Generator Cooling Fan Vibrational Problems: A Design and Operational Case in Jordan

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Abstract

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This project covers certain types of rotor generator vibrational problem, including both conventionally-cooled (indirect copper cooling) windings and direct cooled copper windings as well as those with spindle and body mounted retaining rings. The options for rewinding, modifying, or upgrading are provided for each case type as encountered in the Jordan Petroleum Refinery. Generally; when dealing with generator rotors, problems like oversizing, misalignment, rotor imbalance, off-design operation, lubrication problems, shaft seals, cavitation, balancing system, and vibration are encountered. Methods of resolving vibration problem diagnosis were discussed using different techniques like vibration design analysis which will enhance root cause problems. We have proposed different maintenance best practice like rotor dynamic balance, appropriate operation, seals selection, and lubricant selection. At the end we have proposed a design solution appropriate for failure prevention using rotor balancing after rework or modifications

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Abbreviations

JPR	Jordan Petroleum Refinery, Jordan
Acceleration	The rate of change of velocity often depicted as "g's" or in "mm/s2".
Alignment	A condition where components within a drivetrain are parallel or perpendicular, according to design requirements.
Failure	The event, or inoperable state, in which any item or part of an item does not, or would not, perform as specified.
Forced vibration	the vibration of a machine caused by some mechanical excitation. If the excitation is periodic and continuous, the response motion eventually becomes steady-state.
Sensitivity	The ratio between electrical signal (output) and mechanical quantity (input).
Accelerometer	A transducer whose electrical output responds directly to acceleration. Accelerometers typically cover a much wider frequency range, along them to pick up signals not present with other types of transducers. Due to the frequency range, accelerometers are ideal for most types of rotating equipment, making them the most used transducer for vibration measurements.

Chapter 1

Introduction

1.1 What is a Generator Rotor?

The generator rotor represents an excellent combination of electrical, mechanical and manufacturing skills in which the field coils are well insulated, supported and ventilated in a compound structure rotating at very high speed (typically 1800 or 3600 rpm). Furthermore, though the rotor experiences great mechanical stress and high temperatures (in some cases up to 155˚C) while subjected to electrical voltage and current, it is expected to function in this manner for years without failure.

The three design constraints that limit the size and life of generator rotors are temperature, mechanical force and electrical insulation.

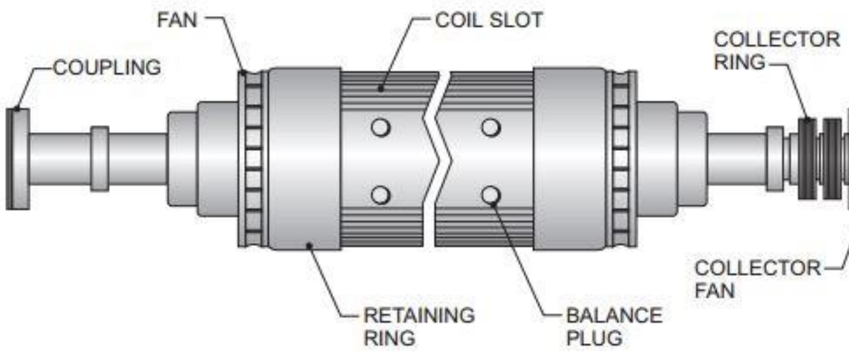

Figure 1. Generator Field.

Figure 1 shows a basic mechanical outline for a typical generator field. Note the major components:

- ■ Turbine coupling
- ■ Main cooling fans

■ Retaining rings

■ Coil slot

■ Balance plug

■ Collector rings

■ Collector fans

There are, of course, variations on this configuration. For example, while the illustrated design uses radial fans, other designs use axial fans.

A typical collector end configuration is shown in Figure 2, which also shows a cutaway view of vital electrical components such as:

■ Collectors

■ Collector terminals

■ Bore copper

■ Main terminal

■ Main lead

■ Retaining ring

■ Coil end windings (shown from the side)

■ Axial fan

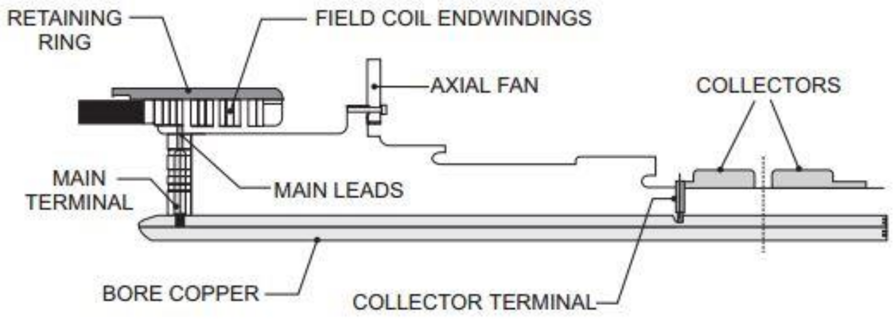

Figure 2. Collector end of generator field.

1.2 Problem Statement

As a generator rotor ages, its insulation can be affected by temperature, mechanical wear and operating incidents. Rotor forging and other rotor components are also at risk. The most common problems occurring with generator rotors are shorted turns and breakdown in ground wall insulation. These two concerns will be discussed in this project and it relevancy to vibration aspects [1].

Chapter 2

Literature Review

In plain bearings such vibrations are the consequence of the intermittent mechanical contact between the shaft and the bearing surface. When a lubricant film carries the load between the shaft and bearing around the full circumference, such contact does not occur, nor any noise. However, contact does occur upon start-up, when the radial forces on the bearing are excessive (belt drive, gear, air gap field), when the shaft and/or the bearing sleeve is not round or if they are crooked, if the sintered bearing surfaces do not have sufficient porosity, if the shaft running surface is too smooth, or if there is not enough lubricant in the bearing (mixed friction). The consequence is vibrations at the roughness peaks of the bearing surfaces, which are dependent on the elasticity of the shaft or bearing, at numerous frequencies (frequency band in spectrum) in the audible range. The frequency of rotation and multiples thereof are particularly pronounced. The amplitudes of the higher frequency vibrations are often simply modulated, which causes the friction noises to become particularly objectionable. As the lack of lubricant worsens, the mechanical contacts increase, and mechanical friction occurs and wear increases sharply [2]. This results in natural vibrations in the bearing (with the self-induced addition of energy from the bearing friction), which is perceived as a squeaking or squealing noise. In plain bearings, the deformations are primary and the resulting deformation forces are secondary (displacement excitation).

In roller bearings, rolling elements – in small motors these are generally ball bearings – roll in inner and outer rings, usually with mechanical contact. They are surrounded by a layer of lubricant, which has a slight cushioning effect and enlarges the contact surface somewhat. If the rolling elements and the ring raceways are sufficiently round and undamaged, only a broad frequency band (spectrum) of oscillations results as a consequence of the circumferential elastic deformations at the contact points caused by the compression forces (force excitation) and as a consequence of lubricant movements (displacement excitation). The lubricant dampens the vibrations. So if not enough lubricant is present or if the viscosity of the lubricant is incorrect, vibration will increase ("metallic, hard-sounding" noise) because the pressures increase at the contact points. Material fatigue increases. Raceway damage, in particular that caused by axial overloading of the bearings, also leads to faster material fatigue. Radial ball bearings have radial bearing play due to how they are manufactured and how they operate. If this radial play is "compressed to zero" as a result of an improper elastic axial preload on the outer or inner ring, the bearing balls cannot transfer the radial forces optimally. The installation fits of the bearing and the assembly quality play an important role in this case. At certain rotational speeds the balls actually run in synch instead of in a circular, wavy path. As a result, self-induced axial vibrations, which are heard as howling noises, occur in the bearing bracket [2].

Thus in bearings, we mainly encounter motions that generate elastic forces, which in turn lead to vibrations, as well as forces that lead to oscillating motions. This means that there is displacement excitation as well as force excitation, so that both must be considered when the individual case is being analyzed in theoretical terms. Stator bar temperature will increase as electrical current flows in the copper of the winding. The relationship of temperature to electrical current is well known as $T \propto I^2$. Therefore, if the generator is at full load where the stator current is theoretically at its maximum (I_{ref}), then the temperature of the stator bar hose outlets will be some temperature above the cooling water inlet temperature. The difference between the cooling water inlet and outlet temperatures will be the temperature rise, dT_{ref}, at this reference load, due to the heat input from the stator bar I^2R losses. The temperature difference between T_{out} and T_{in} will obviously change as the generator loading (operating stator current, I_s) is increased and decreased. Applying the relationship $T \propto I^2$, we can use I_s and I_{ref} in the form $(I_s/I_{ref})^2$ to account for generator load changes. Therefore the basic formula to calculate stator winding hose outlet temperatures can be written as

$$\text{Tout} = T_{\text{in}} + dT_{\text{ref}} \left(\frac{I_s}{I_{\text{ref}}} \right)^2$$

In the relationship above, we can see that the portion of the function $(I_s/I_{ref})^2$ is equal to one, as it should be, when fingerprinting of the stator winding temperatures is done at the reference load. As I_s becomes lower, at lower loads, the temperature calculated for T_{out} will decrease proportionally [1]. Using the formula, the difference between the measured reading and the calculated value can be closely monitored. An alarm value (e.g., 5°C) can then be added to the calculated value to produce the dynamic alarm limit as follows [3]:

$$\text{Talarm} = \text{Tout} + 5^{\circ}\text{C}$$

2.1 Rotor Vibration Failures: Causes and Solutions

Steam turbine rotors bend during operation, but the bearing and supports are designed to keep the static and dynamic forces under control. However, bending can cause impact between stationary and rotating parts—often cascading impacts. An operator of many utility-scale steam turbines like the one in Jordan Petroleum Refinary shares its extensive field experience identifying the root cause of failures as well as successful solutions.

Rotor bending that results in premature failure of steam turbine blades and other internal components is one of the most serious problems experienced in power plant operations. The problems often reduce plant availability by limiting generation and increase plant operation and maintenance cost. Extreme rotor bending problems often involve interaction between the turbine's rotor and stationary parts. Rotor bending may be caused by a variety of static and dynamic factors, many of which will be explored in this research project [4].

We begin with mechanical factors related to the rotor, the largest rotating assembly in the turbine. Working from the inside out, we next look at rotor balance issues, followed by rotor and casing misalignment problems, and problems caused by the casing. The discussion is based on the authors' experiences Jordan Petroleum Refinery located in Jordan.

It almost goes without saying that rubbing caused by insufficient clearances, disrupts the end sealing of the rotor. This situation commonly occurs when the high-mass rotor at operating speed comes in contact with a stationary surface, typically caused by a too-small clearance between the gland seals and the rotor. Secondarily, there may be a localized temperature increase at the point of contact, causing increased metal temperatures at the point of contact due to friction.

The forces produced by the impact of the large rotating rotor mass with the poorly functioning stationary seals often impress a layer of metal on the surface of the rotor. The rub can cause elastic deformation of the rotor at the point of impact and temporary rotor shaft bending. The shaft bending will usually cause increased vibration levels (Figure 3-b below) [5].

2.2 Core Vibration

Vibration in the stator core is naturally produced by the unbalanced magnetic pull in the airgap, originating from the unequal magnetic field distribution of the rotor. The pole or direct axis carries the main flux while the winding or quadrature axis carries only the leakage and stray fluxes. Therefore a large difference in magnetic force is inherent between the two axes [6].

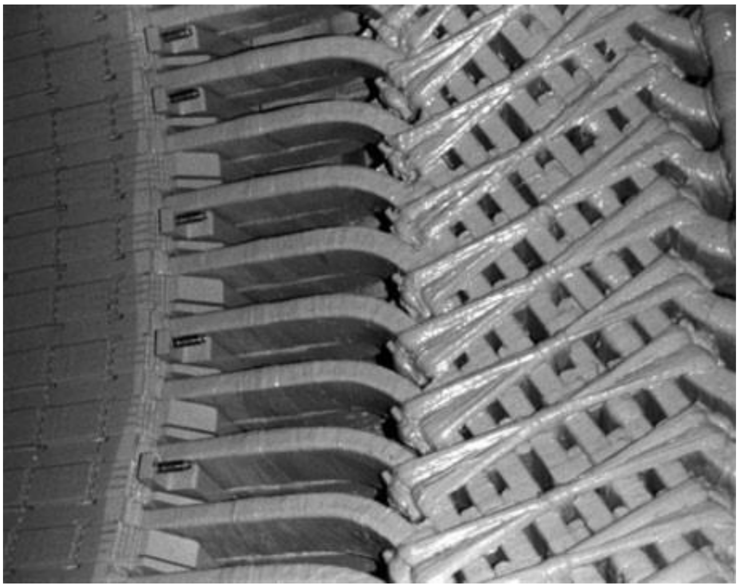

Fig. 3-a Color-coded tagging compounds reflecting vibration intensity.

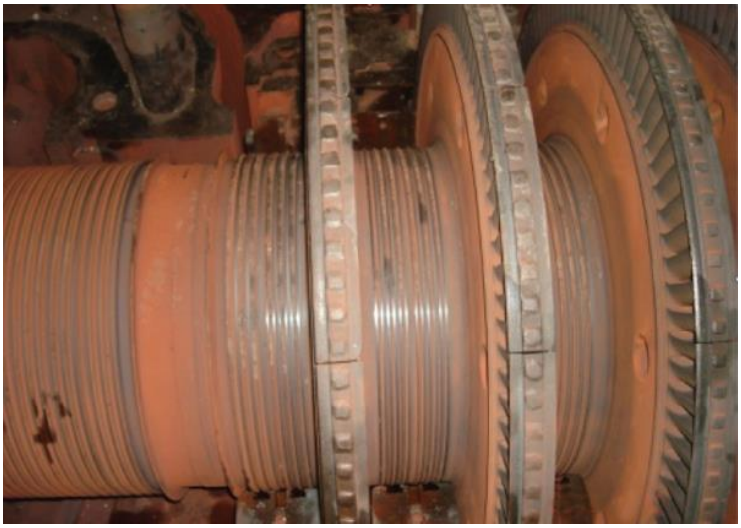

Fig. 3-b Blade rubs cause bending. Rubbing in the sealing of a high-pressure rotor caused bending of this rotor and blade tip rubs. Source: Photo taken at Jordan Petroleum Refinery (2017).

Uneven cooling of the rotor, particularly after shutdown, also causes the rotor to contact stationary parts. After a unit shuts down, the relatively high-temperature rotor may bend solely due to the mass of the rotor and the distance between bearing supports, if left in a stationary position to cool. This situation can cause permanent shaft bending.

The effect of a permanent shaft bend caused by uneven cooling will immediately appear as high rotor vibration at the next startup. The vibration is caused by insufficient clearance between stationary and rotating parts, as well as a shaft located off-center in its bearing. Even if the clearance change is small, there may be significant rubbing along the rotor to cause damage. Again, the rubbing causes friction between stationary and rotating parts, localized heating of the rotor metal at the contact point, and shaft bending [7].

Furthermore, uneven shaft warming caused by rubbing between rotating and stationary parts can cause further bending of the shaft in the same direction of the existing bow and cause additional contact with stationary parts, increasing temperatures and therefore causing more bending. The effect cascades if allowed to continue. If the bending is allowed to continue, it is possible that the yield strength of the metal could be exceeded, causing a permanent deformation of the shaft. The allowed bending in 3,000-rpm turbines is up to 0.02–0.03 mm in each section. When on turning gearing, the limit is 0.05 mm.

Magnetic force is generated in the pole axes, and a weak magnetic force is present in the winding axes. Since each pole has a north and a south (or a plus and a minus) associated with it, an unbalanced magnetic pull is generated at twice the line frequency.

The core must be maintained tight or fretting will occur between the laminates. Minor fretting will tend to deteriorate the inter-laminar insulation, but if the core becomes too loose, the laminates and or the space blocks may even fatigue with the result being pieces of loose core material breaking off and causing damage. Monitoring of core vibrations can be done with accelerometers mounted on the core back in strategic locations to determine the magnitude and phase of both radial and tangential vibration modes (see above figure) [7].

To avoid rotor bending during cooling, turbine vendors provide very specific instructions on the allowable rate of cooling. For example, the turbine should remain on turning gear until the high-pressure (HP) cylinder temperature is below 150C and the oil temperature is below 75C. The turbine vendor also defines the rotational speed of the turning gear.

Misalignment of the coupling between two shafts or between a shaft and bearing may cause bending in the system. Misalignment between two shafts of an integrated rotor can cause an eccentricity of the mass center of the rotor, and this eccentricity at high rotational speed will produce a centrifugal force in the radial direction, causing bending of the rotor. Misalignment between the axis of rotation and the axis of the shaft can also cause bending in the rotor. There are six primary factors that can cause misalignment.

A poor connection between the turbine casing and the bearing pads on the foundation frame is one cause. If a pad experiences increased friction or stops sliding during thermal expansion (usually during startup) in the axial direction, the result is a tipping torque on the casing. This torque can cause a misalignment between the casing and bearing surface, causing vibration in the forward end of the turbine, foundation frame surface support deformation, and bearing pad stall.

Also pay close attention to the foundation frame—including bolts, keys, and pads—so that free movement of the bearing surfaces is possible, particularly while undergoing startup and load changes. The extent of the longitudinal and lateral thermal expansion bore centers of the cylinders and pad travel should be recorded for future comparisons. This process should be part of routine maintenance equipment inspections [7].

Another factor concerns the difficulty in assembling the HP turbine front bearing. While the shaft is rotating in its journal bearing, the shaft pushes oil from the bottom of the bearing, causing the oil film thickness to change. When this happens, the centerline of the shaft moves up and to one side. To account for this shaft movement, the segmented bearing should automatically adjust and the contact surface of the journal bearing will remain in a good position. If there is too much contact surface, friction will increase on the bearing surface, causing increased rubbing and corrosion of the bearing surface and increasing vibration and rotor eccentricity. The result will be bearing oil leakage and rubbing in the sealing glands. On the other hand, if the bearing contact area decreases, the oil film will cause uneven movement of the rotor within the segmented bearing, and the oil film will not form, also resulting in increased vibration (Fig. 3-c).

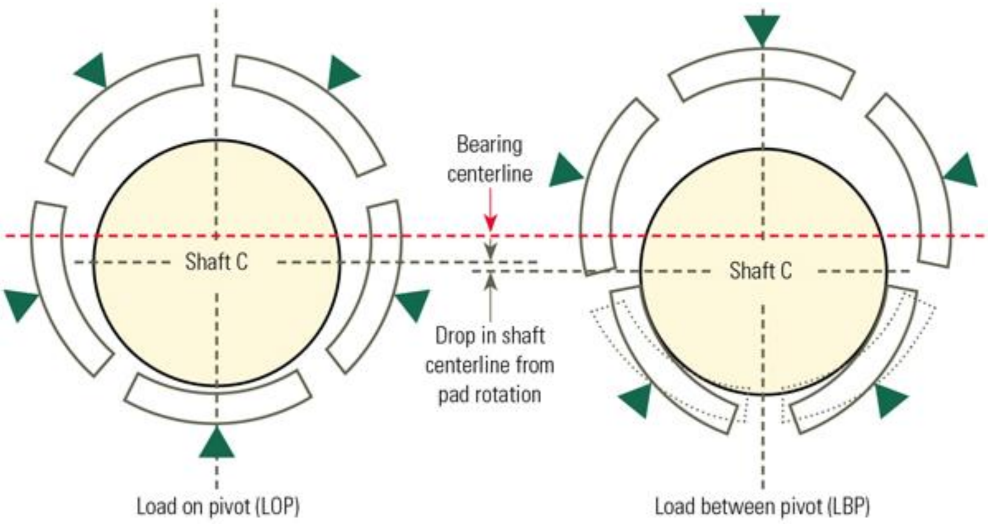

Figure 3-c Shaft must stay centered. Relocation of shaft center in a segmented bearing while rotating can cause vibration [5].

We should not overlook the concentricity of the rotor with the bores and couplings. Correct rotor alignment is lost when the axis of one rotor is not continuous with the rotor in the following casing, in multi-casing steam turbines. The individually connected drivelines must operate as one long continuous, yet flexible driveline. After major steam turbine maintenance, it is important to confirm the alignment of the rotor to the couplings as well as any other factors that may cause change to the primary positions of individual casings, bearing, and rotors. During maintenance, if rubbing is observed at the end or intermediate sealing of the rotor or eccentricity of couplings, it is necessary to realign the driveline to avoid high turbine vibration, contact and rubbing of glands and so on (Figure 3-c).

Shaft curvature also shifts the rotation axis of the shaft by moving the mass center of the rotor, creating vibration. This vibration affects blades in three significant ways.

First, the vibration causes blade structural problems [7]. The centrifugal forces encountered during operation are significant, causing an increase in the tensile forces in the blade cross-section and, if the center of mass is not on the radial line, bending stresses also occur. Additionally, bending stresses are created in blade joints under the pressure effects of the HP steam flowing axially through the turbine cylinder. The magnitude of these stresses is dependent on the flow rate of steam, the temperature drop across the blade stage, the rotational speed of the blades, and the blade weight. The temperature of the steam, superheated in the first stage and saturated in the final stages, will have an effect on the mechanical properties and corrosion of the blade materials (Figure 3-d).

Figure 3-d Corrosion causes imbalance. Failure of this high-pressure rotor control stage was caused by uneven distribution of steam due to corrosion [6].

2.3 Frame Vibration

Frame vibration is also excited by unbalanced magnetic pull and by any vibration produced in the core. There are known cases of vibration resonance occurring on the frame as a result of the frame having a resonant frequency near line or twice line frequency. Resonant frequencies may be corrected by either adding mass to the frame to bring the natural frequency down, or by stiffening the frame to drive the natural frequency higher; the object being to move the frame natural frequency away from the exciting frequencies by at least 10%.

Severe damage to the frame can occur by initiating cracks in the frame welds or in the frame members themselves. Residual damage from the high vibrations associated with frame vibration is likely to be transmitted to other components of the generator if the situation becomes severe [6].

Good core-to-frame coupling is required to ensure that the core and the frame move together. There is evidence of numerous cases where core frames became “uncoupled” from the core and impacting damage found at the core to keybar core

Fig. 4 Core and frame accelerometers [6].

and Frame Vibration Testing The maximum vibration of the stator core and core frame should be less than 50 µm (about 2 mils) peak to peak (unfiltered), with no natural resonance within the frequency ranges of 50–75 Hz and 100–140 Hz for 60 Hz systems, and about 40–65 Hz and 80–120 Hz for 50 Hz systems. The problems associated with high vibrations are premature stator core inter-laminar and stator winding insulation wear, and structural problems with the core and frame. To maintain low vibration and avoid problems, it is best to have a tight core and to have good mechanical coupling between the core and frame. This ensures that no wear occurs between the two components at the key bars. Low absolute vibrations and low relative friction.

Chapter 3

Vibration in Rotor Generator: Challenges

3.1 Generator Vibration and Maintenance Testing

Vibration between the core and frame, with the two components in phase, is a good indicator that the core and frame structure is sound. Testing can be done off line or on line. However, for either type of test, vibration transducers (accelerometers) must be mounted on the core and frame, internal to the machine. This includes the stator center, both ends, and locations on the circumference based on the nodal vibration patterns of the stator (four nodes for two-pole machines and eight nodes for four-pole machines). The vibrations are generally measured in the radial and tangential directions when looking toward the end of the machine. In addition it is desirable to use a number of portable, magnetic based transducers, which may be moved around the outside of the generator casing, to ensure complete analysis of the machine. Off-line testing is not generally done unless there is a known problem with core and frame vibration. The excitation source must be artificially applied in this method, and there are a couple of ways to accomplish this. One is to simply strike the frame with a heavy rubber hammer and measure the frequencies where the vibrations peak. However, this does not usually produce a significant result because the stimulus is so small. The other method is to attach a shaker device to the frame to stimulate the stator at a fixed frequency, and then measure the frequencies where the vibrations peak. The shaker method generally produces good results because there is a significant stimulus and it can be controlled. The problem with this type of testing is that it does not give an accurate picture of the true vibration of the core and frame in operation. On-line testing is generally the best way to get a complete vibration analysis of the core and frame. With on-line testing it is possible to look at all operating modes of the stator and determine which parameter has the most influence on vibration. During testing the variable parameters are stator current, field current, hydrogen pressure, and hydrogen temperature. In some cases it has also been useful to valve out individual hydrogen coolers, successively, to change the cooling pattern and determine its effect on the core and frame vibration. When core and/or frame vibrations are present, it is necessary to determine if the vibrations are most prevalent on the core or frame, and if the two components require better coupling to each other. The possibilities of what may be found on any individual machine are too vast to cover here. However, as a rule-of-thumb, it is often found that the core and frame will have a natural frequency that is too close to the forbidden zones, and that they do require some artificial means to ensure better mechanical coupling between the two. In such cases vibration damping is also usually required. When trying to dampen excessive vibrations, there are two methods employed. The first method requires adding mass to lower the natural frequency. The second method entails stiffening the structure to raise the natural frequency. Adding mass can be difficult if there is no good place to attach it, and stiffening can sometimes cause problems with overstressing the frame welds, causing them to crack. Stator core and frame vibration problems are very complex to analyze, and even more complex and expensive to solve. They should be addressed on a machine-to-machine basis, as each case will be unique.

3.2 Rotor Vibration Testing

Rotor Vibration measurement is one the most important on-line measurements taken on the machine. Each manufacturer gives its own recommendations for alarm and trip. The information in the following table is from an O&M engineering specification:

- Machine Maximum Amplitude: 0–999 rpm 3 mils 1000–1499 rpm 2.5 mils 1500–2999 rpm 2 mils 3000 rpm & above 1 mil

Vibration is monitored continuously, and vibration charts are normally available at the control room. Vibration is monitored in all turbine and generator bearings.

3.3 Rotor Mechanical Testing

Although vibration monitoring is generally considered an on-line monitoring function, there are many occasions where it is necessary to carry out additional and specific vibration testing. This would be to look for such problems as component rubs or rotor winding shorted turns to determine the correct course of action. This type of detailed vibration testing is very specialized and requires additional equipment to be connected to the vibration probes installed on the generator. The type of additional testing inferred would be to allow characterization of the vibration measurements into both magnitude and phase relation, and to allow frequency spectrum analysis during cold and hot run up to speed and run downs from speed. In addition load changes and field current changes allow the differentiation between mechanically and thermally induced vibrations. This is a very detailed topic for which entire books have been written and the literature that can be found is substantial. Refer to Chapter 6 for an additional discussion of the subject.

3.4 Fan Shaft Critical Speed

In the fan industry has been some confusion as to the exact definitions of critical speed and resonant speed; therefore, to clarify the problem, the Air Movement and Control Association (AMCA) [3] has adopted the following definitions:

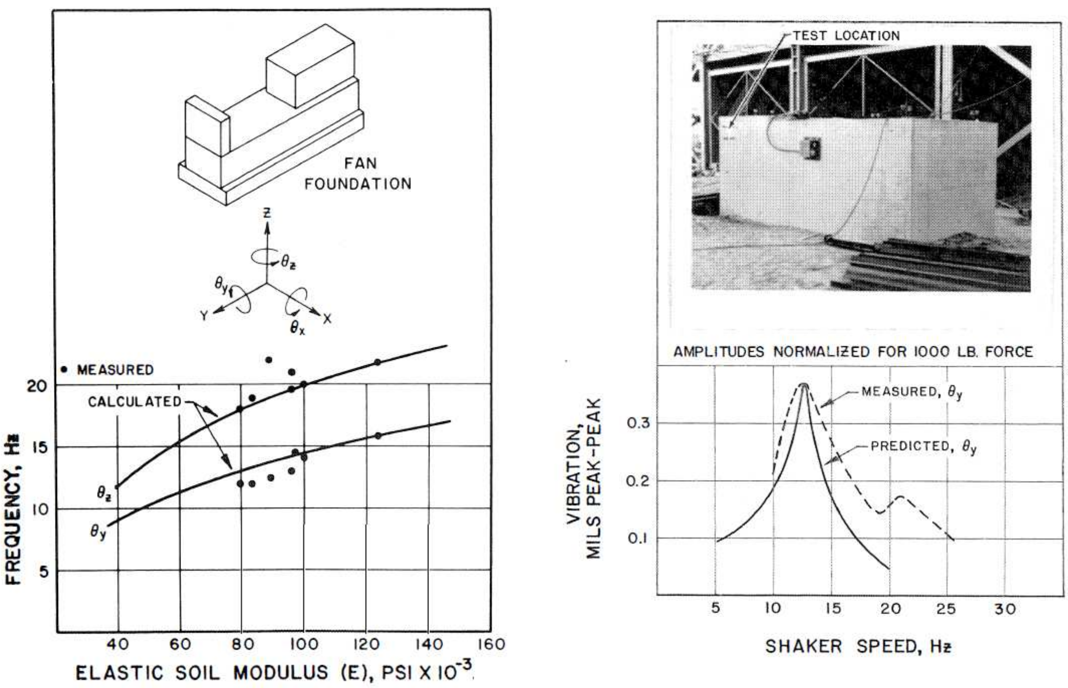

Figure 5. Comparison between measured and calibrated vibration [4].

Critical Speed: A critical speed is that speed which corresponds to the natural frequency of the rotating element (impeller and shaft assembly) when mounted on rigid supports. (Note: This is generally referred to as the rigid-bearing critical speed)

Design Resonant Speed: Design resonant speed is that speed which corresponds to the natural frequency of the combined spring-mass system of the rotating element, oil film, bearing housing, and bearing supports but excluding the foundation (foundation stiffness is considered as infinite).

Installed Resonant Speed: Installed resonant speed is that speed which corresponds to the natural frequency of the combined spring-mass system of the rotating element, oil film, bearing housing, bearing supports, and includes the effect of foundation stiffness. These ID fans mounted on the resonant foundations also had increased vibration amplitudes due to critical speed effects. The rigid-bearing critical speed and installed resonant speed were calculated to be 1180 rpm and 960 rpm, respectively.

The lateral critical speed should be at least 20% from the running speed to prevent excessive vibration amplification. In this case, the calculated installed resonant speed was only 7% above the running speed; therefore, amplification could be expected. It is important that fan users be aware of the critical speed definitions used in the fan industry and refer to the installed resonant speed when writing design specifications.

Major modifications such as shortening the bearing span would be required to increase the shaft critical speed to 20% above the running speed. These modifications were not possible and it was decided to reduce the shaft vibrations by improving the fan balance.

Chapter 4

Research Methodology

The generally accepted methods for vibration problem of industrial equipment include; Force Reduction, Mass Addition, Tuning, Isolation, and Damping. This research will focus on damping related method, and describe practical methods for their application. The Jordan Petroleum case study will be presented, with emphasis on pragmatic solutions to their rotor vibration problems.

Chapter 5

Proposed Rotor Damper Vibration Design

The proposed design provides highly engineered damping and stiffness to shift critical speeds and increase the dynamic stability of the rotor/bearing system.

In the above proposed design, stiffness and damping are independent of each other and can be precisely controlled. This produces higher and more accurate damping capability than a conventional squeeze film damper and makes. This technology a leading solution for controlling vibrations. Stiffness and damping are each optimized for the application through rigorous rotor dynamic analysis.

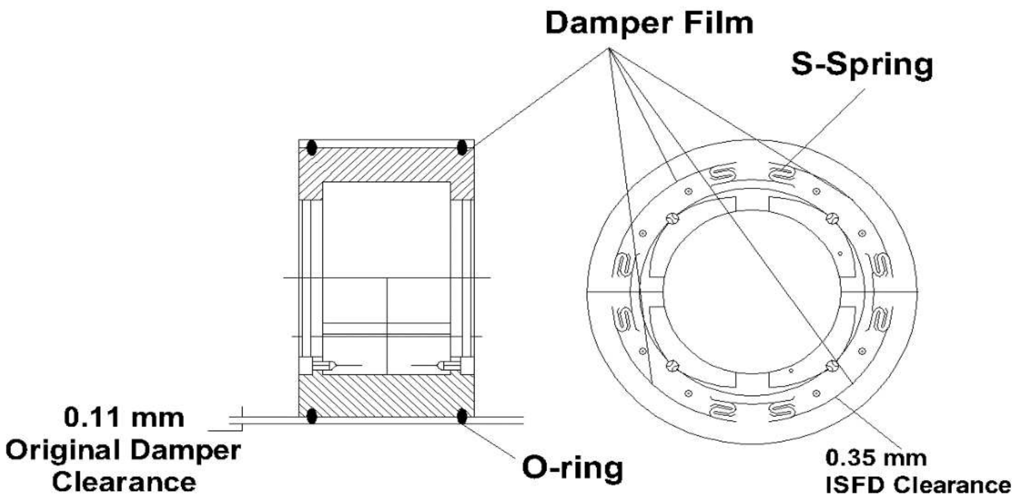

Figure 6. Proposed Damper Design (Front View).

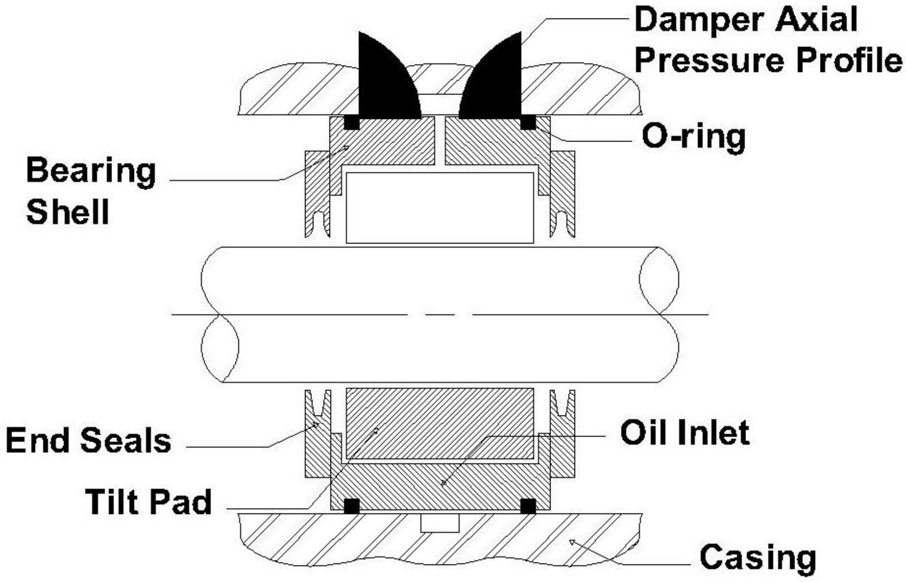

Figure 7. Proposed Damper Bearing Shel Design (Front View).

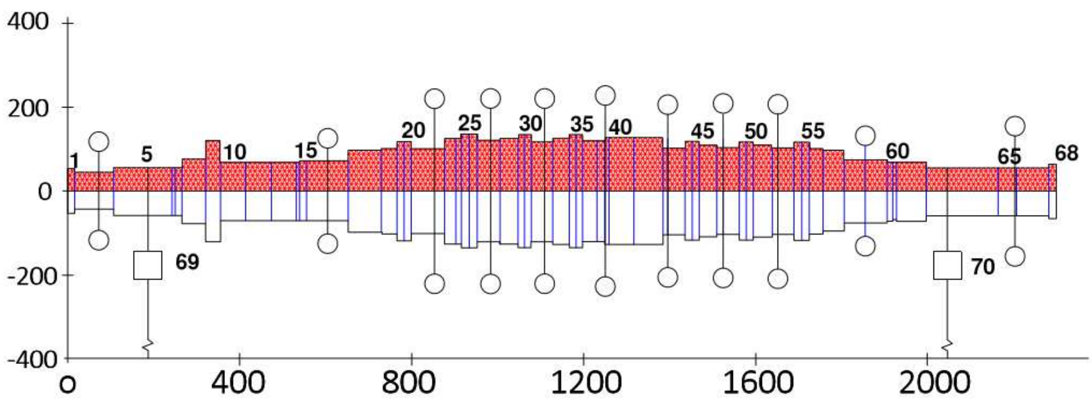

Figure 8. Rotor Load (x-axis in Newton) vs. longitudinal Displacement as a result of vibration (in mm).

$$k = \frac{2\mu R L^3 \varepsilon\omega}{c^3(1-\varepsilon^2)^2}$$

$$C = \frac{\pi\mu R L^3}{c^3(1-\varepsilon^2)^{3/2}}$$

- Damper radial clearance (c) = 0.110 mm
- Damper radius (R) = 95.25 mm
- Effective damper length (L) = 37.85 mm
- Stability is very sensitive to damper eccentricity ratio (ε)
- Added O-ring stiffness

Damper stiffness and damping coefficients (without O-ring stiffness):

K=1.94E+07 N/m, C=1.88E+05 Ns/m at ε=*0.25*

K=6.06E+07 N/m, C=2.62+05 Ns/m at ε=*0.50*

K=1.70E+09 N/m, C=2.06E+06 Ns/m at ε=*0.9*

Level l stability predicts that the rotor is unstable prior of the proposed design:

$$Q_A = \frac{(\mathrm{HP})\mathrm{B_c}C}{\mathrm{D_c}H_c N}\left(\frac{\rho_d}{\rho_s}\right) = \frac{(15000)*(3)*(63)}{(19.64)*(0.78)*(12142)} * (8)$$

$$= 121.932\frac{klb}{in} or\ 2.13E07\ N/m$$

The proposed design, manufactured through mechanical discharge machining, can integrate the bearing and damper into one unit for a space-saving solution suitable for new and retrofit installations. This technology can be used with tilt pad, Flexure Pivot® tilt pad, fixed profile or rolling element bearings.

By introducing flexibility into the rotor/bearing system and providing optimum damping, this technology maximizes the energy dissipation at the bearing locations and significantly improves the stability of the system. Shift Critical Speeds and Reduce Amplification Factor.

The design can shift critical speeds and significantly reduce the amplification factor. With the reduction in amplification factor, machine seal clearances can be tightened to reduce gas or steam leakage.

It also reduces dynamic Bearing (Transmitting) Forces. This technology reduces the dynamic load that is transmitted to the bearings, which reduces pedestal vibration and increases bearing life, particularly for rolling element bearings. For fluid film bearings, the technology can mitigate pivot wear and reduce lateral fatigue.

The proposed decrease Unbalance Sensitivity. It reduce the sensitivity to unbalance, protecting impellers and seals from rubbing and increasing maintenance intervals.

Chapter 6

Results and Conclusion

We found from the previous sections the following general rules for maintaining the vibration level to its minimum level:

• Rule # 1: Match Design Point to System Head & Flow Requirements

• Rule # 2: Use a Long Straight Piping Run to the Inlet

• Rule # 3: Careful When & How You Throttle

• Rule # 4: Minimize Nozzle Loads

• Rule # 5: Avoid Structural Natural Frequencies

• Rule # 6: Minimize Load Cycling, if Practical

• Rule # 7: Select Materials Based on Corrosion, Fatigue , and alignments

- During the initial startup of the centrifugal fan at a JPR, the bearing housing vibrations exceeded the 2 mils at 900 rpm specified in the Japanese contract. It was reported that the fans could not be balanced satisfactorily; when the vibrations were reduced on the fan bearings, the vibrations would increase on the motor bearing.
- Detailed investigations revealed the following problems which made balancing difficult:
- 1. Foundation resonance.
- 2. Fan shaft critical speed.
- 3. Interaction of adjacent fans.
- 4. Corrosion Imbalances.

Foundation Resonance Vibrations measured on the fan bearings during startup and coast downs of the fans (Figure 6) revealed what appeared to be a foundation natural frequency near the running speed.

These fans were designed to have their lowest foundation natural frequencies at least 70% above running speed, which should have been adequate. Further conservatism was included in the design calculations by using a lower limit value of Young's Modulus (E) of 500,000 psi for the sandstone formation beneath the foundation.

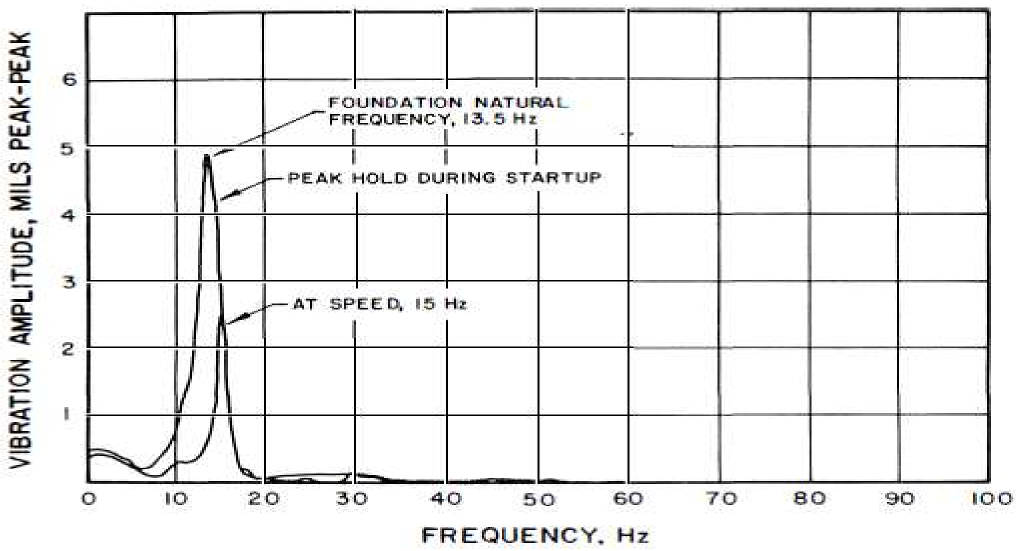

Figure 9. Spectral Analysis of Bearing Housing Vibrations during Startup [7].

On three of the fans, the natural frequency was just below the running speed, but on the fourth fan the resonance was slightly above the running speed. To verify that these were foundation resonances, a variable speed mechanical shaker was attached to the foundation and run through a speed range of zero to 30 Hz. The shaker vibration response agreed with the coast down data and verified that the first foundation natural frequency ranged from 12 to 16 Hz for these "identical" four fans.

To evaluate the discrepancy between measured and calculated foundation resonances, the foundation-soil dynamic system was modeled with a computer program developed by Southwest Research Institute (SwRI). Foundation natural frequencies were calculated for a range of soil moduli and compared with shaker data. As shown in Figure 7, the effective soil modulus must be from 80, 000 to 120, 000 psi to match the measured foundation natural frequencies. The lower than expected soil stiffness beneath the foundations was probably due to blasting, over-excavation, and backfill during construction.

A shaker test was conducted on the partially completed foundation block of an adjacent unit (Figure 8) to determine natural frequencies and vibration mode shapes. These tests showed that the effective soil modulus was similar to that obtained for Unit 1, which indicated that the Unit 2 ID fans would also have foundation natural frequencies near the running speed. Foundation modifications to reduce the vibration amplitudes were analyzed using the computer program. However, it was determined that most modifications investigated were impractical based on cost or space limitations and other methods were investigated to reduce the vibrations. Shaker tests on one foundation revealed that the vibrations were increased by a factor of six when dirt was removed from the side of the foundation. Based upon this test, sand bags were temporarily placed against the side of the foundation to increase damping and lateral restraint on the foundation. The vibrations were reduced by a factor of approximately two to one.

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