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DEVELOPMENT AND TESTING OF THE BACKUP BEARING SYSTEM FOR AN AMB ENERGY STORAGE FLYWHEEL

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ABSTRACT

A 140 kW energy storage flywheel has been developed to provide 15 seconds of ride-through power for industrial UPS applications. The flywheel, which operates in a vacuum, is supported by Active Magnetic Bearings (AMB) to minimize bearing losses, and has a high power motor/generator coupled to an efficient power conversion module. The backup bearing system was tested extensively due to the potential for very long spin down times in the event of a failure in or loss of power to the magnetic bearing system. Key issues encountered during testing are discussed and solutions identified. High fidelity orbit and time-history data from a full speed drop and spin down are presented and discussed in detail. The final backup bearing system is capable of three 2.75 hour spin downs from 36,000 rpm.

INTRODUCTION

A flywheel energy storage system (FESS) has been developed for industrial applications offering advantages over other forms of energy storage such as chemical batteries and ultracapacitors. This system utilizes a flywheel module composed of a high speed rotor levitated on AMBs. Integral on the rotor is a permanent magnet which is used in conjunction with a wound stator to act as a motor to increase rotor energy and as a generator to remove energy. The module operates in a vacuum to minimize windage losses, thereby maximizing operating efficiency. The module operates as part of a FESS with power conversion electronics, system controller, user interface, and supporting systems. The flywheel module forms the core energy storage portion of the product. The system has completed an eight month field test program which followed extensive in-house testing for performance validation, with operating field units accumulating more than of 36,000 hours.

The flywheel module includes a backup bearing system to support the high speed rotor in case of a failure or fault in the primary magnetic bearing system. The design and testing of the backup bearing system is reported here. For this flywheel, the duration of spin down can vary as the integral generator can usually be electrically loaded in different ways to change the deceleration rate of the rotor. However, one fault scenario for the system requires an unassisted spin down without braking from the generator. This scenario would result in a spin down time on the backup bearings of up to three hours. A typical verification test for a backup bearing system is to deactivate the magnetic bearings at operating speed, causing the rotor to drop onto the backup bearings and spin down to rest. There is a substantial body of work in the open literature that investigates AMB rotors on backup bearings. Schmied [1] presented results for a 8.9 kN (2000 lbf) compressor rotor. Several authors have also described full five axis drop tests for test rigs or machines for industrial service. Kirk [2] and Swanson [3] have presented numerous test results and analysis from a full scale, AMB rotor drop test stand. Caprio [4] presented results for drop testing on a large, vertical energy storage flywheel. Hawkins [5] presented results for drop testing of a 30,000 rpm expander generator. However, all of these drop tests except for [5] are for machines considerably heavier and slower than the flywheel described here, and all but [4 and 5] are for electromagnetic (EM) bias magnetic bearings. All of the mentioned test results are for short duration tests. Thus the available literature was not able to provide much insight to guide the design of the backup bearing system for long duration spin downs. Sun [6] presents a detailed backup bearing analytical model and simulation results for a vertical flywheel that provides some insight into expected behaviour. McMullen [7] presented some of the results reported here; however, this paper presents additional results and adds more detailed discussion.

THE ENERGY STORAGE FLYWHEEL

The flywheel module, shown in Figure 1, is designed to store a total energy of 1.25 kWh at 36,000 rpm and deliver 140 kW for 15 seconds (0.58 kWh). The AMB system supports the rotor during normal operation. A magnetic bearing controller (MBC) is powered primarily by power available at the user site, and secondarily by power from the flywheel generator when site power is unavailable. Radial position sensors are located adjacent to the magnetic bearings. The axial sensor is located at the bottom center of the shaft. The steel flywheel rotor weighs 1070 N (240 lbf). A passive axial lifter is used to offset approximately half the rotor weight, reducing the steady force that must be supplied by the axial AMB. This passive lift force is created by a permanent magnet circuit that pulls the rotor vertically up across a 1.0 mm (0.04 in) air gap. Since the passive lift is also present when the AMBs are not active, the weight load on the backup bearings is also reduced by half to approximately 535 N. All key surfaces of the flywheel rotor are ground to a tight tolerance eliminating the need to balance the rotor.

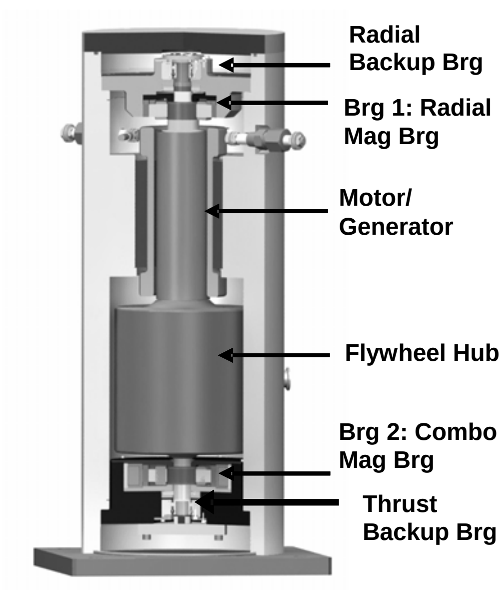

Figure 1. 140kw Flywheel Module

Backup Bearing System

A backup or auxiliary bearing system is used for rotor support during non-operation of the AMB and for emergency spin down of the rotor in the event of an overload or failure in some part of the magnetic bearing system. The upper backup bearing set provides radial support only and the lower set provides both radial and thrust support for the rotor. The baseline configuration for each bearing set is a duplex pair of face/face mounted angular contact ball bearings (Figure 2). The baseline bearings have standard 52100 steel races, ceramic balls, phenolic cage, shields and vacuum compatible grease. The thrust backup bearing must carry about 55% of the rotor weight or 587 N (132 lbf). The rotor weight is transmitted through a replaceable, rotor mounted thrust washer to the axial face of the thrust bearing inner race. The axial clearance is ± 0.18 mm (0.007 in). The radial clearance between the backup bearing inner race and the shaft sleeve is 0.18 mm (0.007 in).

The backup bearings are mounted in a compliant mount that sets the desired support stiffness since the duplex bearing pair is over 20 times as stiff as the mount. The primary motivation for using a compliant mount is to set the lowest natural frequency as low as possible while limiting maximum displacement within machine requirements. The compliant mount also reduces the impact loads carried by the bearings during a drop. In a horizontal machine with a properly designed backup bearing system, the rotor motion in the backup bearing clearance space will be a rocking or pendulum motion if the unbalance forces are low relative to the static load. Forward whirl can occur if the unbalance forces are higher than the static load. A destructive backward whirl of the rotor around the clearance space is assumed to be triggered by a friction mechanism, usually by rubbing or by very large bearing loads. In a vertical machine, such as this FESS, the rotor motion will almost always be a full whirl around the clearance space. Caprio [4] found that using the magnetic bearing actuator to impose a large static force can arrest this whirl, but this is not practical in a production machine due to limits on amplifier power. The whirl frequency apparently locks to the lower of either the spin frequency or the lowest natural frequency of the housing or support (Caprio [4], Hawkins [5], Bartha [8]). Since the load reacted by the bearings of a rotor whirling at a particular frequency will vary roughly with the square of the whirl frequency, it is desirable to keep the whirl frequency low. This is the motivation for including compliance in the backup bearing support.

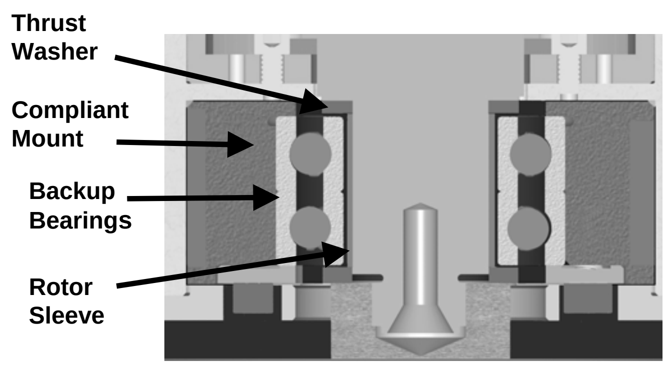

Figure 2. Thrust Backup Bearing Arrangement

The basic arrangement of the FESS backup bearing support was shown in Figure 2, although some important details have been obscured for proprietary reasons. Radial flexibility is provided between the mount and housing, and a hard stop limits radial deflection. The net radial stiffness is 5.0e6 N/m (28,000 lbf/in) per bearing pair, resulting in a lowest lateral natural frequency of 40 Hz. Dynamic data presented below show rotor whirl frequencies of 45-50 Hz at various spin speeds during a spin down on the backup bearings.

Fault Scenarios for Backup Bearing Operation

The backup bearing testing program described here was implemented to insure the integrity and safety of the flywheel system during backup bearing operation. In the event that the primary AMB system can not fully sustain rotor levitation, the rotor may either temporarily contact the backup bearings, or drop onto the backup bearings completely. The duration of operation on the backup bearings is dependent on the cause of the backup bearing contact. Three fault scenarios were identified that would result in backup bearing operation and these scenarios were used to guide the test program.

MBC fault: The MBC will send a fault signal to the FESS due to a number of different conditions including: excess rotor displacement, excess AMB coil current (high load), loss of power to the AMB, and failure of AMB control hardware or cabling. When the FESS system controller receives an MBC fault, it commands a powered shutdown, which will decelerate the rotor from full speed to rest in 10 – 30 minutes (installation dependant).

Loss of primary AC power: The primary application of the FESS is to backup the primary AC power supply for some type of load – e.g., a data center, building, or other critical system. Of course, the FESS has an auxiliary power system - the critical power supply (CPS) – to provide power to the MBC when the primary AC power supply fails. The CPS can provide enough power to the MBC to maintain levitation down to approximately 8,000 rpm. At this speed the flywheel rotor drops to the backup bearings and spins down to rest unassisted.

Multipoint failure: The type of failure that would place the most stress on the backup bearing system is a multipoint failure of the CPS/MBC and the FESS controller at the same time. In this event, the rotor would drop to the backup bearings and would spin down to rest unassisted. The magnetic iron loss of the motor/generator is the only significant drag torque in this situation, resulting in the rotor taking 2.5 to 3 hours to coast from full speed to zero speed. No multi-point failures have been observed in development testing or in over 36,000 hours of Beta testing in the field. Although the chances of this failure are remote, it represents the limiting case for backup bearing performance so considerable effort was spent to address it.

BACKUP BEARING TESTING

The test plan was developed to address the three fault cases defined above. The final objective of the test program was to develop the backup bearing system to survive the most difficult condition, the unassisted spin down from full speed that could be required due to a multipoint failure. An important milestone was reached when the system was shown to be capable of achieving five successful powered spin downs of 10 minute duration from full speed. Meeting this target showed that the system could tolerate several MBC fault conditions with margin.

Procedure

The general procedure during testing was to drop the rotor onto the backup bearings at a given target speed by delevitating all five axes. The rotor would then spin down under braking to a lower target speed and then be relevitated at a designated pickup speed. The typical sequence of tests is given in Table 1. After a series of partial spin down drop tests, one or more full spin down drop tests (test 8 in Table 1) would be performed. A prototype MBC was used that had enough amplifier power to allow the rotor to be relevitated at any rotor speed under most conditions. Magnetic bearing position sensor signals were monitored during backup bearing operation to assess the dynamic performance during the testing. These signals were recorded at a 5 kHz sample rate for post test evaluation of whirl frequency and other dynamic characteristics.

Table 1. Typical Drop Test Sequence

Test No.	Drop Speed (rpm)	Pickup Speed (rpm)
1	5,000	1,000
2	10,000	5,000
3	15,000	10,000
4	20,000	15,000
5	25,000	20,000
6	30,000	25,000
7	36,000	30,000
8	36,000	0

Each time the system was reassembled after inspection or replacement of bearing system components, a series of partial spin down drop tests (Tests 1-7 in Table 1) was performed starting at low speed and continuing to progressively higher speeds. This series of short drop tests helped to establish confidence in the system and also to properly distribute grease in the bearings when new bearings were installed. Subsequent to this initial test cycle, the drop tests typically covered the full speed range, 36,000 rpm to 0 rpm (Test 8 in Table 1). After each drop test, observed results were assessed and a decision made to proceed with additional tests or to partially disassemble the flywheel for inspection. Three different flywheel rotors were used interchangeably during the course of testing. Each rotor was used in the same nominal (field ready) condition with no intentional changes in balance quality or clearances.

All of the early testing was done with assisted spin downs – where the rotor speed was decelerated by electrically loading the flywheel generator. As the test program progressed, the spin down time was gradually increased by adjusting the loading on the generator. Average deceleration rates used in various tests are summarized and identified by number in Table 2. Also listed in the table is total time on the backup bearings from 36,000 rpm to 0 for each braking rate. Braking rate 6, the slowest rate, is the unassisted spin down. In this case, no external loading is used and the drag torque is provided mostly by iron losses as a result of the PM rotor spinning in the generator stator. During the full unassisted spin down, the flywheel spends 9600 sec (2.7 hours) on the backup bearings with speed decreasing linearly with time.

Table 2. Braking Rates Used in Testing

Braking Rate	Deceleration Rate (rpm/s)	Full Spin Down Time (sec) 36,000 -> 0 rpm
1	276.9	130
2	150.0	240
3	66.7	540
4	35.3	1020
5	16.4	2200
6	3.8	9600

The full development series consisted of 46 full spin down drop tests on multiple units, and over 200 drops in different parts of the speed range. The overall test matrix is summarized in Table 3. The matrix is divided into 4 test series to define the major breakpoints in the overall testing. The partial spin down drop tests were generally decelerated with braking rate 1 or 2. The braking rates for the full spin down tests are in Table 3.

Table 3. Test Matrix for Backup Bearing Testing.

Test Series	# Partial Spindowns	Full Spin downs		# Aborted Tests
		Braking Rate	Number	
1	7	1	1	0
	7	1	1	1
	7	1, 1	2	0
	7	1, 2	2	0
2	90	3 (all tests)	12	4
	7	3, 3	2	0
3	7	2, 3, 4, 5	4	0
	10	2, 2, 3, 4 5, 6, 6, 6	8	0
	7	2, 6, 6, 6	4	0
	7	2	1	1
	7	2	1	1
	7	2, 6, 6, 6	4	0
	7	2, 6, 6, 6	4	0
4	24		0	0
Total	**201**		**46**	**7**

Test Series 1: Initial tests in this series were executed at the fastest spin down rate, 130 seconds from 36,000 rpm to rest. The testing started with a full series of partial spin down drops followed by 2 full spin down drops. These tests were used to validate the dynamic performance of the system with the new mount and to gain confidence in the overall system. After disassembly and inspection, a new set of bearings was installed and testing was continued to further evaluate rotor thrust washer and sleeve wear. The next test, a full spin down with the new bearings, was expected to be uneventful but resulted in a failure of one of the bearing sets at about 32,000 rpm. Inspection showed that most of the grease had been pushed out from under the bearing shields. It was then realized that the partial spin down drops had been omitted after installing the new bearings. The partial spin down drops performed after a rebuild was planned to have two functions: 1) gain confidence in the assembly at successively higher speeds, and 2) to serve as the grease run-in procedure for the bearings. Operating new bearings at increasingly higher speeds serves to distribute the grease properly around the outer race and shields by warming and channeling. This is a standard procedure for high speed machines with ball bearings whether they the bearings are main bearings or backup bearings. The test procedure was clarified and failures on the initial drop of a new set of bearings were eliminated.

Test Series 2: The spin down time was increased to 540 seconds (9 min) for the next series of tests. During the first test series, very minimal wear was observed on the original metal rotor thrust washer (Figure 2). However, when spin down times were extended, it became apparent that the thrust washer was actually wearing at approximately 0.05 mm (0.002 in) every thirty minutes (2.78e-5 mm/s). This wear rate was too high to allow even a single unassisted spin down. Further, wear particles were produced that could potentially contaminate the bearings and cause a future failure. A picture of the wear track is shown in Figure 3. The wear track has a radial height equal to the radial height of the bearing thrust face plus about 0.36 mm (0.014 in). The extra 0.36 mm is equal to the diameter of the whirl orbit. Even though the backup bearing inner race speed quickly accelerates to nominally the rotor speed, the rotor thrust washer is still subject to sliding friction with the backup bearing inner race as the rotor whirls at the 40-50 Hz whirl frequency. Several different materials and coatings were tested, and of those, the one with the lowest coefficient of friction gave the best performance. The new thrust washer material was then used for the remainder of the testing. Subsequent testing showed the wear rate of the new material to be a much more acceptable 2.6e-6 mm/s, slow enough to allow more than 3 unassisted spin downs.

Figure 3. Wear Track on Rotor Thrust Washer

Test Series 3: During this group of tests, the spin down time was steadily increased as identified in Table 3. Four successful full speed spin downs were completed on the first flywheel build with spin down times of 240, 540, 1020 and 2200 seconds. The flywheel was disassembled and the bearings, sleeves, and thrust washers carefully inspected. Thrust washer wear was acceptable and the bearings were found in good condition. The flywheel was then reassembled with new bearings and thrust washer and a set of 8 full speed spin downs was completed (after the usual partial speed spin downs). The last three tests were all unassisted, marking a major program milestone.

Subsequent testing in this series was expected to continue to validate the backup bearing performance. However, on two seemingly random occasions, failure of the thrust end backup bearing pair occurred within 60 seconds of the beginning of an unassisted spin-down, just following the completion of a successful assisted (4 minute) full speed spin down on the same bearing set. It was observed that at the instant of failure, the rotor axial position dropped rapidly, either due to accelerated thrust washer wear or internal damage to the bearing. This caused damage to the axial sensor thereby preventing relevitation. Since the rotor then had to spin down to rest on a damaged bearing, the bearing sustained additional severe damage that prevented diagnosing the failure. Following this, an auto-relevitation feature was added to the MBC code. This feature used an axial position threshold to trigger re-levitation, allowing the MBC to quickly re-activate the magnetic bearings and preserve the ball bearings right at the point of failure. This allowed for a more accurate study of the ball bearing failure mode and determination of modifications for improved performance.

Investigation of the bearings after the next such failure showed Brinell marks and metal smearing on the inner race (Figure 4). The Brinelling clearly occurred prior to the smearing of metal, thus indicating the bearings were Brinelled prior to the testing. However, the Brinell damage served to initiate the failure of the bearing during operation. It was determined that the Brinell problem was created by the testing procedure. Prior to the successful unassisted spin downs, quite a bit of time was taken between each drop test to allow examination of the test data. After a number of unassisted spin down tests were successfully completed, the confidence level in the system was significant and groups of tests were performed in quick succession. Prior to each full speed unassisted spin-down test a full speed assisted spin-down test would be run using a 240 sec spin down time. This would heat the bearing inner races very quickly, yet not bring the rest of the bearing to a uniform temperature, thus significantly increasing the bearing preload. The Brinelling then is believed to have occurred upon the hard de-levitation at the start of the drop test in which the failure occurred. The test procedure was then changed to allow a 30 minute cool down between drop tests. The random failures have ceased since the procedure change and Brinell damage has not been found in subsequent post test inspections.

Test Series 4: This final group of tests consisted of 24 rotor drops onto the backup bearings at 8,000 rpm followed by an unassisted spin down to rest. After completion of this series the flywheel was disassembled and inspected. The bearings were in excellent condition and thrust washer wear was less than 0.075 mm (0.003 in). This test series validated the ability of the flywheel to sustain multiple low speed spin downs following long term loss of AC power.

Figure 4. Failed Backup Bearing Inner Race

Dynamic Performance

A selection of dynamic data from a representative drop test is given in Figures 5 - 11. This displacement data was measured by the magnetic bearing position sensors during the drop transient and subsequent spin down.

Drop Transient Data. Position data from the drop transient is shown in Figures 5 - 7. Figure 5 shows the time history from the axial position sensor (vertical direction). After the drop occurs at 0.0 seconds, the rotor takes 8 msec to drop past the backup bearing nominal position at 0.18 mm (0.007 in). Peak deflection is about 0.23 mm (0.009 in). The rotor bounces a few times and then after 70 msec settles into a 50 Hz vertical oscillation. This frequency is the same as the radial whirl orbit frequency observed in the following figures.

Figure 6 shows a time history from the *x* and *y* position sensors from 30 msec before the drop command until 170 msec after the drop. From this figure it is clear that the whirl around the clearance space is at 50 Hz and is a forward whirl since the rotor spin direction is clockwise (*y* then *x*). The main high frequency component is synchronous (~600 Hz).

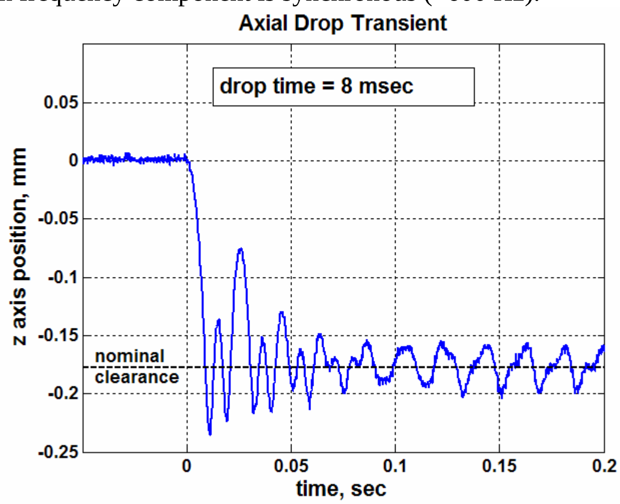

Figure 5. Time history from axial (vertical) position sensors during drop transient at 36,000 rpm.

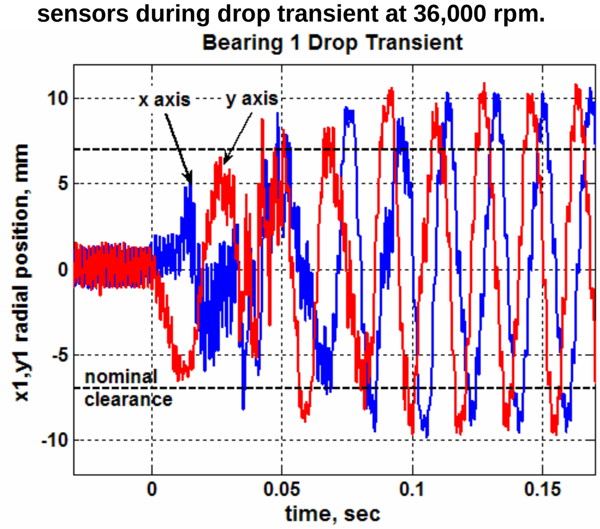

Figure 6. Time history from X & Y Position Sensors during drop transient at 36,000 rpm.

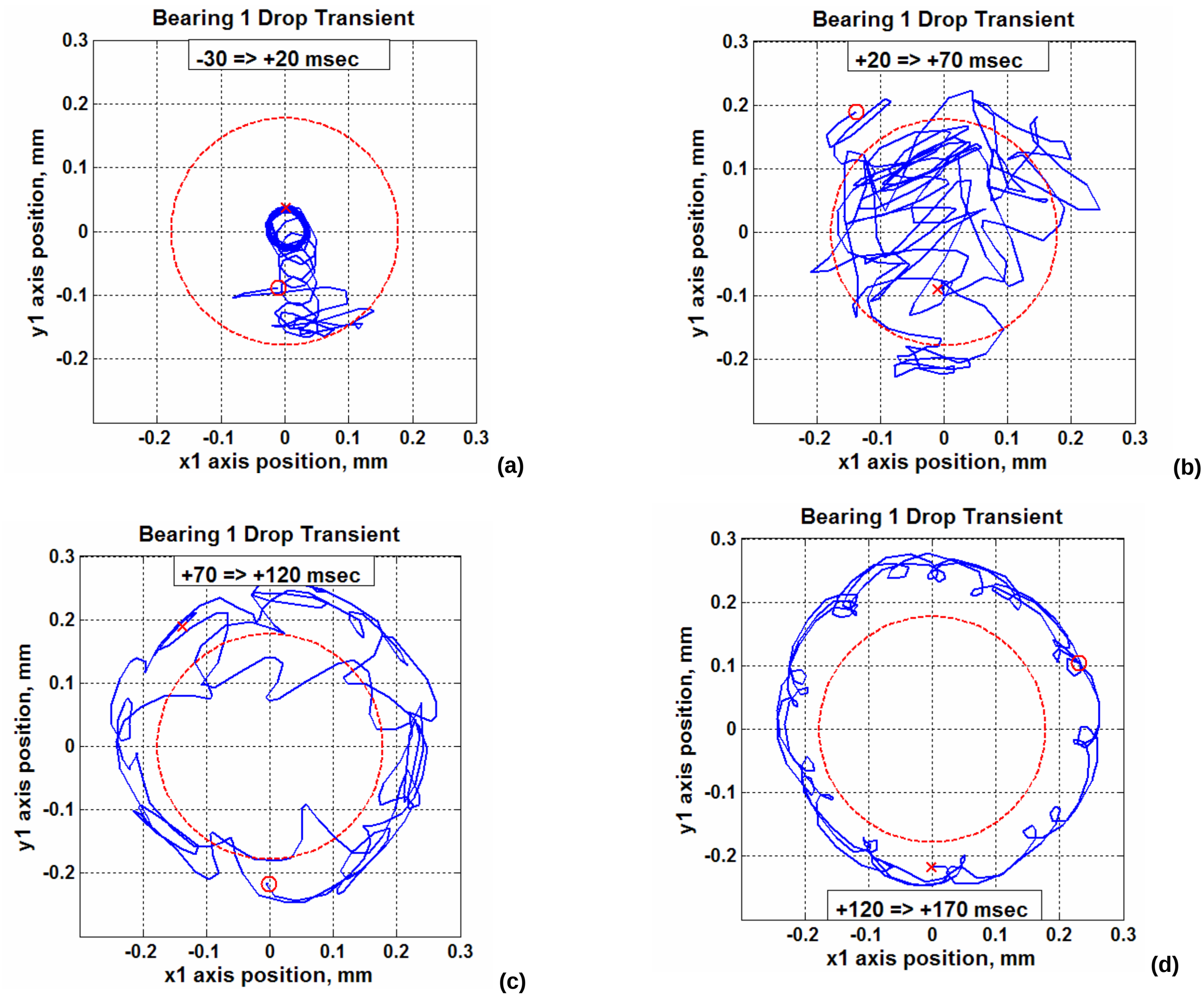

Figure 7. Orbit plot X & Y Position Sensors during drop transient at 36,000 Rpm: (a) -30 -> +20 msec, (b) +20 -> +70 msec, (c) +70 -> +120 msec, (d) +120 -> +170 msec,

Figure 7 (a – d) shows successive orbit plots from the upper radial bearing position sensors from just before the drop until the rotor settles into a steady whirl orbit. The four orbit plots together cover the same time period as the time history of Figure 6. All of the figures have an 'X' symbol marking the first data point and an 'O' symbol marking the final data point. Figure 7a is shows the rotor orbit from 30 msec before the drop command until 20 msec after the drop command. The rotor is initially supported by the magnetic bearings and has a synchronous orbit (600 Hz) of about 0.025 mm (0.001 in) 0-pk. After the magnetic bearing is turned off the rotor spins out toward the backup bearing and bounces away. Since the AMB control was using synchronous cancellation to ignore the synchronous motion, the original synchronous orbit is roughly maintained after the magnetic bearing is turned off until the rotor hits the backup bearing. Figure 7b shows the rotor orbit from 20 msec to 70 msec after the drop. The motion in this phase is chaotic bouncing around and in the clearance space. The backup bearing inner race is being spun up toward the rotor spin speed during this time. Figure 7c shows the rotor orbit from 70 msec to 120 msec after the drop. The rotor is beginning to form an orbit but still shows a number of excursions into the clearance space. Figure 7d shows the rotor orbit from 120 msec to 170 msec after the drop. The orbit has formed into a definitive 50 Hz whirl around the clearance space with small inner loops representing the synchronous orbit.

Spin Down Data. Data taken from the position sensors of the upper radial bearing from four different time slices of the spin down is shown in Figures 8 - 11. Each figure has a part *a)* that shows a 0.1 second time slice in time history form, and a part *b)* that is an *x* versus *y* orbit taken from the same time slice. In the Figures *a)*, the dashed lines at ±0.18 mm (0.007 in) represent the nominal backup bearing clearance. Likewise, the dashed circle at 0.18 mm (0.007 in) in the Figures *b)* represents the nominal backup bearing clearance. Excursions past the nominal clearance represent deflection of the compliant mount and any bending of the shaft between the position sensors and the backup bearings. In the Figure 8 shows data for a spin speed of 32,700 rpm. The primary whirl orbit is forward whirl at 45 Hz with a much smaller synchronous component. The characteristic dynamic behavior in all tests was consistently a full circle forward whirl at 35-50 Hz around the backup clearance for all spin speeds above 2400 rpm. This is an important result because the low whirl frequency reduces the load reacted by the backup bearings.

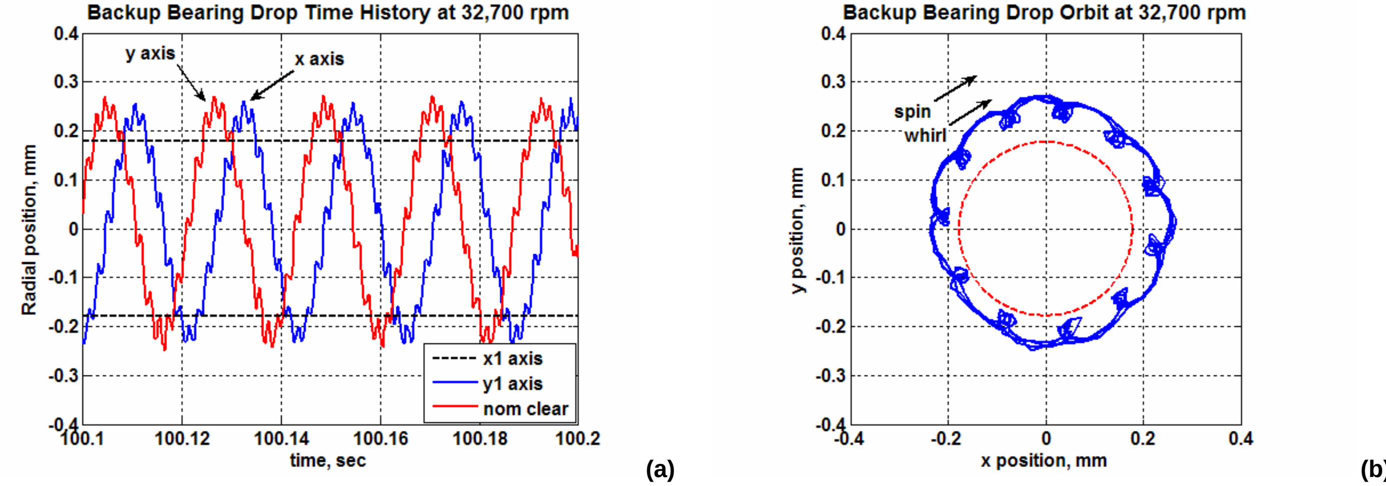

Figure 8. Time History (a) and Orbit (b) from X & Y Position Sensors at 32,700 Rpm

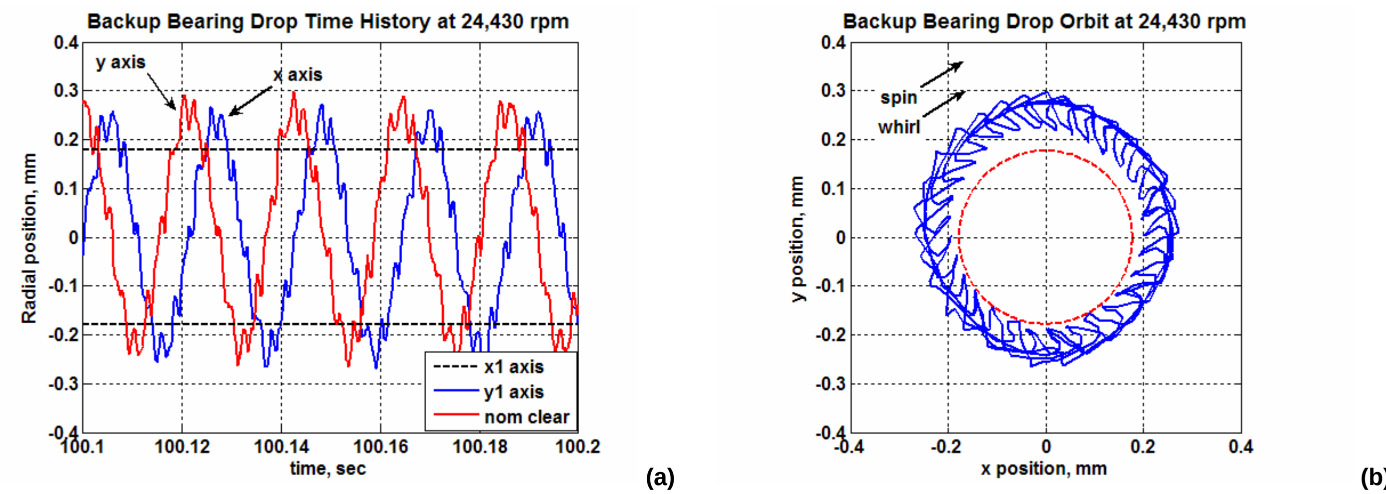

Figure 9. Time History (a) and Orbit (b) from X & Y Position Sensors at 24,430 Rpm

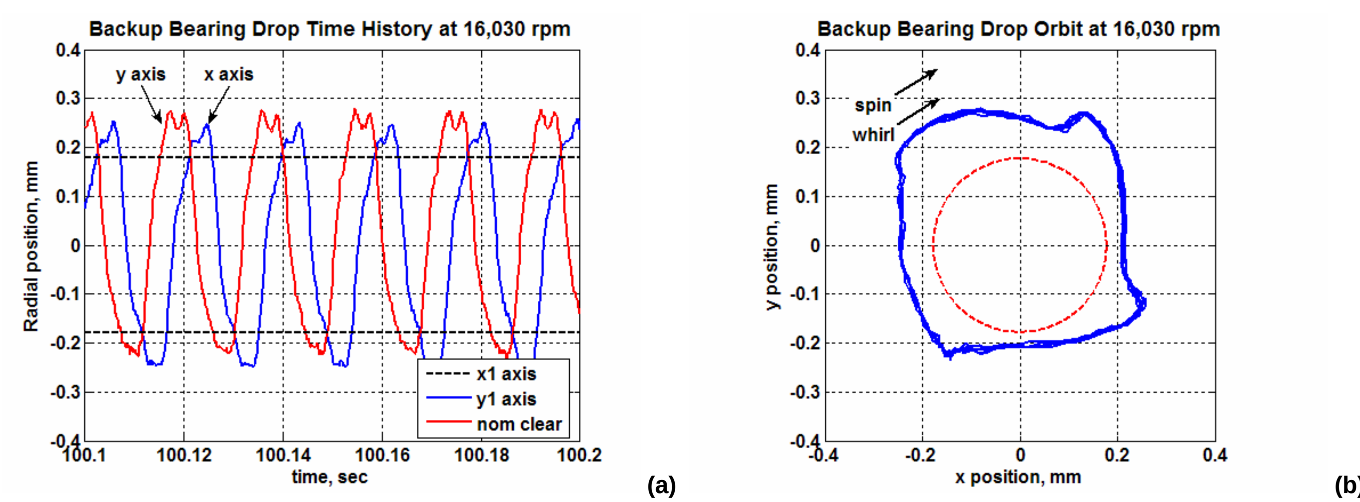

Figure 10. Time History (a) and Orbit (b) from X & Y Position Sensors at 16,030 Rpm

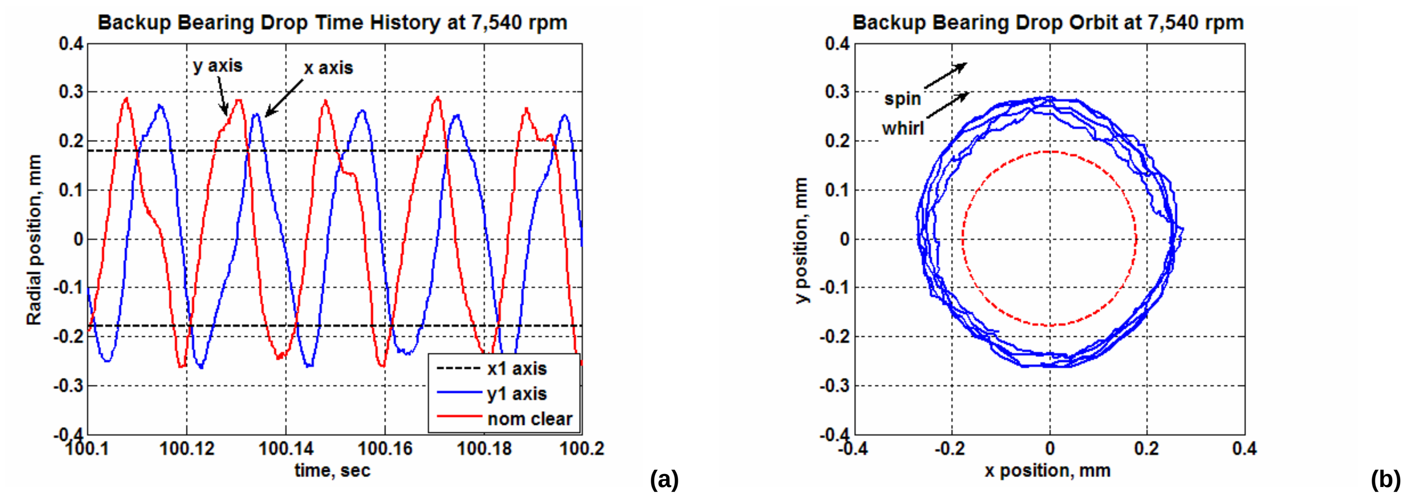

Figure 11. Time History (a) and Orbit (b) from X & Y Position Sensors at 7,540 RpM

The small loops in Figure 8b represent the synchronous orbit which is about 0.012 mm (0.0005 in) at this speed. The synchronous loops on successive cycles fall on top of each other because the selected spin speed happens to be an integer multiple of the whirl frequency. Figures 9 - 11 show dynamic data for speeds of 24,430, 16,030, and 7,540 rpm. In each case, the predominant rotor motion is a forward whirl orbit around the clearance space at 45 - 50 Hz.

CONCLUSION

Extensive design and testing has been done to verify the ability of the backup bearing system in the FESS. Multiple unassisted coast-downs from full speed of 36,000 rpm to zero rpm have been tested on one single unit successfully. And many more full speed unassisted coast-downs have been performed.

1) The compliant backup bearing mount designed for the system performed as expected, reducing the whirl frequency of the rotor on the backup bearings to 40 – 50 Hz. This low whirl frequency is important for limiting the loads that must be reacted by the backup bearings.
2) A key factor in allowing long duration spin downs in this vertical spin axis application was reducing the friction coefficient of the thrust washer.
3) Several bearing failures during testing were initiated by Brinell damage to the raceways. In the case of the testing reported here, the damage was caused by the test procedure that was more severe than expected field requirements. However, the results still highlight the importance of Brinell damage as a failure mechanism.
4) The automatic relevitation feature that was implemented in the MBC control proved to be an excellent tool for aiding diagnosis of bearing issues in a testing environment.

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