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DEVELOPMENT OF LONG-LIFE AUXILIARY BEARINGS FOR CRITICAL SERVICE TURBOMACHINERY AND HIGH-SPEED MOTORS

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ABSTRACT

The development program undertaken by SKF/Revolve Magnetic Bearings Inc. (SKF/Revolve), to develop long life auxiliary bearings for critical service turbomachinery and high-speed motors, is described. The outcome of the program was the development of auxiliary bearing systems capable of multiple shaft de-levitations for machines that experience extended coast down conditions. The targeted machines were large shaft turbomachinery and high-speed motors with either rigid or flexible shaft designs in the 5 to 20 MW range. The program involved the development of simulation and modeling tools, identification of auxiliary bearing system requirements and design considerations, building of scaled and full size test rigs, optimization of the auxiliary bearing designs and validation testing. De-levitation tests on all three test rigs, a compressor demonstrator, a steam turbine/generator demonstrator and a high-speed motor demonstrator, concluded that there was more damping in the test rigs than predicted by the model resulting in better rotodynamic behavior and confirmation that the modeling tools are conservative. In all cases, shaft de-levitations resulted in acceptable rotordynamic behavior with no evidence of shaft whirl instability. The rolling element design proved to be the superior design for long life auxiliary bearing applications. Identification of rolling element auxiliary bearing design failure modes and ways to detect them were also investigated. All failure detection methods used during the test program were effective in identifying when a bearing or landing sleeve failure had occurred. However, only shaft de-levitated position monitoring and auxiliary bearing clearance monitoring showed some potential for providing predictive maintenance capabilities.

INTRODUCTION

In many turbomachinery and high-speed motor applications, catastrophic power loss may result in long coast downs from full speed. If magnetic bearings are used in these machines the auxiliary bearings have to be able to withstand such coast-downs while preventing internal machine damage. Because of the costs involved with facility downtime it is desirable to have auxiliary bearings capable of withstanding multiple catastrophic power loss events. This paper describes the auxiliary bearing development program that was undertaken to address this requirement.

PROJECT DESCRIPTION

The development program undertaken by SKF/Revolve consisted of the development of simulation and modeling tools, identification of auxiliary bearing system requirements and design considerations, building of scaled and full size test rigs and validation of the designs.

Modeling Tool Development

The first phase of the project involved the development of simulation and modeling tools that predict the rotordynamic behavior of a shaft during de-levitation. Static de-levitation data from the test rigs were used to calibrate the model and the observed behavior of the machine during dynamic testing was compared to the predictions of the model. The goal was to determine the extent to which the model can predict the machine's behavior so that new machines could be designed without having to do extensive de-levitation testing to validate the auxiliary bearing system design.

The auxiliary bearing and rotor models take into account damping and stiffness characteristics of all the components in the auxiliary bearing system and support structure.

Auxiliary Bearing System Design

The service life requirements for the steam turbine/generator demonstrator were 8 full speed de-levitations of 10-minute coast down time each at a bearing D_N equal to 1.4 million. The service life requirements for the high-speed motor demonstrator were 5 full speed de-levitations of 90 seconds coast down time each at a bearing D_N equal to 2.14 million. The initial modeling and bearing design tests on the compressor demonstrator were completed in an effort to develop the bearing designs capable of achieving these service life requirements.

The design process typically requires several iterations before all aspects of the design, magnetic bearing air gap, auxiliary bearing clearance, damping ring clearance, damping ring stiffness, etc., are satisfied. Consideration must also be given to the rotational stiffness of the bearing housing end wall or pedestal supports so that critical magnetic bearing air gaps are maintained during shaft de-levitations.

From experience, initial values for auxiliary bearing system stiffness, damping and clearances are entered into the model. Model outputs of shaft maximum impact forces, shaft deflections and remaining magnetic bearing air gaps, are derived from bearing centerline drop simulations which consider worst case conditions of unbalance, operating temperature, auxiliary bearing clearances and shaft orbit size. If the remaining magnetic bearing air gap is not sufficient, then a change of the auxiliary bearing assembly stiffness and clearances are required and the simulations are rerun until the air gap and auxiliary bearing clearances are satisfied. Damping ring stiffness must be checked to ensure that there is sufficient stiffness to minimize the number of times the damping ring housing clearance is taken up during a shaft drop simulation. Once the stiffness and damping properties are defined then the shaft dynamic drop behavior can be analyzed.

Auxiliary Bearing Design Considerations. The precision angular contact bearing pairs and the sleeve bearings used in the de-levitation testing programs were designed under a development program undertaken by the specialty bearing products group in SKF. The development program consisted of a design requirement evaluation, materials selection and evaluation, thermal analysis, mechanical characteristics analysis, modeling and scale testing.

Optimization of the bearing design for maximum service life was achieved by doing sensitivity analysis on bearing design issues such as operating temperature, race differential temperature, shaft speed, bearing pre-load, percent curvature, contact angle and applied load. The outcome of SKFs' sensitivity analysis formed the basis for the full-scale steam turbine/generator and high-speed motor demonstrator auxiliary bearing designs.

Damping Ring Design Considerations. The function of the damping ring in an auxiliary bearing system is to absorb the impact energy of the shaft during shaft de-levitation to minimize shaft bounce and also provide damping to the system to avoid bending mode excitation during shaft coast-down. The latter is particularly important on flexible shaft designs. Elastomeric o-ring and composite mesh damping ring designs were tested during the de-levitation program. Both designs have advantages and disadvantages, and the application of each material is based on the amount of damping and stiffness identified in model simulation plus factors such as operating temperature, static load and in some cases space limitations. For most heavy shaft applications there are significant advantages in using the composite mesh design developed by SKF/Revolve.

The composite mesh damping ring consists of four essential components: an outer housing, compression ring, compliant damping material and an inner housing. The composite mesh used as the damping material is compressed between the compression rings and the inner housing. The stiffness of the damping ring assembly is a function of the amount of compression used on the composite mesh and the radial clearance between the inner housing and the outer housing. Damping ring radial clearances range from 50 to 200 μm depending on the application. The compression can be adjusted by changing the thickness of the compression ring. The stiffness of the final assembly is verified by a load vs. deflection test and adjusted to match the required stiffness determined in the simulation model.

Landing Sleeve Design Considerations. The primary purpose of landing sleeves is to provide a sacrificial surface so the underlying shaft does not get damaged. Landing sleeve designs are influenced by the following factors:

- Extreme bearing-race skidding during the time that the auxiliary bearing inner race is accelerating up to shaft speed
- High shaft impact and coast down forces
- High mechanical stresses due to shaft rotational speed, interference fits and different thermal expansion rates of the shaft and sleeve at their extreme hot and cold operating conditions
- Compatibility with the bearing race materials

- Possible harsh, corrosive environments
- Possible solid particle contamination

Several landing sleeve materials were tested during the auxiliary bearing development program. Special gall resistant coatings were also employed to increase landing sleeve service life. To maximize reliability it was found that the a landing sleeve solution must have a combination of the following properties:

- High gall resistance
- High yield strength
- High fatigue strength
- Corrosion resistance
- Preferably, a softer material than the bearing race so that solid particles can imbed in the landing sleeve to avoid damage to the bearing race.

Test Rig Descriptions

The three active magnetic bearing test rigs used during the development program included a compressor demonstrator, a steam turbine/generator demonstrator, and a high-speed motor demonstrator. The auxiliary bearing service life requirements identified for the end use machines provided the design requirements for test rigs and defined the development program.

Compressor Demonstrator. This scaled rotor rig was used to test sleeve and rolling element bearing designs. The data collected was used to validate the modeling tools and to partially validate the proposed bearing designs for the other demonstrator rig applications.

The rig used a variable speed electric motor to drive a brake shaft flexibly coupled to the test rotor. The motor design was a high-speed spindle on an active conical magnetic bearing system. The motor speed was controlled by a variable frequency drive, with limited regenerative braking capability. The main brake shaft provided the emergency and controlled stopping requirements. Mechanical stress in the thrust disk design limits the test rig speed to 10,300 RPM.

The Compressor Demonstrator rotor was approximately 2 meters in length with a mass of 360 kg and a free-free first bending mode at 6,400 CPM. The test rig design allows for flexible auxiliary bearing location so different configurations can be evaluated. The test rig design could also accommodate the testing of an "inside the shaft" auxiliary bearing concept whereby the bearing is fixed at the ID of the inner race and the shaft lands on the OD of the outer race.

For each auxiliary bearing configuration a set of rotor drop tests were done at increasing speed increments of 2000 rpm (equivalent bearing D_N numbers for specific end user applications) until the bearings failed or the maximum test rig speed of 10,300 RPM was reached.

The following data was gathered for each drop:

- Rotor radial position in two axes at five rotor locations
- Temperature of the balls on the non-drive end auxiliary bearing
- Speed of auxiliary bearing inner race using optical sensor
- Ball passing frequency
- Shaft speed

Steam Turbine/Generator Demonstrator. This full sized test rig was used to validate the final auxiliary bearing design for an end use application. The rotor was approximately 2 meters in length with a mass of 1,200 kg. The journal weight at the driven end was approximately twice as that at the non-drive end. The maximum operating speed was 8,000 RPM with a shaft first free-free bending mode at 11,220 CPM. The auxiliary bearing arrangement on the DE consisted of the outer race mounted in a housing with the shaft landing on the ID of the inner race during de-levitation. The auxiliary bearing arrangement on the NDE consisted of the inner race mounted on a housing with the shaft landing on the OD of the outer race during de-levitation. The auxiliary bearing arrangement on the NDE was referred to as the "inside the shaft" design. Special high temperature, corrosion resistant auxiliary bearing race materials were required due to the end use application.

A conventional variable speed electric motor was used to drive the brake shaft through a speed increasing pulley arrangement. The brake shaft was flexibly coupled to the test rotor. The motor speed was controlled by a variable frequency drive, which had some regenerative braking capability. The main brake shaft provided the emergency and controlled stopping requirement.

Rotor drop tests were done at increasing speed increments of 2,000 RPM up to the maximum speed of 8,000 RPM. The following data was gathered for each drop:

- Rotor radial position in two axes at the magnetic bearing sensor planes
- Trend of shaft positions and speed during run-down
- Shaft de-levitated position after a drop test
- General observations of the condition of the auxiliary bearing during a hand roll test immediately after a de-levitation coast-down

High-speed Motor Demonstrator. This full sized test rig was used to validate the final auxiliary bearing design for an end use application. The rotor was approximately 2.5 meters in length with a mass of 1,850 kg. The maximum operating speed was 12,000 RPM with a shaft first free-free bending mode at 13,100 CPM. The auxiliary bearing arrangement for both the DE and NDE consisted of the outer race mounted in a housing with the shaft landing on the ID of the inner race during de-levitation.

A conventional variable speed electric motor was used to drive the rig through a speed increaser gearbox. The gearbox high-speed pinion was flexibly coupled to the brake shaft, which in-turn was flexibly coupled to the test rotor. The motor speed was controlled by a variable frequency drive, which had limited regenerative braking capability. The main brake shaft provided the emergency and controlled stopping requirements.

Rotor drop tests were done at increasing speed increments of 2,000 RPM up to 10,000 RPM and every 1,000 RPM to the maximum speed of 12,000 RPM. The following data was gathered for each drop:

- Rotor radial position in two axes at five rotor locations
- Temperature of the balls on the DE auxiliary bearing
- Speed of the DE auxiliary bearing inner race
- Ball passing frequency of the DE auxiliary bearing
- Vibration of the DE auxiliary bearing inner housing
- Trend of shaft positions and speed during run-down
- Shaft de-levitated position after a drop test
- General observations of the condition of the auxiliary bearing during a hand roll test immediately after a de-levitation coast-down

RESULTS

The auxiliary bearing system designs for the steam turbine/generator shaft and the high-speed motor demonstrators in terms of meeting acceptable rotordynamic behavior and service life goals were validated. The total number of drops on the respective bearings for each system exceeded their respective bearing life requirements and acceptable rotordynamic behavior was also demonstrated.

Modeling Tool Development

Although the correlation between the static drop simulations and the test results during model calibration was very good, test results showed that the modeling tool consistently underestimates the overall level of damping in the system during dynamic drop testing.

Compressor Demonstrator Results

Several iterations on precision angular contact bearing design were undertaken until a successful design was found for both configurations, the shaft contacting the inner race, and the shaft contacting the outer race during de-levitation. The final designs were able to withstand multiple full speed de-levitations at bearing D_N values equal to 1.4 million. Coast down times for each drop were less than 60 seconds each.

The sleeve bearing design tested on the test rig failed after several de-levitations at a D_N value of 0.6 million. As a result, no further testing of sleeve bearing designs was done.

Steam Turbine/Generator Demonstrator Results

The auxiliary bearings experienced 11 de-levitations before the test program was stopped. The first 5 de-levitations were 1 minute coast-downs at 2,000, 2,000, 4,000, 6,000 and 8,000 RPM. The last 6 de-levitations were 10-minute coast-downs from full speed (8,000 RPM; D_N = 1.4 million). All drop tests showed that the system was very well damped and there were no signs of shaft whirl.

The shaft de-levitated positions indicated a problem after the eighth de-levitation, corresponding to the fourth full speed test. Further investigation found that the drive-end landing sleeve suffered galling damage and had to be replaced. Since the auxiliary bearings were in good condition, the full speed de-levitation testing continued. The shaft de-levitated positions started to indicate a problem after the eleventh de-levitation so the auxiliary bearings were again removed for inspection. The non-drive end bearings showed signs of false brinelling, outer race misalignment and minor galling on outer ring OD (landing surface during drops). The drive end bearings showed signs of galling on the inner ring ID (shaft contact surface during drops). The drive-end landing sleeve showed signs of minor galling damage as well. Although the bearings appeared to be capable of handling further de-levitations, the test program was halted due to the customer's delivery requirements for the demonstrator.

A typical 10-minute shaft coast-down trend is shown in Figure 1. It demonstrates that the energy in the shaft from the 200 μm drop from bearing centerline is effectively dampened. There is some evidence of minor shaft rigid mode excitation as the shaft speed decreases.

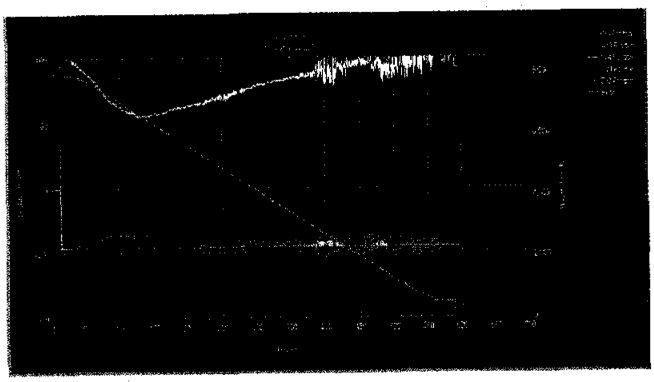

Figure 1 – Typical coast-down trend of shaft radial position on the sensor axis

Figure 2 - Typical coast down trend of shaft radial position on the sensor axis

High-speed Motor Demonstrator Results

The auxiliary bearings experienced 16 de-levitations before a DE auxiliary bearing failure stopped the test program. The first 12 de-levitations were 90 seconds or less, with coast-downs from 2,000, 2,000, 2,000, 4,000, 4,000, 4,000, 6,000, 6,000, 6,000, 8,000, 10,000, and 11,000 RPM. The last 4 de-levitations were 90 second coast-downs from the maximum speed of 12,000 RPM (D_N = 2.1 million). All drop tests showed that the system was very well damped and there were no signs of shaft whirl.

Failure analysis identified the cause to be dirt and abrasive material contamination from the adjacent brake system. This is a viable conclusion because the dust seal on the DE bearing was not installed during the test program so ball temperature could be monitored. Unfortunately, this also allowed abrasive materials to contaminate the bearing. It is important to note that the magnetic bearing components were not damaged as a result of catastrophic failure of the auxiliary bearings. The customer bearing life requirements were met and the test program was considered to be complete.

A typical 90-second shaft coast down trend is shown in Figure 2. It demonstrates that the energy in the shaft from the 250 μm drop from bearing centerline is effectively dampened. On all de-levitations above 4,000 RPM, there was an increase in shaft displacement at 3,500 RPM. This is most likely due to the shaft decelerating through either a system resonance or a shaft rigid mode. Again, note that shaft whirl was not observed at any time during the de-levitation.

DISCUSSION

To maximize service life in an auxiliary bearing design, careful consideration must be taken in the selection of materials (bearing races, balls, landing sleeve), bearing race contact stress, peripheral rotor speed (D_N), internal bearing radial and axial clearances, bearing protection from contamination, and the ability of the support structure to maintain acceptable internal alignments and concentricity toloerances. To achieve acceptable rotordynamic behavior, careful consideration must be taken in the modeling of the entire system (auxiliary bearing assembly, the auxiliary bearing housing support structure, and machine foundation).

Modeling Tool Validation

The test data demonstrated that there was more damping in the system than was predicted by the models. This resulted in more stable shaft behavior than expected. Additional sources of damping may come from the following areas:

- Experimental data from independent tests on the composite mesh damping ring material suggested that the damping coefficient is higher than expected and it varies with the amount of the stiffness required in the damping ring assembly for the application
- Uncertainty on how to model the damping of the system due to the complexity of the design, i.e. viscous, hysteretic or Coulomb damping or a combination
- Underestimating the contribution of overall damping from the bearing housing structural support and the machine foundation

As a result, the risks associated with rotordynamic stability of future designs is reduced due to the conservatism of the current modeling tool to estimate the overall level of damping in the system. It is conceivable that if a system is conservatively modeled, an extensive dynamic drop test and validation program on future machines may be avoided. The advantages of this would be to avoid exposing the machines to potential damage and to avoid the costs associated with an extensive test programs.

Auxiliary Bearing System Design

For acceptable rotordynamic behavior during de-levitation, defined as maintaining the shaft below the bearing centerline with no observable shaft whirl, the impact energy of the shaft must be quickly dissipated and the tangential contact surface forces that can cause shaft whirl conditions must be minimized. The performance of the auxiliary bearings system on all three-test rigs demonstrated the validity of their designs. The damping ring and system design was able to control the shaft rotordynamic behavior at de-levitations above and below the shaft 1st bending mode and through the shaft rigid modes as well.

The auxiliary bearing designs tested were capable of surviving multiple drops at high speed, for both long and short coast down periods. To maximize bearing life and machine protection careful attention was paid to the auxiliary bearing housing support structure and the machine casing designs. They were sufficiently rigid so critical internal alignments and concentricity's of the auxiliary bearings and magnetic bearing components were maintained. The bearing failure on the high-speed motor demonstrator showed the importance of protecting the bearings from contamination. The catastrophic failure that occurred also demonstrated the advantages of the ceramic ball design over a sleeve bearing design in terms of maintaining bearing integrity and its ability to protect the magnetic bearing components during an auxiliary bearing failure.

The landing sleeve design for the high-speed motor demonstrator successfully met the application requirements and service life of the bearings. Even during the catastrophic failure of the DE auxiliary bearing the interference fit of the sleeve proved to be sufficient to maintain positive contact with the shaft so further damage to the shaft was avoided. As shaft weight increases sleeve materials with greater surface hardness should be considered.

The landing sleeve design for the steam turbine/generator demonstrator did not have sufficient material strength to withstand the contact stresses that occurred during the long coast down times. As a result the landing sleeve material was changed to material of superior strength properties, however to date further drop testing has not been done to validate the new sleeve material.

Failure Identification and Detection

One of the goals of the development program was to identify auxiliary bearing system failure modes and determine a reliable way of detecting them. Some failure modes were experienced during the drop test program while others are listed as possible failure modes for an auxiliary bearing application. Although, factors that influence the bearings' ultimate life will vary for each application, some of them can be avoided by applying proper design and maintenance techniques while others are a result of design limitations or ultimate component service life.

None of the failures experienced during the development program resulted in magnetic bearing component damage. The damage was always isolated to auxiliary bearing components only. This is an important result because it suggests that the risks of using an auxiliary bearing beyond its useful life are reduced. It also suggests that the need for a reliable predictive maintenance tool is diminished. A major part of this is attributed to the strength of the ceramic balls and the fact that the ball elements do not deteriorate/collapse during a bearing failure, as would be the case for steel ball designs. This feature helps contain shaft movement during a bearing failure and provides added protection to all the internal components of the machine.

The auxiliary bearing system failures can in general be put into the following categories:

1. Bearing related failures
2. Landing sleeve related failures

Bearing Related Failures. Auxiliary bearings are subject to extreme impact forces, high race acceleration forces and high differential race temperatures. This makes their application very unique in the bearing industry. Their ultimate service life is measured in minutes or hours instead of years, as is the case for other precision bearing applications. However, if the operating conditions have been properly identified and an appropriate bearing design has been selected then the bearings should be capable of handling multiple shaft de-levitations and protect internal components of the machine from damage. To achieve the service life goals the bearings must be protected from poor handling practices, must be properly supported, installed and aligned, must have adequate dry and/or grease lubrication and must be protected from contamination and corrosive environments. The main causes of failure are the following:

- Overheating
- Inadequate lubrication
- Misalignment
- Contamination

- Corrosion
- Raceway spalling
- False brinelling
- True brinelling
- Fretting

Landing Sleeve Related Failures. As mentioned earlier, landing sleeve material selection and design are influenced by a number of different factors. If the design has taken these factors into consideration then the landing sleeve service life should exceed the auxiliary bearings. To achieve adequate service life the landing sleeves must be properly installed, protected from corrosion and contamination and internal alignments must be maintained. The main causes of failure are the following:

- Galling
- Material deformation/tearing
- Overheating
- Misalignment
- Contamination
- Corrosion

Failure Detection. The following methods were tried during the test programs:

- Ball temperature monitoring
- Seismic vibration monitoring on the outer housing
- Shaft coast-down trending
- Shaft de-levitated position trending
- Auxiliary bearing clearance checks
- De-levitated shaft hand roll checks

In summary, all of the failure detection methods used were effective in identifying when a bearing or a sleeve failure had occurred. However, only shaft de-levitated position trending and auxiliary bearing clearance checks had some potential of providing predictive maintenance capabilities. It is still questionable if alarm limits on shaft de-levitated position could provide effective predictive maintenance capabilities due to the difference between the data from consecutive drops that was experienced during the test program. Plus the fact that some failure modes cause loss of clearance and a rise in shaft de-levitated position while others cause clearances to increase and a drop in shaft de-levitated position. It is recommended to keep records of the shaft positions and clearance checks before and after each de-levitation event and do further investigation if there are significant increasing or decreasing changes.

SUMMARY

The following conclusions of the long life auxiliary bearing system development program can be made:

- The rolling element design proved to be the superior design for long life auxiliary bearing applications.
- The auxiliary bearing program successfully developed auxiliary bearing systems capable of exceeding the project requirements for long life auxiliary bearing applications.
- The most critical considerations in bearing design were found to be materials (bearing races, balls, landing sleeve), bearing race contact stress, peripheral rotor speed (D_N), internal bearing radial and axial clearances, bearing protection from contamination, and the support structures ability to maintain internal alignments and concentricity's.
- To achieve good rotordynamic behavior the critical considerations in the system design were found to be conservative modeling of the system, auxiliary bearing assembly stiffness and damping and the auxiliary bearing housing support structure and machine foundation design stiffness and damping.
- All of the failure detection methods used during the test program were effective in identifying when a bearing or landing sleeve failure had occurred. However, only shaft de-levitated position monitoring and auxiliary bearing clearance monitoring showed some potential of providing predictive maintenance capabilities. It is recommended to keep records of the shaft positions and clearance checks before and after each de-levitation event and do further investigation if there are significant increasing or decreasing changes.
- Results of the model validation program indicate that the developed modeling tools provide a conservative design. The program also demonstrated that the model could be calibrated with a simple static drop test. It was then capable of effectively predicting the rotordynamic behavior of a shaft de-levitation at speed.
- With proper modeling, dynamic testing of similar machine designs may not be necessary.