Failure Analysis of 7500 HP Induction Motors Driving Reciprocating Compressors with Three Years Service

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Abstract – This paper presents the failure analysis and field measurements of five 7500 HP induction motors driving reciprocating compressors for a natural gas compression station. The motors were in service for less than three years when two of them suffered cooling fan failures. Four of the five motors were individually sent one at a time to a motor repair shop to be fitted with a modified cooling fan per the manufacturer's recommendation. It was during one of the fan replacement processes that the repair shop, accidentally, discovered that many of the magnetic wedges were missing. The end user is not always aware if a motor is provided with magnetic or non-magnetic wedges unless identified in the motor engineering specification.

This paper presents the findings on the failure analysis of both the cooling fans and magnetic wedges for these motors. This failure analysis examined fan design, vibration, motor operating temperature, compressor loading profile, number of motor starts, and ambient temperature. The paper also presents a comparison between direct on line motor starting and soft starting using an adjustable speed drive and the impact on motor performance. The settings and historical data gathered from the microprocessor protection relay for the motors will also be covered and discussed.

Index Terms — Motor failure analysis, magnetic wedges, cooling fans, vibration, microprocessor protection relay, soft starting, adjustable speed drive.

I. INTRODUCTION

From 2005 to 2007 five 7500 HP, 4160 V, 1189 rpm induction motors were installed at compressor station "A" all having direct on line starting. A sixth motor of the same rating was installed at compressor station "B" using 3000 HP adjustable speed drive (ASD) for the purposes of a soft start application. The soft start application of the ASD was necessary due to utility power system requirements. The two compressor stations are less than 20 miles apart. In May 2007 one of the five motors in station "A" suffered a cooling fan failure and it was sent to a motor shop for repair. In November 2007 a second motor at the same location experienced a similar fan failure that had to be shipped to the same motor shop for repair. As a precautionary measure and under planned conditions, the end-user decided to take the remaining 7500 HP induction motors (with the original fan design) to the motor shop one at a time and retrofit them with a modified cooling fan design. In August 2008 during a fan replacement program of the last motor, it was noticed at the motor repair shop that there was evidence of winding material close to the non-drive end. The rotor had to be removed in each case for fan replacement. Due to the presence of the winding debris, the stator was thoroughly examined. The subsequent inspection revealed that many of the magnetic wedges, fitted in the stator slots, were either partially missing or loose.

This paper presents findings for premature failure of both the cooling fans and magnetic wedges for 7500 HP induction motors with three years in service.

II. FAN FAILURE

In 2007, two motors had serious cooling fan failures which required the motors to be removed from service and repaired. These events called for a review of the fan design and engaging the original motor manufacturer. The fan failure investigation, findings and subsequent modifications are covered in the following sub-sections.

A. *Fan Failure Mechanical Analysis*

In May and again in November 2007, two separate fan failures occurred on two of the then five installed electric driven gas compressor packages at a gas gathering compressor station in western Colorado. These two installations have identical 7500 HP electric motors each driving a two stage, six throw natural gas compressor. The failure investigation determined that the likely sequence of events that led to the eventual fan failure were:

- Fan bolts became loose
- Fan becomes loose
- Vibration levels begin to escalate
- Bolts begin to fail and some vibrate out
- Vibration levels continue to escalate
- All remaining bolts fail and fan falls onto steel boss
- Unit shuts down due to very high vibration levels

The design of the fan for this motor is a bolted joint where a cast aluminum fan is bolted to a machined steel boss that has a shrink fit onto the steel rotor shaft. In both cases, when the fan failed, many of the bolts in the assembly were sheared and the cast aluminum fan had significant damage at the bolt holes. The failure modes for the two cooling fans were essentially identical. The aluminum fan housing had many bolt holes that were 'key slotted' whereas others had no damage

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due to the bolt backing out completely prior to failure, see Fig. 1. There was also significant evidence of fretting corrosion on the mating surface between the fan and the steel boss indicating substantial and/or frequent relative motion, see Fig. 2.

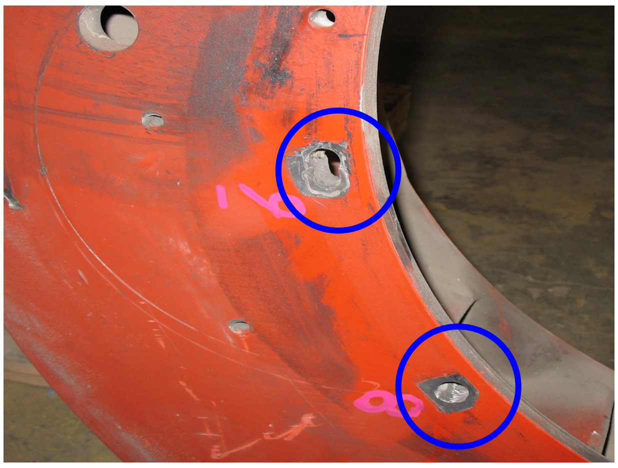

Fig. 1 One fan bolt hole damaged and another with no damage

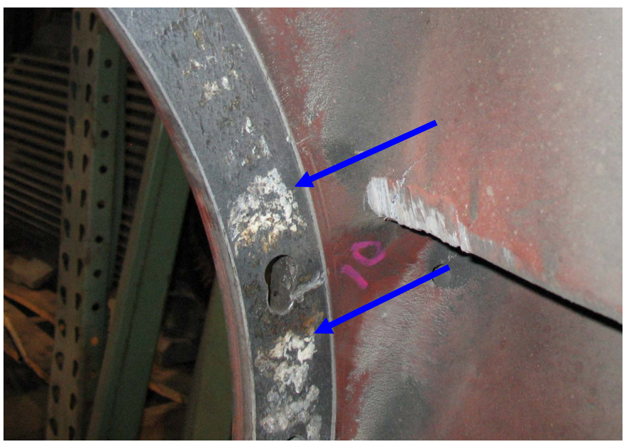

Fig. 2 Fan mounting surface fretting corrosion damage

These 1200 rpm units are started at full voltage with the compressor unloaded and, at the time of the failures, were shut down with the compressor fully loaded. These operational conditions led to very high acceleration during start up (2.5 seconds) and approximately twice the startup acceleration magnitude during shut down. This combination of inertial forces caused the fan to slip during startup and then fully reverse during shutdown. It is believed that this very small rotational translation is what led to the development of fretting corrosion. As the surface asperities grew in number and size, the effectiveness of the bolted joint decreased. Total rotational slip of the fan with respect to the fan boss likely increased. This relative motion combined with some inherent machine vibration and a low initial bolt torque ultimately allowed some of the bolts to become loose and vibrate out.

An engineering mechanics and dynamics analysis was performed for the original fan assembly design. This analysis revealed that the fan's bolted joint factor of safety (S_f), defined as the ratio of the total clamping force to the inertial force of the fan, had the potential to less than 1.

The motor manufacturer already had an 'improved' fan design that was available. This improved design used 33% more bolts with 78% more cross sectional area and included an additional steel ring as part of the improved design. This steel ring was installed on the inside of the fan which created a steel-aluminum-steel bolted joint assembly. This, in combination with more and larger bolts, increased S_f to approximately 5.

In addition to the improved design, the choice of bolt was also evaluated. The original and replacement bolts provided by the motor manufacturer were property class 4.8 (approx. SAE grade 2) which is a low strength, mild steel bolt. For the fan upgrade process, two changes were made to the bolts. Property class 8.8 (approx. SAE grade 5) bolts were selected for use and all of the bolt heads were drilled to accept stainless steel safety locking wire, see Fig. 3. Changing the bolt property class allowed the S_f to be increased yet again to approximately 8.5.

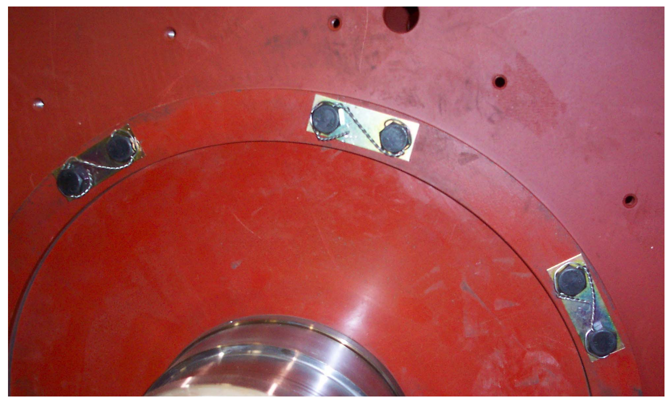

Fig. 3 Example of new fan bolt layout and safety lock wire

Although the hardware changes increased the safety factor for the fan assembly, operational process changes were also made. Instead of leaving the compressor fully loaded during any shutdown event, a shutdown matrix was created and all shutdown conditions were evaluated. For all but emergency shutdown conditions, the compressor package now initiates a controlled shutdown sequence whereby all the compression stages are unloaded prior to removing power from the motor. This process change decreases the load on the motor and increases the unpowered spin down time for the motor and compressor by a factor of approximately 12 which eliminates the high inertial forces that contributed to the original fan failures.

B. Motor Vibration Analysis

As part of the failure analysis process, potential energy sources that could have contributed or caused the fan failures were analyzed. High or excessive vibration was identified as a possible cause for the bolts in the fan becoming loose. A series of vibration analyses were conducted in an attempt to understand the vibration characteristics of the motor as a standalone unit and also the motor/compressor package system. Additionally, all pulsation and torsional vibration design study reports were reviewed.

By the time the motor vibration analyses were completed, the build out of the compressor stations were nearly complete. This allowed for eight motors at compressor station "A" and one motor at compressor station "B" to be included in the vibration study.

At compressor station “A”, the first five motors were installed on a common skid design whereby steel adjustable chocks were used for the motor support. The last three were installed on steel sole plates with shims. At compressor station “B” the motor was mounted on steel adjustable chocks.

Table 1 shows the motor manufacturer’s vibration guidelines.

TABLE 1
Recommended motor manufacturer’s vibration Limits

	Target	Alarm	Shutdown
Vibration Level (in/s pk)	0.25	0.31	0.50

Motor fan mechanical natural frequency (MNF) vibration was investigated by the manufacturer. By way of finite element analysis (FEA), it was determined that the MNF of the fan and fan assembly was 220 Hz to 310 Hz, more than 10X the running speed.

During the field vibration studies, 22 points on the motor housing were measured. These points included drive end and non-drive end bearing housings, frame locations, frame support (adjustable chock) locations, mounting skid/pedestal, and compressor. In general all vibration measurements were well below the motor and compressor manufacturer’s vibration guidelines. Only the bearing housings (not the rotor shaft) in the axial direction indicated any elevated vibration levels. The units that had elevated vibration magnitudes were random but when the vibration was above the target level, none were above the shutdown limit. When the vibration was above the target level, it was most often very close to the 3X running speed. This led the vibration investigation to examine the bearing housings more closely.

The mechanical natural frequency of the bearing housings was measured with the motor in the following configurations:

- Uncoupled, not running
- Coupled, not running
- Uncoupled, running
- Coupled, running and loaded

The motor MNF when uncoupled was consistently very close to 30 Hz. When the motor was coupled, running or not, the MNF was consistently between 64-68 Hz, see Fig. 4 and Fig. 5. The 3X running speed is very near to 60 Hz. Based upon the nature of the vibration amplification, as it approaches the first harmonic, it appears that there is a strong likelihood of the high axial bearing housing vibration being amplified by the mechanical natural frequency of the bearing housing and support structure itself. Mechanical modifications to the bearing housing have been investigated; however, none have been implemented to date.

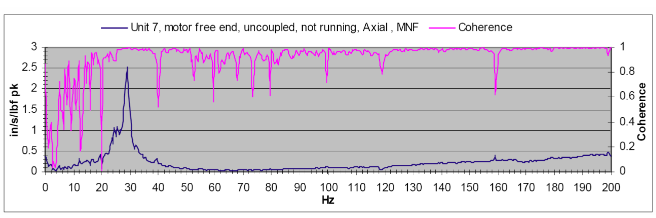

Fig. 4 Mechanical natural frequency of uncoupled motor, not running

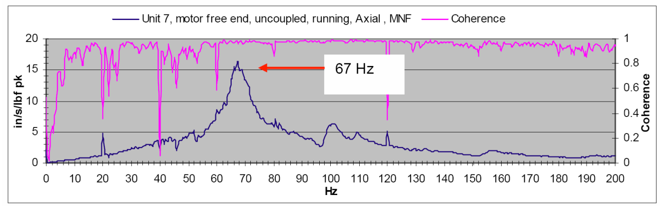

Fig. 5 Mechanical natural frequency of uncoupled motor running

III. MOTOR FAILURE

Compressor station “A” receives its incoming power at 230 kV single overhead line and “B” at 69 kV. At Compressor station “A”, the power is stepped down to 4160 V via two oil type transformers each rated 20/26.6/28 MVA. Two 7500 HP motors are connected to one side of the bus and the remaining three motors to the other side with the tie-breaker in normally open position. All five induction motors were installed at compressor station “A” over a period of three years from 2005 to 2007. In November 2008 an expansion project was underway to install an additional three motors to meet process requirements. Each motor is rated at 7500 HP, 4160 V, 872 A, 1189 rpm, 1.15 SF, class F insulation and direct on-line start. All motors underwent Vacuum Pressure Impregnation (VPI) processing. At compressor station “B” an adjustable speed drive is used to soft start a similar motor and automatically transfers it to 60 Hz bypass mode.

In August 2008, during a scheduled repair work to upgrade the cooling fan for the fourth 7500 HP motor from compressor station “A”, it was accidentally discovered that about 50% of the magnetic wedges were either partially missing or loose, see Fig. 6. The stator has 90 slots and each slot has three magnetic wedges. The slot length is 38 ½”. The use of magnetic wedges in the stator winding widely depends upon the manufacturer’s preference and their design practices. In this case, it is the motor manufacturer’s common practice to use magnetic wedges for induction motors rated 7500 HP, 1200 rpm and it has been used for almost 20 years. On the other hand, motor manufacturers in North America widely use non-magnetic wedges for similar induction motor ratings. It should be noted that information regarding the type of wedges used in stator slots is not normally published, shared, or provided by any motor manufacturer at the bidding stage unless it is specifically requested or called for in the engineering specification. This information is considered detail design and privy to the manufacturer’s practices.

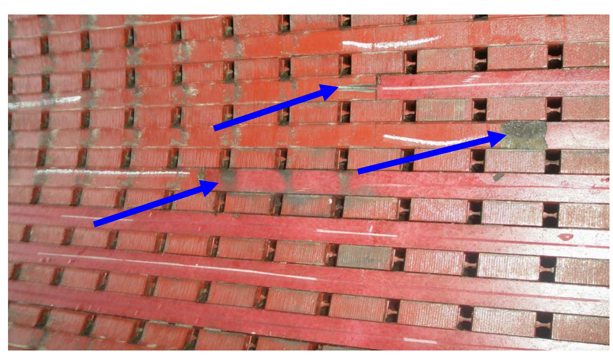

Fig. 6: Stator winding at the motor repair shop. White chalk

marks identify partially damaged or loose magnetic wedges.

Magnetic wedges when compared to non-magnetic wedges have pros and cons [1], [2]. The key advantages of using magnetic wedges are:

- Lower stator temperature rise and reduced core losses
- Slightly higher motor efficiency
- Reduced inrush current
- Reduced noise level

A typical magnetic wedge material comprises of 75% iron powder, 7% glass mat and 18% epoxy resin. Magnetic wedges are more susceptible to failure when compared to non-magnetic wedges because they are more brittle due to the high percentage of iron powder. In addition, for reciprocating type loads, the magnetic wedges by their very nature are subject to cyclic mechanical forces (120 times per second) so that if there is any freedom to move in the stator slot, fretting can occur and the movement will slowly increase until wedge failure occurs. Upon failure, magnetic wedges normally disintegrate into very small pieces and get crushed in the very small air-gap (2.8 mm) between the stator and rotor. Typically, upon failure, magnetic wedges leave a distinct color mark on the rotor surface. The reliability and longevity of magnetic wedges are greatly impacted by the following factors:

- The number of full voltage motor starts should be kept to a minimum as these subjects the magnetic wedges to excessive magnetic and thermal stresses. In this case, the data retrieved from motor protection microprocessor relay show that the number of motor starts is high and the motor is started on average once every three days.
- Motor vibration should be kept to within design limits. High vibration could result in magnetic wedges weakening and possible failure.
- Motors should normally run at or below 80º C (176º F) temperature rise. In this case, the motors were running at an elevated temperature rise although the actual measured motor loading is at approximately 80% of rating. This situation is compounded because of poor air ventilation in the compressor building for station "A" and elevated ambient temperature during summer months. On the other hand, forced ventilation fans were installed at compressor building 'B".
- The surrounding area to the compressor building "A" is fairly dusty and all the motors have WPII enclosures without a provision to add air filters.
- It is common that a reciprocating compressor load profile exerts greater electro-magnetic forces on the magnetic wedges due to the pulsating currents and when the motor is running at rated load. This was not a factor in this case because the stator current profile, see Fig. 7 is uniform (non-pulsating) and the motor was running at 80% load. MG1–2003 standard [3] calls for the current variation in induction motors driving a reciprocating type load not to exceed 66% of full load value. In this case, the variation between minimum and maximum current measurements was insignificant, less than 2%, well within MG1 requirements.

There is no known definitive field measuring technique or computer modeling available to positively predict loss of magnetic wedges of the stator winding without removing the rotor to visually inspect the stator. However, for this site, during a scheduled station shutdown, a special borescope was inserted in the motor air-gap (2.8 mm) to look for any marks (wear bands) on the rotor surface to establish loss of magnetic wedges. This inspection method proved to be reasonably effective to asses the condition of magnetic wedges. To date, none of the borescope inspections have found any new wear (rub) marks on the rotors indicating that there have not likely been any additional magnetic wedge failures.

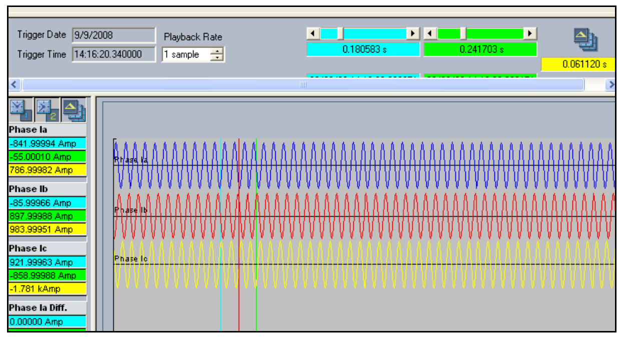

Fig. 7: Sample of stator current waveforms for phase A, B and C. FLA is 872 A rms.

The motor manufacturer cautioned that if the motor is repaired using non-magnetic wedges the stator temperature rise would be approximately 125% higher than that of magnetic wedges. This could potentially impose operational limitations and not to be able to run the motor at the current loading conditions.

The repair work was carried out using magnetic wedges with the following dimensions: 5 3/8" long X 3/4" wide X 1/8" deep (see Fig. 8). The replacement magnetic wedges are half of the length of the original wedges to facilitate easier insertion at the repair shop. In this case, the total wedges to be installed were 270. The motor manufacturer approved the product technical specification for the two epoxies used by the motor repair shop for installing the new magnetic wedges. The first epoxy was applied at room temperature in the stator slots before inserting the magnetic wedges. The second product was applied over the magnetic wedges after the stator is removed from the oven. The motor was subjected to a core loss test before and after re-wedging and the results were acceptable.

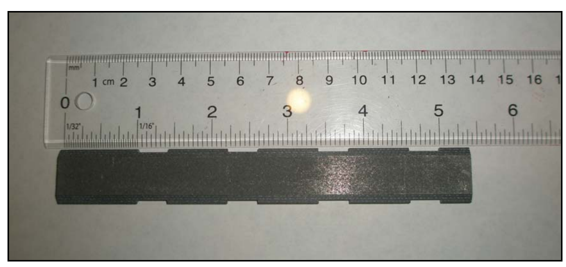

Fig. 8: Sample of new magnetic wedge replacement.

During the course of the magnetic wedge failure investigation, the motor manufacturer provided a copy of a factory heat run test that was conducted for one of the 7500 HP motors in September 2005. The factory test results showed that the stator winding temperature rise was recorded at 53.1º C, which is considerably less than the design value of 80º C. Accordingly, the motor temperature rise when running at 1.15 service factor is expected to be $(1.15)^2$ X 53.1 = 70.2 º C. If the same motor was rewound using non-magnetic wedges, the motor temperature rise, according to the manufacturer, is expected to increase by 25%, or 1.25 X 53.1 = 66.4º C. Likewise, at 1.15 service factor it would be $(1.15)^2$ X 66.4 = 87.8 ºC.

IV. MOTOR PROTECTION

A microprocessor-based protection relay is used for all 7500 HP at compressor stations "A" and "B". The digital relay is equipped with several features to provide comprehensive motor protection during starting and running conditions. The relay has the capability to display actual field measurements, historical data, current waveforms, and past events. This information was important to assist in determining the premature failure of the magnetic wedges and overall motor performance. A laptop was used to communicate with all the relays and retrieved all pertinent data including actual motor loading, stator and bearing RTD temperatures, maximum stator and bearing RTD temperatures, total number of motor starts, total number of hours in service, starting current, and acceleration time. The data for the five relays at station "A" were almost identical and are summarized as follows:

- The average loading was 81% rated value.
- The motor stator RTD is set to alarm at 125º C, high alarm at 130 º C and trip at 150º C.
- The bearings are set to alarm at 80º C, high alarm at 85º C and trip at 90º C.
- The highest stator RTD temperature was recorded at 146º C and for the bearings it was 79º C. This indicates that the motor stator winding had experienced a high stator temperature close to the tripping point despite that the average motor load was 81%. It is unlikely that these motors would be able to run at service factor rating because the stator temperature would exceed the trip settings of 150º C, especially during the summer months.
- Average motor acceleration time was 2.5 seconds and motor starting current was 5.15 PU.
- Average power factor was 90%.
- On average, the motor was started every three days and in some cases, more frequently.
- Most of the captured events were either motor starts or stator RTD high temperature.

Historical data shows that the number of motor starts is high and the motor was exhibiting many over temperature stator RTD alarms although it was running only at an 81% average load. Typical reasons for a motor to run hot are:

- Overload condition
- Excessive number of starts
- High ambient temperature around the motor
- Dirty air filters

These factors were investigated and in this case the overload factor is disregarded because the motor was running at 81% rating. The ambient temperature was measured throughout the compressor building "A" in the proximity of the five compressors using an infrared temperature instrument. The measurement showed a considerable difference between the outside and inside ambient temperatures and this difference was greater in the afternoon. On the other hand, at compressor building "B", the inside and outside ambient temperatures were close because the building is equipped with forced ventilation using several fans. The ventilation scheme that exists in compressor building "B" is very effective in cooling the compressor building. There was a close correlation between the ambient air temperature and motor temperature rise. The higher the ambient temperature adjacent to the motor, the higher was the motor stator temperature rise. Fig. 9 shows data retrieved from the distributed control system (DCS) during March 2009 for a two week period.

It was discovered that none of the 7500 HP motors had an air filter or even had a provision to install them. The motors have a standard WPII enclosure.

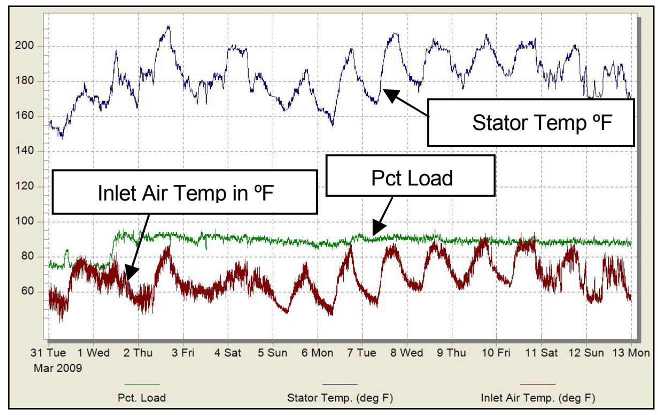

Fig. 9: Correlation between stator and inlet temperature

V. FIELD MEASUREMENTS

For a period of two weeks in September 2008, two portable on line power quality monitors were installed at compressor station "A" to monitor the voltage and current profile and motor starting characteristics for two 7500 HP motors. The starting current was measured at 4563 A, or 4563/872 = 5.23 PU. The starting voltage drop was measured at 15.5% and starting time at almost 3 seconds. The measured voltage drop is marginally high and might cause nuisance tripping of 480 V small adjustable speed drives. The magnitude of the motor starting voltage drop would be worse under abnormal conditions with a single main transformer in service and tie breaker closed. The steady state voltage fluctuation at 4.16 KV bus was within ± 1% and this is very acceptable. The same on line measurements were repeated for compressor station "B". At this location, the 7500 HP motor is started using an adjustable speed drive and the acceleration time is set at 40 seconds. The motor is automatically transferred to bypass power. The motor starting hardly caused any voltage drop because the motor is started in a "soft mode".

VI. CONCLUSIONS

A. Fan Failure

As part of the fan replacement process, all fans have been upgraded to the manufacturer's improved design and all motors (at both compressor stations) have had property class 8.8 bolts installed with safety lock wired heads. To date, there have been no additional fan failures.

Field vibration measurements have been collected and efforts are being made to ensure that values are within motor design limits. A review of the motor vibration data has indicated that mounting the motor on adjustable chocks or on sole plates has had no attributable effect on the vibration amplitudes. It is recognized that high vibrations could accelerate the damage and ultimate loss of magnetic wedges.

Elevated axial vibrations at the bearing housings appear to be caused by the amplification effect of approaching the mechanical natural frequency of the bearing housing structure. No modifications to the motor bearing housings have been pursued.

As a result of the mechanical analysis, the shutdown procedures for all the compressors at both stations have been modified to unload all stages of compression (except for emergencies) prior to removal of power to the motor. This process change increased the spin down time by a factor of at least ten times.

B. Magnetic Wedges Failure

Several factors are attributed to the premature failure of the magnetic wedges for the 7500 HP motor at compressor station “A” with less than three years in service. These are elevated motor temperatures, excessive motor starts and high vibration. All five motors at compressor station “A” exhibited high RTD stator temperatures measured at 144º C despite running at approximately 80% rated load. Primary causes for high stator temperatures were inadequate compressor building ventilation, dusty surrounding with no motor air filters and excessive motor starts.

Several measures were undertaken to address the factors impacting the longevity of magnetic wedges. In June 2009 the ventilation scheme for compressor building “A” was improved as that in compressor building “B”. The motor enclosures were modified to install air filters with differential pressure gauges. Serious consideration is being given to installing two adjustable speed drives at compressor station “A”, similar in concept to that at compressor station “B”, to soft start the existing five 7500 HP.

Information regarding the type of wedges used in stator slots is not normally published, shared, or provided by any motor manufacturer at the bidding stage unless it is specifically requested or called for in the engineering specification. The use of magnetic versus non-magnetic wedges in stator slots depends on the manufacturer’s experience and preference. In this particular case, the motor manufacturer has been using magnetic wedges for over 20 years but in North America is it more common to use non-magnetic wedges.

There is no known conclusive field measuring technique available to predict the loss of magnetic wedges without dismantling the motor to inspect the stator. A technique using a special borescope inserted in the 2.8 mm air-gap was tested to check for discoloration (wear band) on the rotor surface as a result of magnetic wedges failure. This inspection method appears to be a reasonably effective method to establish advance warning of magnetic wedge failures.

VII. REFERENCES

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VIII. VITA

Robert A. Hanna received the B.Sc. degree from the University of Basra, Iraq in 1971, M.Sc. Degree (with Distinction) from Queen Mary College, University of London, England in 1973 and Ph. D. degree from Imperial College of Science and Technology, University of London, England in 1977, all in electrical engineering. Following a short teaching career, in 1981 he joined Petro Canada (formerly Gulf Canada) as a central engineering specialist providing technical support to the refineries in implementing capital projects and equipment selection. In 1995, he founded RPM Engineering Ltd., a certified consulting company in Ontario, Canada specializing in Adjustable Speed Drive applications, power quality studies, emergency shutdown equipment, renewable energy and equipment failure investigations.

Dr. Hanna is registered professional engineer in the provinces of Ontario, Alberta and British Columbia, Fellow of the Institute of Electrical and Electronics Engineers (FIEEE), a Fellow of the Institution of Engineering and Technology (FIET), UK and a Fellow of the Engineering Institute of Canada (FEIC). He was President of IEEE Canada and IEEE Director (Region 7) in 2006-2007.

Dennis W. Schmitt received an A.A.S. degree in Electrical Engineering Technology in 1991 and B.Sc. and M.Sc. degrees in Mechanical Engineering from Colorado State University in 2005. Dennis worked for Hewlett-Packard and Agilent Technologies in the consumer electronics and semiconductor industry for more than 10 years where he focused on maintenance and reliability programs for a wide variety of automated manufacturing equipment. After graduating with his mechanical engineering degrees, he joined EnCana Oil & Gas (USA) Inc. in early 2006 and ultimately assumed the role of rotating equipment engineer within their natural gas gathering operations, with responsibilities encompassing their compression assets within the Rocky Mountain region of the United States. Within that role, Dennis led many failure investigations predominantly related to gas compression equipment and implemented many procedural and hardware improvements which has led to an overall increase in equipment reliability.

Dennis is a registered professional engineer in the state of Colorado. Currently, Dennis is the Group Lead of Engineering and Technical Support within the Facility Asset Management team, based out of the EnCana corporate headquarters in Calgary, Alberta.