ROTOR DROP TEST STAND FOR AMB ROTATING MACHINERY PART I : DESCRIPTION OF TEST STAND AND INITIAL RESULTS

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ABSTRACT

The recent increase in the number of critical path rotating machinery applications using active magnetic bearing (AMB) technology has focused awareness and necessity for proper design of the auxiliary or backup bearings. These emergency bearings are essential for protection of the AMB stator in the event of control system failure or limited operation during momentary overload conditions wherein the AMB control system is still active. The current research project is concerned with the former design requirement, which is referred to as rotor drop. The rotor system and the auxiliary bearing support structure are equally important and influence the nature of the resulting rotor drop transient response. Limited testing of production machinery has demonstrated both successful drop tests and limited cases of auxiliary bearing failure. The reliability required for critical path machinery makes it essential to completely understand what parameters control the nature of the rotor drop transients. The design and construction of a full scale research test stand at Virginia Tech will be documented in this paper. The overall goals are summarized and initial test results of rotor drops on the rigidly supported auxiliary bearings are presented.

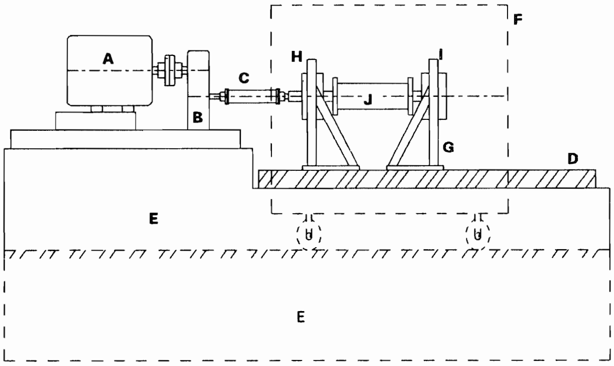
INTRODUCTION

The application of AMB for turbomachinery in North America has been steadily increasing in recent years. One of the major benefits of this kind of bearing support system over fluid-film bearings that is found in the majority of applications to date for land base turbo machinery, is the oil free environment [1]. More recent development work for aircraft jet engines and the power utility industry is targeted to eliminate anti-friction bearings currently used for these designs. The other major selling point of AMB technology is the reduced maintenance and longer run times between mandatory shutdowns to replace worn or defective components [2]. The advantage of this particular claim is somewhat reduced by the fact that the majority of industrial applications have relied upon a dry lubricated anti-friction backup overload bearing. These backup bearings are used for momentary overloads or for less likely power failures that would cause total loss of the load capacity of the AMB support system. For either situation, the backup bearings are expected to provide the load support and thereby allow either continuous running for short duration momentary overloads or immediate restart in the event of a power loss mandatory shutdown. These expectations can be realized in most situations. However, the desired number of restarts have not been obtained without bearing failure. Thus the claim of extended run times between major machine tear downs is questionable. The time transient response of machinery dropping onto backup bearings during operation at full speed have confirmed that very high instantaneous loading, and possibly large amplitude vibration can occur [3-6]. The duration of the vibration necessary to damage the backup bearings is very short under certain machine operating conditions. The exact cause of the damaging vibration is not certain. This fact has resulted in the development of a full scale test stand to examine the possible causes of the destructive vibration and develop a new backup support system that will give extended operation by assuring repeated restarts after total power failure without bearing failure.

Part I of this paper will detail the new test stand and the custom PC based data acquisition system. Rotor drop transients will be shown for the balanced condition to verify the operation of the data acquisition system. Limited testing has also been conducted for various levels of rotor unbalance. A small portion of the test data for these drops will be discussed in Part I. The steady state operation of the test rotor will be documented in Part II of this paper.

DESCRIPTION OF THE TEST STAND

The AMB drop test rig, shown schematically in Figure 1, is based around a large general purpose test stand that is configured for these tests with the magnetic bearings and related equipment. The details of the rig are given below.



A - Motor D - Steel Base Table G - Bearing Pedestal J - Drum Rotor
B - Gearbox E - Concrete Foundation H - Inboard Bearing
C - Coupling F - Safety Cover I - Outboard Bearing

FIGURE 1 : Details of Test Stand

Driver

It is a 150 kW AC variable frequency drive motor coupled to a speed increasing (5.2:1) gearbox with a diaphragm type coupling.

Test Bed

It is 3.6 m long, 1 m wide, steel table bolted/grouted to a 34,000 kg reinforced concrete foundation.

Safety Cover

It consists of a 25 mm thick, mild steel plate enclosure, cover is designed to be readily moved back from test area to allow free access to the test rotor/bearings.

Bearing Mounts

Two pedestals, fabricated from 102 mm steel plate uprights, bored to match magnetic bearing housings. The pedestals are intended to simulate a machine with very rigidly mounted bearings.

Magnetic Bearings

The test rotor is supported by standard industrial magnetic bearings. The bearing at the inboard end is a radial type with a length of 60 mm and ID of 190 mm. It is rated for a static load of 4400 N. The outboard end radial bearing has a length of 50 mm and an ID of 190 mm, and is rated for a static load of 3690 N. The outboard end also contains a magnetic thrust bearing of 192 mm ID and 246 mm OD, rated for a static load of 2311 N. The bearings are controlled by a digital controller which implements a PID type algorithm. The controller has been modified to allow selected bearings to be remotely de-energized. Provision has been made to synchronize disabling of the bearing power amplifiers with the key-phasor, thus assuring that the test drop occurs at the same shaft angular position for each drop at a given shaft speed.

Backup Bearings

The backup bearing system consists of a 150 mm ID, 190 mm OD, 20 mm wide, deep groove roller bearing at the inboard end of the rotor, and a 152 mm ID, 203 mm OD, duplex angular contact bearing at the outboard (thrust) end of the rotor, each 25.4 mm wide. The bearings are hard mounted for initial test drops.

Test Shaft

The test rotor is a drum type rotor, originally constructed to simulate an aircraft gas turbine compressor section. The rotor is 1.75 m long and weighs 1379 N. It has static bearing reactions of 618 N and 627 N at the drive end and thrust ends respectively. A new solid rotor test shaft has been designed and is currently in manufacture for the second year of testing.

DETAILS OF DATA ACQUISITION SYSTEM

The test rig is supported by a custom designed PC based data acquisition system, depicted schematically in Figure 2. The details of the data acquisition system are provided below.

A/D Board

It is a standard 12 bit, 16 channel A/D board with a maximum throughput of 100 kHz plugged into a 486-33 MHz computer. Simultaneous sampling is accomplished using an external peripheral board.

Data Acquisition Software

Custom programs are used for data acquisition under steady state and transient conditions. The transient data acquisition programs utilize multiple DMA buffers for recording the transient rotor drop phenomena. Tests are under manual control, with actual bearing power trip synchronized to the key-phasor to ensure consistent tests.

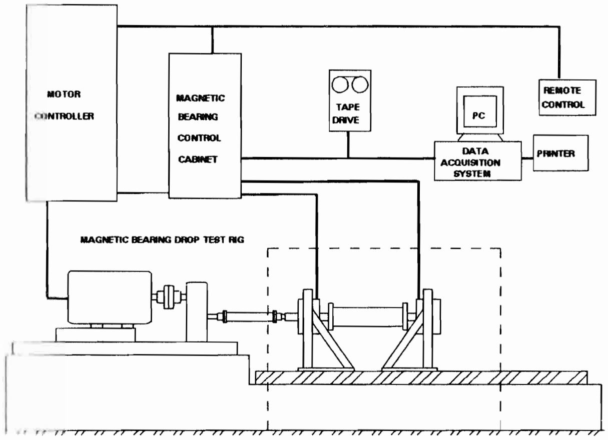


FIGURE 2 : Details of Data Acquisition System

Data Reduction Software

A series of custom written programs are utilized to reduce and analyze the large amount of data generated from the experiments. This software can generate orbit, response, FFT, cascade, magnitude and attitude angle plots.

Data Storage

A tape drive is used to store data from the experiments for archival purposes.

ROTOR DROP TEST RESULTS

The initial test drops are for a balanced rotor condition at operation speeds of 1000, 2000 and 4000 RPM. The rotor was balanced by a 4-plane, 3-speed, 4-probe least squares procedure. The rotor job probe response levels are less than 13 μm (peak-to-peak) over the speed range of interest. The initial drop orbit at 1000 RPM for the plain bearing is shown in Figure 3. A repeat drop demonstrated very good repeatability. The data for the two tests are almost indistinguishable for the first several impacts, subsequently diverging. The continuation and stable decay of the orbit is shown in Figure 4. Another presentation of the drop of Figure 3 is given in Figure 5 as a magnitude and attitude angle measured from bottom dead center. This presentation allows for rapid evaluation of forward and reverse whirl [3]. The data before the drop shows the phase for forward whirl about the center of the bearing. A negative whirl would have phase slopes just the opposite of that shown. After the drop, the phase should show an oscillation and eventually settle to a 0° position for a stable non-whirling condition. Figure 6 shows the time trace from additional non-contact probes located at the rotor coupling hub and the drive pinion hub.

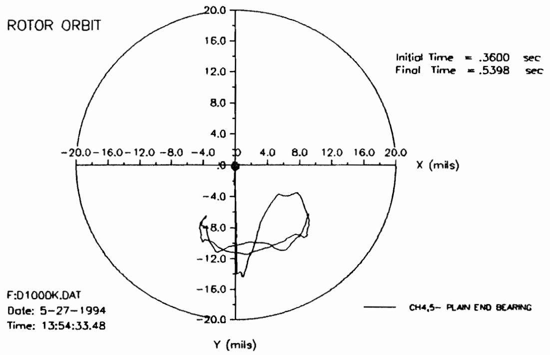


FIGURE 3 : 1000 RPM, Balanced, Inboard Bearing

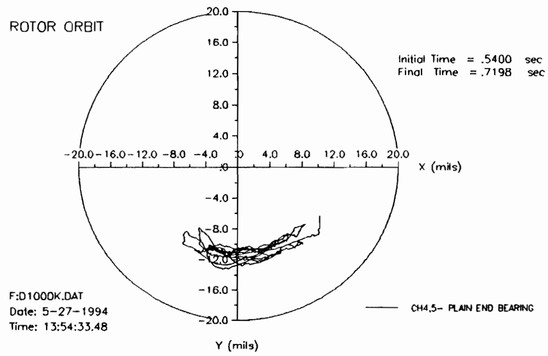


FIGURE 4 : 1000 RPM, Continuation of Figure 3

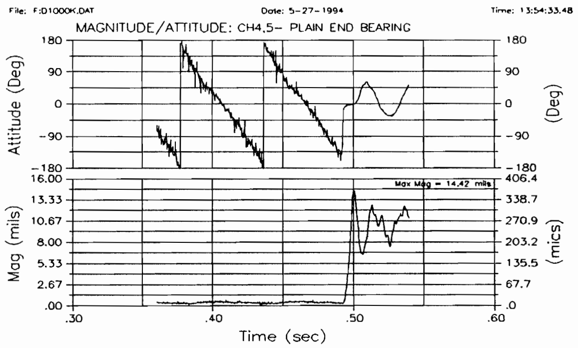


FIGURE 5 : 1000 RPM, Balanced, Inboard Bearing

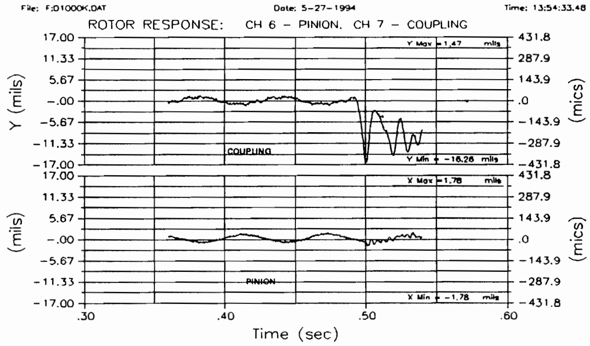


FIGURE 6 : 1000 RPM, Balanced, Coupling & Pinion

A drop at 2000 RPM at the inboard bearing is shown in Figures 7 & 8. The horizontal and vertical sensor time traces are shown in Figures 9 & 10. The magnitude and attitude plot for this case is given in Figure 11. The attitude angle is seen to settle near 0° position for this stable drop. Figure 12 shows the timing marks from the backup bearing cage (25/rev.) and the inner race (60/rev.). The increasing angular velocity of the cage is clearly indicated by the shortening period near the end of the time block. Evaluation of such information can be useful for determination of bearing performance, especially in regards to ball/roller skid which can cause excessive heating and premature failure of the backup bearing.

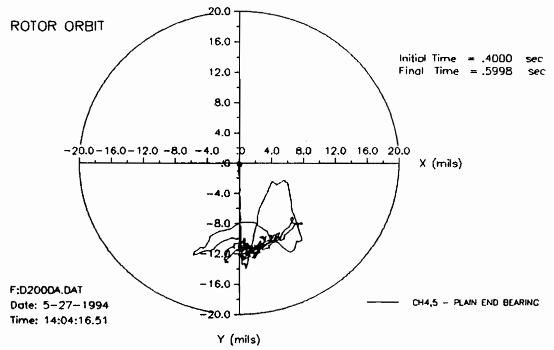


FIGURE 7 : 2000 RPM, Balanced, Inboard Bearing

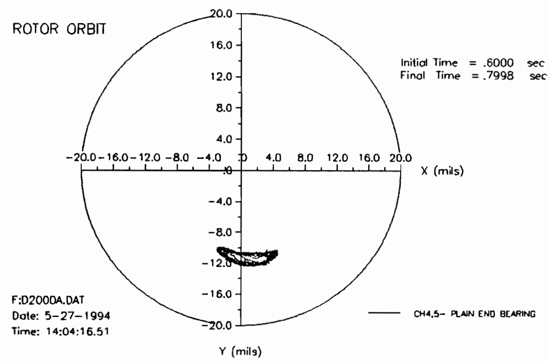


FIGURE 8 : 2000 RPM, Continuation of Figure 7

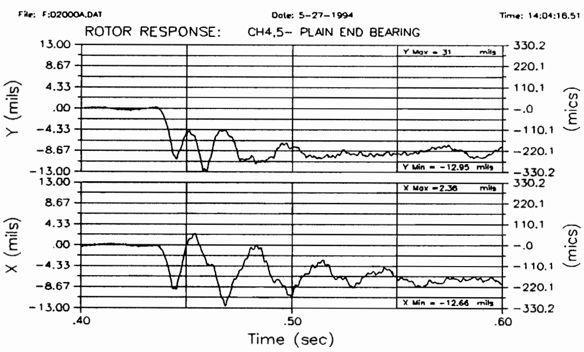


FIGURE 9 : 2000 RPM, Balanced, Inboard Bearing

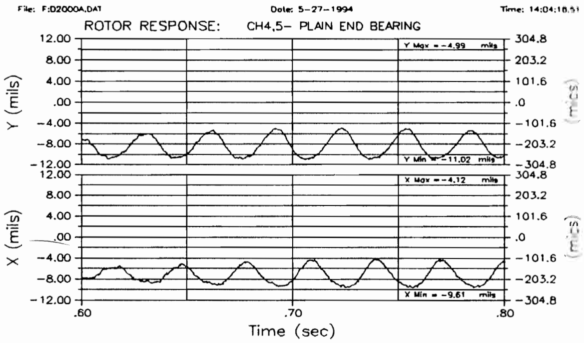


FIGURE 10 : 2000 RPM, Continuation of Figure 9

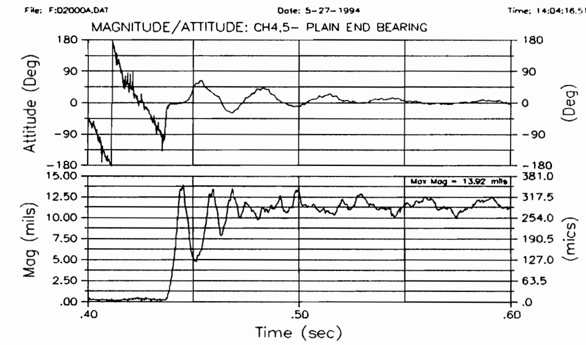


FIGURE 11 : 2000 RPM, Balanced, Inboard Bearing

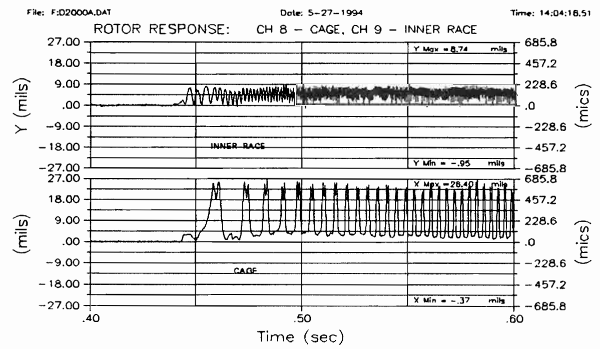


FIGURE 12 : 2000 RPM, Backup Bearing Spin Up

The drop at 4000 RPM is shown in Figures 13 and 14 for the inboard and outboard bearing locations. Only the inboard magnetic bearing is deactivated for this drop. There is a response on the thrust end due to the impact of a drop at the inboard end. This effect is shown in Figure 14.

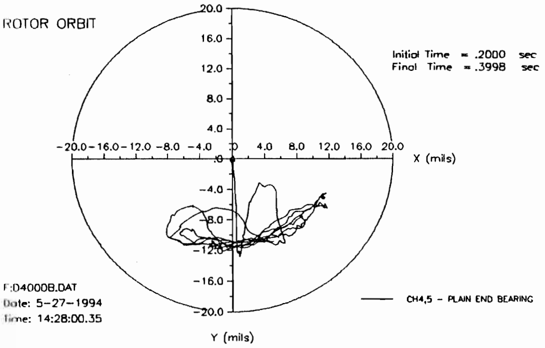


FIGURE 13 : 4000 RPM, Balanced, Inboard Bearing

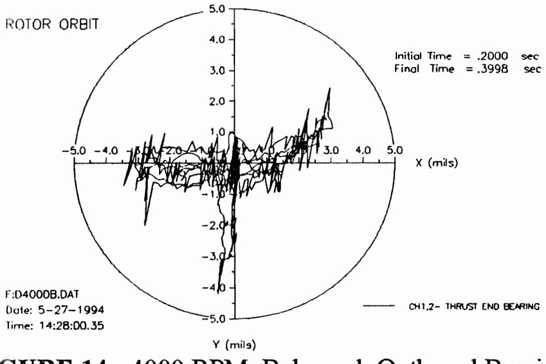


FIGURE 14 : 4000 RPM, Balanced, Outboard Bearing

The fact that this response is a reaction to the impact was verified by examining the time traces of the drop. The response plots for a drop at both inboard and outboard radial bearings are given in Figures 15-18. While the initial drop and impact is similar to the previous drops at only the inboard end, the total response transient is different as would be expected.

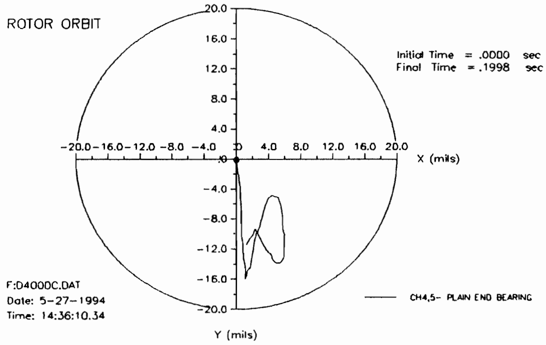


FIGURE 15 : 4000 RPM, Drop at Both Bearings

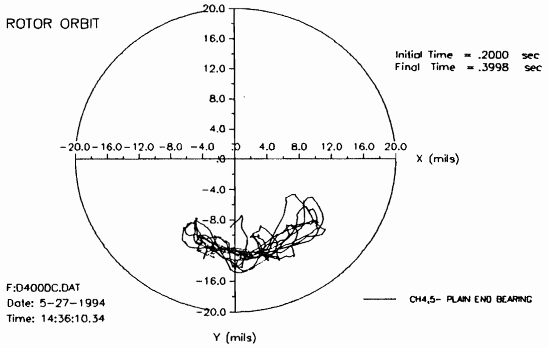


FIGURE 16 : 4000 RPM, Continuation of Figure 15

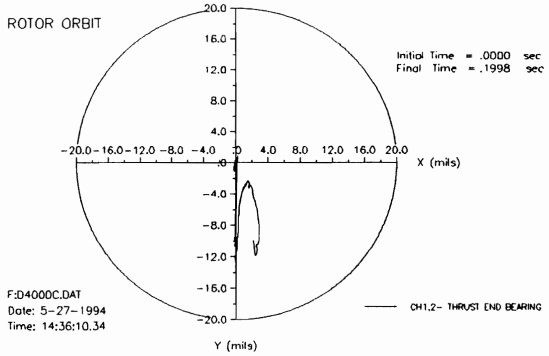


FIGURE 17 : 4000 RPM, Outboard Bearing

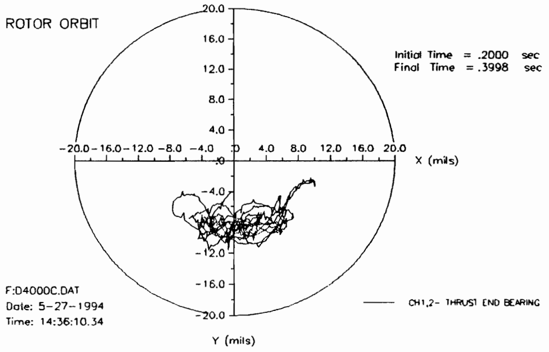


FIGURE 18 : 4000 RPM, Continuation of Figure 17

A major influence on the nature of the resulting drop and transient response is thought to be the balance level of the rotor. The results for increasing levels of unbalance (0.16 g and 0.3 g) are shown in Figures 19-22. The initial orbits are clearly increasingly larger and the transient responses more active. Tests at higher speeds and larger levels of unbalance will be conducted and results reported in future reports.

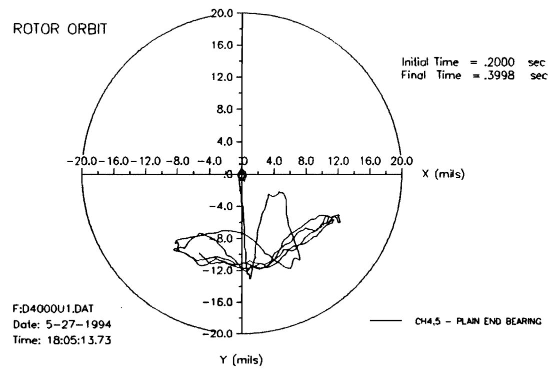


FIGURE 19 : 4000 RPM, Inboard Brg., Unb. = 4.66 g

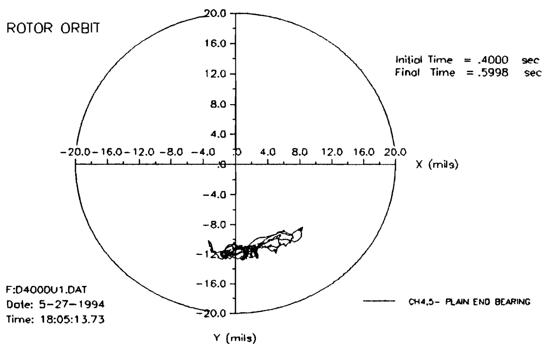


FIGURE 20 : 4000 RPM, Continuation of Figure 19

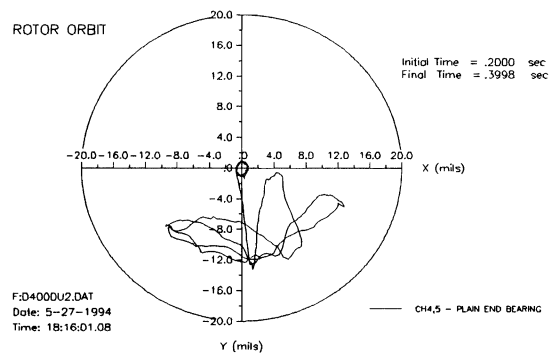


FIGURE 21 : 4000 RPM, Inboard Brg., Unb. = 8.51 g

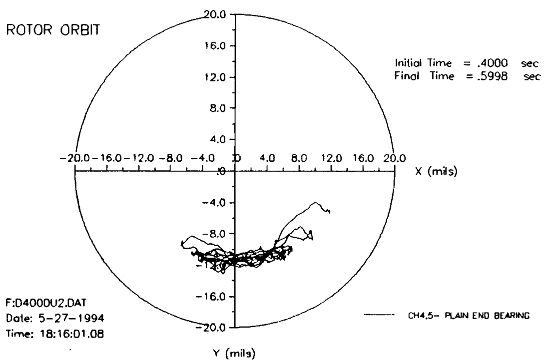


FIGURE 22 : 4000 RPM, Continuation of Figure 19

CONCLUSIONS

From the limited testing to date, the following conclusions have been reached :

1. A balanced rotor will drop and spin on the backup bearings if they are in a good condition
2. Increased levels of imbalance give the rotor more potential for larger motion in the backup bearings.
3. Unbalance levels as high as 0.2 g's have not caused large whirling motion.

These results are in agreement with previous analytical results of similar rotors [6]. It is expected that the dynamics of the rotor drop will be worsened by:

- Unbalance levels approaching 1 g.
- Operation near free-free modes will amplify the response upon rotor drop.

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