ELECTRIFICATION OF ALKYLATION PROCESS WITH HIGH SPEED MOTOR DRIVE SYSTEM IN A REFINERY

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Abstract — Directly coupled, variable frequency driven electric motors are replacing turbines driving process compressors due to advances in increased efficiency, precise process control and quick return on investment. In the oil and gas industry, high speed motors directly coupled to centrifugal compressors are also filling the need for more megawatts at higher speeds. In applications with demanding structure and space limitations, the smaller motors are the driver of choice when compared to turbines or conventional two pole motors with speed increasing gears. Motor manufacturers have developed specifically designed motors for these and other demanding applications instead of using higher-speed 2 pole or 4 pole standard motors. For the discussed application reliable and optimized motor designs, especially the motor frame, rotor-bearing, enclosure, bearing shields and cooling, are designed to meet stringent vibration and noise requirements, which are defined in specifications such as API, IEC and ISO. This paper discusses design and testing of 3750 kW induction motor with oil film bearings at a speed range of 3880-5300 rpm to replace an existing steam turbine in an United States refinery.

Index Terms — Structural dynamics, flexible foundation, rotor dynamics, bearings, high speed, electric motors, vibration, API, oil film bearings, direct drives, induction motor, solid rotor.

I. INTRODUCTION

Reducing operating costs, increasing energy savings and minimizing the carbon footprint are leading initiatives in today's industrial processes toward precise process control, higher efficiencies, lower total cost of ownership and reliable systems. The driver choice for large fans, pumps and compressors in the oil and gas industry is shifting to electric drive systems to meet environmental emission requirements. Each industrial process and application is unique to site and process conditions that lead to engineered to order (ETO) solutions. The biggest bang for the buck could be achieved if the ETO is designed based on standard, available technology and products. This would reduce lead times for engineering, manufacturing, procurements and spare parts. When selecting an individual component to integrate into an industrial drive system, the user has a significant opportunity to improve the overall performance and reliability of the package by properly selecting components that work together at the system level.

A high-speed motor that is operating on an adjustable speed drive (ASD) with a wide operating speed range offers significant flexibility to meet process requirements. The motor design can be optimized to fulfill two kinds of interface requirements; one is for the end user and the other is for the mechanical design. The end-user requirements include the foundation and cooling, and the design requirements include the rotor, enclosure, and mechanical and dynamic parameters.

Foundations are already available for revamping projects and often have size restrictions. A foundation in general - especially for revamping projects usually does not meet the high standards for foundation stiffness that is required for high-speed motors. The motor must be designed so that it operates well on a wide range of foundations according to the stiffness parameters. Instead of specifying a special foundation, the motor design should be flexible and able to be adapted for various foundations. This must be integrated in the motor design if specified by the customer. This paper discusses the concept, design and testing of a 3750 kW high speed motor that was commissioned by a refinery to drive a compressor with a ASD.

II. ELECTRIFICATION OF THE COMPRESSION PROCESS

The decision to select an electric drive compression over traditional steam or a gas turbine drive compression depends on various factors, including,

- Availability of electricity on site
- Flexibility in current and future process requirements
- Increase process efficiency
- Meeting emissions requirements
- Required training of maintenance personnel

Higher efficiency and availability and less drive train maintenance make the electric compression more attractive over steam or gas turbines. Initial costs involved with the electric compression design complexity and engineering customization may be higher than traditional compression. . However, the return on investment could be realized within short period of time due to cost savings coming from increased process efficiency, reduced maintenance costs and higher availability. Appendix A shows a decision matrix for various drivers. For this alkylation unit, return on investment was realized within two years after installation.

III. APPLICATION, SITE & DESIGN SPECIFICATION

In this application, a 3750 kW drive motor was required with a specific power, torque and speed range to replace the steam turbine driving a process compressor. A variable-speed electric motor with a power rating of 3750 kW and with a speed range of 3800-5300 rpm was built and tested. The compression application also demanded a constant torque for a speed range of 3800-5150 rpm and constant power for a speed range of 5150-5300 rpm. Besides these two operating points, the drive system required two duty points during startup at 1000 rpm and 2400 rpm. The customer specification also required having oil film bearings instead of magnetic bearings.

After the site inspection, it was concluded that the compressor and motor could be mounted on the non-massive foundation. The foundation structure is a part of a three-floor concrete building with the motor and compressor located on the top floor that is 9 m above the ground level. Due to the existing process and machinery layout requirements, the motor was partly overhanging the building with the non-drive end supported by the steel cantilever supported by cross beams as shown in Fig.1. The foundation structure was located at the seismic zone and at ocean front. Since the compressor had critical speed at 3100 rpm and 10,400 rpm, motor rotor-bearing were required to have critical speed not matching these speeds to avoid the interactions between compressor and motor rotors critical speeds.

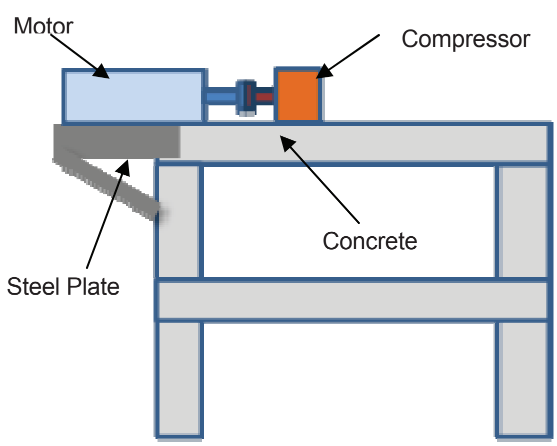

Fig 1 Schematic of Site Location of Motor

A) Motor Specification

Power 3750 kW; Voltage 4.8 kV; Stator Insulation: Class F
Area Classification: Class 1 Division II; Duty Cycle S1
Ambient Temp: 40° C; Bearings: Oil Film Bearings
Specification: API 541

To ensure the highest motor reliability, industry standard API 541[1] was specified for the application. API standards clearly define design criteria for the separation margin between critical speeds and the operating speed range. They also define the maximum allowable shaft and bearing housing vibration levels. Besides vibration level, API 541 [1] also states that the lateral natural frequencies of the motor, which can lead to resonance amplification of vibration amplitudes, shall be removed from the operating speed frequency and other significant exciting frequencies by at least 15%. Motors intended for continuous operation with adjustable speed drives shall meet this requirement over the specified speed range. API 541 standard also defines the criteria for factory testing of the motor for degree of residual unbalance and thermal stability of the rotors. In order to have thermal vibration stability, API 541 standard states that the magnitude of vibration vector changes from no-load to rated temperature shall not exceed 50% of values of the maximum allowed vibration. To ensure bearing reliability, API 541 standard states that the bearing temperatures measured with bearing temperature detectors shall not exceed 93°C (200°F) under rated operating.

IV. HIGH-SPEED MOTOR CONCEPT

In order to fulfill the application, site and design requirements as discussed in section III for the motor application, the design concept was selected from large high-speed electric motors that were successfully applied for the power ratings from 3 MW-15 MW. The motors operated in the speed range from 4000 rpm up to 15000 rpm, driving compressors equipped with magnetic or oil film bearings [2-3]. This high-speed drive systems concept has been successfully applied to onshore projects as well as offshore platforms where the machine foundation was not as massive as specified in API standard [1]. To increase reliability of drive system, the high-speed motors were designed so that the foundation dynamics along with rotor dynamics and structure dynamics of the motor were considered as a system when designing the enclosure and motor frame. This lead to the vibration design which was suitable for the site foundation stiffness requirements. This section discusses some of the basic requirements of designing components for high-speed motors.

A. Rotor Design

When designing high-speed motors, two design areas were first considered the rotor and the bearings. The rotor was critical because it rotates at a high circumferential speed and must withstand the dynamic forces. It is also subject to high material stress. Further, it was also a part of the motor that generated the unbalanced forces and must ensure stability and serviceability over the service life. The high-speed rotor design employed different design concepts. Each rotor design concept should comply with following fundamental design requirements

1) The shaft should never change its modal properties over its lifetime
2) The rotor should never change its residual unbalance over its lifetime
3) The rotor should be rugged to withstand the centrifugal forces, with respect to external influences, such as brief overheating or very cold starting conditions
4) The rotor should have low losses (electrical as well as the higher friction losses due to high surface speed)

All of these requirements must be compliant, because the bearings, irrespective of which rotor technology is selected, require stable rotor parameters to fulfill their role in ensuring low vibration levels. This is especially true for specifically designed high-speed oil bearings, which are optimized to meet vibration requirements over the complete speed range. In addition to the requirements listed, the rotor should be easy to manufacture, machine, repair, service and purchase.
Various technologies are available to realize an electric motor shaft design. For the drive systems or high power applications in the range of 3 MW to 15 MW, operating at between 4000 rpm and 15,000 rpm with magnetic and sleeve bearings. The three most common rotor designs for electric motors are:

1) Induction motor with laminated rotor design
2) Synchronous motor with permanent excited rotor design
3) Induction motor with solid rotor design

The solid shaft design utilizes a technology where the copper used for the rotor squirrel cage and the steel body of the shaft as shown in Fig. 2 is bonded on an atomic base (where the copper atoms are fused into the steel at an atomic level) so that the shaft can be considered as one piece. Another solid shaft design employs open milled slots in the shaft without copper. The rotor technology has been tested up to circumferential speeds of approximately 250 m/sec for maximum continuous operations [2].

Fig. 2: Solid Rotor Design

B. Bearing Housing Design

Housings for large, high-speed motors were designed to prevent bearing damage. Bearings were securely mounted. Flanged bearing housings, those with faces that bolt onto the machine frame, gave the most reliable support to the bearings. In machines with hydrodynamic bearings, bearing support stiffness was highest when the center of the bearing intersects the plane of the end shield. That is because it allows rotational forces transfer from rotor to bearing supports via oil film without transmitting vibrations. On the other hand, bearing forces at a distance from the end-shield plane made moment forces that could excite end-shield vibration modes. Bearing housings should be isolated forces from the motor structure, because such forces can induce damaging forces and vibration in the bearing.

C. Housing Design

In order to minimize the influence of the frame dynamics on the bearing and the influence of the electrical excitation the housing vibration is minimized by decoupling the stator core assembly from the housing using a low-pass spring suspension.

D. Cooling Design

In order to have a flexible cooling design, high-speed motors are designed with low pressure resistance housings and internal systems, which require wide cooling ducts. A large volume, low pressure air flow, which suits the axial fan on the shaft as well as external cooling fans that blow air into the motor are usually required for high-speed motors. Induction motors can be designed with a relatively large air gap, which means that the low pressure requirements can be fulfilled. If shaft mounted fans are used, fans with an adjustable pitch are preferable to minimize fan losses. Also, a very low weight is preferable, so that even in the unlikely event of a blade being lost, no significant unbalance occurs and the motor can remain operational. The low pressure cooling system allows axial fans to be used for the self-cooled design, which are located next to the rotor core as shown in Fig. 3. It also allows external fans to be installed for force-cooled designs. This allows a shorter shaft to be used, which can improve the rotor dynamics.

V. OPTIMIZATION OF MECHANICAL DESIGN TO MEET SPECIFICATION

Once the motor concept is established the mechanical design components discussed in section IV are optimized to meet the application and vibration specification discussed in section III. The schematic of the optimized final design is shown in Fig 4. This section discusses the components which are optimized to meet application and vibration specification.

Fig.3 Variable Pitch Axial Fans

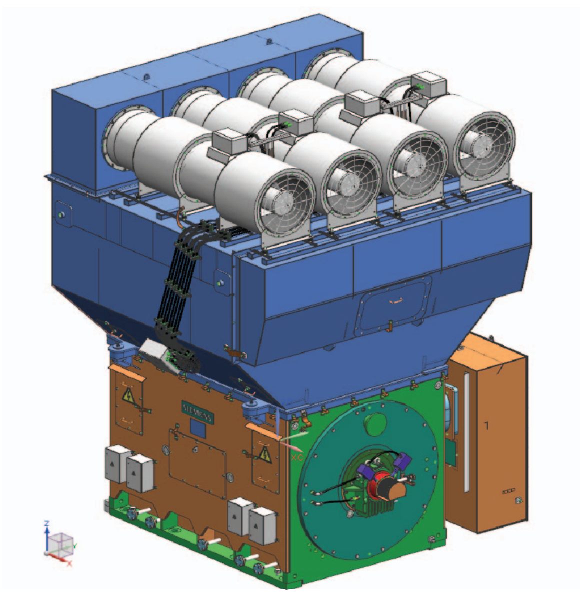

Fig 4: 3750 kW, 5300 rpm Induction Motor

A) Optimized Rotor Design

Good rotor design is a fundamental requirement of low vibration motors. For the compressor speed range, the rotor was designed in such a way that its bending critical frequencies do not fall within the speed range. To meet API specifications critical speeds were at least 15% removed from an operational speed range. The rotor bending natural frequencies depends on the bearing span (distance between the bearings) and shaft diameter. A shorter bearing span or higher shaft diameter increases the bending natural frequency. For a motor with variable speed and torque, shaft diameters and bearing span are optimized to meet power ratings, speed range, rotor dynamics and cooling airflow.

B) Optimized Oil Film Bearing Design

A good rotordynamic design is required for reliable and efficient rotating machinery. In order to meet compressor speed requirements, for variable-speed motors operating at high speeds, the rotor bearings are designed to have critical speed which will have a separation margin from the operating speed – or will be well-damped in order to prevent an amplification of the vibration level. Bearings at the rotor supports play an important role regarding the location of the critical speed and damping of vibration. API specifications also limit the bearing temperature for babbitt metals. For most applications, conventional hydrodynamic fluid film bearings with special geometry – such as two-lobe, three-lobe, four-lobe and tilt-pad bearings – can be optimized to fulfill the rotordynamic criteria for the particular speed range. In these applications a variable-speed electric motor with a power rating of 3750 kW and a speed range of 3800-5300 rpm is required to meet special power and torque requirements. In order to fulfill API criteria regarding vibration values and the requirement relating to rotordynamic separation margin, an optimized 4-lobe bearing provided the best rotordynamics solutions over 3 lobe, 2 lobe and tilt pad bearings. The bearings were designed to provide the required stiffness and damping for the rotordynamic separation margin. The 4 lobe bearing has better stability over 2 lobe and 3 lobe bearings as it has higher oil film stiffness in horizontal and vertical direction. Besides rotordynamics and vibration considerations, bearing design and lubrication systems had to be optimized to meet API temperature limits for bearing babbitt metals.

C) Rotordynamic Design Per 541 Standard

The finite element rotordynamic program was used to calculate the natural frequency and forced vibration response of the induction rotor. This rotordynamic model consists of the rotor modeled as beam elements with gyroscopic effects. Magnetic attraction between rotor and stator was included for the worst case calculation. Oil-lubricated hydrodynamic bearings are also included in the critical speed and vibration response calculation. Bearing housing and support stiffness based on experimental analysis are included in the rotor dynamic model. Both direct and cross-coupled stiffness and damping of the rotordynamic oil film are calculated using a calculation program based on the hydrodynamic theory.

Foundation stiffness for the rotor dynamics model is based on API 541 description of massive foundation. The foundation is considered massive if the vibration amplitudes of the foundation near the machine feet are less than 30 percent of the vibration of the bearing housing. The rotor dynamic model also considers the stiffening effect due to the presence of the copper bars and end rings embedded in the rotor as well as the mass and inertia of couplings and axial fans. Residual unbalances are applied at locations where correction weights would be applied during an actual balancing process.

A damped critical speed & stability analysis is the first step in the rotor dynamic design calculation is to calculate natural frequencies of the rotor-bearing system for the entire speed range and the separation margin required. A natural frequency along with the damping is plotted with respect to critical speed. The natural frequencies associated with oil film and coupling modes are strongly influenced by the rotational speed of the rotor. Hence, the natural frequencies of the modes and associated damping should be evaluated to identify critical and noncritical natural frequencies. API 684 [4] defines the criteria to distinguish between critical and noncritical natural frequencies based on the amplification factor. Natural frequency is noncritical if the amplification factor is less than 2.5 or 20% damping and hence does not require a separation margin. For high-speed motors running on fluid film bearings, it is also important to look at the stability of lower natural frequencies at higher speeds. All the modes should have positive modal damping for the entire speed range including the over speed conditions.

D) Optimized Structural Dynamics Design

Electric motors are a special class of rotating machinery, where there is strong interaction between the rotating shaft and non-rotating structures, such as stators and frame due to the presence of electromagnetic forces in the air gap. These forces are exerted on both the rotor and stator. In order to have low vibration levels, the motor should be free of combined rotor-structural resonance points in the operating speed range. Rotating rotor and non-rotating structures are coupled through oil film forces at the bearing locations and also through the electromagnetic forces in the rotor-stator air gap. Electromagnetic forces generated in the air gap rotate at the line frequency and twice the line frequency. Forces at twice the line frequency can cause ovalization of the stator and frame, which is manifested as vibration and noise. It is also possible that these 2x deformations are transmitted through the frame to bearings. These then result in axial bearing housing velocities that can exceed the specified limit. These vibrations are related to second-order resonance amplification in the stator structure. Bearing housing vibration resulting from electromagnetic forces and stator natural frequencies can be minimized by implementing decoupling springs between the stator and frame. To estimate the vibration behavior and minimize the risk of vibration problems in the field, structural dynamic calculations should be performed, including the non-rotating structures in addition to rotor dynamic calculations. The resonance points of a rotor can be easily identified in the combined rotor-structure model. This is more complicated for the non-rotating parts as a sheet metal construction has many modes, and it is not so easy to filter out the critical ones. The modal mass is one indication of the relevant modes. Only modes with sufficiently high modal mass have to be taken into account for structural vibration purposes. To identify the most critical modes, a forced vibration response calculation is normally required. The most sensitive factor for such types of calculation is the assumption of modal damping factors for the individual modes and exciting frequencies. Comparison of calculation and measured response values leads to modal damping factors of between one and three %. For the forced vibration calculation, the rotor unbalance level should be considered. The calculation should be performed for unbalance forces acting at phase angles of 0° and 180°. In addition to the unbalance forces from the rotor, rotating electromagnetic forces acting on the stator must also be considered for forced vibration calculations.

A forced vibration calculation of the complete rotor-structure showed that the calculated vibration is higher than the API vibration limits. The design of the structure needs to be optimized to meet API vibration requirements. Optimized structure design leads to flange type design of the bearing housing as shown in Fig. 5. Very tight tolerances, precise machining and alignment of stator, frame and bearing housing were required to meet vibration limits for the wide speed range required for the motor.

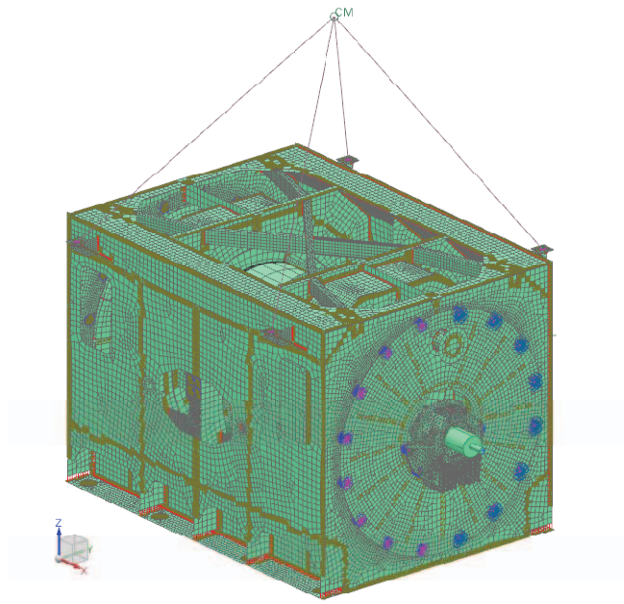

Fig 5: Structural Dynamics Model of the Motor

E) Design Optimization For Non Massive Foundation

The alkylation unit compressor was located on the top floor of the three floor concrete structure. Hence the high speed driver motor was located on the third floor. The half of the motor portion was supported with steel frame which was supported by the steel cross beams attached to the main concrete structure as shown in the Fig 1 and Fig. 6 respectively. The motor system on the field does not satisfy the massive foundation requirement specified by the API 541 specification. Hence it becomes important to evaluate the influence of flexible foundation on the motor vibration. Although there is no straight forward methods available or defined in standards to evaluate such cases. The foundation parameters such as mass, mechanical stiffness derived from FEM calculations are included in the rotor dynamics model as a mass spring element. Inclusion of foundation parameters significantly affects the predicted vibration behavior of the motor. The difference in the vibration amplitude in the calculation arises due to the fact that system's natural frequency may be different between massive and non-massive foundation. If the system natural frequency is close to the operating speed range of the motor, then this will amplify the vibration amplitudes leading to unexpected shutdowns. In order to get more realistic predictions expected at the site conditions finite element calculation of the complete system which includes the foundation and motor structure were performed. The motor and compressor mass and inertia was added to the foundation structure as point mass at the shaft level and they were connected to the structure with massless rigid bars and springs to represent the mounting conditions.

The foundation structures usually have several natural frequencies and modes of vibrations which will occur within the speed range [6]. The mode shape and mass participation factors where used to sort out the vibration modes which have the highest influence on the motor vibration. The response of the foundation at the motor connection locations was calculated for the dynamic forces resulting from the motor unbalance distribution. Based on a dynamic analysis the structure of the motor and foundation base plate was optimized to reduce the vibration. The base plate in the foundation structure was optimized. The motor structure was optimized until the bearing housing response was reduced to less than 2.54 mm/s (0.1 in/s) for complete speed range with non-massive foundation parameters. Figure. B1 and B2 show the shaft vibration and bearing housing vibration after the optimization of the mechanical design. Figure. B1 and B2 also shows that by optimization of mechanical components the location of critical speed and vibration amplitude could be achieved for non-massive foundation.

Fig. 6 : Site Fondations Structure

F) Optimized Cooling Design

A totally enclosed air-to-air cooling (TEAAC) was the customer requirement for the motor. In order to effectively utilize the available air flow for the length of machine, a four-sided ventilation scheme was chosen. The air flow and associated distribution are shown in Fig. 7. The arrows show the path of the air inside the machine. The frame of the machine was designed so that some of the cool air at the inlet can be routed to cool the center part of the stator.

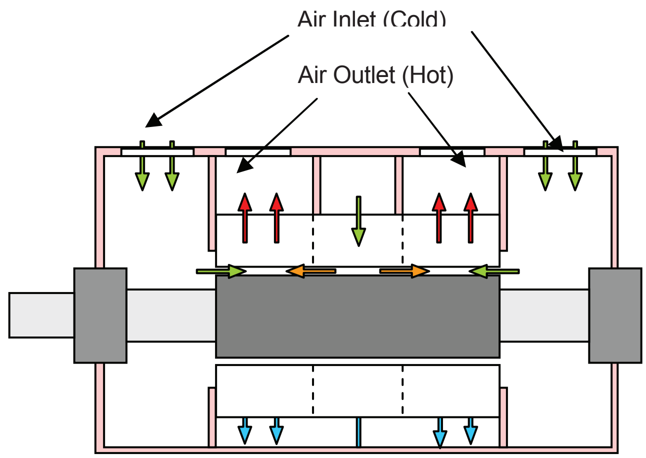

Fig. 7. Airflow Schematic of Machine

VI. FACTORY ACCEPTANCE TESTING

Load coupled testing at a rated voltage under full load was performed to evaluate the machine's electrical and mechanical performance. Vibration and bearing temperature readings were collected per API 541 standard on the test field foundation which is a massive foundation. Machine performance data at rated voltage and nominal speed from the factory test are listed in TABLE I.

TABLE I
TESTED MACHINE PERFORMANCE DATA AT RATED VOLTAGE UNDER LOAD CONDITIONS

	V = 4800 V, f = 88.5 Hz, N = 5300 rpm	
Rated Power	3750	kW
Stator Current	589.8	A
Power Factor	.79	
Efficiency	96.8 %	
Bearing Temperature:		
Drive End	68	°C
Non Drive End	72	°C

In order to verify the location of critical speed and vibration behavior of the rotor over the complete speed range, the unbalance response test was performed as required by API 541. Appendix F shows a mechanical coast down from over speed 6300 rpm to rest for the rotor in its final state, with balance weights applied in phase and with weights moved to new positions 90 degrees from the previous positions (calculation in accordance to API 541, 4.3.5.3 a).The shaft displacement of the rotor in the final state shall not exceed 38 µm peak-to-peak per API 541.

TABLE 2
COLD TO HOT VIBRATION DATA AT RATED VOLTAGE UNDER LOAD CONDITIONS

V = 4800 V, f = 88.5 Hz, N = 5300 rpm		
	Cold	Hot
Shaft Vibration 1X [µm pp]		
DE-Y	6.92	7.58
DE-X	5.22	3.66
NDE-Y	1.31	3.92
NDE-X	4.96	5.09
Housing Vibration 1X [mm/s]		
DE –Hor	0.072	0.252
DE- Ver	0.159	0.257
NDE-Hor	0.283	0.308
NDE-Ver	0.293	0.334

Thermal vector vibration change per API 541 test was also performed to test the sensitivity of the rotor with maximum operating temperature. API 541 allowed 38 µm (0.6 mils) p-p vibration changes from cold to hot condition. TABLE 2 shows the thermal vibration vector change at full load conditions. As seen from the measurements results the influence of the temperature on the vibration is well below API 541 limits. The test result shows that the rotor is thermally very stable and vibration levels were well within the specification.

VII. Installation and Commissioning

The drive and motor was commissioned in early 2013 as shown in Fig. 8. The baseline motor shaft vibration data was collected for the following drive train configurations

1) Uncoupled shaft vibration
2) Coupled no load vibration
3) Coupled loaded vibration

All vibrations were found to be well below the allowable alarm and shut down values. The coupled drive train vibration with motor installed on the site was shown in Appendix E. As observed from figure E1 the vibrations of the coupled drive train is 17 micro-meters (0.5 mils) from peak to peak. Vibration trending collected over several days also shows that the vibration is very stable.

Fig. 8. Installation of Motor in 2013

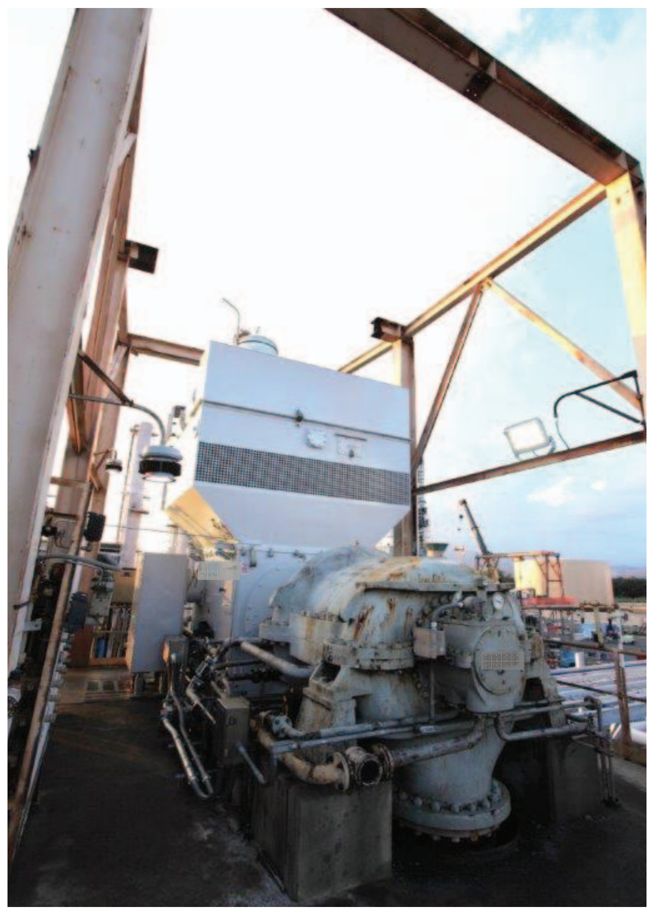

Fig. 9. Motor Coupled to Compressor

VIII. CONCLUSION

The high speed motor driven by adjustable speed drive was applied to replace the steam turbine driving compressor in an alkalization unit. The high speed induction motor design with massive rotor was selected to increase mechanical reliability. Mechanical components such as bearings, frame and bearing housings of the motor were optimized to reduce the vibration amplitude at the customer site. The advanced rotor dynamics and structure dynamics were performed to predict the vibration behavior on the site conditions. Based on the unbalances forces from the drive train, foundation design was optimized to reduce the vibration of motor and foundation. The measured data shows that vibration behavior could be optimized for placement of critical speeds and vibration amplitude for non-massive foundation.

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X. VITA

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APPENDIX A: Decision Matrix for Choosing Prime Mover

Decision Factor	Steam Turbine	Variable Speed Electric Drive	Fixed Speed Electric Drive
CAPEX	Higher (Comparable in very large systems)	Lower (Comparable in very large systems)	Much lower
OPEX	Much higher	Lower	Much lower
Installation	Longer, higher cost	Simple, lower cost	Fast, much lower cost
Availability	Lower than VSDS, less frequent overhaul than GT	Higher, no major overhaul required	Higher, no major overhaul required
Applicability (Power)	Available in all powers up to 100+ MW	Available in all powers up to 80 MW	Available in all powers up to 80 MW
Applicability (Speed)	Some units have mismatch with driven load speed	Flexible, as high as 15,000 rpm (depends on power)	Limited (Up to 3,600 rpm)
Speed Control	Slow response	Faster response	N/A
Maintenance cost	Much higher	No periodic maintenance required	No periodic maintenance required
Adjustable Speed Capability	Narrow speed range (80-105%	Wide speed range (<50-105%)	N/A
Foot print & weight	Much bigger considering auxiliary systems	Bigger than turbine only	Comparable to turbine only
Emissions	High (Depends on fuel in boiler)	No emissions (Power utility maybe subject to penalties)	No emissions (Power utility maybe subject to penalties)
Noise	Higher	Lower (Comparable in large systems with gearbox)	Lower (Comparable in large systems with gearbox)
Starting	No extra equipment required. Slow starting	No extra equipment needed. Fast starting	Large motors need starting equipment. Fast starting
Thermal efficiency	Very low (28%-35%)	System itself has high efficiency (>90%). Overall system eff. considering power source is 35% to 55%)	System itself has high efficiency (>90%). Overall system eff. considering power source is 35% to 55%)
Delivery time	Longer (> 18 months)	Shorter (<12 months)	Shorter (<12 months)
Return on Investment	Low	High	High

APPENDIX B: Predicted Vibration Response from Rotor dynamics Calculations

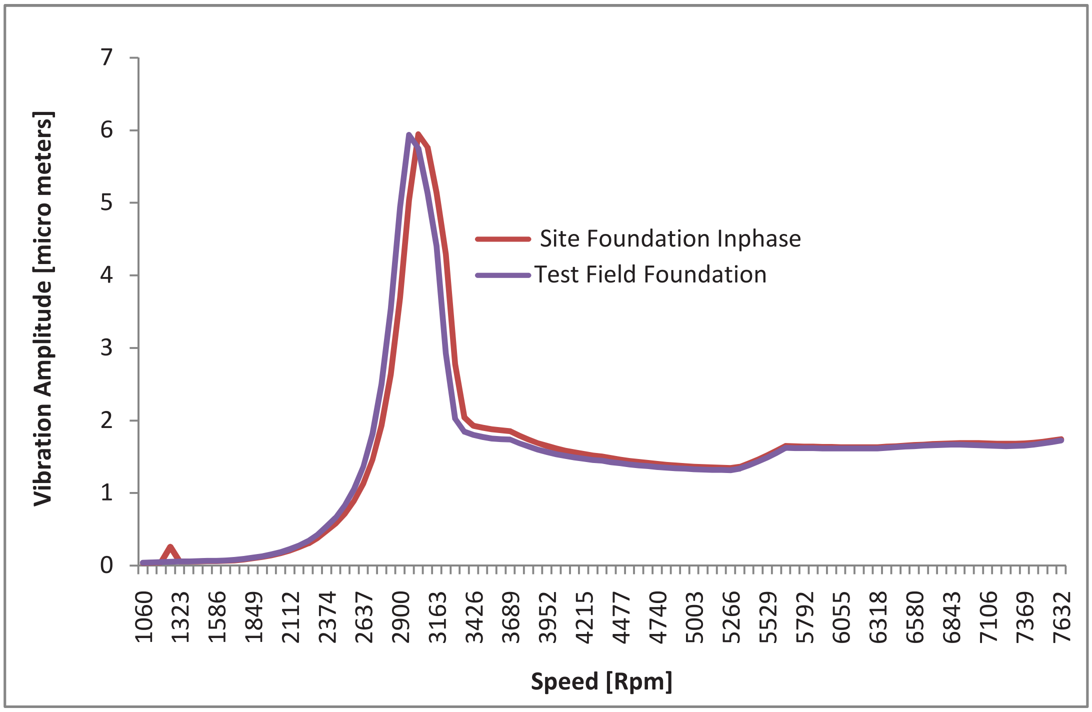


Fig. B1: Relative Shaft-Bearing Housing Vibration

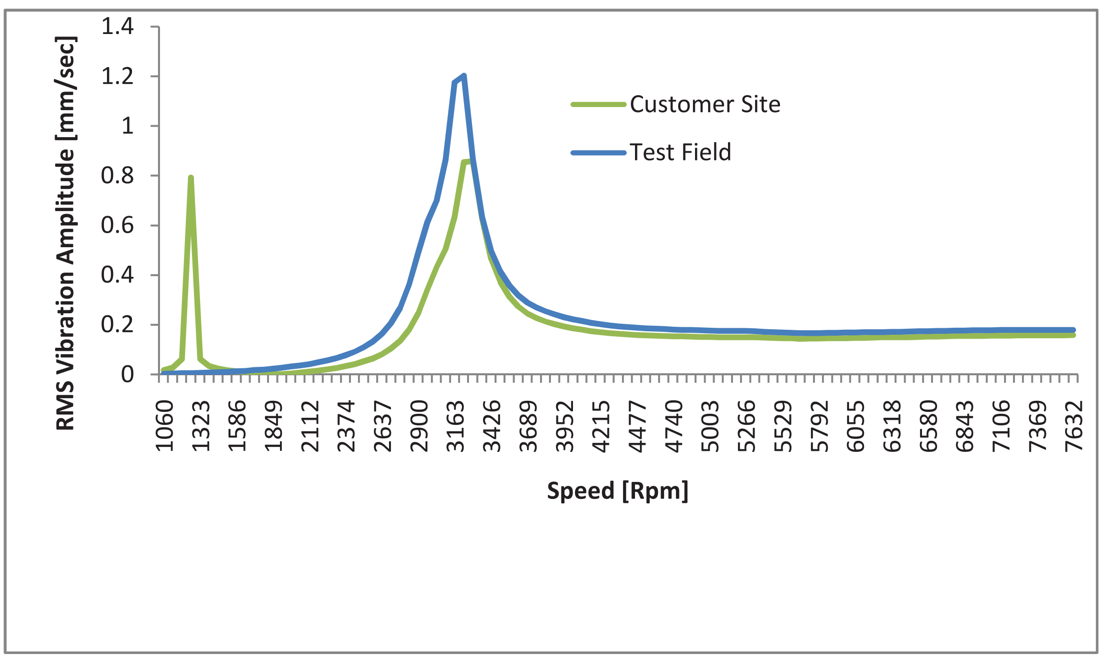
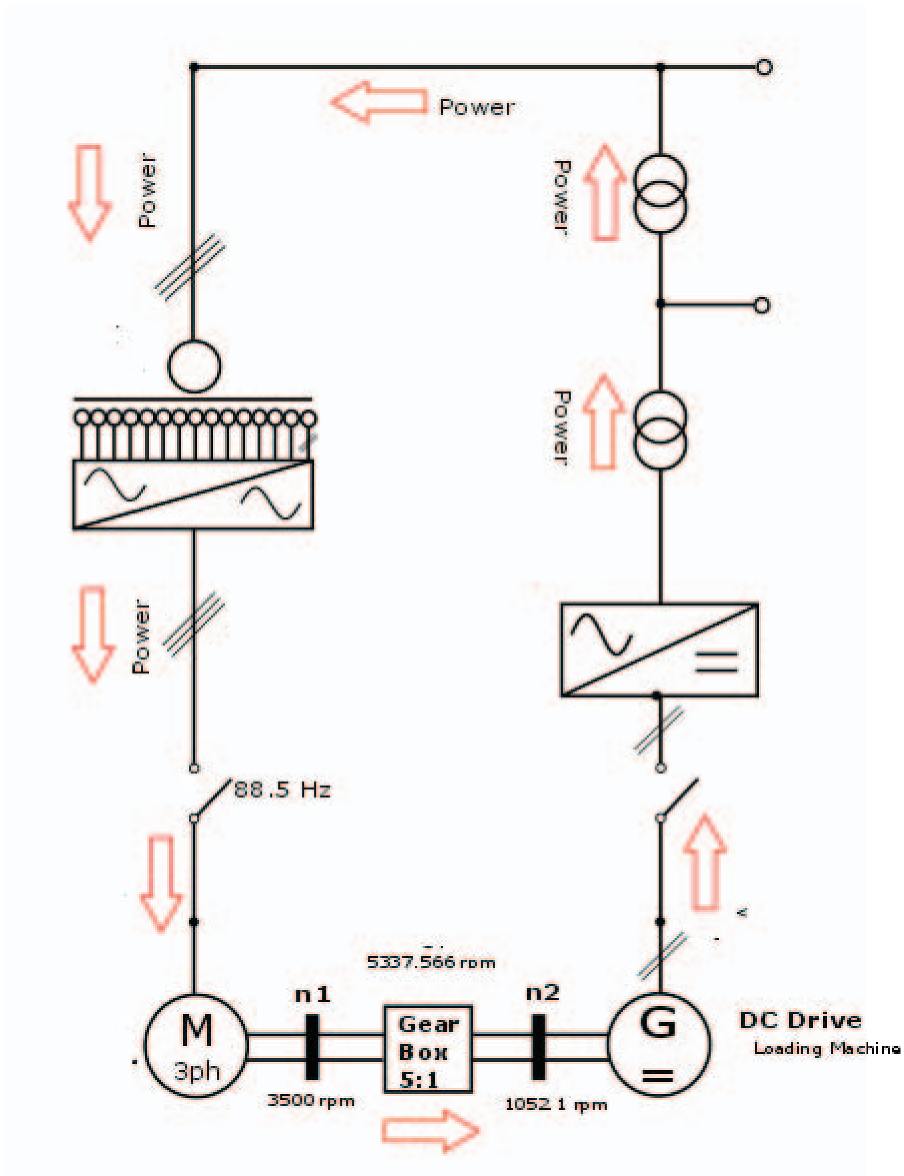


Fig. B2: Absolute Bearing Housing Vibration

APPENDIX C: Test Setup for Load Testing



APPENDIX D: Measured Shaft Vibration for the Speed Range during Factory Acceptance Test

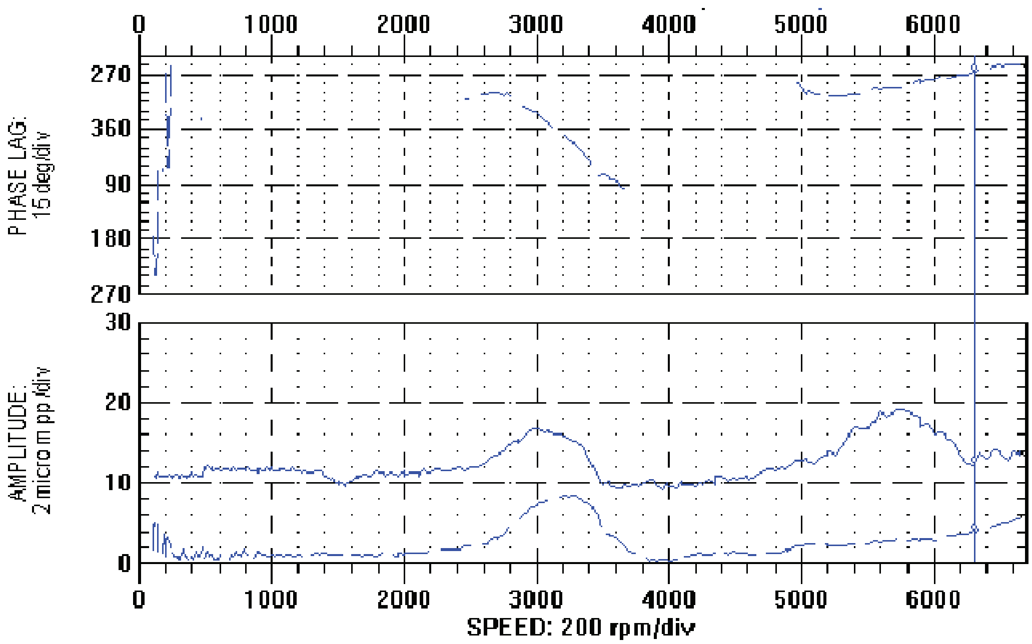


Fig. D-1. Shaft Vibration Measured at Probes at 45 °

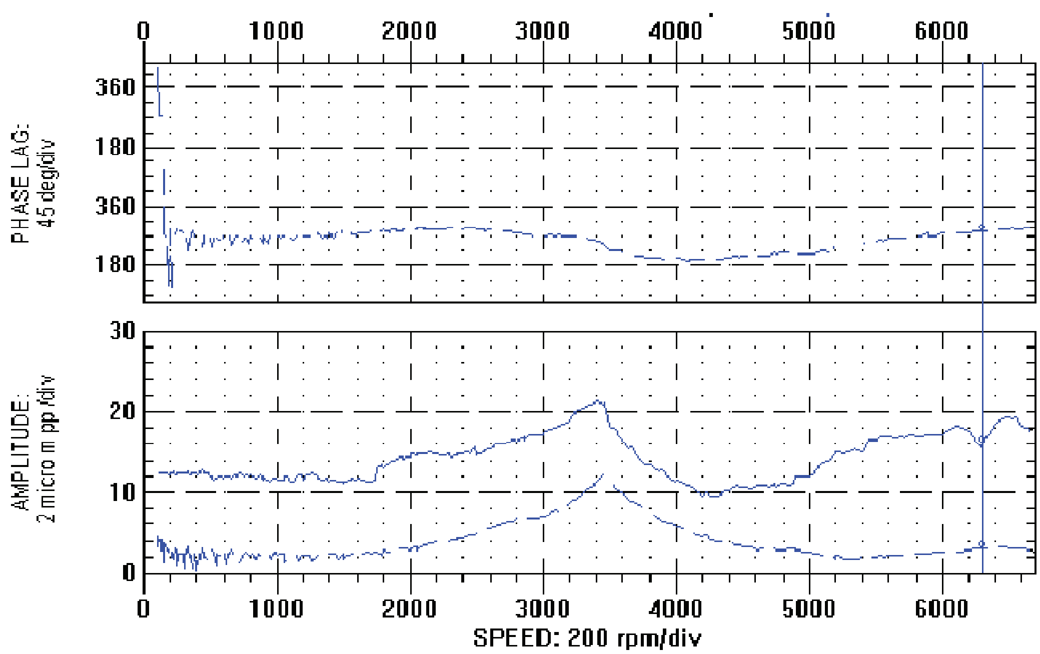


Fig. D-2. Shaft Vibration Measured at Probes at 135 °

APPENDIX E: Measured Shaft Vibration at Site Installed Location

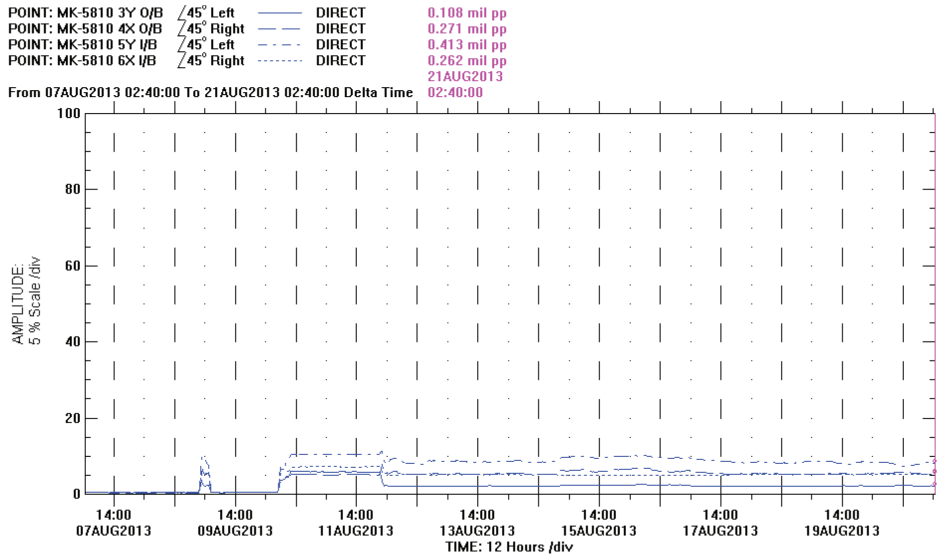


Fig. E-1. 5 Day Vibration Trend During Commissioning

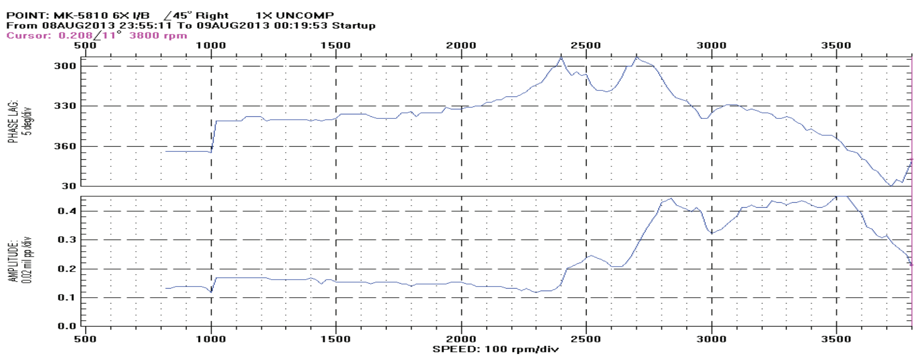


Fig. E-3. Shaft Vibration Measured at Motor Probes at 45 °

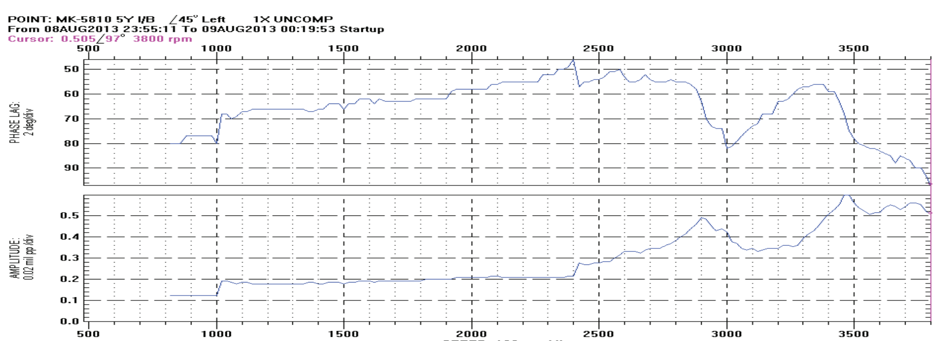


Fig. E-3. Shaft Vibration Measured at Motor Probes at 135 °