to meet the load demand at the site, the drive manufacturer allowed for an increase in the current limit at lower speeds, which would give the motor more torque delivery at lower speeds. Shortly after this modification was made, the motor vibration on the new replacement rotor began to increase.

No conclusive analysis has been performed, to date, which suggests the reason for this behavior. It appears, however, that once the rotor vibration begins to increase, nothing restores the vibration behavior to its original state, which suggests that permanent changes occur when the rotor is operated with rated torque at less than rated speeds. The difference between operation at 100 percent speed and lower speeds, say, 75 percent speed, should normally not cause more rotor heating in an externally ventilated machine. The variable frequency drive current waveform for this type of drive (filtered current source) contains a higher percentage of harmonic current at less than rated frequency. The reason for this is that the drive utilizes capacitive filtering to reduce the harmonic current content in the motor. The capacitive filter serves as both a low pass filter and a necessary element for inverter switch operation. In general, the higher the frequency, the better job the capacitive filter does at filtering the harmonic currents that the motor would otherwise have to absorb. At lower operating frequencies, the capacitive filter offers less filtering, but what is more important, at lower frequencies and higher loads, the capacitive filter does not provide the conditions for inverter "natural commutation" (switching by circuit voltage bias), and extra harmonics are introduced in the motor as other circuitry operates to provide "forced commutation." It is possible to have fifth harmonic current levels in the motor operating at 75 percent speed and rated fundamental current that are twice those found when the motor is operating at full speed and load. This creates more harmonic eddy current and rotor bar loading than would otherwise be present at higher operating frequencies. It is quite possible that this loading created more localized heating than could be cleared by external cooling means. As of this writing, the problem still exists, and the only known method of maintaining the vibration at acceptable levels is to reduce the torque limits on the machine to keep the rotor harmonic current loading down to loads acceptable for continued operation.

Case Example 3

This next example involves two more motors for gas transmission service. The rating is 8800 hp at 4200 rpm, continuous to 5200 rpm. These motors also were purchased with a back-to-back test as part of the contract. A spare rotor was also purchased. The coupling was specified for a hydraulic fit, so that it could be removed and placed on the spare rotor, if required at a later date. Vibration levels were low, approximately 0.6 mils peak-to-peak.

During commissioning, high drive-end bearing temperatures were noted on one of the machines. Both machines were running well, with vibration levels just as low as those recorded during factory testing. After oil flow to the bearings was checked and corrected, the problem persisted in one machine and not the other. Finally, it was noticed that the location of the bearing babbitt RTD on the machine with the high reading was axially located closer to the coupling end of the bearing than the RTD on the other machine. It was surmised and later verified that each bearing babbitt was reaching similar temperatures during operation. The bearing babbitt RTD closest to the coupling end of the machine reflected a temperature rise associated with thrust. The drive-end bearings had not been designed for high levels of thrust, and the calculated levels of thrust were higher than the rating of the nominal thrust faces that were provided, especially for the oil flowrates that were provided to the bearings. The motor manufacturer had designed the drive-end bearing to be a locating bearing, with small axial clearances that were much less than the nominal axial travel and expansion expected for the compressor. As a result, the axial load on the motor drive-end bearing was usually higher than necessary for nominal temperature operation. Some attempts were made at finding an axial alignment that would reduce the thrust loading on the bearing, but the overall thrust of the compressor had too much variability. An adjustment to the bearing axial clearance had to be made. The clearance was increased from ë1 mm to ë10 mm, and a new axial alignment was chosen to minimize the axial loading. Some concern over the rotordynamics of the system was expressed by the customer. It was not felt that the modification would have a significant influence over the rotordynamic behavior of the motor. After modification to the bearings was made, the bearing temperature readings returned to normal under all operating conditions.

An operational glitch occurred as a result of a unit shutdown unrelated to motor performance. The shutdown turned out to be unintentional, and a unit start sequence was immediately initiated. The unit start sequence involves a wait of more than five minutes while the lube oil rundown tank fills to its preset level. During that wait, the cooling system blower continues to run. The first time this occurred, the motor tripped on a vibration alarm while ramping through the first critical speed at approximately 1800 rpm. A consultant was brought in to help ascertain the problem. His findings were that the phenomenon was repeatable, and that the severity of the vibration depended on length of time that the motor sat still—an immediate restart without delay gave no increase in vibration through the first critical, while a wait of minutes gave significant increase, and a wait of an hour gave no increase in vibration through the first critical. This was a different slant on thermal sensitivity. This type of problem was not easily explained, except in general terms. The motor rotor at standstill was being unevenly cooled by the blower. Because of the uneven cooling and the high temperature gradients associated with the hot rotor, a transient thermal bow develops, which creates an effective unbalance in the rotor. After a long enough period of time, the temperature gradients in the rotor dissipate, and the thermal bow disappears. An immediate restart does not give the rotor time to develop significant temperature gradients, and the motor spinup reintroduces uniform cooling to the rotor surface. The solution to the problem was to implement a slow roll sequence to the unit start logic that was based on time from the last stop. If the unit start request came within a half-hour from the last stop, the variable frequency drive was requested to slow roll the machine at 1600 rpm for 10 minutes, determined from experimentation at the site. After this modification to the control logic, the motor exhibited no more radial vibration problems.

Another application concern appeared after a number of months of operation: the customer noticed significant axial movement in the coupling, on the order of 100 mils peak-to-peak. The vibration was not constant, coming and going with rhythmic regularity. The frequency of the vibration, when it occurred, was approximately 5 Hz. The compressor thrust vibration was very low by comparison. Of course, the customer was not inclined to believe that the vibration could come from anywhere other than the motor, and the drive and motor manufacturers were equally disinclined to believe that an axial vibration could be initiated within the motor. Quick calculations indicated that the 5 Hz component of vibration coincided well with the natural frequency of the motor mass with the coupling axial spring constant combination. Attempts to correlate known excitation mechanisms with the vibration proved futile. Since no cause could be found that identified a means of correction, the coupling manufacturer was consulted to determine whether the coupling was in imminent danger of fatigue failure. The coupling fatigue limit was approximately 10 million cycles with the observed displacements, so the units could afford to continue running. Weeks later, when the units were not running, and the compressor case was pressurized and exposed to the line pressures, the customer noted a sound he described as pounding coming from the compressor case, very similar in rhythm to the vibration onset rhythm experienced by the motor and coupling. A qualitative judgement was made that the previously noted vibration was pipeline induced. No other action has been taken since that time, and no failures have been reported.

Case Example 4

These were two units for gas transmission service with a need for quick delivery. The motor rating is 6700 hp at 6000 rpm. Extensive testing was not purchased. The machines ran very well from a vibrational performance standpoint. However, the bearing rigid labyrinth seals performed poorly. The seals provided as standard with the purchased bearings were made of aluminum. The motor manufacturer replaced the aluminum seal with brass seals of their own manufacture, to reduce the potential for sparking in the classified area with seal-to-shaft contact. The location of the drainage holes for oil return in the innermost labyrinth stages, however, were too high to allow sufficient drainage. The grooves in the seals would fill with oil under normal operation and oil flows. The problem was resolved by modifying the drainage hole size.

Case Example 5

This is a single gas transmission compressor unit. The rating of the machine is 12,800 hp, with constant power capability between 6600 rpm and 7800 rpm. The variable frequency drive for the machine has a filtered 12-pulse output, which creates a very low harmonic excitation for the motor. During manufacture of the rotor for this machine, one of the retaining rings, a critical element for successful high speed operation, was found to have a poor fit that heavily influenced its vibration in the upper end of the speed range (>7200 rpm). This defect was found after the entire rotor had been assembled and underwent high-speed balancing, where a successful balance configuration could not be found. This posed a significant problem, because the retaining ring could not be replaced without removing the integrally mounted coupling hub, introducing some risk to the shaft. For the sake of the schedule, the motor manufacturer offered to send the motor to site, and to manufacture a new rotor for field retrofit at a later date. Once in place, the unit performed well, with a limited speed range. The replacement rotor was installed, when it became available, and the unit continues to run well.

Case Example 6

These are two gas transmission compressor units with machines rated 12,000 hp at 6950 rpm, with constant power capability out to 7300 rpm. The machines were purchased with a back-to-back test. Shortly after the beginning of back-to-back load testing, one of the motors failed catastrophically with the rotor bars melting in the center of the rotor core. The conclusions drawn at the time, regarding the meltdown, were that the cooling circuit for the motor was insufficient. The blower sizing was marginal, and the ducting arrangement (three 90 degree bends) introduced too much pressure drop. The other machine sustained no visual damage, and due to schedule pressures, was sent to site for commissioning. Due to the failure of the one machine, testing was limited and very few data were available to verify the performance of the machines. The failed machine was reconditioned with new stator coils, and a new rotor was manufactured. The failed rotor was reconditioned with new rotor bars, rebalanced, and delivered as a spare rotor.

During commissioning of the first machine, the motor vibrational behavior was found to be poor during stop decelerations in the speed range of the first lateral critical speed (approximately 1600 rpm). In the operating speed range, however, the vibrational behavior of the machine was good. Motor plant engineers treated the problem as a transient thermal sensitivity, and asked the drive manufacturer to implement a higher deceleration ramp rate for planned shutdowns. This would serve to minimize time spent operating in the critical speed range during planned unit shutdowns, and, therefore, the maximum vibration levels. This produced acceptable results under normal operating conditions, but it did not help with unplanned conditions, such as system trips.

The motor manufacturer offered to attempt to improve the overall performance of the machine by changing the balance of the machine, striking a compromise between operating range balance and critical speed coastdown balance, to improve the coastdown vibration performance (by compromising the balance in the operating speed range). Their attempts at balancing were thwarted, however, by vibrational behavior that continued to change with each successive balance run. Vibration began to climb in the operating range during this activity. The behavior of the machine changed rapidly after each start, suggesting that the rotor cage, with its low thermal mass, was creating an unbalance when loaded. The rotor was pulled, and unexpectedly found to have significant rotor bar radial movement (protruding out of the slots).

The reconditioned rotor involved in the back-to-back test was now available and was installed. Initial vibration behavior was acceptable, but with vibration levels much higher than those recorded on the test stand. A design review of the motor that concentrated on the rotor design was undertaken. Interleaved in the design review was a failure analysis of the failed rotor. At this same time, commissioning of the second machine, with a virgin rotor in it, was performed. Performance was acceptable from the first run, with very low vibration in the operating speed range, and good vibration response transiting the first critical speed range.

Design Review—The motor design team reviewed all aspects of the rotor design. They were looking for a problem that could cause thermal stresses and/or mechanical stresses in the rotor cage, such that rotor bar extrusion may occur. Electromagnetic design and loss distribution calculations were reviewed for errors and a comparison of the design of this machine with that of a very successful similar design at another installation (Case Example 5). The two designs had comparable current densities, losses, and centrifugal forces. A finite-element parametric analysis of the rotor surface eddy currents, and temperature distribution at simulated operating conditions, indicated that the rotor was designed properly from an electromagnetic point of view. Review of the rotor mechanical design details included a review of the cooling circuit, a finite-element stress analysis on the rotor bars, the operating interferences between materials, and the dimensional tolerances. None of the mechanical design details were, in any way, different from that of the other successful installation.

Failure Analysis—A visual inspection of the rotor surface indicated that rotor bar movement was nonuniform in the radial direction along the length of the rotor body, corresponding to locations on the rotor surface where air velocity was the lowest. Axial movement was nonuniform, as well. The rotor bars appeared to have been restricted from further axial movement toward the end of the rotor. Mechanical tests on the rotor bar sections indicated that the strength of the rotor bars were greatly reduced from their original levels, and especially weak in sections where the bar had begun to extrude. Chemical analysis verified that the alloy was proper. The best indication of what actually happened to the rotor bars, however, came from microphotographs of the rotor bars, prepared and interpreted by an independent materials laboratory. These data indicated that the material had undergone thermal fatigue, for two reasons:

- The crystalline structure of the material had been modified to that of a sample that had been heated above 300%C. This temperature is far above the annealing point of the material, which in turn is far above the design operating temperature.
- The microfractures of the material went across grain boundaries, eliminating the possibility of creep (mechanical slip under stress) as a mode of failure.

This led the design team to conclude that the failure was related to an initiating event, such as the back-to-back test, as the cause of rotor bar overtemperature, fatigue, and ultimate failure.

The newer unit was run, logging as many hours as possible, in advance of the cold weather season, to seek out any problems. Within a short period of time, the vibrational behavior of this machine began to worsen, suggesting that operating conditions may be responsible for the changing rotor behavior, rather than a single test event. By early this year, the machine failed catastrophically, with the rotor bars in the center of the rotor having extruded enough to rub against the stator. As of this writing, a happy ending to this story has yet to be written.

Some facts known about this machine are worth exploring. A study of the logs of the operating units indicated that the vibration on the machines began to climb after the machine was run at rated torques, but at less than rated speed, for a number of hours. When the rotor was pulled shortly after the vibration began to increase, the color of the rotor surface sealant was not uniform as expected, but was darker at one end of the rotor than at the other. This suggested nonuniform air flow in the machine (and nonuniform cooling), while total actual air flow measured for the machine exceeded the manufacturer prescribed level. The resulting behavior of these machines resembles what was discussed in Case Example 2, although on a more accelerated failure timeline. The specific adjustable frequency drive configuration for these machines contains greater harmonic current loading at high loads and reduced speed. It is almost impossible to test such a hypothesis in the field, because the thermal effects in the machines have shown to be irreversible. It should be possible, with finite-element modelling, to demonstrate a direct correlation between the rotor bar and rotor steel temperature with different levels of harmonic current injection; however, such analysis has not been performed to date.

Case Example 7

These three very compact, integral motor/compressor units for gas transmission service are rated 4000 hp, 10,000 rpm. This type of integral unit is quite different from the previous cases, representing a very different kind of machine. The motor and compressor are built on a common shaft, with the motor cooled by the natural gas stream, and magnetic bearings actively to control the machine rotordynamics. The only mechanical problems that the units have had are related to loading. The per unit loading has sometimes been greater than the variable frequency drive was designed to provide. When the station is started up without any existing head, high loads due to stonewalling are present. Because this condition was not understood or precisely defined at the beginning of the contract, it remains an application problem.

CONCLUSIONS

High speed electric drive systems have much to offer industry. There are already a significant number of installations (a partial installation list is located in APPENDIX A). Much has already been learned by the manufacturers of this equipment, as can be seen from the case studies presented herein. Where problems have persisted, there has been an incomplete definition or lack of understanding of the operational requirements and parameters. One lesson is clear: the applications must be understood by the motor and drive system supplier, in terms of the required torque *vs* speed requirements of centrifugal compressors. Early evidence suggests that some machine designs exhibit behavior that is sensitive to these requirements. A full understanding of the mechanisms of these sensitivities is necessary to move this technology forward toward general acceptance. The drive/motor/compressor assemblage must be treated as a "system," and sized/selected to match the needs of the application. The maturity level of the gas turbine market has not been reached, where standard packages with predefined operation envelopes are available. The traditional "arms-length" relationship between compressor and motor manufacturers must evolve to promote closer association and joint development. Some participants in the marketplace have already begun to move in this direction. Motor design engineers have learned, and borrowed greatly from, the mechanical design tools and techniques of this industry, and are beginning to develop some.

APPENDIX A

INSTALLATION LIST OF HIGH SPEED ELECTRIC APPLICATIONS

Quantity	HP	RPM	Type	Year
2	16100	5100	Synchronous	1980
2	16000	5100	Synchronous	1980
1	25000	4500	Synchronous	1982
1	25000	5200	Synchronous	1982
1	15000	5100	Synchronous	1983
2	13400	6000	Synchronous	1983
4	3600	5900	Synchronous	1985
19	12900	6000	Synchronous	1985
19	10700	5520	Synchronous	1986
1	21000	5100	Synchronous	1986
1	15300	5100	Synchronous	1986
1	24000	5100	Synchronous	1986
1	24000	6060	Synchronous	1986
1	21000	5100	Synchronous	1986
1	17400	6400	Synchronous	1986
1	7100	6000	Synchronous	1986
1	22800	6060	Synchronous	1986
1	22800	4850	Synchronous	1986
1	3200	6000	Synchronous	1987
2	17400	5700	Synchronous	1988
2	10200	5200	Synchronous	1988
3	27000	5400	Synchronous	1989
3	15800	5000	Synchronous	1989
2	15000	4750	Synchronous	1989
1	9000	4655	Synchronous	1990
18	13400	6000	Synchronous	1990
2	20100	5500	Synchronous	1990
3	26700	5400	Synchronous	1990
1	8000	10000	Induction	1991
2	10200	5200	Synchronous	1991
3	15800	5050	Synchronous	1991
2	14900	4750	Synchronous	1991
18	17400	5800	Synchronous	1991
2	7000	5500	Induction	1991
2	8800	5200	Induction	1992
2	15000	4750	Synchronous	1992
1	14700	5800	Synchronous	1992
1	24100	5500	Synchronous	1992
1	31000	4300	Synchronous	1993
2	51000	4200	Synchronous	1993
1	8000	4655	Synchronous	1993
1	2000	20000	Induction	1993
1	600	6800	Induction	1993
2	900	15000	Induction	1993
3	3500	11160	Induction	1994
2	6700	6000	Induction	1994
2	7000	6000	Induction	1994
2	47000	4500	Synchronous	1994
5	55000	3750	Synchronous	1994
4	16100	5150	Synchronous	1994
1	16360	4650	Synchronous	1994
1	16500	4650	Synchronous	1994
1	38000	5000	Synchronous	1995
2	12000	7300	Induction	1995
1	12800	8000	Induction	1995
4	4500	10000	Induction	1995

REFERENCES

1. Clean Air Act Amendment of 1990, United States Government, Washington, D.C. (1990).

2. Hewson, T., Stamberg, J., Thumb, S., "NO_x Compliance Options: The Use of Electric Motors to Replace Reciprocating Engines at Pipeline Compressor Stations," EPRI Chemical & Petroleum Office Draft Report (March 1994).

3. EPA, "Alternative Control Techniques Document—NO_x Emissions From Stationary Reciprocating Internal Combustion Engines," 435/R-93-032 (July 1993).

4. Kempers, G. and Oliver, J. A., "Electric Utilities Look at Gas Pipeline Source," IEEE Winter Power Conference Proceeding (1994).

5. Oliver, J. A. and Poteet, D., "High-Speed, High-Horsepower Electric Motors for Pipeline Compressors; Available ASD Technology, Reliability, Harmonic Control, IEEE Winter Power Conference Proceedings (1995).

6. Wood, B., Olsen, C., Hartzo, G., Rama, J., and Szenasi, F., "Development of an 11,000 RPM, 3500 HP Induction Motor and Adjustable Speed Drive for Refinery Service," IEEE IAS Petroleum and Chemical Industry Conference, pp. 55-63 (1995).

7. La Grone, S. C., Griggs, M. C., and Bressani, M., "Applications of a 5500 RPM High Speed Induction Motor and Drive in a 7000 HP Natural Gas Pipeline Installation," IEEE PCIC Paper 92-42, (1992).

8. Bressani, M., Odorico, M., and Sica, M., "High-Speed, Large-Power Induction Motors for Direct Coupling to Variable-Speed Gas Compressors," Proceedings of the Fifth European Conference on Power Electronics and Applications, EPE , pp. 366-371 (1993).

BIBLIOGRAPHY

Billinski, G., "Electric-Driven Compression: A Cycle of Opportunity," Presentation to the "Power for Compression into the Next Century, the Electric Option" Symposium, Houston, Texas (September 1994).

Bird, R. L., "Replacement of a Gas Turbine Compressor with an Electric Motor Drive in a Natural Gas Compression Station," Presentation to the "Power for Compression into the Next Century, The Electric Option" Symposium, Houston, Texas (September 1994).

Clifton, S. "The Electric Viewpoint," Presentation to the "Power for Compression into the Next Century, The Electric Option" Symposium, Houston, Texas (September 1994).

Department of Energy, "Emissions of Greenhouse Gases from Use of Transportation Fuels and Electricity," Washington, D.C. (November 1991).

Duncan, N., "Houston and the 1990 Revisions to the Clean Air Act," Center of Public Policy, Publication 93-09, University of Houston, Houston, Texas, (March 1993).

Energy Policy Act of 1992, United States Government, Washington, D.C. (1992).

Fagg, J. P., "Technology Options for Electric Compression," Power for Compression into the Next Century, The Electric Option" Symposium, Houston, Texas (September 1994).

Foster Natural Gas Report, "Columbia Cleared to Install Electric Motor-Powered Compressor Units in Maryland to Appease Residents Environmental Concerns" (December 1993).

Graham, A. D. and Wimhurst, M., "The High-Speed Oil Free Intelligent Motor Compressor," Proceedings of the Fifth European Conference on Power Electronics and Applications, EPE '93, pp. 384-389 (1993).

Hare, R., "Emission Control Regulations—Now and In the Future," Presentation to the "Power for Compression into the Next Century, The Electric Option" Symposium, Houston, Texas (September 1994).

Odegard, B., Stulz, C. A., and Steimer, P.K., "Industrial Application of Variable Speed Drive System for High Speed in Megawatt Power Range," IEEE (1989).

Oliver, J. A., Samotyj, M. and Ferrier, R., "Application of High Speed, High Horsepower, ASD Controlled Induction Motors to Gas Pipelines," Proceedings of the Fifth European Conference on Power Electronics and Applications, EPE '93, pp. 430-434 (1993).

PG & T Staff, "Unique Compressor Passes Field Test," Pipeline & Gas Journal, International Pipeline & Utility Operations, Design & Maintenance (October 1992).

Rama, J., "Supplemental Guidelines for Use with API Standard 541 When Specifying Motors for Use with Adjustable Speed Drives," IEEE IAS Petroleum and Chemical Industry Conference, pp. 81-88 (1993).

Solar Turbines, "Gas Turbine Systems for Compression and Mechanical-Drive Applications," Bulletin Number T76/1294/10M (1994).

Wickman, W. E., "Finding the Economic Niche—Dollars and Sense," Presentation to the "Power for Compression into the Next Century, The Electric Option" Symposium, Houston, Texas (September 1994).

Wrenn, Jr., K. F., "Electric Powered Compression at Columbia Gas Transmission," Presentation to the "Power for Compression into the Next Century, The Electric Option" Symposium, Houston, Texas (September 1994).