Landing Test Results for a 6 MW Motor Compressor and Auxiliary Bearing Systems for Corrosive Environments

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Abstract

Auxiliary bearing landing tests have been conducted on a 6 MW vertical motor compressor. The corrosion resistant auxiliary bearing system for the motor compressor is a unique combination design with one auxiliary bearing of bushing type design and the other, a rolling element type design. The results of the landing test were good, with 5 landings completed and with the auxiliary bearings fit for further service. Corrosion resistant auxiliary bearing designs are described. Analysis methods and design considerations for auxiliary bearing systems are discussed.

1 Introduction

Rotating equipment end users are using Active Magnetic Bearing (AMB) systems in an increasing variety of applications. Many of these applications are in capital intensive industries such as Oil and Gas and Power Generation. These large machines require robust investment protection to avoid costly down time. The Auxiliary Bearing (AB) system plays a key role in this investment protection. AMB systems achieve 99.9% availability values in industrial applications, but the AB system is still required as a last line of defense for unusual events.

An AMB system is usually equipped with an Auxiliary Bearing system to support the rotor when in the delevitated state or to protect the machine during process overload or emergency shutdown situations. The primary function of the AB system is to provide a sacrificial, replaceable means of supporting the rotor while the machine is coasting down in a predictable manner until the delevitated state is achieved. This has to be achieved without damaging the rotor or the Magnetic Bearings / Machine Housing. A secondary function of the AB system is to provide load sharing with the AMB to maintain uninterrupted operation during overload transients.

2 Auxiliary Bearing Systems

Waukesha Magnetic Bearing's has developed several auxiliary bearing solutions over the last 20 years. This development process has been in step with the market trend where AMB were first adopted for use in near ambient air environments and then steadily progressed to more and more challenging environments. This progression also included moving the bearings into the process fluid, with greater pressures and increasingly aggressive corrosive environments. Waukesha deploys the optimum auxiliary bearing system design for each application.

Waukesha has two primary design options for auxiliary bearings. These are bushing type AB and rolling element type AB. The bushing type AB is known as the 'Rotor De-levitation System' (RDS) and typically uses an articulated multi-pad bushing type plain bearing which makes contact with a metallic landing sleeve which protects the rotor itself. The pad material is both self lubricating and robust.

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The dry lubricated bushing type design is shown on Figure 1. The radial auxiliary bearing components are the non-rotating stator part fixed to the machine casing and the bearing landing sleeve fixed to the shaft of the machine. An engineered matching of the materials for the stator pad lining and the rotor landing sleeve surface yields an optimum sliding coefficient of friction to maintain good rotordynamic performance.

The radial auxiliary bearing stator is comprised of multiple articulated pads. Each pad is lined with a dry lubricated bushing material. The bushing material is selected based on the bearing compartment environment, and is comprised of a base material matrix with embedded lubricants. The mixture of several types of embedded lubricants in the bushing material ensures a consistent coefficient of friction over the load and temperature operating range.

The articulated pad design provides features that affect the rotordynamic performance of the machine during a landing event. This compliant mounting of the pads provides:

- Provision for pad alignment to shaft slope
- Attenuation of impact force between pad and landing sleeve due to a shaft drop
- Preload force applied to pad. The articulated pad design can easily be preloaded, to minimize the static deflection and contact with seals when the rotor is in contact with the AB
- Adjustable stiffness of compliance travel
- Adjustable damping of compliance travel

The RDS type AB designs are available for both radial and axial functions, with a typical axial design shown in Figure 2.

The RDS is fully adaptable to corrosive environments. The metallic pad backing and housings can be fabricated from highly corrosion resistant alloys. Ceramics and polymers are included in the corrosion resistant material options for the pad lining matrix. Figure 3 shows a corrosion resistant RDS type auxiliary bearing.

A rolling element based auxiliary bearing system is shown on Figure 4. A duplex pair of angular contact ball bearings are mounted in a fixed housing, with a landing sleeve mounted to the shaft of the machine.

A compliant mounting is provided for the ball bearings at the outer races. This compliant system provides an adjustable stiffness and damping characteristic for the mounting and also provides a means for attenuation of the impact force during the landing.

For corrosion resistant applications the material options for the races include stainless steels and nickel alloys. The rolling elements can be made of ceramic material.

Comparing the different types of auxiliary bearings (RDS and rolling element types) shows the RDS can be more easily protected against fouling. Fouling may be in the form of entrained particles, such as sand, or other solids either plating out or condensing out of a gas flow. The RDS does not have any rolling elements that may seize due to fouling, and the compliance gap behind the pads may be easily isolated from the process.

Again comparing the different types of auxiliary bearings (RDS and rolling element types) shows the rolling element type can be packaged in a smaller space envelope. The duplex ball bearing can accommodate the radial and axial AB functions in a compact envelope. The RDS type requires an increased space envelope for the axial and radial AB functions.

3 Corrosive Gas Application and Landing Test Results

Waukesha has provided magnetic bearings and auxiliary bearings for high speed integrated motor compressors. The motor compressors are hermetically sealed vertical machines with a power rating of 6 MW. The rotor mass is approximately 1.5 tons with speeds up to 12800 RPM. The motor compressors are designed for highly corrosive and contaminated applications, using canned magnetic bearings and corrosion resistant auxiliary bearings immersed in the process gas. The process gas can be contaminated with sour gas, mercury, carbonic acid, sand, and condensates.

The auxiliary bearing system design for this vertical motor compressor made use of many of the unique features of the Waukesha AB designs. An integrated design approach was used to design an optimized AB system. The specified environment for the lower bearing compartment has the harshest combination of corrosives and fouling elements. Gravity and pressure effects could create full immersion conditions of liquids and sand in this lower compartment. The RDS type bearing is the only type of AB that would be suitable for this environment. This RDS bearing is shown on Figure 5 and has a bearing bore of 180 mm.

The expected environment for the upper bearing compartment is corrosive gases, without immersion in liquids or solids. The upper compartment was the most sensitive area for rotordynamic considerations, where any added mass and added axial length would have the largest rotordynamic penalty. For these reasons, a corrosion resistant rolling element type AB is used in the upper bearing compartment. This bearing is shown on Figure 6 and has a bearing bore of 120 mm.

Following the successful completion of the integrated motor compressor factory acceptance test program, a full auxiliary bearing landing test program was carried out.

Starting the testing with a new set of Auxiliary Bearings fitted, a series of five landing tests were performed from the maximum speed of 10000 rpm with varying gas flow conditions. Each test was a full coastdown from 10000 RPM to standstill (0 RPM). These tests were carried out in the worst case scenario, without any active braking from the motor. Without braking, the usual 15-18 second emergency stop braking time was increased to approximately 50 seconds, to bring the rotor to a halt. As a consequence, in each of the five tests, significant amount of energy had to be absorbed by the Auxiliary Bearing system.

After each landing event an Automatic Clearance Check (ACC) was performed to assess the condition of the auxiliary bearings. The ACC function uses the AMB system to automatically measures the clearances within the auxiliary bearings of a stationary machine. The AMB system traverses the rotor until the shaft contacts the auxiliary bearings and then the AMB position sensors are used to measure the clearance within the AB. The ACC function allows the condition of the auxiliary bearings to be assessed without any disassembly of mechanical parts. This function minimizes the down time, before the machine can be re-started.

Following the completion of the test (five landing events) Waukesha Magnetic Bearings are pleased to report the machine and auxiliary bearings were in good condition. After the five landings, the auxiliary bearings were visually inspected. Figure 5 shows the RDS AB following the landing tests.

The wear was measured on the RDS AB and it was determined that a further 10 landings would be possible. The measured wear after 5 landings was 0.070 mm, with a wear allowance of 0.270 mm. Assessment of the rolling element AB determined that one additional landing would be possible, before required bearing replacement.
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The conclusion is that the Waukesha auxiliary bearing system performed well, proving to be sufficiently robust to protect the motor compressor during the series of extraordinary landing events, with the number of landings achieved significantly exceeding the specification requirements.

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4 Design and Analysis Methods

Reference 1 shows the results of a previous auxiliary bearing landing test program that demonstrated that the rotordynamics of auxiliary bearing landing events are analytically tractable, giving results with reasonable accuracy. That test program used a horizontal rotor, while the integrated motor compressor is a vertical machine.

A vertical motor compressor rotor is analyzed to illustrate the rotordynamic analysis method. The motor compressor model is shown on Figure 7. The axial bearings are located at the top of the machine, with a single impeller at the bottom of the machine.

A full finite element representation of the rotor is used to generate the degrees of freedom for the rotor and the bearing supports. Additional equations of motion are considered for the radial and auxiliary bearings to generate the additional degrees of freedom to describe the non-linear behaviour of these components. The time transient simulation performs a time integration using the Newmark constant acceleration method. The model uses the calculated bearing loads at each time step, as well as inputs of the external braking and initial rotational energy to determine a calculated rotational speed at each time step. In this way the model generates a speed versus time curve as well as the complete response of the rotor during a coastdown event.

The non-linear radial auxiliary bearing characteristics are fully considered by the model. Figures 8 and 9 illustrate these non-linearities. Figure 8 shows the contact model for the RDS type AB. The model describing the interface of the shaft landing sleeve and dry lubricated pad lining considers friction effects as well as rolling resistance effects. The non-linear characteristics of the RDS stator include stiffness that varies with displacement and pad preload effects.

Figure 9 shows the non-linear model for the rolling element type AB. This model includes the lateral and tangential characteristics of the compliance mechanism acting between the outer race of the rolling element bearing and its retaining housing.

Figure10 shows the model coordinates and forces for the representation of the axial auxiliary bearing. As shown, application of axial loads can cause lateral forces that affect the lateral rotor behaviour.

The time transient model also includes representation of the magnetic bearing effects. The transients are started with the rotor in the ‘levitated’ condition where it is supported at the magnetic bearing actuator centrelines. The rotor is ‘operated’ in this condition for a short period of time before initiating the coastdown event to allow the rotor to settle into the static deflected shape due to gravity loads as well as the dynamic displaced shape due to imbalance loads. The postulated failures are initiated by disabling the required portions of the magnetic bearing support.

Figure 11 shows displacement plots of a typical simulated auxiliary bearing coastdown event where all magnetic bearings have been disabled. The response plots show the rotor displacement versus time (seconds) for one axis of the upper bearing and one axis of the lower bearing. The plots show the transition from fully levitated operation on the magnetic bearings to the landing on the auxiliary bearings. The plot shows the transition to a subsynchronous forward whirl as the rotor contacts the auxiliary bearings.

There are many design features and performance parameters that determine the rotor behaviour during an auxiliary bearing coastdown event. Rotor design considerations include the location of the auxiliary bearing on the machine rotor determining bearing span, natural frequencies and node locations. The bearing foundation characteristics can have a significant impact on the rotordynamics. The balance quality of the rotor also has a major influence. The important auxiliary bearing characteristics are coefficent of friction, clearances, and the stiffness and damping characteristics. Any aerodynamic loads on the rotor from seal or impeller effects should also be included. For compressors, the characteristics of these aerodynamic forces may change rapidly, depending on the coastdown rate, and during this period the operating point of the compressor may move to the extents of the pressure/flow envelope.

The test results and the simulated rotor results show similar dynamic behavior. The vertical rotor supported on the AB, experiences a nearly constant frequency subsynchronous forward whirl during the coastdown. The whirl is at a low frequency and the resultant dynamic loads are well within the capacity of the auxiliary bearings.

The key points in designing a successful auxiliary bearing system for vertical applications is to predict the whirl frequency and then ensure the dynamic load capacity of the auxiliary bearings is sufficient for the resultant loads. The integrated design of the auxiliary bearings and the rotating machine is crucial to ensure the whirl frequency is low enough to avoid overloading the auxiliary bearings.

The analysis methods used for the load prediction should be a benchmarked method that has been previously compared to actual test results. The test results should include rig test results for the individual auxiliary bearing component, as well as overall string landing test results.

5 Conclusion

-A Waukesha auxiliary bearing system has successfully provided robust operation for 5 full speed landings in a 6 MW high speed vertical integrated motor compressor. After 5 high speed landings the RDS type auxiliary bearing still had significant service life available for an estimated 10 further landings.

-A mixed auxiliary bearing system of bushing type auxiliary bearing and rolling element auxiliary bearing can provide robust service when used in combination on a common rotor.

-Robust corrosion resistant auxiliary bearing are available for immersion in the process fluid in harsh environments, such as sour gas applications.

-Successful auxiliary bearing systems are available for vertical machines.

-An integrated design process between the auxiliary bearing design and the rotating machine design is required to realize a robust system.

-Rotordynamic time transient analysis results can be used as a reliable guideline for the design of vertical machines with auxiliary bearing systems.

-The analysis process should have been previously benchmarked against test results. The test results should include data from individual auxiliary bearing rig testing as well as landing test results using the entire machine string.

-For vertical systems, it may not be possible to avoid whirl during a landing event. In this case the design of the system should ensure that any whirl is forward in direction, and the whirl frequency is predicted. The auxiliary bearing load capacity can then be designed to accommodate the resultant loading during the coastdown.

References

[1] Richard R. Shultz and Emiliano Lucchetta; Time Transient Simulation Model and Full Scale Experimental Verification Test for Coastdown Events with a 1.5 ton Supercritical Rotor Supported by Active Magnetic Bearings and Dry Lubricated Bushing Type Auxiliary Bearings; *IMechE International Conference on Compressors and Their Systems*, September 2005, London

A Appendix A

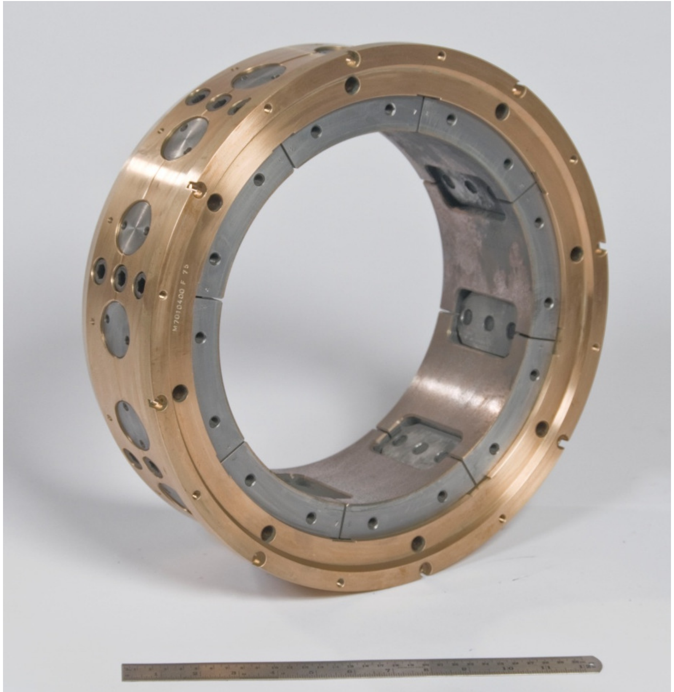

Figure 1- RDS Radial Auxiliary Bearing

Figure 2- RDS Axial Auxiliary Bearing

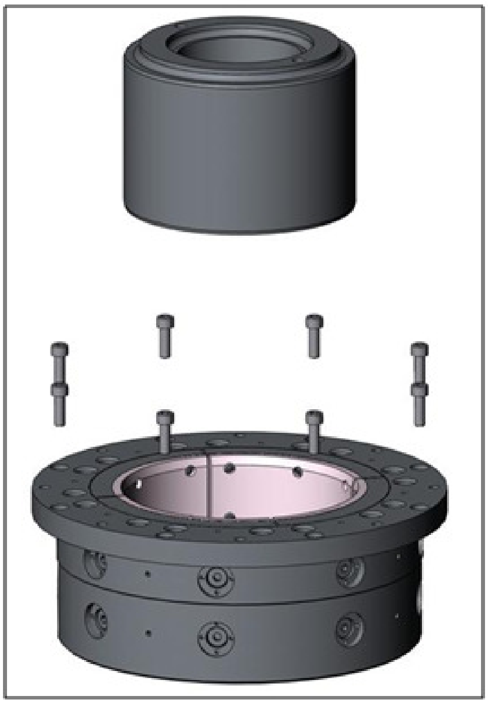

Figure 3- RDS Radial Auxiliary Bearing

Figure 4- RE Auxiliary Bearing

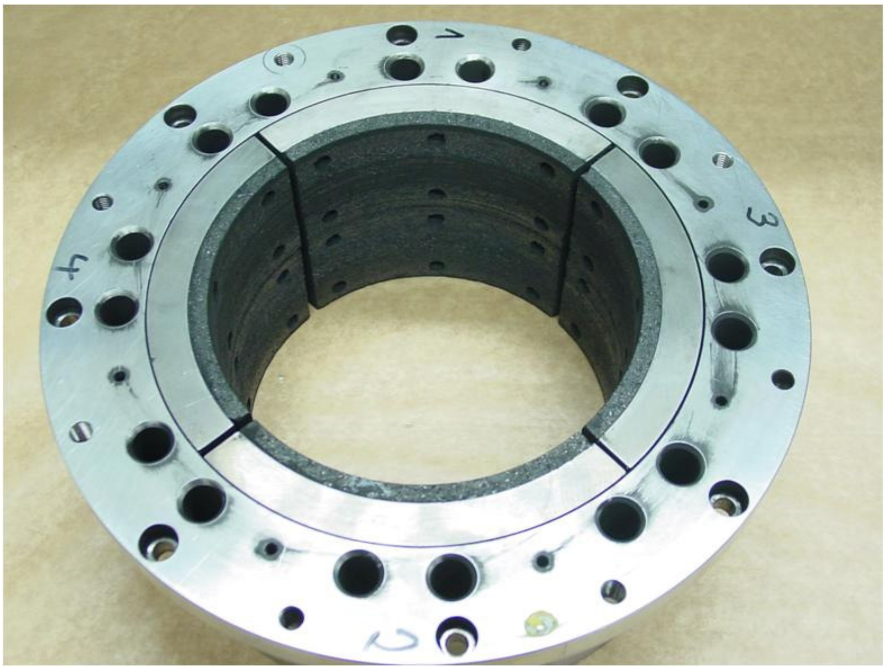

Figure 5- RDS Radial Auxiliary Bearing, Corrosion Resistant

Figure 6- RE Radial/Axial Auxiliary Bearing, Corrosion Resistant

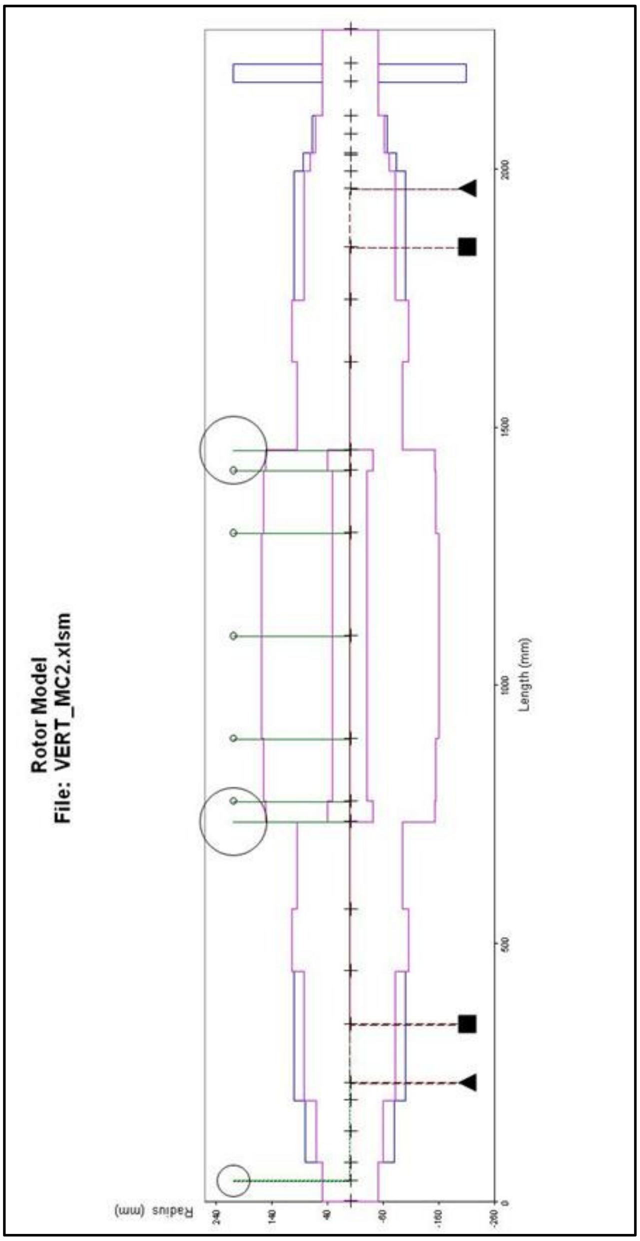

Figure 7- Vertical Motor Compressor, Rotordynamic Model

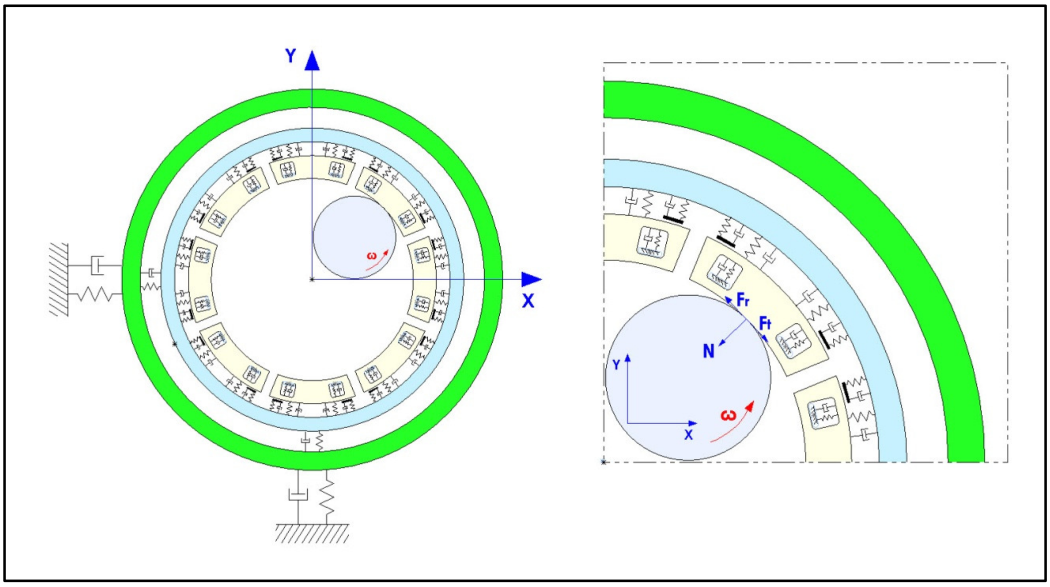

Figure 8- RDS Radial Auxiliary Bearing, Modeling Diagram

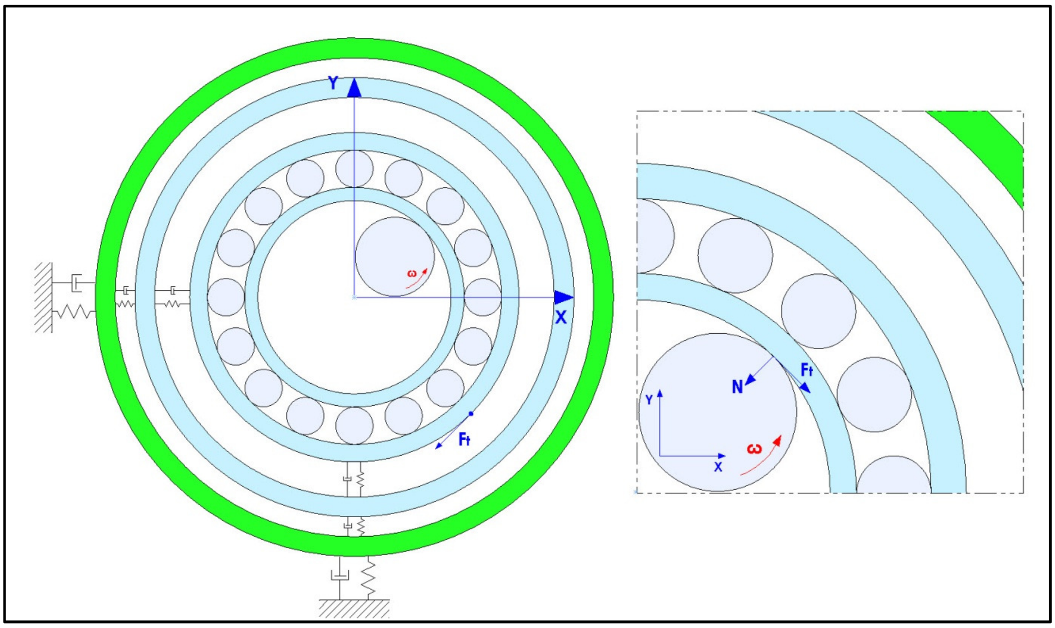

Figure 9- RE Radial/Axial Auxiliary Bearing, Modeling Diagram

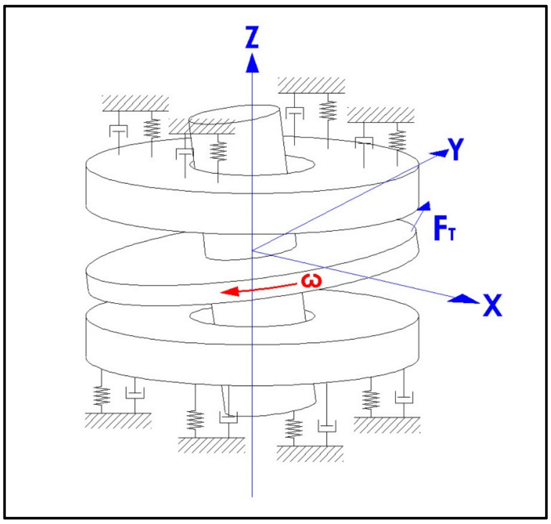

Figure 10- Axial Auxiliary Bearing, Modeling Diagram

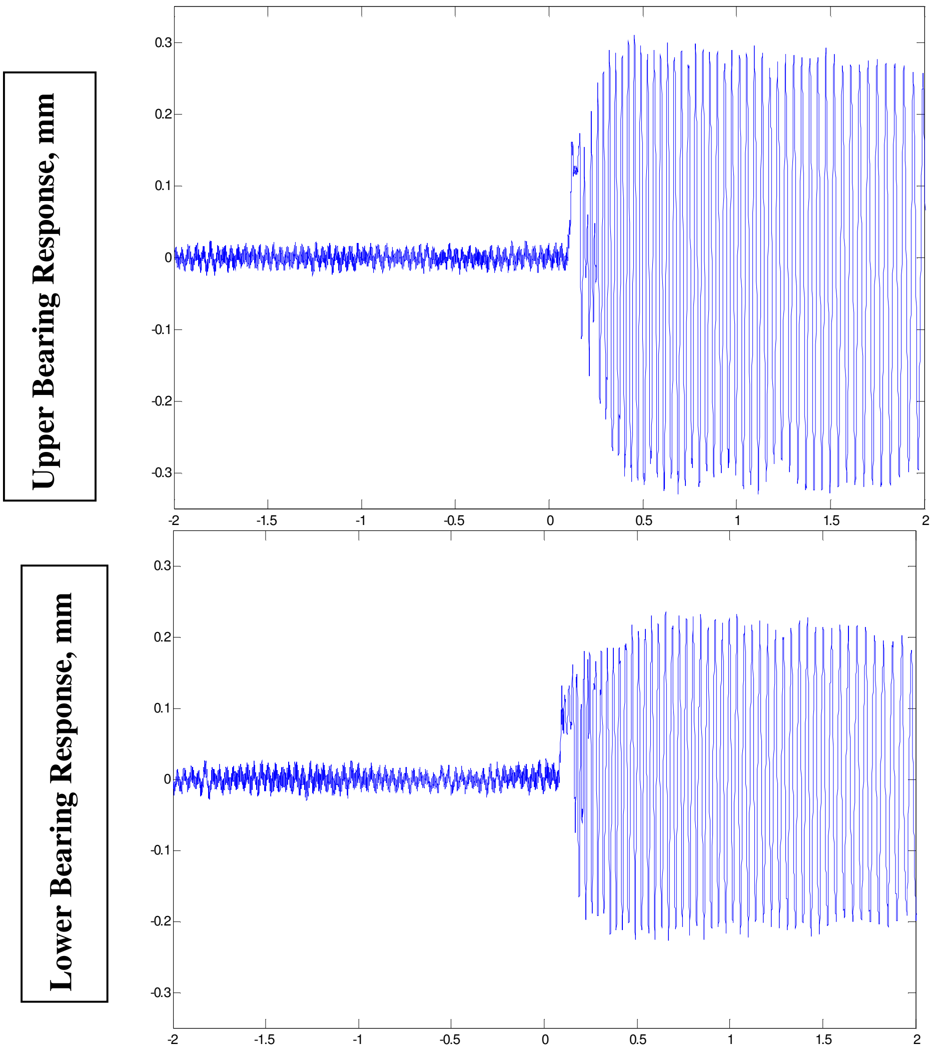

Figure 11- Upper and Lower Rotor Response during Landing Event