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Review of Mechanical Vibration Tests Conducted on Control Moment Gyros and Life Test Fixtures

Prepared for

George C. Marshall Space Flight Center
National Aeronautics and Space Administration
Marshall Space Flight Center

August 24, 1973

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TECHNICAL REPORT

Review of Mechanical Vibration Tests Conducted on Control Moment Gyros and Life Test Fixtures

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Prepared for

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INTRODUCTION

This report is a summary of experimental vibration studies performed on a number of flight control moment gyros and bearing life test fixtures for the National Aeronautics and Space Administration, Marshall Space Flight Center, Huntsville, Alabama. Tests were performed at MSFC, at Wyle Laboratories, Huntsville, Alabama, and at the Bendix Corporation facilities in Teterboro, New Jersey. Test period covered is from January, 1971, through July, 1972. A description of test and analysis equipment is included as well as test procedures and overall performance rankings. Advanced ultrasonic rolling element bearing fault detection techniques were applied for bearing analysis along with conventional vibration and sound analysis procedures.

TEST RESULTS AND CONCLUSIONS

I. A tabulation of results from all CMG testing is included as sheets D-1 through D-6. Based upon bearing condition, unbalance levels, bearing misalignment, and acoustic noise, the overall performance ranking is as follows:

Rank	Serial No.	Comments
Best	0008	Smooth bearings, low unbalance, low sound level, good alignment.
2.	0007	As above, nearly as good as 008
3.	0009	Slightly more noisy than units above but very good.
4.	0002	Bearings rough, probably due to ball wear. Unbalance fair - no mounting problems. Noise fair.
5.	0010	Unit bearings good until retainer squeal - Apparent mounting problems produced 2 per rev vibration. Unbalance low, noise high.
6.	0004	Bearings rough, outer gimbal damaged by shake tests. Resonant response of structure to rotation frequency - even with best IG (0008).

II. Test results from engineering IGRA units E-2 and E-3 are tabulated on sheet E. Based upon selected parameters at 7900 rpm, ranking is as follows:

Rank	Serial No.	Comments
Best	E-2	Bearings fair, unbalance very good. Sound level low, bearing mounting good.
2.	E-3	Bearings fair to poor, unbalance good, sound fair to poor. Bearing mounting problems present

Both units rank below worst flight IGRA's but better overall than CMG 0010 and CMG0004.

III A tabulation of results from LTF units is included as sheets F-1 and F-2.

Rank	Serial No.	Comments
Best	1	Very good performance
2	4	nearly equal
3	5	
4	3	Very nearly the same - good performance
5.	2	
6	6	Bearings rough - worn - balance fair

IV. Further conclusions are as follows:

1. The bearing fault detection technique developed under NASA Contract NAS8-25706 can be applied to the analysis of problems occurring in Life Test Fixtures and Control Moment Gyros.

2. High endurance hour test vehicles show increases in high frequency resonant responses characteristic of general wear rather than from discrete faults.

3. Preflight noise and vibration tests appear to inflict more damage upon outer gimbal components than upon gyro rotor support bearings.

4. Bearing retainer squeal and the resulting material removal appears to be the most likely failure mode of CMG bearings.

5. Bearing retainer squeal produces significant response at 900 H_z and 3100 H_z which can be used for two possible uses:
   a. To indicate the presence of squeal in a particular bearing.
   b. To aid in research to define the mechanism of retainer squeal and techniques to minimize or eliminate the occurrence.

6. Dynamic loads generated within CMG bearings during sweeps from one angular position to another might produce structural problems not predicted by analysis techniques. Apparent loads in excess of 100 pounds at 300 H_z were demonstrated for CMG 0010 at 3° per second sweep rate.

DISCUSSION

Definition of Bearing Failure

The application of rolling element bearings to machinery support systems often produces a number of significant operating advantages: starting and running torque requirements are minimal, lubricant flow demands are low, load capacity is large for steady state and transient conditions, generated temperatures are reasonable, and vibration levels are low. Failure of the bearings typically is caused by or results in a change in the above conditions. Torque requirements go up as surfaces deteriorate or debris builds up. Interruption of oil flow usually results in unusual wear and friction with a resulting increase in torque and temperature. Increasing roughness produces greater amounts of bearing generated vibration which may interfere with the use of the complete machine. The ultimate failure of a particular bearing may be from a number of possible modes. The classic failure is fatigue, where the surface of one of the elements of the bearing is stressed beyond its ability to resist and a crater is formed as material pops off. A typical fatigue fault is 0.008 to 0.012 inches in diameter and 0.001 to 0.004 inches deep. As wear progresses, additional faults occur and general deterioration is accelerated. Improvement in material properties due to such techniques as vacuum degassing of steel has produced lower statistical failure scatter and has extended fatigue life beyond the hours predicted.

An increasingly more common failure mode of rolling element bearings is due to retainer or separator failure. Advances in ball and race materials and in lubricant properties have permitted increased speeds, loads, and temperatures which have sometimes exceeded retainer capabilities. Fracture or rapid wear often occur during a retainer failure to produce large changes in bearing torque, excessive heating, high vibration levels, and audible noise. Failure of the bearing may be very sudden and dramatic.

A third bearing failure mode involves the general deterioration of rolling contact surfaces due to wear. Deterioration progresses from the first revolution of the bearing until at some point the increased roughness of the surfaces produces increased torque, temperature, and vibration beyond permissable levels.

The rate of wear is dependent upon time, lubricant performance, and foreign material present within the contact region.

A fourth failure mode is termed lubrication failure. The gradual or sudden cesation of lubrication usually does not cause instantaneous failure, but in time leads to retainer difficulties or to increasing wear rates. If the bearing is dependent upon the lube supply for cooling, deterioration will progress more rapidly as components lose strength with increasing temperature, noise, or vibration limits.

Other bearing failure modes would typically result in rapid advancement of one of the failure modes indicated above. Improper mounting, for example, might produce local high stress as a cocked race forces a few balls to carry the total bearing load. Greatly increased retainer load follows any distortion of the normal stress distribution within the bearing.

The limits which are applied to bearing condition must be set by the application. Obviously, when bearing torque exceeds driving torque then the machine will slow down or stop. Other limits are more subtle, depending upon such criteria as, the importance of complete availability or the permissable level of acoustic or mechanical noise.

Bearing Fault Detection

A bearing fault detection technique based upon ultrasonic frequency range vibration has been developed under NASA, contract NAS 8-25706 and reported in Mechanical Technology Report No. 71TR-1. This fault detection technique has been applied to condition monitoring of control moment gyro inner gimbal rotor assembly bearings and to life test fixtures used to develop bearing systems for the flight gyros.

The principal of operation of this fault detector is quite straight-forward. As a ball rolls between the inner and outer race of a new bearing, the smooth surface of the ball "sees" an equally smooth track which offers minimum surface irregularity. The resulting vibration levels are very low. During the life of the bearing these contacting surfaces gradually roughen and the higher peaks of roughness contact one another such that local high stress regions exist. In time, repeated high stress contact will result in the development of a spall which will be a discrete gap or void in the rolling track, and each ball passing over that gap produces an impact as the ball load is relieved and then suddenly re-applied much as the tire of a vehicle is shocked by contact with a chuck-hole in a road. The energy of the impact is a pulse input to the system which causes the components of the system to resonate or "ring" at natural frequencies of vibration. As the components of a properly operating rolling element bearing are very regular, discrete repetitions of the impact occur as subsequent balls hit a race defect or as a pitted ball alternately contacts inner and outer race. (The ball defect may not always be in the track of rotation of the ball, but under conditions of uniform speed and load a ball tends to run in one preferred plane. A ball defect then will appear for some period and then disappear as loads or speeds change.) It usually is expected that a struck part will resonate strongest at its first natural mode of vibration, and this is true for free unmounted components with minimum damping, however, lower modes of vibration are apparently suppressed while high modes are quite readily transmitted. Further emphasis of high frequency components is accomplished by measuring acceleration (which is related to force) rather than the often used displacement vibration limits. Acceleration is increased by the square of the frequency (Acceleration = $0.0511 X (\text{frequency})^2$ X Displacement) so resolution is enhanced.

A primary problem with high frequency vibration analysis in the past has been the availability of suitable sensors. About the time of the original bearing fault detection program, several accelerometers with capability of response to $40 KH_z$ and above became available so these have been used to allow evaluation of the ultra-sonic region. The 107 size ball bearing used in initial Life Test Fixtures and Inner Gimbal Rotor Assemblies produced a major response at 28,000 H_z when an artificial flaw was inserted. This frequency was later resolved to be approximately the third ring mode resonance of the inner race the fifth ring mode resonance of the outer race, and, depending upon load, possibly the resonance of the ball on its oil film. The ring mode resonances were evaluated experimentally and were found to correspond well to computed values. It was found that race mounting conditions significantly affected lower mode response amplitudes but that higher modes were quite insensitive to the fit between shaft and race or race and housing. This may explain the greater response of the $28KH_z$ signal...the lower modes were suppressed by external influences, and the component resonances combined to produce the superior output. It should be noted also that the levels of vibration measured, ten to thirty G's peak (gravity units) are displacements of 0.00000025 inches to 0.00000075 inches peak-to-peak. Most fluid and friction damping mechanisms require significant deflections to be effective so this may explain the good transmissibility of the high frequency data with only moderate interface loss.

The high frequency resonant response of the bearing components is treated as a communications wave carrier to extract additional information about the source of impact response. A single spall in the inner race ball track will produce regular impulse - and - decay responses as each ball in turn contact is shown as modulation of the resonant response frequency of the bearing, and demodulation produces a sine-like wave which clearly shows the ball-defect contact frequency. For the 107 size bearings used in the CMG program, an inner race defect contact occurs at 8.7 times inner race rotation frequency, an outer race defect contact occurs at 6.3 times inner race rotation frequency, and a ball defect contact occurs at 6.0 times inner race rotation frequency. (These frequencies are computed from ball and race dimensions and will vary depending upon geometry. A fair rule of thumb is that retainer rotation is approximately 40% of inner race rotation frequency so that in one revolution an inner race spot will overtake 60% of the balls in the complement. For this bearing there are 15 balls, so inner race fault frequency is about 9 times rotation.)

To minimize resolution, only the demonstrated bearing resonant frequency is demodulated. A band pass filter centered at $28KH_z$ attenuates other high frequency components while passing those which define fault character. The Bearing Fault Detector can be applied directly to raw bearing data or it can be used to aid in analysis of tape recorded accelerometer responses.

Application of Fault Detection Techniques

Soon after the high frequency bearing fault detection technique was demonstrated it was applied to operating control moment gyro assemblies to determine the effects of pre-flight vibration and noise tests upon bearing condition. Tests were done at MSFC in Hartsville, Bendix test facilities in Teterborough, N.J., and at Wyle Laboratories in Huntsville on complete control moment gyros, inner gimbal rotor assemblies, and life test fixtures. Individual task results were reported by memos and by verbal presentations at Huntsville and at Teterborough, but this report will consolidate test procedures, results and conclusions, and attempt to relate the various tasks to a common performance base.

Test Sensors

Analysis of complete mechanical system problems is best accomplished by monitoring a number of appropriate outputs. High frequency response accelerometers, Bruel and Kjaer Model 4344 units with selected response characteristics, were attached to the external housings as near the bearings as possible. The accelerometers were stud attached to a one inch by one inch by one-fourth inch aluminum block which was glued to the unit using brittle cyano-acrylate adhesive (Eastman 910 or equivalent) at a location in the radial plane of the bearing being tested. For the CMG and IGRA units, this location was on the main body of the inner gimbal frame as shown on sheet A. For life test fixtures, the accelerometer mounting blocks were glued to the hexagonal end pieces which support each bearing mount assembly. These model 4344 accelerometers have mounted resonance frequencies near 85KH_z and so the usable frequency range is greater than 50 KH_z with only minor amplitude errors. This frequency band includes the selected bearing resonance frequency of 28 KH_z.

Other test sensors used to define overall system performance included the built-in Kistler accelerometers which were mounted directly on the housing and sleeve assemblies which hold the gyro rotor bearings. The use of these sensors was limited by a major problem: the mounted resonance of the accelerometers occurs in the range between 32,000 and 40,000 H_z, and very often the built-in electronics of the accelerometers were saturated by large responses at accelerometer resonance. Because the accelerometer charge conditioning equipment was located within the unit, it was not possible to filter out this resonant response before amplifier overload occurred, so results often were questionable.

Additional low frequency response accelerometers were mounted on the inner gimbal frame to measure axial vibrations of the gyro rotor as shown also on sheet A. Bruel and Kjaer Model 4333 or Kistler Piezotron Model 568 units were used to define frequency components to 5000 H_z. Major usage of these sensors was for component measurements at rotational and twice rotational frequencies.

Several outer gimbal locations were used to monitor low frequency vibrations under specific test problem conditions. The Bruel and Kjaer 4333 units and the Kistler 568 Accelerometers were used alternately at these sites.

A significant indicator of overall machine performance is the acoustic output, so a Bruel and Kjaer Model 2203 precision sound level meter with one inch condenser microphone was used to monitor sound levels. The microphone was placed next to test bearings for LTF and IGRA tests (two inches to 8 inches away from individual bearing locations) and was inserted into the port in the cover of the complete CMG for those tests. Octave filter levels were tabulated for initial tests, but it was found that narrow band frequency analysis was necessary to discriminate pure tones generated at rotation and two times rotation frequency.

Data Record

All test signals plus gyro speed indications were recorded on magnetic tape with a Lockheed Electronics Model 417D seven channel recorder operating at 30 inch per second tape speed. An edge voice track allowed a running commentary of test conditions and impressions to be recorded along with the test sensor outputs. The Lockheed recorder has plug-in electronics which permit the selection of Direct or FM record capability for each tape channel. At 30 ips, the FM record channels have linear response from DC to 10,000 H_z while the Direct record channels respond from 200 H_z to 100,000 H_z within ±3db.

This latitude permits complete spectrum coverage - the model 4344 high frequency accelerometers were recorded on FM and Direct while other sensors were recorded on FM only.

Between sensor and recorder channel, Encore Electronics Model 501 amplifiers were used to provide adjustable gain capability. Signals need be in the one volt rms range to optimize recorder signal-to-noise levels and the Encore units permit precise gain adjustment from 0.1 to 1000 in 1-2-5 steps. A data log was used to identify recorder input gain and test conditions for each channel.

Test Procedures

An effort was made to standardize on a test plan to minimize possible errors and to provide maximum machine performance identification. A typical CMG test arrangement was as follows:

1. Steady state performance with gyro rotor centerline horizontal
2. Steady state performance with gyro rotor centerline vertical with bearing number 1 down.
3. Sweep from bearing 1 down to bearing 1 up at 3° per second sweep rate
4. Steady state performance with gyro rotor centerline vertical with bearing 1 up.
5. Sweep from bearing 2 down to bearing 2 up at 3° per second, sweep rate
6. Steady state performance at any special axis position (to define an unusual performance condition such as retainer squeal).

Tests on IGRA units followed this plan as closely as possible within the restraints of the support structure for each test vehicle. Life Test Fixtures were operated with the shaft center line horizontal and the machine base set on rubber pads to isolate the unit from other machine vibrations.

Gyro rotor speeds initially were set at 7900 rpm but part way through the test program the need for additional gyro energy pushed operating speeds to 9000 rpm. Many of the units were checked at both speeds to define performance differences.

Test Data Reduction

Tape recorded data were analyzed at Mechanical Technology, Inc. laboratory facilities using a Spectral Dynamics Model 301A Real Time Spectrum Analyzer (with the SD302 Time Averager) as the primary tool. This unit provides narrow band frequency analysis of sensor outputs which allow identification of the source of machine vibration and noise. To permit full frequency band analysis, the Lockheed tape recorder was operated at 7 1/2 inches per second to effectively compress the high frequency response data signals from 200 to 80,000 H_z into a band from 50 to 20,000 H_z, the operating frequency band of the real time analyzer on its highest range setting. Real time analyzer outputs were recorded by Hewlett-Packard 7004 X-Y Recorder.

Performance Analysis

Analysis of overall machinery condition was based upon a number of considerations. The high frequency bearing analysis technique was applied to monitor bearing condition, even though limits of performance have not yet been established. Relative levels can be used, and the bearing fault detector instrument built for NASA on this contract does permit the discrimination of faulted bearing component frequencies. Preliminary testing of a flight type CMG bearing with an induced fault has indicated that that bearing has a resonance at 26,000 H_z which responds to bearing impacts, so both 26KH_z and 28KH_z spectrum response levels were recorded as a measure of performance.

Gyro rotor unbalance response was defined by radial accelerometer outputs at rotational frequency. Comparisons between units permits another input to machine performance ranking. The internal Kistler accelerometers were reviewed when possible, and external B and K accelerometer levels were used otherwise.

Radial accelerometer outputs at 2 times rotational frequency usually are indicative of bearing mounting problems such as race skew or out-of-roundness, so this parameter was evaluated as an additional performance measure.

Axial acceleration at rotational and at two times rotational frequency also provide an indication of bearing mounting condition as these components cannot be generated without irrregularities in the rolling element components. Not all tests had axial accelerometer data available, but where possible these parameters were evaluated in defining performance. It was assumed that mounting errors will produce locally higher bearing stresses and higher ball separating loads which will decrease the life of the overall system (or at least increase the possibility of premature failure). No attempt was made to assess the effects of additional vibratory loads upon other structures or devices in the CMG area.

Overall sound pressure level and the presence of discrete frequency components in the sound spectrum form an additional rather subjective parameter for use in ranking complete machines. The narrow band spectrum analyzer permits resolution of frequency components which otherwise would be lost by normal analysis techniques, such as separating a 63H_z retainer rotation frequency from 60 H_z noise.

Test Conditions For Ranking

It was found that a horizontal shaft position produced the most consistent output of low and high frequency data. It also was theorized that gravity thrust loading was an unrealistic load condition for either bearing to have on it, so the horizontal position was most like a space situation. In the low frequency range the differences between vertical and horizontal positions made only minor response changes, but again optimum bearing loading occurs in the horizontal mounting condition so rating is done with that plane.

Sweep tests were recorded whenever possible with bearing performance recorded as the unit is "rated" from bearing down to bearing up. This produced some rather startline level changes in some machines, and unfortunately quite often produced signal levels in excess of tape recorder level capacity so the record was "clipped" due to saturation. One example, included as Sheet B, shows the two times rotation component for a sweep for bearing No. 1 down to up showing a 20 time increase in level, from 0.020 G up to a maximum of 0.460G at just above horizontal and then slowly dropping to 0.200 "G" just before the sweep is concluded. Included sheet C shows the saturation of the FM record channel record of the same test sensor as for sheet B.

It is theorized that this complex response is made up of some small changes in bearing operating conditions due to gravity and preload washer loadings and operating contact angles within the ball bearings. Some machines, however, produced only minor dynamic sweep level changes. Due to uncertainties about the mechanism of response, no performance ranking was done based upon sweep tests, but it appears that significant dynamic loads are occurring which might produce control or structural problems outside the CMG area. A 1/2 G acceleration of the complete IGRA is equivalent to a load in excess of 100 pounds.

Performance Ranking

Based upon experimental results, an overall performance ranking has been produced for each type of unit tested. For complete Control Moment Gyros, the ranking is as follows:

Rank	Serial No.
Best	0008
2	0007
3	0009
4	0002
5	0010
Worst	0004

Because all performance factors are not equal in their influence upon unit life, any such ranking is open to considerable discussion. CMG unit number 0010 was derated significantly because of excessive response at two times rotation frequency which indicated that problems existed in the bearing mountings.

Inspection by Bendix showed that components were made to blueprint specs and that significant improvement in vibration levels occurred when lubricator nuts were exchanged end for end, but elimination of excessive noise and vibration were not accomplished. The fact that bearing retainer squeal developed within this unit may or may not have been related to the large twice rotation frequency noise and vibration which were present.

The retainer squeal phenomenon appeared to be a common failure mode for the CMG bearings as it occurred in several tested units. The deterioration and removal of retainer material which can accompany squeal would produce excessive drag from ball track "litter" and lead to premature failure. Test measurements indicated that squeal produces significant sound and vibration signals at 900 H_z and 3100 H_z which some investigators[1] consider to be retainer whirl frequencies. Maximum response occurred at the internal Kistler accelerometer and it is concluded that a simple monitor could be built to give warning of the presence of squeal. It may also be possible to detect squeal symptoms ahead of the audible output which could serve as a screening test for flight units.

The bearing fault detection technique did not give any indication of retainer squeal problems, but it is assumed that bearing resonant response would occur

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1 Kingsbury, E.P. "Torque Variations in Instrument Ball Bearings" ASLE Transactions 8-435-441 (1965)

as debris builds up in the rolling track.

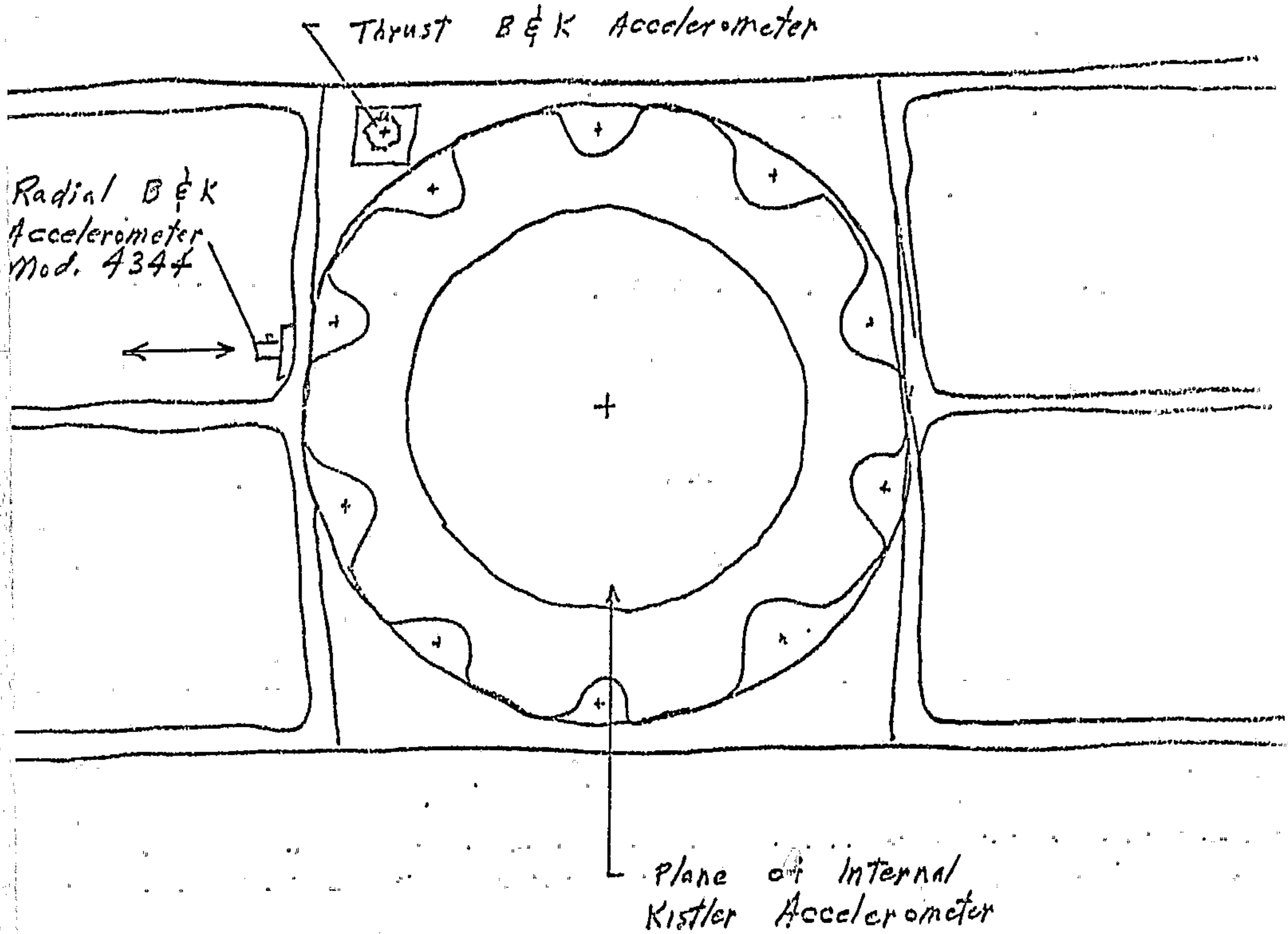
The ranking for CMG 0004 was very low because of apparent damage to the outer gimbal which occurred during vibration tests at Wyle Labs. This damage showed up as an excessive response at rotation frequency when the spin axis of the gyro was vertical. The inner gimbal rotor assembly from CMG number 0008 was inserted into 0004 outer gimbal and it also produced excessive rotational frequency response, indicating that the outer gimbal was at fault. Close inspection by Bendix apparently did not produce a satisfactory source of this low frequency response.

Ranking of the two engineering prototype inner gimbal units indicated that unit E-2 was better than unit E-3, but that both units were poorer in performance than any of the flight IGRA's.

Life Test fixture ranking required some difficult decisions as performance broke down into 3 groups. Unit number 1 was best, but unit number 4 was very close to it. Units 5, 3, and 2 were grouped together in the next three ranked positions with very good performance and little significant difference between them. LTF number 6 was last with significantly lower performance than the others.

Sheet A

CMG Accelerometer Mounting
End View of Gimbal Bearing



Sheet B

TEST LOCATION D-3 TAPE NO. 94 JOB DATE 5-18-71

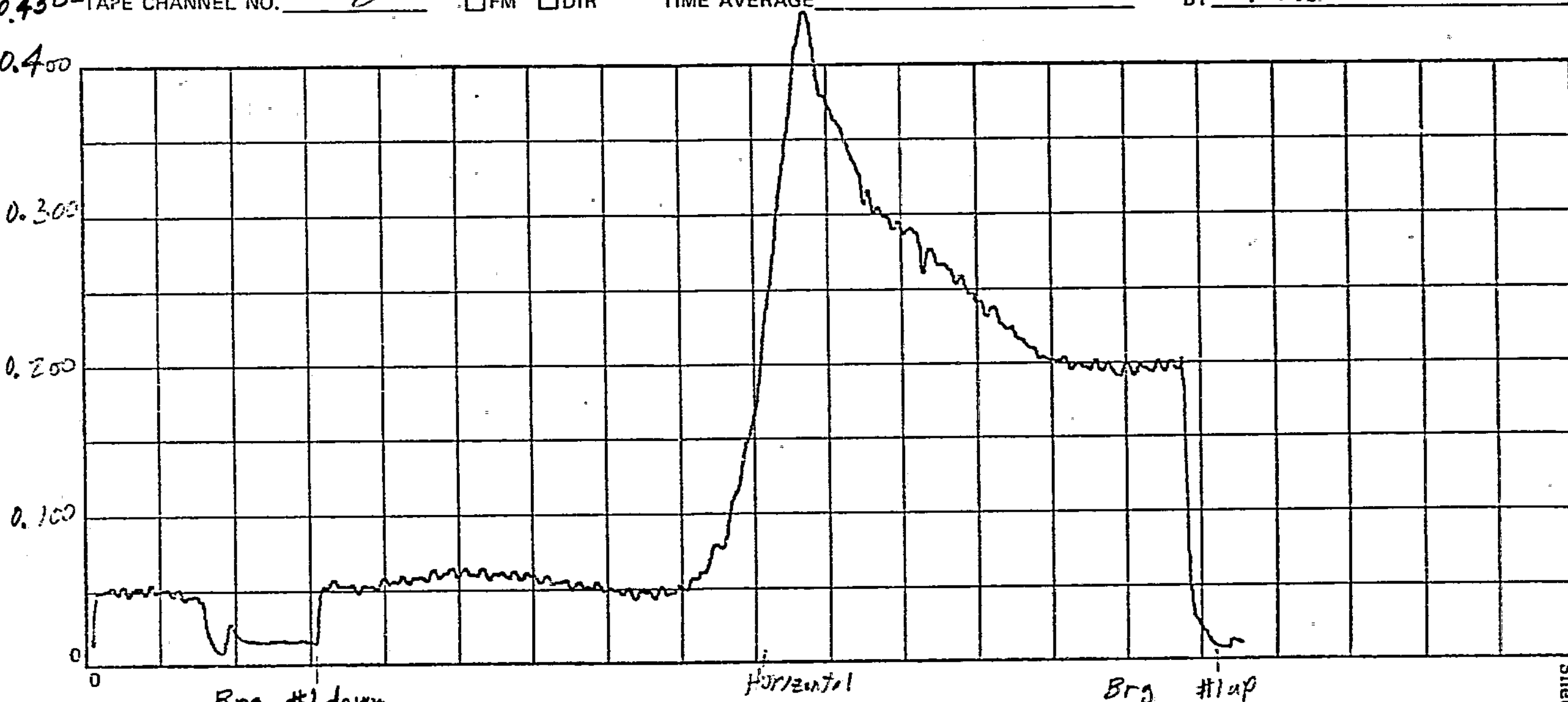
TEST CONDITIONS 9100 RPM Sweep 3°/sec Bearing #1 down To #1 Up Radial Accelerat

INPUT:
TRANSDUCER B+K Radial Bra #1
CALIBRATOR
AMPLIFIER
TAPE CHANNEL NO. 2 ☐FM ☑DIR

ANALYZER:
RANGE-V,RMS 0.1VRMS = 1VDC
FREQUENCY-HZ 300 ± ½ octave
GAIN
TIME AVERAGE

RECORDER:
GAIN 0.5 V/in
X-UNITS 10 in/s
Y-UNITS
BY

Peak G's Accel.



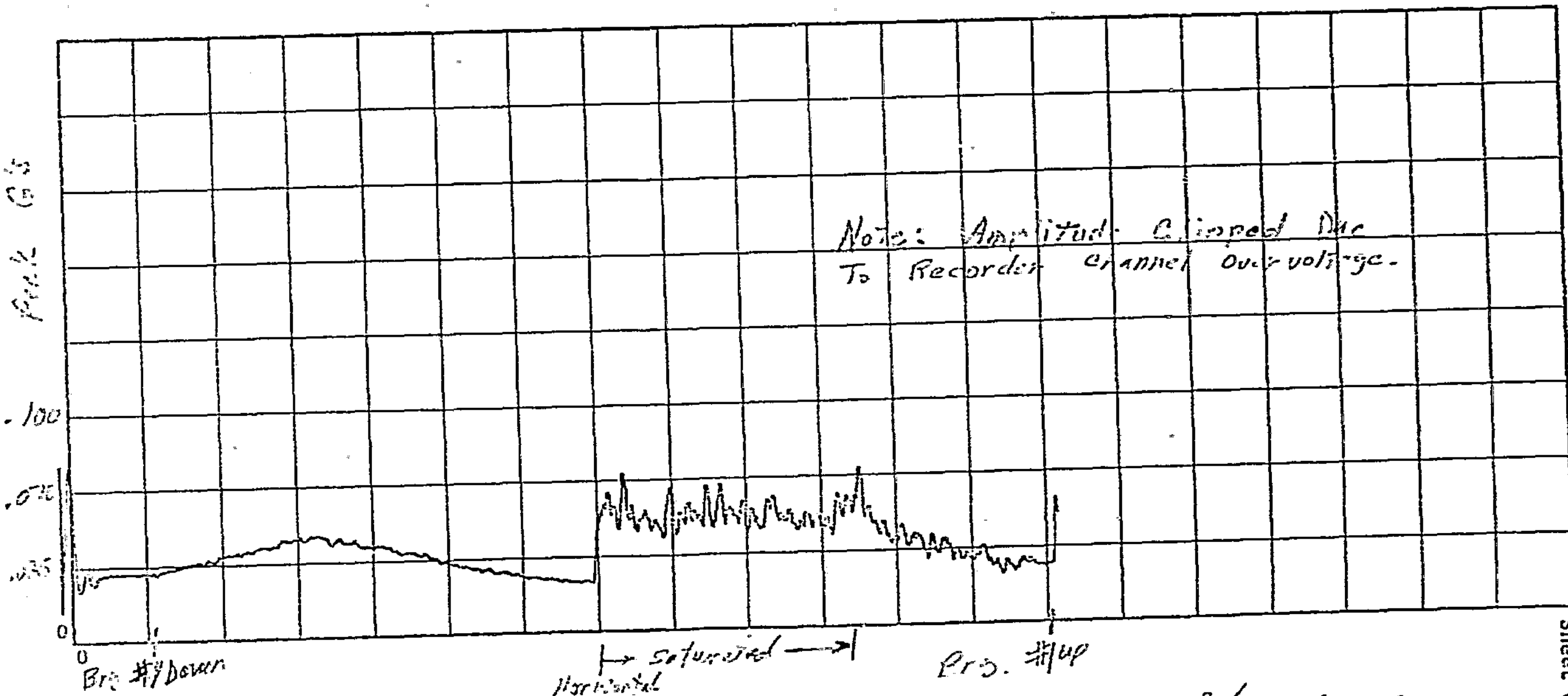
TITLE 2X Rotation Frequency - Radial Acceleration vs Position - 3°/Second Sweep
CMG # 2012 9100 RPM

Sheet C

TEST LOCATION D-3 TAPE NO. 94 JOB ______ DATE 5-13-71

TEST CONDITIONS 9100 RPM Sweep 3°/sec Bearing #1 from to #1 Hor. Radial Accel.

INPUT:		ANALYZER:		RECORDER:	
TRANSDUCER	S+I Radial Brg #1	RANGE-V,RMS	0.1 V RMS = 1Vdc	GAIN	0.5 V/in
CALIBRATOR		FREQUENCY-HZ	150 ± 5%	X-UNITS	
AMPLIFIER	10 X??	GAIN		Y-UNITS	
TAPE CHANNEL NO.	6 ☑FM ☐DIR	TIME AVERAGE		BY	



TITLE Synchronous Frequency Radial Acceleration Vs Position – 3°/Second Sweep ... down

CMG TEST PERFORMANCE SUMMARY 8-15-73

Unit Serial No.	0002				0004				0004		
Test Date/Location	1-12-71 MSFC				1-13-71 MSFC				3-9-71 Wyle		
Wheel Speed-RPM	7800				7850				7900		
Test Mount	Brg. No. 1		Brg. No. 2		Brg. No. 1		Brg. No. 2		Brg. No. 1		Brg. No.
Position of Shaft	Horiz.	Vert.	Horiz.	Vert.	Horiz.	Vert.	Horiz.	Vert.	Horiz.		Horiz.
Low Freq.Vib.-G's											
Rotation Freq-Rad.		0.13		.07	0.08	0.07-0.16	0.13	0.12-0.17	0.067		0.09
2X Rotation-Radial		.02				0.02	0.17	–	0.067		0.04
Other-G's@ H_z											
Rotation Freq-Axial											
2X Rotation-Axial											
Other-G's@ H_z											
Intermed.Freq.Vib.-G's											
900 H_z	–	–		0,16					0.05		
3100 H_z	–	–				.098740			.1587800		1.287800
Other-G's @ H_z		0.128850				.1888400					
High Freq.Vib.-G's											
25 KH_z	0.70	0.05	1.2		0.40		0.90	.05	0.2		0.3
28 KH_z	0.50	0.05	0.6		0.10	0.1-0.15	0.30	0.1	0.2		0.1
Sound Press.Level-dB											
Overall Level	92	94.5	91	92	94		94		94		
Components-H_z	260	1620			130		130		260		
(Highest First)	130	130			720		3000		130		
					3000				3100		
Service Meter Hours	5094				580					613 hrs.	
Comments			Ball Fault Freq. Present-Wear Damage?						After Vertical Shake Test		

	0004 3-10-71 Wyle 7900		0004 3-11-71 Wyle 7900		0004 3-26-71 MSFC 7900				0004 3-26-71 MSFC			
									7800	7500	6,000	
	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1		Brg. #2		Brg. #2		Brg. #2	
	Horiz.	Horiz.	Horiz.	Horiz.	Horiz.	Vert.	Horiz.	Vert.	Vert.	Vert.	Horiz.	Vert.
Low Vib.												
1/Rev.	0.085	0.10	0.065	0.10	0.082	0.21-0.36	.10	0,11-0,25	0.10	.065	.023	
2/Rev.	0.083	0.06	0.045	0.03	—				.01	-.003		
Ax. 1/Rev.												
2/Rev.												
Intermed.												
900Hz												
3150Hz	0.02		0.025									
Other	0.075@7800		0.05@7800	0.6@7800								
Hi Freq.												
26K	0.2		0.4	0.3		<.04		.12			–	–
28K	0.2		0.2	0.1		<.04		–			–	–
Sound OA	95 260 130		96 2350 260 130		97 128	98 130		98 130Hz			02	97
SMH					730							
Comments:	Horizontal radial load test shaking in place to load bearing radially.		Horizontal Load test with load applied horizontally in plane of center line of bearings (Axial)		Perf. Base							

CMG Date	0004 3-27-71 7750 MSFC				7000	0004 3-27-71 6000 MSFC	5920		0004 4-1-71 7920 Bendix				0004 OG/0008 IG 4-1-71 Bendix			
	Brg. #1		Brg. #2		Brg. #2		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2	
	Hor.	Vert.	Hor.	Vert.	Vert.	Vert.	Hor.		Hor.	Vert.	Hor.	—	Hor.	Vert.	Hor.	
Low Vib																
1/Rev	0.110	0.110	0.03	0.05	0.6	0.52	.06		.05	0.08	0.04	.030	0.04	0.280	0.060	.060.
2/Rev	0.02	0.02			.05				.015	0.01	0.01	.007	0.007	–	0.005	–
Ax 1/Rev											0.04	.08	.025	.04		
2/Rev											0.16	.04	.015	.01		
Intermediate																
900 Hz																
3100 Hz																
Other	.1287600				0.568700	0.358630										
Hi Freq.																
26K	0.4		0.7				0.16		0.3	.03	0.7-1.4	.1	.–		.05	
28K	0.1		0.3				0.4		0.2	.03	0.5-1.1	.1	.05			
Sound OA	92		94	92	94 740 Hz	97	96	90	79.8	78						
SMA	751															
Comments	Wyle Base								Inner Gimbal Stand				Complete Performance Stand 0008 IG in 0004 OG Assy.			

S/N Date/Loc. Speed	0007 3-15-72 MSFC 8950				0007 5-2-72 MSFC 8950				0007 5-11-72 Wyle 8950				0008 3-11-71 MSFC 7850		
	Brg. #1		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2
	Hor.		Hor.		Hor.	Vert.	Hor.	Vert.	Hor.	Vert.	Hor.		Hor.		Hor.
Low Vib.															
1/Rev.	.045		.062		0.037	.045	0.55		0.036		0.05		.033		.038
2/Rev.	.005		.028		0.037	.005	0.037		0.007		0.023		.008		.004
Ax. 1/Rev.	0.070		0.080		.10	.08	.10	.08	0.11		0.10				
2/Rev.	0.064		0.085		.09	.02	.10	.01	0.08		0.09				
Intermediate															
900															
3100															
Other															
Hi Frequency 26K		–	–		0.02		0.03		0.10		.20		–		–
28K	.10	.15	.10		.02		0.10		0.10		0.50		0.05		0.10
Sound OA	91	89–94			92		.		90	93.5			89.5		
	150				300		150						1500		
	300				150								133		
													187		
SMH	1916				1989										
Comments										Reglued block on bearing #1 side – reduced levels at 26K					

S/N Date/Place Speed	0008 4-1-71 MSFC 7800				0009 1-7-71 MSFC 7800				0010 1-8-71 MSFC 7750				0010 4-20-71 Bendix 9080			
	Brg. #1		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2	
	Hor.	Vert.	Hor.	Vert.	Hor.		Hor.		Hor.	Vert.	Hor.	Vert.	Hor.	Vert.	Hor.	Vert.
Low Vib. 1/Rev.	0.024	0.048	0.080	0.035	0.03	0.01	0.02	0.06	0.100	0.06	0.08	0.11	0.025	0.03	0.057	0.070
2/Rev.	0.010	0.006	0.001	0.005	0.03		0.02		0.24	0.02	0.27	0.02	0.070	0.0	0.110	—
Ax. 1/Rev.		0.020											0.170	0.150	0.20	0.160
2/Rev.		0.007											0.375	0.010	0.38	
Intermediate 900 Hz														0.050		
3100 Hz																
Other																
Hi Frequency 26K	0.1	—	0.02	—	0.18		0.10	—			0.03	—	0.060	—	0.07	—
28K	0.07	—	.04	—	0.09	0.01	0.08	—	—		0.04	—	0.175	—	0.12	—
Sound OA	89.5				92	96	92	94.5	105	99			104	91.5	104	95
	130				130				260	130			300	150	300	150
	260				260				130				150	300	150	300
SMH	205				127				100				414			
Comments																

	0010 4-20-71 Bendix 7950				0010 IG 7-27-71 Bendix 9000				0010 IG 7-27-71 Bendix 7900			
	Br. #1		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2	
	Hor.	Vert.	Hor.	Vert.	Hor.	30° from Hor.	Hor.	30° from Hor.	Hor.	30° from Hor.	Hor.	30° from Hor.
Low Vib. 1/Rev.	0.065	0.067	0.046	0.090	0.042	0.012	0.133	0.070	0.045	0.027	0.030	0.050
2/Rev.	0.030	-	0.017	-	0.062	0.060	0.075	0.050	0.022	0.015	0.030	0.012
Ax 1/Rev.	0.160	0.05	0.070	0.03	0.100	0.100	0.210	0.120	0.040	0.050	0.062	0.035
2/Rev.	0.055	0.005	0.020	0.025	0.033	0.040	0.130	0.090	0.070	0.020	0.040	0.016
					0.03@2800			0.50@3100			0.05@940	0.10@930
											0.07@3200	0.28@3200
Intermediate 900 Hz					-	0.095	-	0.083		0.050	-	0.067
3100 Hz					-	0.040	-	0.070	0.010	0.020	-	0.030
Other									0.012@2300			
Hi Frequency 26K	0.170		0.17		0.05	-	0.200	0.05	0.120	-	0.160	0.120
28K	0.080		0.16		0.120	0.060	0.050	0.050	0.150	0.060	0.150	0.070
Sound OA	97	90.5	91	98	82	85	80	86	77	84	77	87
	130	130	130	130	150	944	155	940	133	964	2400	954
	260	900	260		300	3100	300	3100	660	133	133	3140
					3600				267	1600	267	
SMI												
Comments	1/Rev. and 2/Rev. modulation of 28KHz.				No.2 Brg.down 30° from Horiz. produced brg.noise- "squeal" inspection showed retainer problems.				Bearing fault detector shows retainer rotation freq.,2X retainer rotat.& shaft rotat.for horiz. check of #2 bearing.			

TEST PERFORMANCE SUMMA

ENDURANCE IGRA

	E-2 1-12-71 MSFC 7800 Endurance Mount		E-2 3-31-71 MSFC 7900 Engr. Mount					E-2 3-31-72 MSFC Engr. Mount		
	Brg. #1	Brg. #2	Brg. #1		Brg. #2		Brg. #1	Brg. #2		
	Hor.	Hor.	Hor.	Vert.	Hor.	Vert.	Hor.	Hor.	Hor.	Hor.
Low Vib. 1/Rev.	0.020	0.015	0.022	0.020	0.015	0.020		0.03		0.05
2/Rev.	0.055	0.010	0.020	0.005	0.0025	.013		0.06		0.13
3/Rev.			0.030		0.055	-		0.08@372		0.14@372
Ax. 1/Rev.			0.008							
2/Rev.			0.038							
Intermediate										
900										
3100										
Other			0.07@800							
Hi Frequency										
26K	0.30	0.27	0.27	0.2	0.2	-	0.10	0.20	0.06	0.20
28 K	0.40	0.13	0.45	-	0.12	-	0.05	0.10	0.05	0.10
Sound OA	87	86.5	74.6	79	81.5	92	81			
	260	1300								
		260								
SMH	19000		21,079				2300			
Comments:			Bearing Squeal present in one position.					Bearings noisy-to be removed.		

FOLDOUT FRAME

TEST PERFORMANCE SUMMARY 8-18-73

E-2 31-71 MSFC 00 Engr. Mount				E-2 5-11-72 MSFC Engr. Mount			E-3 4-20-71 Bendix 7900 Engr. Mount				E-3 4-20-71 Bendix 7650 Hanging in Straps		
	Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2		Brg. #1		Brg. #2
Vert.	Hor.	Vert.	Hor.	Hor.	Hor.	Hor.	Hor.	Vert.	Hor.	Vert.	Hor.		Hor.
0.020	0.015	0.020		0.03		0.05	0.06	.04	0.04	0.03	0.034		0.070
0.005	0.0025	.013		0.06		0.13	0.07	-	0.03	0.004	0.030		0.015
	0.055	-		0.08@372		0.14@372					0.070		0.012
							0.125	0.05	0.080	0.01	0.160		0.040
							-		-		0.050		0.020
							0.25@825	0.55@825	0.22@825	0.00@825	0.18@0390		.08@390
											0.00@1525		0.23@1525
0.2	0.2	-	0.10	0.20	0.06	0.20	0.280	0.02	0.70	0.060	0.120		0.450
-	0.12	-	0.05	0.10	0.05	0.10	0.250	0.002	0.560	0.060	0.150		0.700
79	81.5	92	81				84	88	86	103	86		86
							390	2450	390	825	130		825
							750	825	130	1600			2200
			2300				3200	1600	260				
								3200					
l present in one				Bearings noisy-to be removed.				Bearing wear indicated..					

LIFE TEST FIXTURE PERFORMANCE SUMMARY 8-21-73

Unit Serial No. Test Date/Location Speed-RPM	1 4-21-71 Bendix 7900		1 4-21-71 Bendix 9100		2 1-8-71 MSFC 8340		3 4-21-71 Bendix 7840		3 4-21-71 Bendix 9100		4 1-8-71 MSFC 8070	
	Horizontal		Horizontal		Horizontal		Horizontal		Horizontal		Horizontal	
	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1	Brg. #2
Low Freq.Vib.G'sPeak Rotation Freq.Rad. 2XRotation-Rad.	0.022 –	0.045 –	0.037 – 0.05@800	0.045 0.116@800	0.02 –	0.03 0.02	0.050 –	0.033 –	0.070 –	0.021 –	0.035 –	0.060 0.045
Rotation Freq.Axial 2XRotation-Axial	0.005	0.005	0.016 0.046@520	0.016			0.03 – 0.12@1600	0.03 – 0.11@1630	0.02 –	0.025 –		
Intermed.Freq.Vib. 900 3100 Other G's@H_z					0.2 0.25@4300							
High Freq.Vib.G's 26KH_z 28 KH_z	0.03 0.05	0.035 0.045	0.02 0.02	0.04 0.02	0.2 0.3	0.2 0.2*	0.060 0.220	0.050 0.110	0.060 0.080	0.060 0.140	0.020 0.030	– 0.150
Sound Pressure Overall Load-dB Components-H_z (Highest first)	77-79	78-79	79	79 950	83 1530	82 7200 5200	83 3200	83 3200	84 2.8KH_z	82 5.5KH_z	74 130 1600	77
Service Meter Hrs.	21,123		21,123		19,640		20,488		20,488		14,680	
Comments					*Earlier reported as 0.7G-an error.		Outer race and ball fault frequencies present in fault detected signal.				160° Ambient	

LTF SUMMARY 8-21-73

Unit Serial No.	5		5		6		6	
Test Date/Location	4-21-71 Bendix		4-21-71 Bendix		4-21-71 Bendix		4-21-71 Bendix	
Speed RPM	7800		9030		10,000		9,000	
	Horizontal		Horizontal		Horizontal		Horizontal	
	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1	Brg. #2	Brg. #1	Brg. #2
Low Freq.Vib.G's Peak								
Rotation Freq.-Rad.	0.060	0.167	0.075	0.020	0.130	0.030	0.080	0.100
2X Rotation Rad.	-	-	-	-	0.040	-	0.060	-
Rotation-Axial	0.040	0.010	0.03	0.04	0.07	0.050	0.045	0.045
2X Rotation-Axial					-		-	-
	0.3@1600	0.15@1600	0.28@950			0.14@1935	0.38@1825	0.15@1825
Intermediate Freq.Vib.								
900			0.42	0.35				
3100								
Other G's@H_z								
High Freq.Vib.-G's								
$26KH_z$	0.070	0.060	0.060	0.060	0.30	0.30	0.20	0.50
$28KH_z$	0.060	0.090	0.070	0.140	0.60	0.58	0.26	0.70
Sound Pressure								
Overall-db	82	81	80	84	93	93	89.5	90
Components H_z	1600	1600	950	950	170	170	2000	150
(Highest First)							150	
Service Meter Hrs.	17,335				16,795			
Comments:	High oil flow unit.				Speed signal error. High Ambient Temp. 180°F with heat shield		180°F operation	