Technical Report Documentation Page

1. Report No. FRA/ORD-81/26	2. Government Accession No.	3. Recipient's Catalog No. PB83 110577
4. Title and Subtitle RAILCAR ROLLER BEARING FAILURE PROGRESSION TESTS		5. Report Date April 1982
		6. Performing Organization Code
		8. Performing Organization Report No. DOT-TSC-FRA-81-7 79-TR-48
7. Author(s) Warren D. Waldron		
9. Performing Organization Name and Address Shaker Research Center* Northway 10 Executive Park Ballston Lake NY 12019		10. Work Unit No. (TRAIS) RR028/R0306
		11. Contract or Grant No. DOT-TSC-1536
12. Sponsoring Agency Name and Address U.S. Department of Transportation Federal Railroad Administration Office of Research and Development Washington DC 20590		13. Type of Report and Period Covered Final Report March 1978 - March 1979
		14. Sponsoring Agency Code

15. Supplementary Notes

*Under contract to: U.S. Department of Transportation
Research and Special Programs Administration
Transportation Systems Center
Cambridge MA 02142

16. Abstract

This report describes the laboratory endurance test of six railcar roller bearings that had previously suffered physical damage or were otherwise degraded as a result of actual railroad service. Two different onboard impending bearing failure sensors were also physically evaluated. The objectives of the tests were to obtain a better understanding of the railcar roller bearing failure process(es), the manner in which bearing defects progress, and the effectiveness of the impending failure warning devices. A 150 hour test with 26,250 pounds radial load (equivalent full car load) at 528 rpm (equivalent to approximately 52 mph) was planned for each bearing. Only one bearing actually failed to complete the 140 hour test. All bearings exhibited further measurable degradation of defect progression during the course of the tests. Neither warning device actually gave warning of the one failure experienced.

17. Key Words		18. Distribution Statement
Bearing Failure	Race Cracks	DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161
Failure Progression	Water Etch	
Bearing Defects	Bent Cage	
Spall	Onboard Sensor	
Brinell		

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	130	

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

PREFACE

This study has been conducted for the Federal Railroad Administration through the Transportation Systems Center (TSC) in Cambridge, Massachusetts. Mr. Roger K. Steele was the initial technical monitor and Mr. James M. Morris also served as technical monitor for the program.

New York State derailment data was made available and explained by Mr. Wallace R. Klefbeck of the New York State Division of Traffic and Safety.

Data from several railroads were made available to us in this study. Acknowledgment of their help is made to Mr. R. F. Tuve of the Southern Railway System, Dr. P. E. Rhine of the Union Pacific and Mr. Dale Harrison of the Santa Fe. Southern Railway data formed the basis of the analysis conducted in Appendix D of this report. Mr. Tuve also was a great help in reviewing a draft of this report and making many suggestions for modifications. Acknowledgment of his help in reviewing this report does not imply that he, or Southern Railway necessarily agrees with the content and the conclusions and recommendations presented in this report.

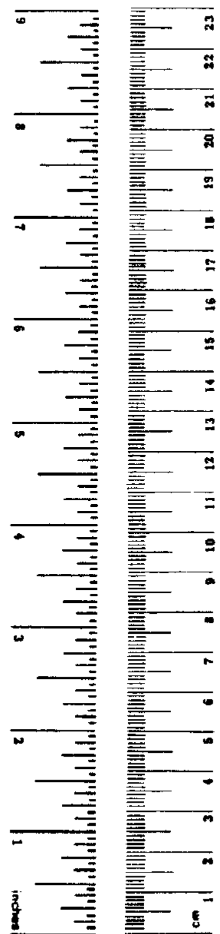
The Southern Railway System made their Coster Shop available for diagnostic testing of bearings while installed on wheel sets. Acknowledgment of their help is made to Messrs. Bible, Hayes, and Trollinger, whose tolerance of our intrusion allowed us to successfully complete the tests.

This study could not have been completed without the active cooperation of the staff of Brenco, Inc., who provided bearing defect data from three of their rework shops. Dr. Gerald Moyar, Vice President, Research and Development for Brenco, Inc. provided invaluable technical guidance.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

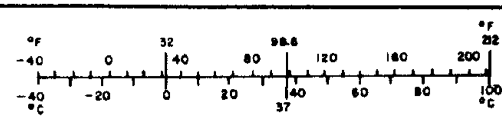


TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF TABLES

1. INTRODUCTION

The railroad freight car roller bearing has been a remarkably reliable mechanism permitting the railroads to operate heavier cars with much reduced likelihood of hot box failures. However, the number of derailments attributable to defective roller bearings has risen in recent years. Thus, roller bearings may be on the way to becoming one of the major vehicle-related causes of derailments. In spite of this, virtually nothing is known of how railroad roller bearings fail in actual service. Furthermore, because of the more rapid rate at which the incipient failure condition can progress to catastrophic failure in the roller bearing as opposed to the friction journal bearing, the need for on-board detection of impending failure has become more evident.

In order to achieve a better understanding of how bearings fail so that cost effective corrective measures can be taken, and in order to evaluate various designs for on-board sensors, the bearing failure progression tests in this report were undertaken in pursuit of the goals of improved railroad safety under sponsorship of the Federal Railroad Administration.

The subject test program had two specific objectives:

1) To better understand or define the manner in which railcar roller bearings fail so that rational measures for failure reduction can be outlined.

2) To "functionally" evaluate three on-board temperature sensing devices: namely, the DOT-STAR adapter temperature sensing device, an RF emitting axle temperature sensing device, and an ultrasonic emitting axle temperature sensing device.

To accomplish the first objective, a number of faulty railcar roller bearings that had been removed from revenue service were to be operated to failure or for a specified time period on a laboratory device which simulated the steady-state load and thermal environment of a fully loaded freight car. During the laboratory operational period, the test bearings' thermal, vibrational, and sound emission behavior were monitored in such a manner that the degradation process could be identified and documented.

During these "failure progression" tests, the available on-board impending failure alarm devices were to be installed on the test apparatus and their functional reliability in terms of false alarm incidence, operational capability, and ability to warn of actual impending failure were to be documented.

All test bearings were 6 x 11 tapered roller (rotating end cap) type and were loaded to 26,250 lbs (radial) -- the maximum design load for this size bearing. They were operated at 528 rpm (approximately 52 mph with a 33-inch wheel) for a minimum of 150 hours (equivalent to 7775 miles) or to failure. The bearings were force-convection-cooled using outside ambient air at 37 ft/sec.

The test bearings were obtained from the Brenco Bearing Service Company (BBS) at Knoxville, Tennessee. They were selected by Shaker Research Corporation personnel out of a lot of approximately 800 used bearings that were being subjected to routine rework procedures.

The following sections of this report describe the test bearing selection process, the test apparatus and test procedures, and the significant events that occurred during the test of each of the bearings. The effectiveness of the available "on-board" warning devices is also described. Conclusions and recommendations are presented in the final section.

2. SELECTION AND INSPECTION OF CANDIDATE TEST BEARINGS

Thirteen bearings out of approximately 800 used bearings had been tentatively selected as candidates for the failure progression tests. These were selected on the basis of a cursory inspection at the initial breakdown (disassembly) point in the Brenco Bearing Service Company Rework Facility in Knoxville, Tennessee. These thirteen bearings (less backing rings) were then shipped to Shaker Research's facilities where they were disassembled and inspected more carefully.

Because of the desire not to lose any significant quantity of grease or unduly disturb the grease and location of metallic debris within the grease, an extensive inspection of the candidate bearings was not possible at Shaker Research. For example, the inner race could not be inspected and only about 45° of the rollers could be viewed. The inspection procedure consisted of the following steps:

1) The seal and cone were removed from one end of the bearing.
2) The cone was visually inspected for cage damage and evidence of metallic debris in the grease between rollers and between the cage I.D. and cone O.D. If cone damage was noted, freedom of cage and roller rotation was checked by hand.
3) The cup race was wiped with a finger to reveal surface condition.
4) Defects of interest were logged and photographed.
5) The cone and seal were replaced and the process was repeated on the opposite side.
6) A sample of grease was taken from the seal case on the side judged to be in poorer overall condition.

For those bearings finally selected for test, three more inspection steps were performed prior to assembly on the test shaft. These were:

1) Measurement of cone bore size.
2) Visual inspection of cone bores.
3) Measurement of lateral clearance.

Table I summarizes the condition of the thirteen candidate test bearings, indicates which ones were selected as test candidates and indicates the order in which they were tested. Table II summarizes the types of defects contained in each bearing and ranks the worst six based upon the number of different defect types contained in each bearing.

However, "number of defect types" alone is an insufficient parameter for test bearing selection, since all defects will not lead to failure unless something else happens. For example, a cracked cup in itself is not failure producing unless the crack propagates to the point that a piece breaks out of the bearing causing a more drastic event: i.e., loss of lubricant or jamming.

Figure 1 is a likely laboratory test failure progression scenario of any of the thirteen candidate bearings. One must now attempt to select the most likely bearings to fail and/or select bearings that have defects of particular interest for defect enlargement rate/mode study.

Six of the bearings were judged "most likely to fail" by a defect type weighting process with rating factors from 0 to 5. The weighting logic is shown in Table III.

The results of this rating process are shown in Table 4. The results of all ratings shown in Tables 2 and 4 are shown in Table 5. Of the thirteen candidates, the bearings shown in Table 5 were the six "worst" by any of the four measures used, and bearing Number 6 was consistently the "leader." For lack of a better criterion, the weighted order shown in the last column of Table 5 was used as the selected test sequence order with one exception. That exception was that bearing Number 8 was selected as the first test bearing since it was not exptected to "fail." Thus, experience could be gained with the test apparatus and the data acquisition system using a bearing with the most benign defects.

TABLE 1

SUMMARY OF FAILURE PROGRESSION TEST CANDIDATE BEARING DEFECTS

Bearing Number	Summary of Defects	Test Candidate	Test Sequence
FP-1	Broken cup at C'bore	No	-
FP-2	Water damage, very black grease	No	-
FP-3	Water damage, slimy black grease	No	-
FP-4	Broken cup at C'bore	No	-
FP-5	Spalled cup 360° both races	No	-
FP-6	Broken cup at C'bore both ends, seal case badly dented, both cages dented, brinelled adjacent to breaks, chunks of cup jammed in cage, debris between rollers, one roller frozen	Yes	2
FP-7	Low on grease, dried between rollers, chips between rollers, both seals loose in C'bore, suspect sides, water damage one side	Yes	3
FP-8	Rusty appearing grease, caked between rollers, metal chips between rollers, moderately spalled both sides, water damage one side	Yes	1
FP-9	Particle indentation, lightly spalled	No	-
FP-10	Broken cup at C'bore one side into race area, metal chips between rollers, cup cracked thru race area other side	Yes	4
FP-11	Similar to FP-10 except heavy brinell at cup crack	Yes	5
FP-12	Water damage on one side, broken cup at C'bore, other side with metal particle in cage area, grease caked between rollers	Yes	6
FP-13	Several brinells and one large spall one side	No	-

Note: Inspection logs for all candidate test bearings are included in Appendix A.

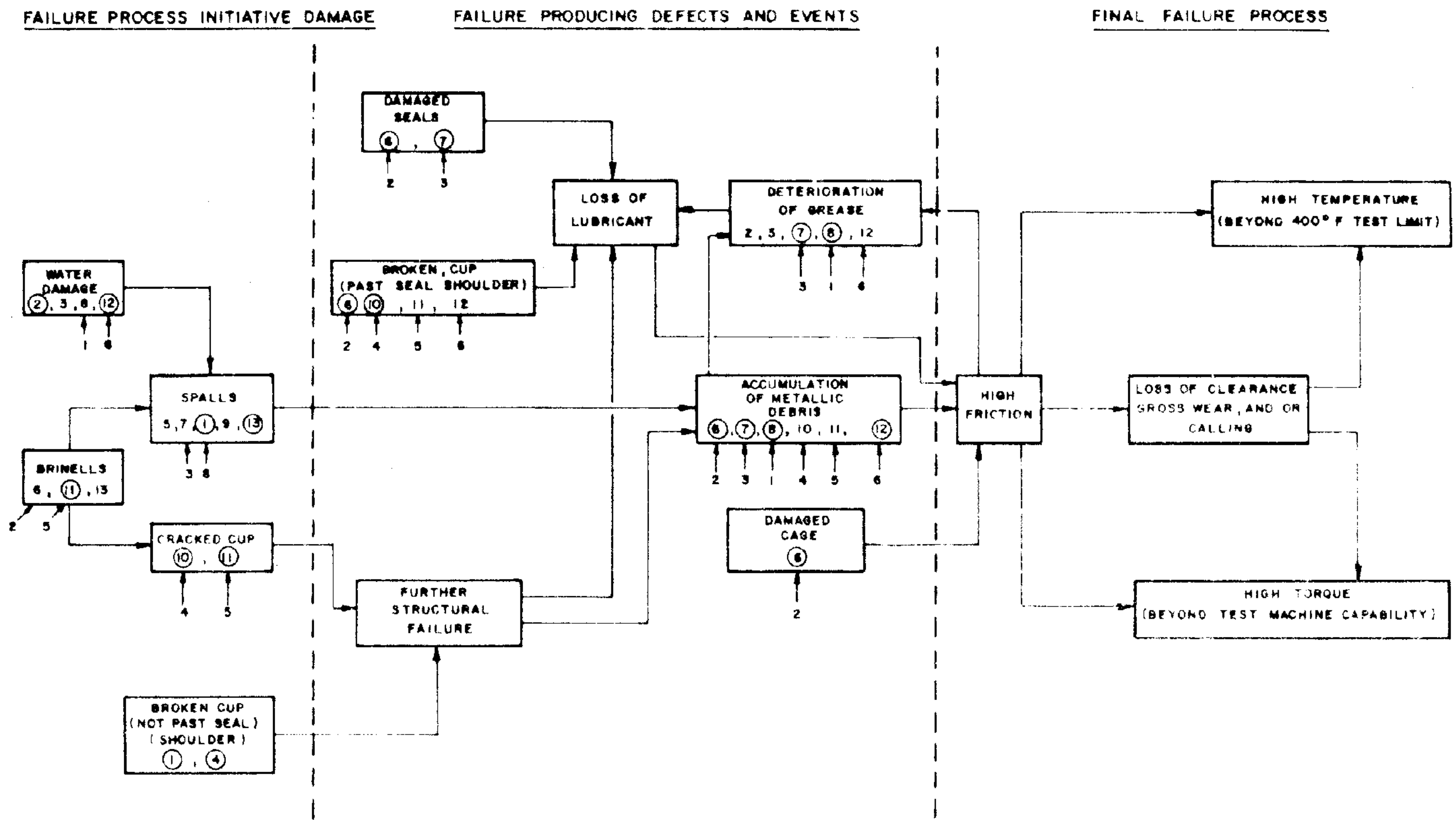
TABLE 2

SUMMARY OF DEFECT TYPES
CONTAINED IN EACH CANDIDATE TEST BEARING

Defect Type	Candidate Bearing Number 1	2	3	4	5	6	7	8	9	10	11	12	13
Lubricant Depletion Causing Defects						Ⓧ	Ⓧ						
Damaged seals						Ⓧ	X						
Broken cups (past seal shoulder)						X				Ⓧ	X	X	
High Friction Causing Defects													
Particles in grease						Ⓧ	Ⓧ	Ⓧ		X	X	Ⓧ	
Damaged cages						Ⓧ							
Poor quality grease		X	X				Ⓧ	Ⓧ				X	
Structural or Surface Damage Type Defects													
Spalls					X		X	Ⓧ	X				Ⓧ
Brinells						X					Ⓧ		X
Water damage		X	X					X				Ⓧ	
Cracked cups										Ⓧ	Ⓧ		
Broken cups (not past seal shoulder)	Ⓧ			X									
Particle Indentation									X				
TOTAL Number of Defect Types	1	2	2	1	1	5	4	4	2	3	4	4	2
Ranking by Number of Defect Types						1	2	3		6	4	5	
TOTAL Number of Severe Defect Types	①	①				④	③	③		②	②	②	①
Ranking by Number of Severe Defect Types						1	2	3		4	5	6	

X indicate presence of indicated defect type

Ⓧ indicate that the defect type is judged to be severe relative to the rest of the bearings



NOTE: Candidate test bearings containing referenced defects are indicated by bearing number. Circled bearing numbers indicate bearings most highly distressed relative to the rest of the candidate bearings. Bearings selected for testing are indicated by an arrow. Numbers adjacent to arrows indicate test sequence number.

FIGURE 1. FAILURE PROGRESSION SCENARIO FOR CANDIDATE TEST BEARINGS

TABLE 3

WEIGHTING LOGIC

Rating Factor	Defect Type
5	Those immediately causing loss of grease; i.e., cups with breaks into race area, damaged or loose seals
4	Severe structural damage leading to high heat generation or possible secondary failures; i.e., damaged cages, cup cracks in race region
3	Severe structural damage outside of roll track; i.e., broken cups in seal C'bore, and deteriorated and contaminated grease
2	Surface structural defects; i.e., spalls, brinells
1	Surface nonstructural defects; i.e., water etch
0	Surface blemishes; i.e., particle indentation

TABLE 4

SUMMARY OF WEIGHTED DEFECT TYPES

CONTAINED IN EACH CANDIDATE TEST BEARING

Defect Type	Defect Severity Rating												
	1	2	3	4	5	6	7	8	9	10	11	12	13
Lubricant Depletion Causing Defects													
Damaged Seals						5	5						
Broken Cups (past seal shoulder)						5				5	5	5	
High Friction Causing Defects													
Particles in Grease						3	3	3		3	3	3	
Damaged Cages						4							
Poor Quality Grease		3	3				3	3				3	
Structural Damage Defects													
Spalls					2		2	2	2				2
Brinells						2					2		2
Water Damage		1	1					1				1	
Cracked Cups										4	4		
Broken Cups (not seal shoulder)	3			3									
Particle Indentation									0				
TOTAL SCORE	3	4	4	3	2	19	13	9	2	12	14	12	4
Rank						1	3	6		4	2	5	
TOTAL SCORE (considering only severe defects)	3	1				17	11	8		9	6	4	2
Rank						1	2	3		3	5	4	

Numbers indicate rating factor (see Table III)
Circled numbers indicate that defect is judged to be severe relative to the rest of the bearings.

TABLE 5

CANDIDATE BEARING RATING SUMMARY

Ranked By:

Bearing	Defect Types	Number of "severe" Defect Types	Weighted Defect Types	Weighted "severe" Defect Types
6	1	1	1	1
7	2	2	3	2
8	3	3	6	4
10	5	6	4	3
11	4	4	2	5
12	6	5	5	6

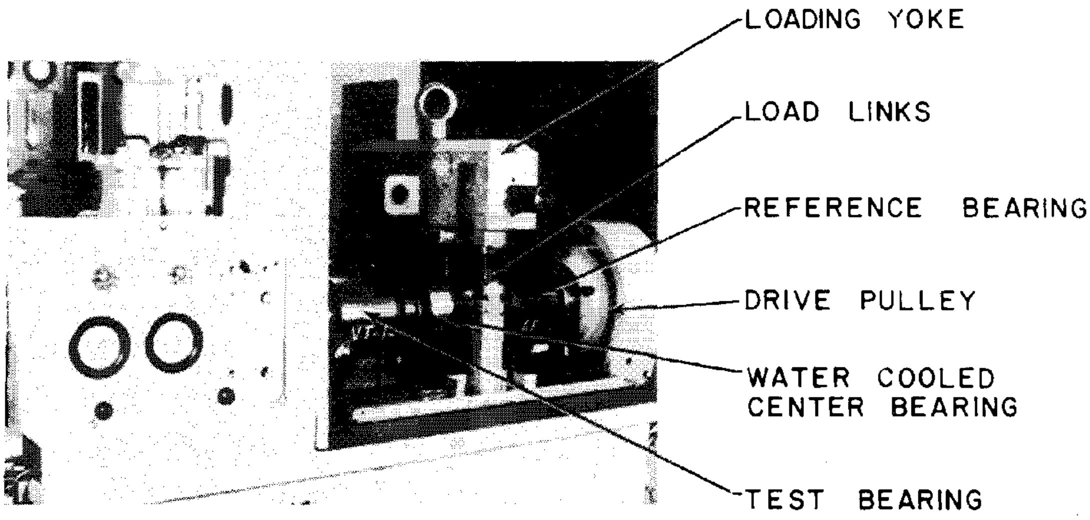
3. TEST APPARATUS

3.1 ROLLER BEARING TESTER

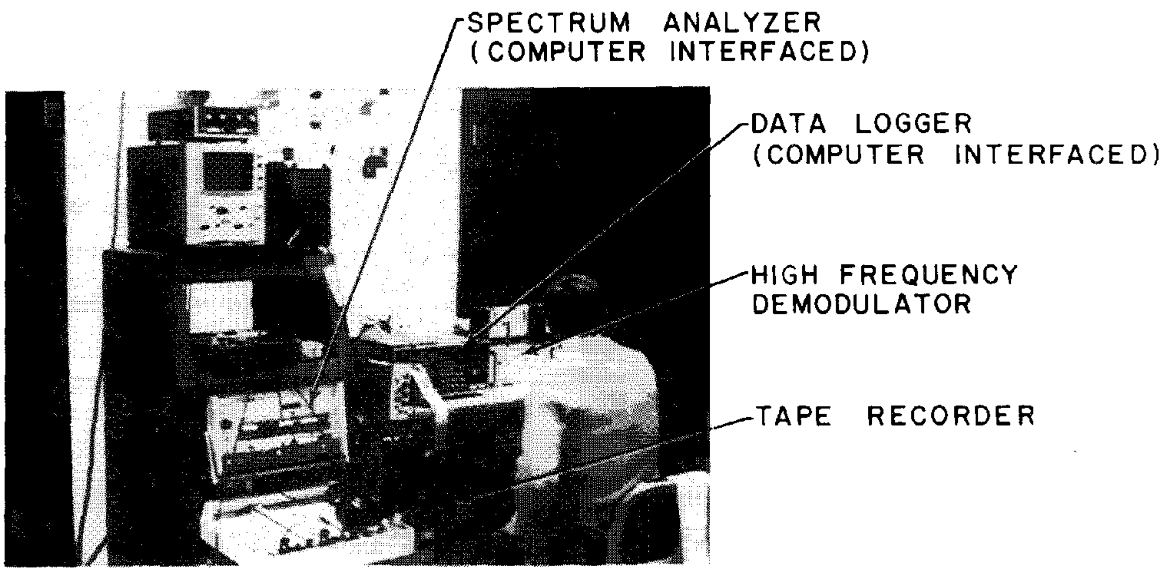
The test rig used to conduct the subject tests is shown in Figure 2. The test rig shaft is driven by a 25 hp electric motor and holds three standard 6 x 11 tapered railcar roller bearings with a standard end cap secured to one end. On the other end is a five V-belt drive pulley. The outboard or support bearings are attached to the test rig base or frame by means of standard railcar bearing adapters and mounting fixtures which closely resemble the truck side frame arrangement. Pure radial and moment loading is imposed upon the center bearing by means of suitably arranged hydraulic pistons and linkages, again through a standard railcar bearing adapter. The outboard bearings are directly air-cooled using outside ambient air forced through a shrouding arrangement in the "as built" configuration. The center or test bearing is cooled by conduction to the outboard bearings, by conduction through the load adapter to the massive loading yoke, and by convection to the air which flows axially from the outboard bearing air shrouds.

For the testing required by the subject program, it was desirable that the support bearing opposite the drive end, instead of the center, be employed as the test bearing. That end of the test shaft exactly duplicates the railcar axle complete with end cap and standard bolts. Thus, it is available for the exact duplication of the installation of the shaft-mounted alarm devices (RF and high frequency sound emitting bolts). Furthermore, the adapter mounting arrangement permits exact duplication of the installation of the NSWC alarm device. The drive end support bearing was a good (undamaged) bearing and was considered as a baseline bearing for making judgments regarding thermal behavior of the test bearing.

In terms of thermal simulation, this interchanging of test and support bearing positions presents a problem. First, from a steady-state point of view, it imposes an undesirable axial thermal gradient upon the test bearing due to the heat input from the center, highly-loaded, bearing. Also, without abnormally high air flows directed on the test bearing, it runs abnormally hot -- again due to center bearing heat input. Secondly, from a thermal inertia point of view, the heat transfer path down the axle is blocked because of both



ROLLER BEARING TESTER



DATA SYSTEM

FIGURE 2. ROLLER BEARINC TEST APPARATUS

the high temperature center bearing and the absence of a thermal mass as large as a railcar wheel.

These problems were alleviated, if not completely eliminated, by providing an auxiliary method of cooling the center bearing. The center bearing cooling scheme consisted of the following major components:

1) A water-cooled heat exchanger around the center bearing through which the load was applied (shown in Figure 2).

2) A reservoir containing a volume of water (approximately 10 gallons) to simulate the heat capacity of a railcar wheel.

3) An air-to-water heat exchanger including variable speed fan and means of regulating water flow.

By properly adjusting the water flow and temperature to the cooling jacket, axial thermal gradients between the two ends of the test bearing, and between the test bearing and the center loader bearing could be consistently held to less than 10°F, which was considered more than adequate for the purposes of the test program. Once the cooling parameters (cooling jacket and heat exchanger bypass flow) were established using a new (undamaged) bearing, they were not changed for the duration of the test program.

3.2 DATA ACQUISITION

To satisfy the program objectives, a quantitative time history of the events leading to failure and the failure process needed to be recorded. The basic questions are:

1) What are the important quantitative parameters?

2) What are the best methods of recording these parameters?

Temperature is obviously a key parameter because of its traditional symptomatic character and because it is the parametric base of the alarm devices of interest. Unfortunately, temperature, like input power, provides only a gross indication of the failure process. It provides little detailed information regarding the physical interaction between the components within the bearing assembly.

Vibration is a frequently used measurement in bearing testing as it quantifies the impacting between mating parts. By using frequency vibration and demodulation techniques, one can postulate the relative damage to the primary components (inner race, outer race, cage, rolling elements). By studying how the vibration signature (frequency and amplitude content) changes with time and temperature, one can deduce many of the internal bearing motion and failure mechanisms including the details of the events leading to ultimate failure. High frequency sound has the potential of being incorporated into a wayside diagnostic scheme or other bearing inspection location if high frequency vibration should prove to be an accurate precursor of failure, and if adequate acoustical coupling can be accomplished. A record of high frequency sound data during the subject tests was an aid in determining the relationship between sound and vibration signatures and the degree of acoustic coupling available.

Although measurement of temperature, vibration and sound is relatively straightforward, recording the data in a meaningful and easy-to-interpret manner must be approached with significant forethought. With a degraded bearing, all parameters of interest are constantly changing with time. Thus, manual recording is inappropriate. Since the tests are likely to last many hours before culminating in failure, relatively slow speed strip chart records are highly desirable so as to discern long term trends without consuming excessive space (paper). On the other hand, the actual failure process is relatively rapid and one would want to see this process in detail. During the failure process a strip chart recorder is not appropriate because of both its slow speed and the desire to have real time vibration data over a wide frequency spectrum. During this period, only magnetic tape recording is truly adequate. Magnetic tape recording is not desirable, however, to record the entire test duration because of economic and data reduction considerations. For these reasons, a system employing continuous strip chart recording, continuous computer logging, and event-triggered magnetic tape recording was utilized.

A summarized schematic of the data acquisition scheme is shown in Figure 3. It consists of essentially four levels of data collection and logging:

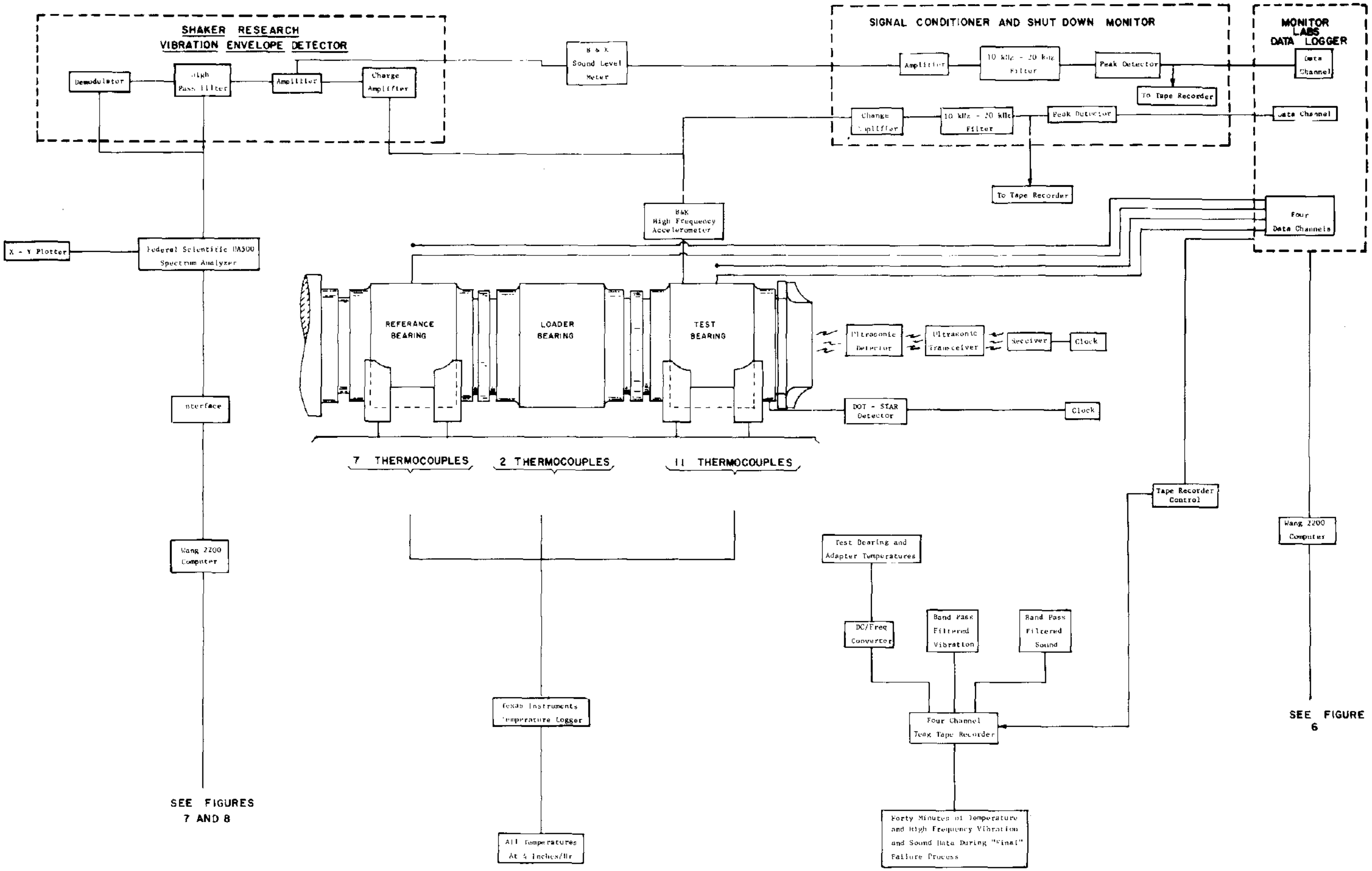


FIGURE 3. DATA SYSTEM SCHEMATIC

1) Continuous logging of all temperatures on a slow speed chart recorder.

2) Continuous sampling of selected temperatures and overall (peak) sound and vibration levels. These parameters were computer logged every 30 minutes, and at any time when one of the measured parameters changed by more than 5% since the last recorded condition.

3) Continuous magnetic tape recording (for a duration of 8 minutes) of selected temperatures and high frequency vibration and sound triggered by temperature exceeding a predetermined level.

4) Continuous sampling of several test rig safety and maintenance parameters which were used for automatic test rig shutdown

The following subsections describe how each of the parameters are monitored and logged.

3.2.1 Continuously Logged Temperatures

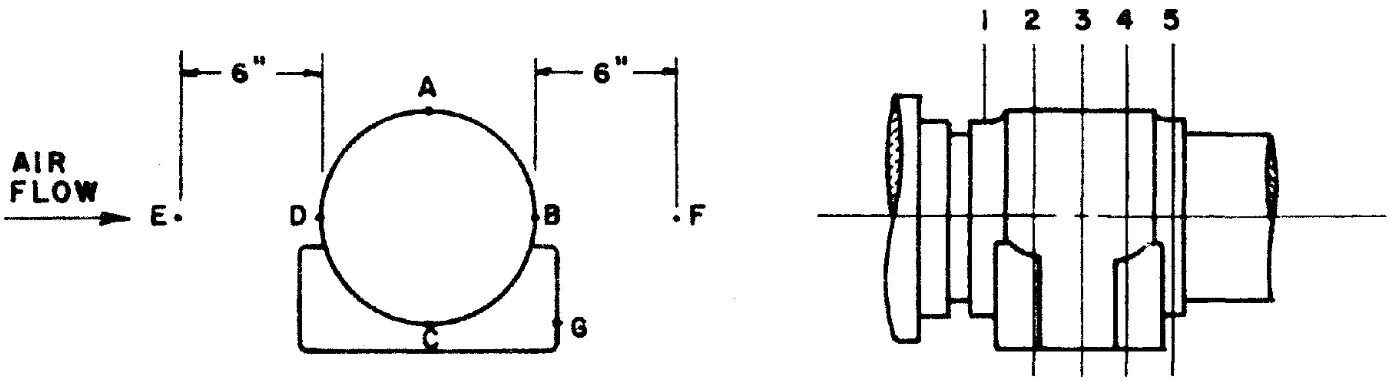
Figure 4 shows the location of the 24 thermocouples which were continuously logged. A temperature switch in the logger itself is set at 390°F and actuates the automatic shutdown system if any of the logged temperatures reach that value.

3.2.2 Selected Temperature Data

Figure 3 shows the schematic for logging and recording the test bearing race, adapter, and ambient temperatures and reference support bearing race and ambient temperatures. These temperatures were sampled approximately every 10 seconds by the Monitor Labs data logger and were also sampled every 30 seconds by the Wang computer and logged* (tabulated) either every 30 minutes or at any time that any computer logged parameter (temperature, vibration, sound) changed in value by 5%, or more compared to the last recorded condition. Temperatures

---

*Section 3.3 describes the computer logging format.



Location	Outboard Test Bearing	Center Loader Bearing	Drive End Support Bearing
1B	1	12	14
2B	2	--	15
3B	3	--	16
4B	4	--	17
5B	5	13	18
3A	6	--	--
3C	7	--	--
3D	8	--	--
3E	9	--	19
3F	10	--	20
2G	11	--	--

- 21 Drive Motor
- 22 Cooling H_2O In
- 23 Cooling H_2O Out
- 24 Fan Motor

Texas Instruments Model Temperature Recorder 0.25 in/hr chart speed

Set 390°F

Automatic Shutdown Cause Indicator

A, B, C, D located on cup surface
E, F air temperature
G adapter temperature at DOT/STAR sensor head

FIGURE 4. THERMOCOUPLE LOCATIONS

3 and 11 were also to be automatically recorded on magnetic tape at the time that T3 reached 300°F, 340°F, 360°F, and 380°F. The tape recorder was to be allowed to operate for eight minutes after any of the aforementioned events. In other words, when T3 reached 300°F, the recorder would start and collect temperature, high frequency sound, and high frequency vibration data for eight minutes, and then stop (if the temperature had not reached 340°F in that eight-minute period). Upon reaching 340°F the tape recorder restarts and the process repeats until all temperature levels have been traversed. After the last eight-minute increment (at which point the tape will have been expended) the test rig automatically shuts down.

3.2.3 Vibration and Sound Data

Figure 3 also shows the schematic for logging and recording vibration and sound data. Peak sound and vibration was sampled and logged in a manner identical to temperature as described in 3.2.2. High frequency sound and vibration (to 20 KHz) also could be recorded on magnetic tape during tape recorder operating periods (as described in 3.2.2).

High frequency and demodulated spectral data were manually acquired and computer tabulated as described in Section 3.3.

3.2.4 Automatic Shutdown

Figure 5 shows the schematic for automatic test rig shutdown. The "automatic shutdown cause indicator" indicated which parameter was the first to exceed its shutdown limits.

3.2.5 "Impending Failure Sensor" Monitoring

Figure 3 shows the schematic for monitoring the "impending failure alarm" devices. These devices were not used to shut down the test but instead were "hooked" to clocks which stop upon signal from the device. A correlation of the stop time and recorded events of the time were used to determine the effectiveness of the device.

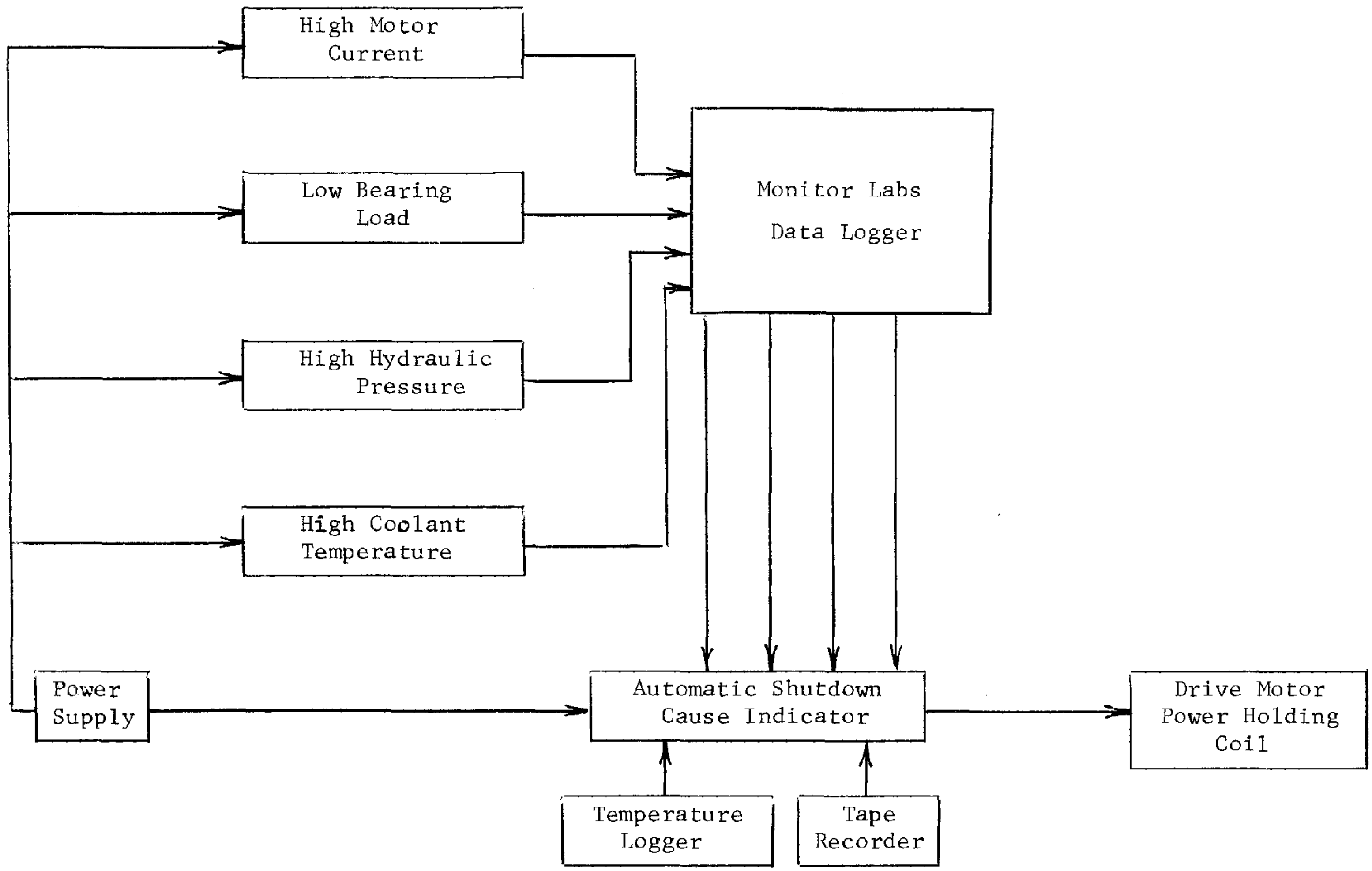


FIGURE 5. AUTOMATIC SHUTDOWN SCHEMATIC

3.3 COMPUTER LOGGED DATA FORMAT

The format for computer logged temperature, vibration, and sound data is shown in Figure 6. The "parameter record cause" column indicates the reason or cause for the data line. A normal 30 minute record is indicated by the letter T and other letters indicate a change of more than 5% in a particular variable since the last record event. The meanings of the different designations in this column are as follows:

V - At least 5% change in vibration parameter

DTT - At least 5% change in Del T for test bearing

DTR - At least 5% change in Del T for reference bearing

S - At least 5% change in sound parameter

The format for computer calculated and logged demodulated frequency/ amplitude data is shown in Figure 7 and 8.

Failure Progression Test
Data Summary
Bearing No. 12

	Test Bearing					Reference Bearing		
Time	Race T3	Del T T3-T9	Adapt T11	AVG Peak Vib	AVG Peak Sound	Race T16	Del T T16-T19	Parameter Record Cause
%DAY.HR.MN	Deg F	Deg F	Deg F	g's	d/cm↑2	Deg F	Deg F	Sym
071:16:18	101	61	86	31.000	4.726	112	67	T
071:16:23	102	64	87	23.702	4.984	113	68	V
071:16:38	105	67	90	21.458	5.360	113	67	DTT
071:16:49	109	71	91	18.914	4.923	112	68	DTT
071:17:08	108	68	93	22.986	4.793	112	68	V
071:17:15	108	70	93	18.782	3.811	112	67	S
071:17:19	109	71	93	16.432	3.668	115	70	DTR
071:17:27	110	72	94	19.806	3.804	115	70	V
071:17:39	108	70	93	19.580	4.575	113	68	S
071:18:09	105	68	91	18.562	4.451	113	69	T
071:18:20	104	66	90	19.532	5.113	116	73	DTR
071:18:31	106	68	90	17.982	5.134	115	69	DTR
071:19:01	107	69	91	18.096	4.739	110	66	T
071:19:15	107	69	91	17.538	4.945	109	62	DTR
071:19:23	106	69	92	18.562	4.968	111	65	DTR
071:19:41	104	66	91	19.544	4.771	112	69	DTR
071:20:11	102	64	88	18.770	4.826	114	69	T
071:20:41	103	65	88	18.972	4.949	114	67	T
071:20:52	103	66	89	18.588	5.109	115	71	DTR
071:21:12	105	68	90	18.226	4.901	112	67	DTR
071:21:31	103	66	89	19.610	5.627	109	63	DTR

FIGURE 6. TYPICAL COMPUTER LOGGED PARAMETRIC DATA

Failure Progression Test
Trend of Characteristic Defect Frequencies
Bearing Number 10

				Demodulated amplitudes (g's)				
				Frequencies				
				4.0	8.8	88.0	96.0	116.0
Rec No	Cum Hrs	Pk-Amp	Pk-Freq	Cage	Rot	Roll	OR	IR
360	0.5	12.5	8.28	0.04	0.12	0.04	0.10	0.02
361	2.6	20.0	12.40	0.12	0.14	0.08	0.41	0.07
362	14.4	20.0	12.08	0.12	0.18	0.06	0.19	0.03
363	30.1	15.0	12.36	0.08	0.21	0.04	0.11	0.03
364	31.5	20.0	13.20	0.38	0.34	0.08	0.52	0.07
365	47.0	15.0	12.56	0.09	0.16	0.06	0.18	0.05
366	48.8	25.0	13.32	0.28	0.74	0.14	0.69	0.07
367	65.3	35.0	13.44	0.52	1.32	0.25	1.18	0.09
368	72.8	35.0	15.48	0.56	0.68	0.19	0.75	0.07
369	90.4	23.0	13.04	0.27	0.44	0.09	0.14	0.06
370	98.3	20.0	12.52	0.15	0.32	0.06	0.24	0.06
371	112.1	25.0	12.56	0.11	0.43	0.07	0.31	0.07
372	113.9	25.0	13.24	0.38	0.61	0.10	0.31	0.06
373	129.7	35.0	13.44	0.63	1.19	0.14	0.87	0.11
374	131.9	32.0	13.52	0.27	1.09	0.16	0.96	0.09
375	148.5	40.0	13.28	0.67	1.16	0.16	1.00	0.07

FIGURE 7. TYPICAL COMPUTER LOGGED VIBRATION DATA

Failure Progression Test
Trend of Characteristic Defect Frequencies
Bearing Number 10

				Demodulated amplitudes (MD/cm↑2)				
				Frequencies				
				4.0	8.8	88.0	96.0	116.0
Rec No	Cum Hrs	Pk-Amp	Pk-Freq	Cage	Rot	Roll	OR	IR
401	0.5	1.2	11.76	15.78	22.27	6.11	22.82	5.87
402	2.6	1.4	12.44	23.38	14.33	7.32	22.05	4.82
403	14.4	1.5	12.08	12.05	16.61	5.30	10.07	3.12
404	30.1	1.5	12.28	10.08	17.74	7.58	6.21	4.13
405	31.5	1.5	13.24	15.89	23.27	14.60	15.04	6.14
406	47.0	1.5	12.00	10.07	9.98	14.14	9.80	4.09
407	48.8	2.2	14.52	15.59	49.26	10.71	31.07	4.15
408	65.3	5.6	13.44	64.61	162.39	19.63	102.71	14.50
409	72.8	3.1	13.48	46.47	60.56	12.91	41.67	8.98
410	90.4	1.9	13.52	11.93	30.79	8.15	7.99	5.25
411	98.3	1.9	11.92	11.97	29.69	13.01	15.68	7.34
412	112.1	2.2	12.72	18.91	47.90	12.62	16.09	6.70
413	113.9	2.2	12.92	38.81	65.23	13.44	37.17	11.21
414	129.7	4.1	13.40	26.80	64.52	16.91	73.68	10.80
415	131.9	3.5	13.00	27.10	76.37	11.66	55.55	7.99
416	148.5	3.6	13.36	75.21	84.79	13.55	60.98	9.76

FIGURE 8. TYPICAL COMPUTER LOGGED SOUND DATA

4. SUMMARY OF TEST RESULTS

The condition of each of the selected test bearings prior to failure progression was summarized in Section 2.0 (Table 1).[1] Each bearing was subjected to the same load (26,250 pounds) and speed (528 rpm) conditions. Cooling air velocity (37 ft/sec) was also the same for each. Cooling air temperature was, however, dependent upon the outside air temperature which varied between approximately -10°F and +50°F. The bearings were generally operated from approximately 3:30 p.m. to 8:30 a.m. of the following day except on weekends when they were operated continuously from 3:30 p.m. on Friday to 8:30 a.m. the following Monday, if possible. During the off period, the radial load of 26,250 pounds was maintained.

Only bearing number FP-6 failed to run the full 150 hours. It exceeded the maximum outer race temperature limit of 390°F after 69.5 hours. Although the other five test bearings (FP-8, 7, 10, 11, and 12) completed the 150-hour test, each showed signs of further degradation and progression of prior noted defects.

The principal changes in bearing condition noted in each bearing as a result of the failure progression tests are summarized in Table VI. Detailed descriptions of conditions are given in the inspection logs included in Appendix A and in the following section.

4.1 DISCUSSION OF CHANGES IN BEARING CONDITION

The changes in test bearing condition and an overall description of parametric behavior for each bearing are described in the following subsections in chronological order. Comparison of significant parametric data for each bearing will be discussed in Section 4.2.

4.1.1 Bearing FP-8

The pertinent information concerning bearing FP-8 is summarized in Figure 9. Significant condition photographs are shown in Figures 10 through 13. This bearing was the least damaged of the six bearings. It originally possessed a number of small and moderate outer race spalls, had metallic particles in the grease and one side was water etched.

TABLE 6

SUMMARY OF BEARING CONDITION AFTER TEST

Test Bearing	Hours Tested	Principal Changes in Condition Noted After Test	Typical or Significant Photograph Figure Numbers
FP-8	150.1	One new spall	11
		All spalls slightly larger in size	10
		Ten of eleven spalls had peculiar appearing adjacent surface areas	10, 12
		Grease and water-etched areas lost their rusty appearance	13
		Increased fragment indentation	12, 13
FP-6	69.5	Considerable grease leakage at outboard end; moderate grease leakage at inboard end	16
		Grease caked and hard in seal case	17
		Grease dry and burnt with roller ends blue from heat (outboard end)	18
		Five bent separators had rubbed on outer race	19, 20
		Cage worn at seal edge two places 120^{o} away from separator wear from contact with bent seal	22
		Approximately 180^{o} of cage is rolled down at seal end in area of separator wear	20
		Seal on inboard end had rotated 90^{o} and jammed in place	24
		Grease on inboard end shows no significant evidence of overheating	23
FP-7	150.5	Metal chips between rollers finer in size	25
		Small circumferential scratches on rollers	25
		Several spalled rollers on outboard end	25
		Brinell-like marks on outboard cup	26
		Some deep particle marks	26
		Rollers on inboard cone have brownish discoloration	27
		Inboard cup has mottled appearing race with smooth "track" near seal C' bore	27 28
		Seal lip very worn (but smooth) seal case O.D. smooth and sharp edged from rotation	28
		Both inner races spalled	-
FP-10	150.0	Crack on outside of inboard end propagated and chipped out, three new cracks visible	32
		Large spall in cracked area with stringer-like cracks	33
		Fragment indentation CCW from spall (trailing edge side)	33
		Race 180^{o} from spall/crack shows roller imprint but no discernible indentation	34

TABLE 6 (cont.)

Test Bearing	Hours Tested	Principal Changes in Condition Noted After Test	Typical or Significant Photograph Figure Numbers
FP-10 (cont.)	150.0	Rollers have deep circumferential scratches	34
		Particle indentation in area adjacent to cup break on outboard side	35
		Cone on outboard end seems more depleted of grease	36
		Circumferential scratches on rollers corresponding to broken area in cup	36
FP-11	151.6	No discernible difference in broken cup (outboard end)	38
		Crack propagated toward center of bearing and second crack became visible	39
		Brinell appears deeper with additional small cracks near center of bearing	40
		Small chip broken out of inner edge at brinell	40
FP-12	156.7	Chip at edge of break increased in size	42
		Wear track on small diameter of cup	42
LB-1	676.8	Grease in seal case that interfaces cone rusty in appearance -- like new under surfaces	43
		Metallic particles in grease between rollers on drive end giving a glittered appearance	43
		Rollers on drive end show severe fragment indentation	43
		Very wide spall in load zone of drive end cup	44
		Fragment indentation adjacent to trailing edge of spall	44
		Outer race 180° from spall shows roller imprints with no discernible indentation	44
		Slight wear tracks on rollers and outer race on test bearing end	45
LB-2	308.3	Grease in seal case that interfaces cone rusty in appearance -- like new under surface	46
		Metallic particles in grease between rollers on test bearing end	46
		Rollers on test bearing end show same fragment indentation and circumferential scratches	46
		Wide spall in load zone of test bearing end outer face	47
		Fragment indentation on trailing edge of spall	47

TABLE 6 (cont.)

Test Bearing	Hours Tested	Principal Condition Noted After Test	Typical or Significant Photograph Figure Numbers
LB-2 (cont.)	308.3	Metallic particles in grease between rollers on drive end	48
		Drive end rollers show slight circumfer- scratches and rust-like spots and stains	48
		Drive end cup has moderate spall with rust-like staining adjacent to trailing edge of spall	48

SUMMARY OF INITIAL CONDITION

- Several small and moderate outer race spalls
- Metallic debris in grease
- Water-etched

SUMMARY OF EFFECT OF ADDITIONAL RUNNING

- Slight spall size growth -- one new spall
- Increase in degree of fragment indentation
- Less rusty looking in general

QUALITATIVE EVALUATION OF DEGRADATION

· Temperature rise over ambient:	45 - 55^{o}F
· Vibration level:	10 - 20 g's
· Sound pressure level:	102 DYNE/cm^2
· Demodulated 96 HZ amplitude (O.R.)	0.1 - 0.5 g's 10 - 80 md/cm^2

GENERAL DESCRIPTION OF DATA

- Temperature mildly erratic during about half of test
- One temperature excursion starting at 93 hours and lasting for $\approx$ 5 hours peaking at 68^{o}F ΔT
- One large sound excursion starting at $\approx$ 130 hours and lasting for $\approx$ 6 hours peaking at 3 DYNE/cm^2
- Sound and vibration levels during last two runs 2 x level of previous runs

FIGURE 9. BEARING FP-8, SUMMARY OF PERTINENT INFORMATION

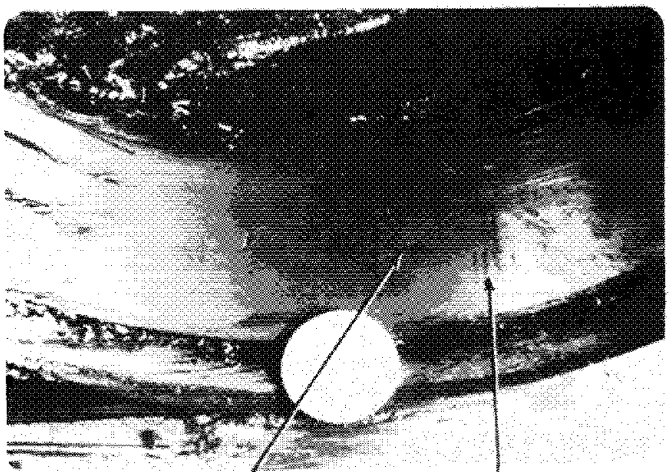
During the course of the 150-hour test, some of the spalls increased slightly in size (Figure 10) and one new spall developed (Figure 11). The area around some of the spalls developed a brinell-like appearance or an appearance indicating that the subsurface was severely damaged and that large pieces were soon to break out (Figures 10 and 12). These areas may have been originally slightly brinelled and the polishing action of the additional running may have simply made the slight brinells more evident or significant. Additional subsurface fatigue may have occurred which gave the area around the spalls their "disturbed appearance."

In addition to the changes in the spalled condition, bearing FP-8 experienced rather significant fragment indentation during the test (Figures 12 and 13). Polishing or wearing action also removed the surface rust from the heavily water-etched areas (Figure 13).

4.1.2 Bearing FP-6

The pertinent information concerning bearing FP-6 is summarized in Figure 14. This was the most severely damaged of all six test bearings in that it not only had a broken cup but its seals and one cage were also badly dented. During the course of the test, its thermal behavior was both erratic and high. Sound and vibration signatures at outer race frequency as well as thermal behavior showed evidence of binding. During the last few hours of testing (before failure at 69.5 hours) very large vibration and sound amplitudes at cage frequency were also noted.

The cup temperatures as recorded on the multipoint chart recorder just before "failure" (390°F shutdown point) are shown in Figure 15. Unfortunately, the thermocouple at the center of the bearing was the one being computer-monitored, and it reached the 300°F tape recorder trip point only a few seconds before the highest temperature reached the rig shutdown temperature of 390°F. Thus, no meaningful magnetic tape recorded data was produced during the final failure process.



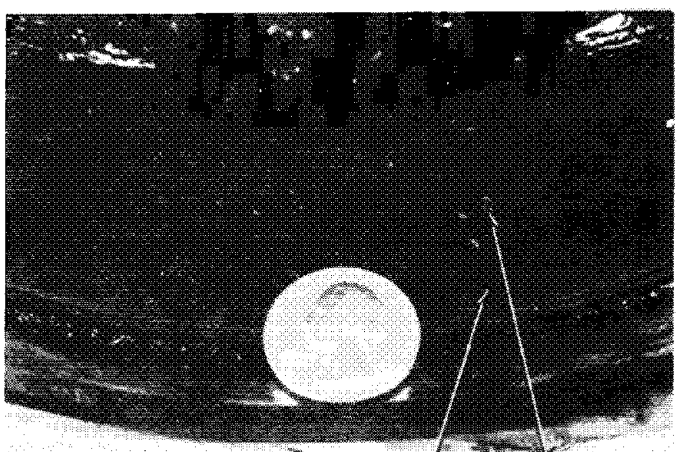


FIGURE 10. BEARING FP-8 SHOWING INCREASED SPALL SIZE AND MEASUREMENT

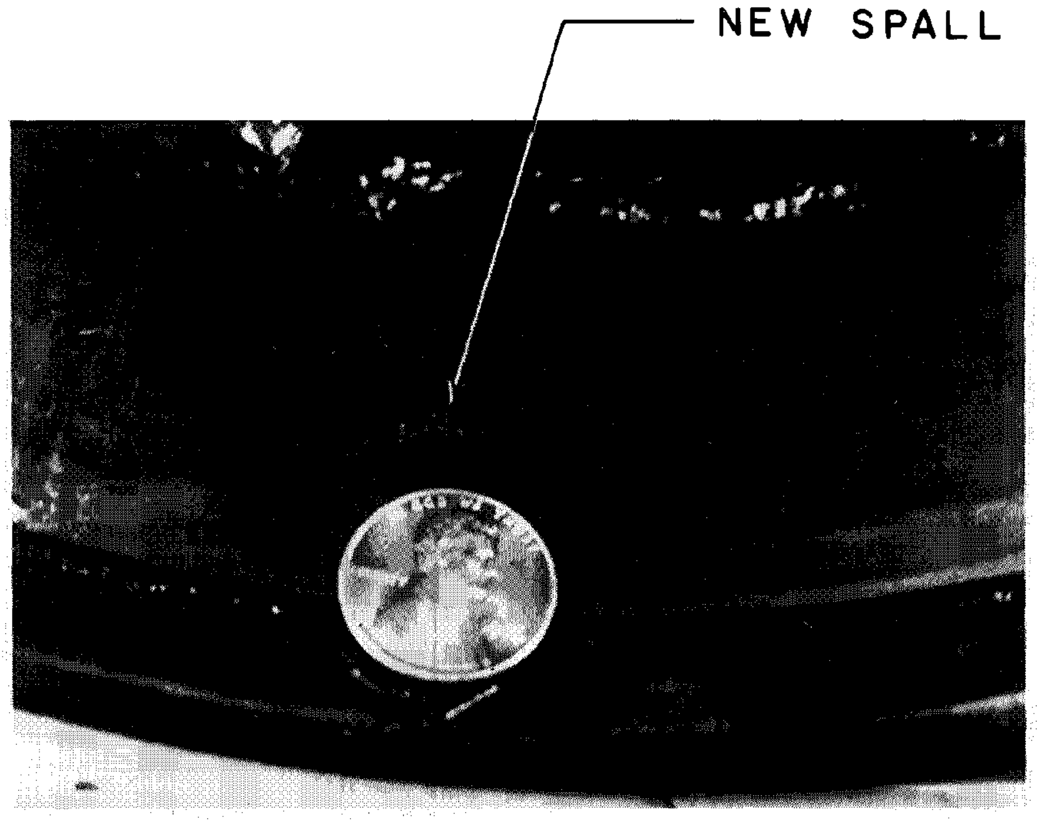


FIGURE 11. BEARING FP-8 AFTER 150.1 TEST HOURS SHOWING NEW SPALL

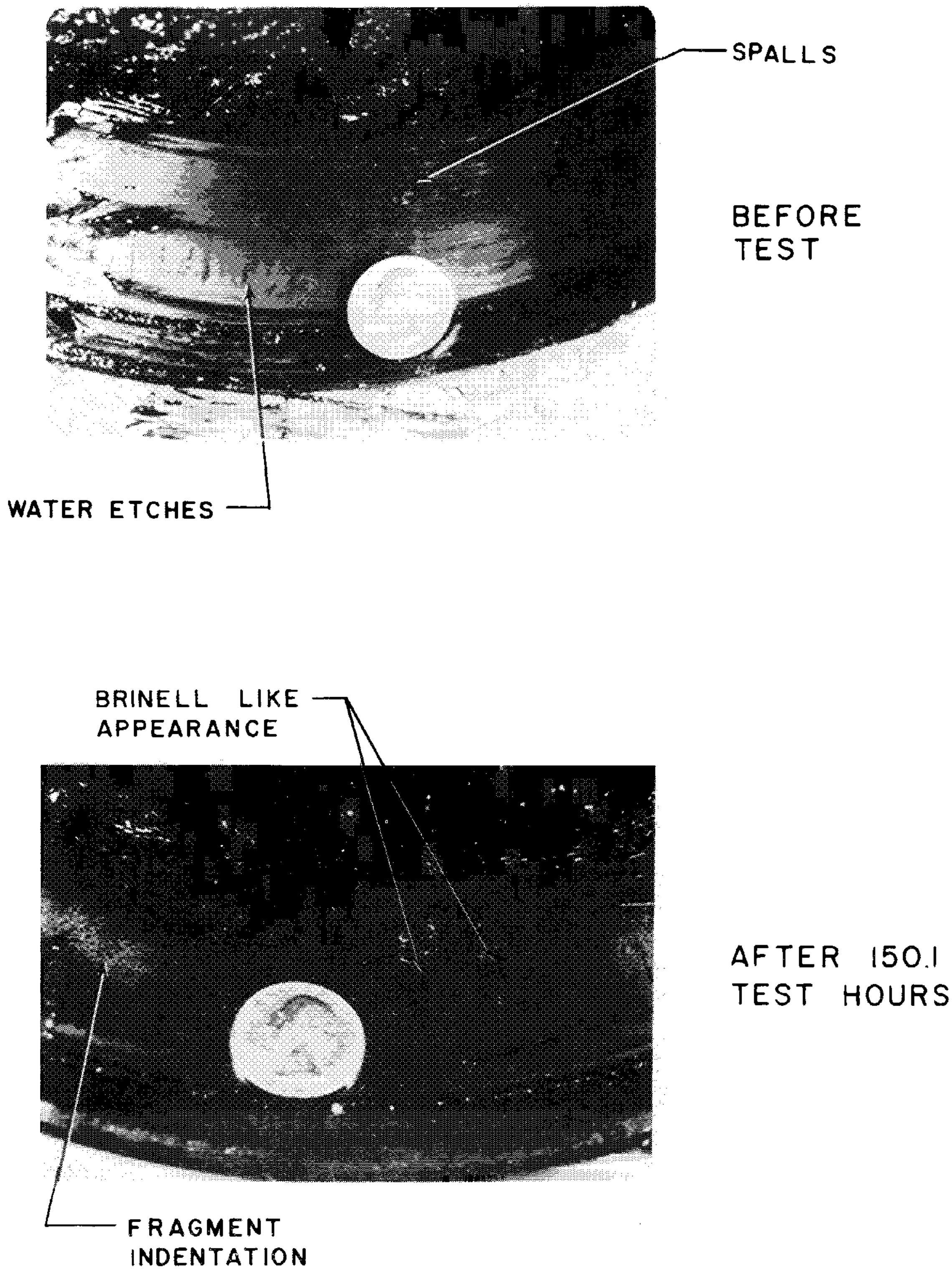


FIGURE 12. BEARING FP-8 SHOWING INCREASED FRAGMENT INDENTATION

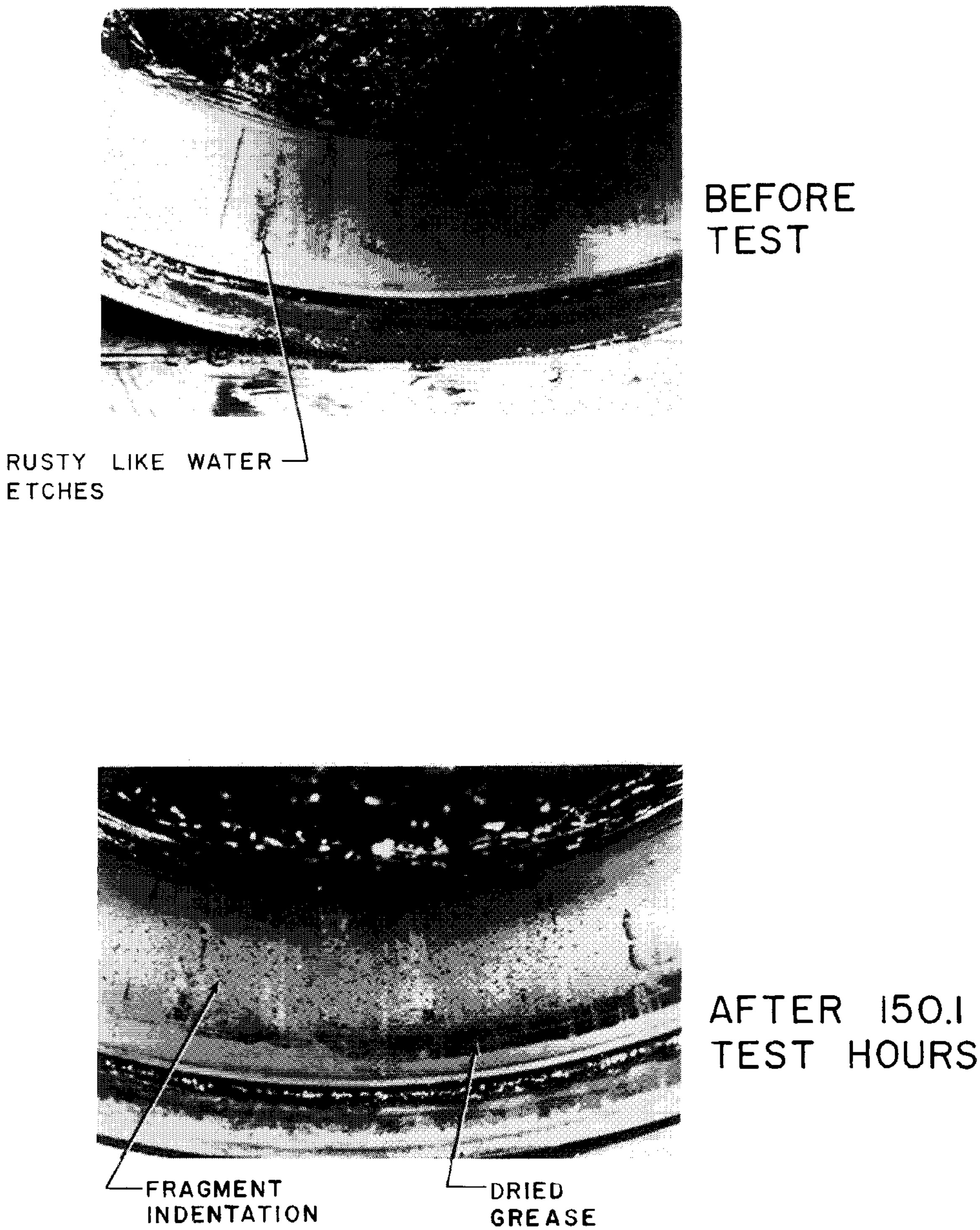


FIGURE 13. BEARING FP-8 SHOWING INCREASED FRAGEMENT INDENTATION AND LOSS OF RUSTY APPEARANCE OF WATER ETCHES

SUMMARY OF INITIAL CONDITION

- Cup broken on both ends
- Badly dented seal cases
- Moderate brinells
- One cage dented and bent

SUMMARY OF EFFECT OF ADDITIONAL RUNNING

- Cage had run eccentrically as a result of distorted shape and rubbing on damaged seal
- Rollers had rolled under separators causing jamming, wear and overheating
- Rollers blue and grease-dried as a result of heat on affected cone

QUALITATIVE EVALUATION OF DEGRADATION

- Severe, near catastrophic

OVERALL PARAMETRIC BEHAVIOR

- Temperature rise above ambient: 100 - 200°F
- Vibration Level: 20 - 40 g's
- Sound pressure level: .5 - 4 DYNE/cm^2
- Demodulated sound and vibration amplitudes

	4 Hz (cage)	8 Hz (2 x Cage)	96 Hz (Outer Race)
Vibration	2.7 - 133	6.9 - 40	2.7 - 18.4
Sound	0.9 - 99	0.3 - 75	1.5 - 3.7

GENERAL DESCRIPTION OF DATA

- After the first 14 hours of steady state thermal operation ($\approx$ 100°F Δt) temperature became erratic and generally increased until shutdown at $\approx$ 24 hours. This was preceded by moderately erratic vibration and sound behavior.

- Steady operation lasted for about 9 hours after second start (load had been removed during idle period) at which point temperature rise abruptly increased from $\approx$ 100°F to $\approx$ 160°F. Temperature rise then rose gradually (and erratically) to $\approx$ 165°F. at which point the test was routinely interrupted. During this period sound and vibration levels became mildly erratic.

FIGURE 14. BEARING FP-6, SUMMARY OF PERTINENT INFORMATION

- The fourth run started where the third ended as far as temperature, vibration, and sound were concerned. Cage frequency amplitudes in the demodulated sound and vibration data were considerably above those noted at the end of the third run.

- Temperature continued to rise erratically throughout remainder of run ($\approx$ 9 hours). Sound and vibration levels continued to be very erratic. Suspect that vibration amplitudes were electronically clipped. Rig shut down automatically due to over temperature (390^{o}F.) on temperature logger. (Computer monitored thermocouple only reached 305^{o}F.

Figure 14 (Continued)

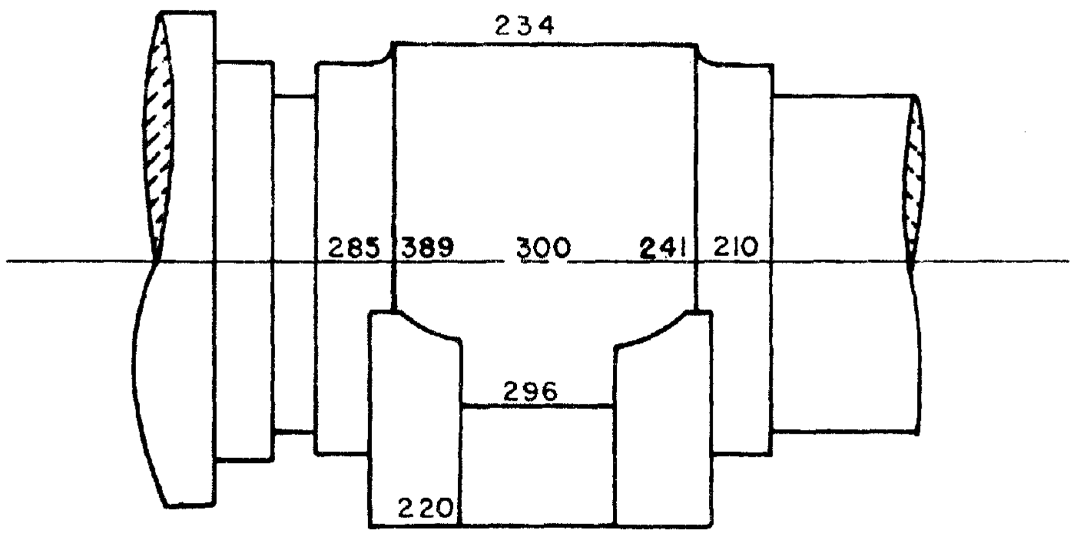


FIGURE 15. FP-6 CUP TEMPERATURES JUST BEFORE AUTOMATIC SHUTDOWN

Figure 16 shows the bearing after the completion of the 69.5 hours prior to removal from the test rig. Here it can be seen that some grease had leaked from the poorly fitting seal cases. Figure 17 shows the dried and burnt grease in the outboard end seal case. Figure 18 shows the outboard cone before and after the test. The grease was charred hard and black, and the worn and blued roller ends indicated that they had reached temperatures in excess of 500°F.

Figures 19 and 20 show the separator/race wear that occurred during the test which was the cause of the large vibration and sound signature that was noted late in the test. Note that the separator wear occurred on those separators that had been bent outward as a result of the rolled under cage. This rolling under had been caused by a previous impact that had also smashed the seal case, broken the cup and cracked the wear ring (Figure 21). The separator wear was actually caused by eccentric cage operation resulting from interference between cage and the damaged seal case (Figure 22). Here it is seen that the cage actually contacted the seal in two areas approximately 120° apart and 120° from the center of the worn separator area shown in Figure 19. It is suspected that the eccentric cage operation was caused and accentuated by, the rollers' tendency to become jammed and cocked under the bent separators. This then caused the rollers to become forced against the inner race shoulder resulting in the major portion of the high heat generation.

Although the outboard end of the bearing suffered a traumatic degree of distress, the inboard end was essentially unaffected by the excessive temperatures only a few inches away, as can be seen in Figure 23. The only thing of significance was that the seal case had rotated by about 45° in the direction of rotation and had become cocked and jammed in that position. (See Figure 24.) Even so, only a minimal amount of grease leaked from the seal area. (See Figures 16 and 24.)

4.1.3 Bearing FP-7

Pertinent information regarding bearing FP-7 is summarized in Figure 25. This bearing had very loose fitting seals and had roller and inner race spalls, only one of which was discovered during the pretest inspection. Furthermore, it appeared to have less and drier grease in the cone than the other bearings tested and a larger lateral clearance.

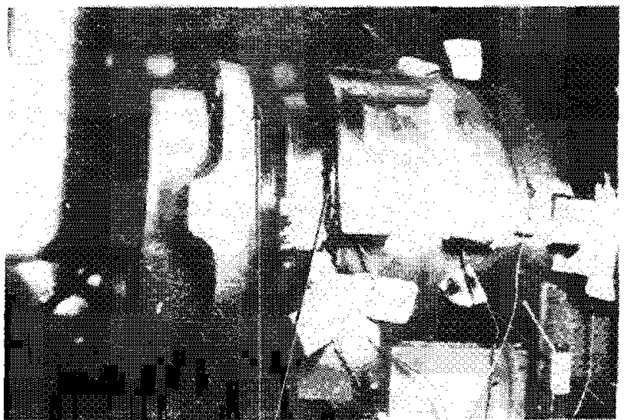


FIGURE 16. BEARING FP-6 AFTER 69.5 TEST HOURS SHOWING GREASE LEAKAGE FROM DAMAGED SEAL CASES

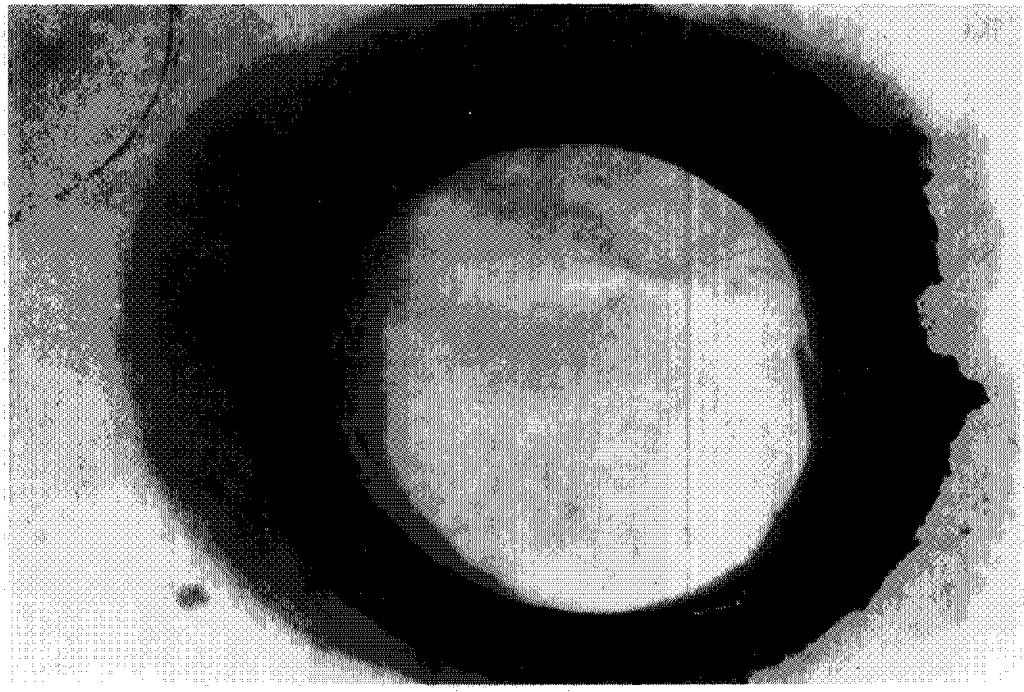


FIGURE 17. OUTBOARD SEAL FROM BEARING FP-6 SHOWING DRIED AND BURNT GREASE

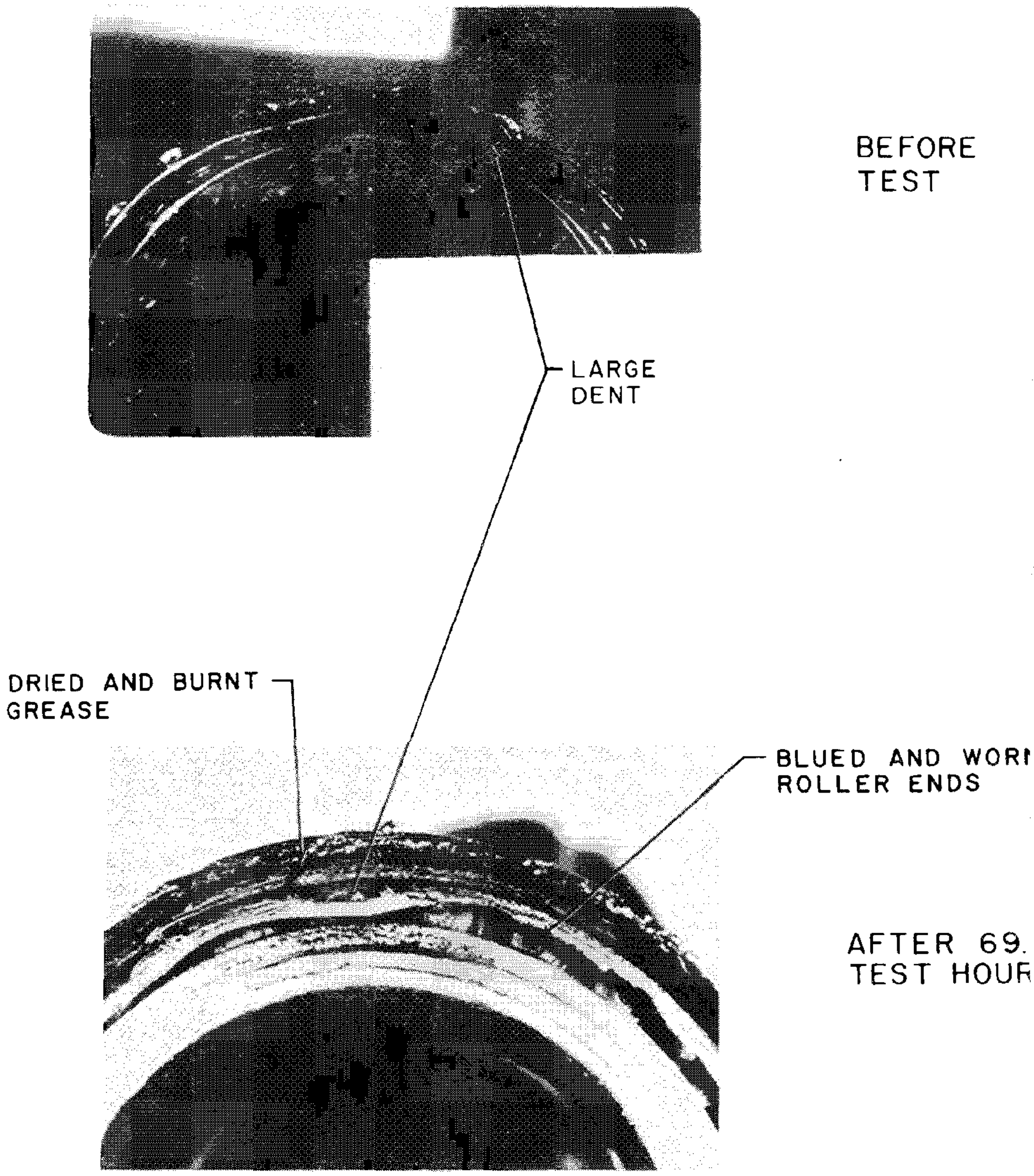


FIGURE 18. BEARING FP-6 SHOWING HEAT DAMAGE

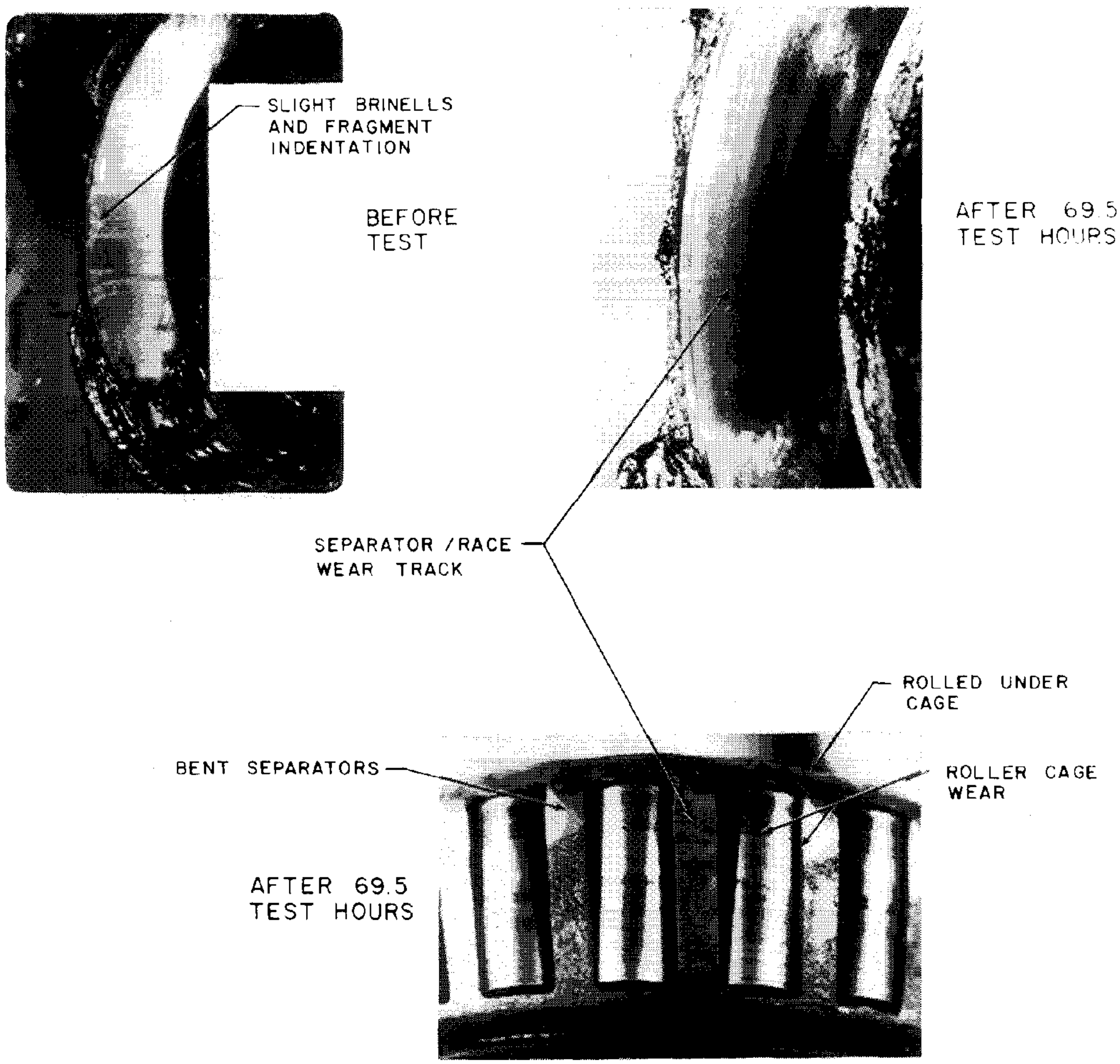


FIGURE 19. BEARING FP-6 SHOWING EFFECTS OF BENT AND ECCENTRIC CAGE

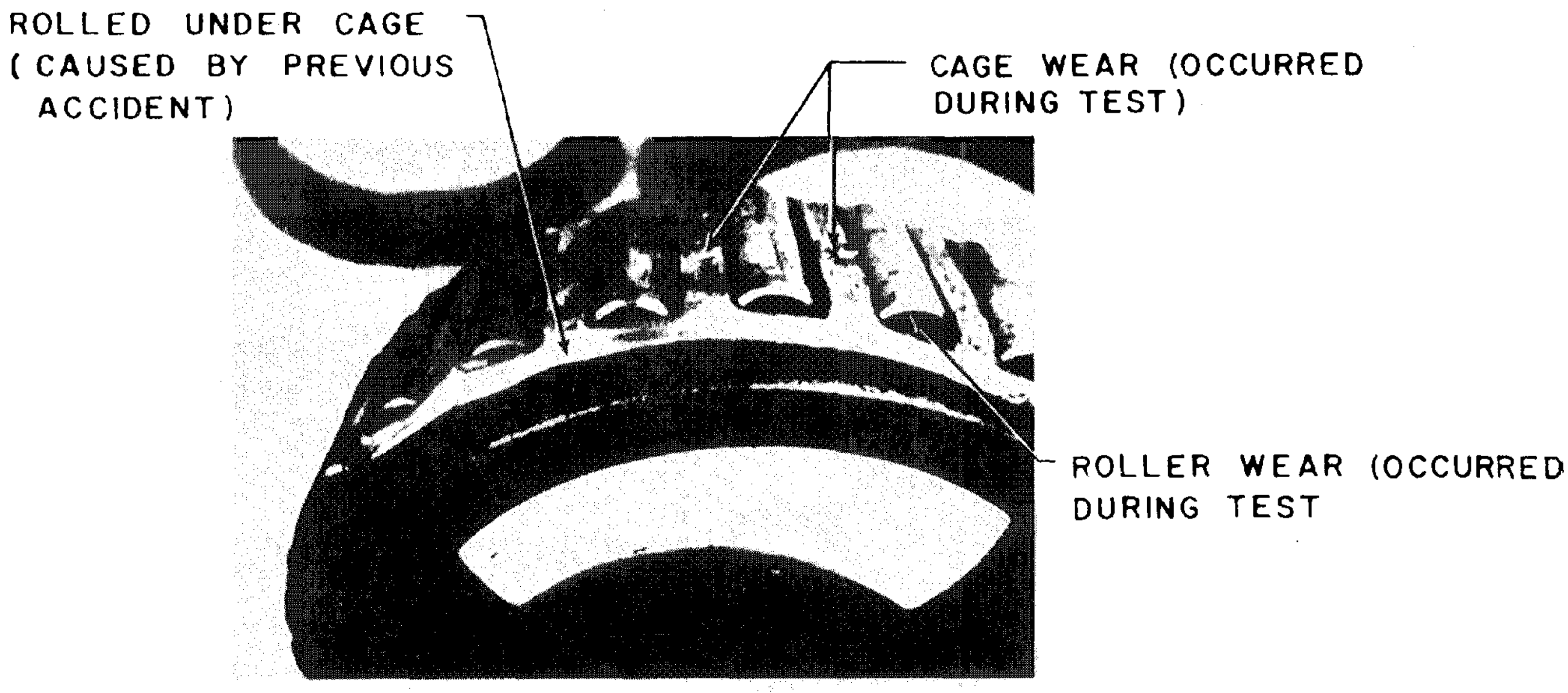


FIGURE 20. BEARING FP-6 AFTER 69.5 TEST HOURS SHOWING OUTBOARD CONE DAMAGE

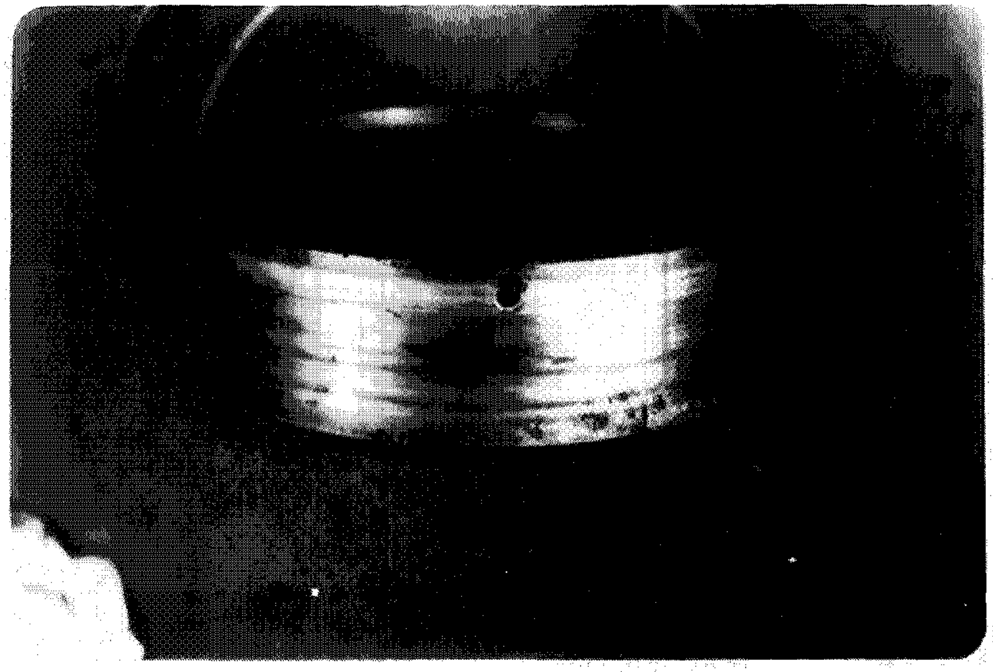


FIGURE 21. BEARING FP-6 AFTER 69.5 TEST HOURS SHOWING CRACKED OUTBOARD END WEAR RING

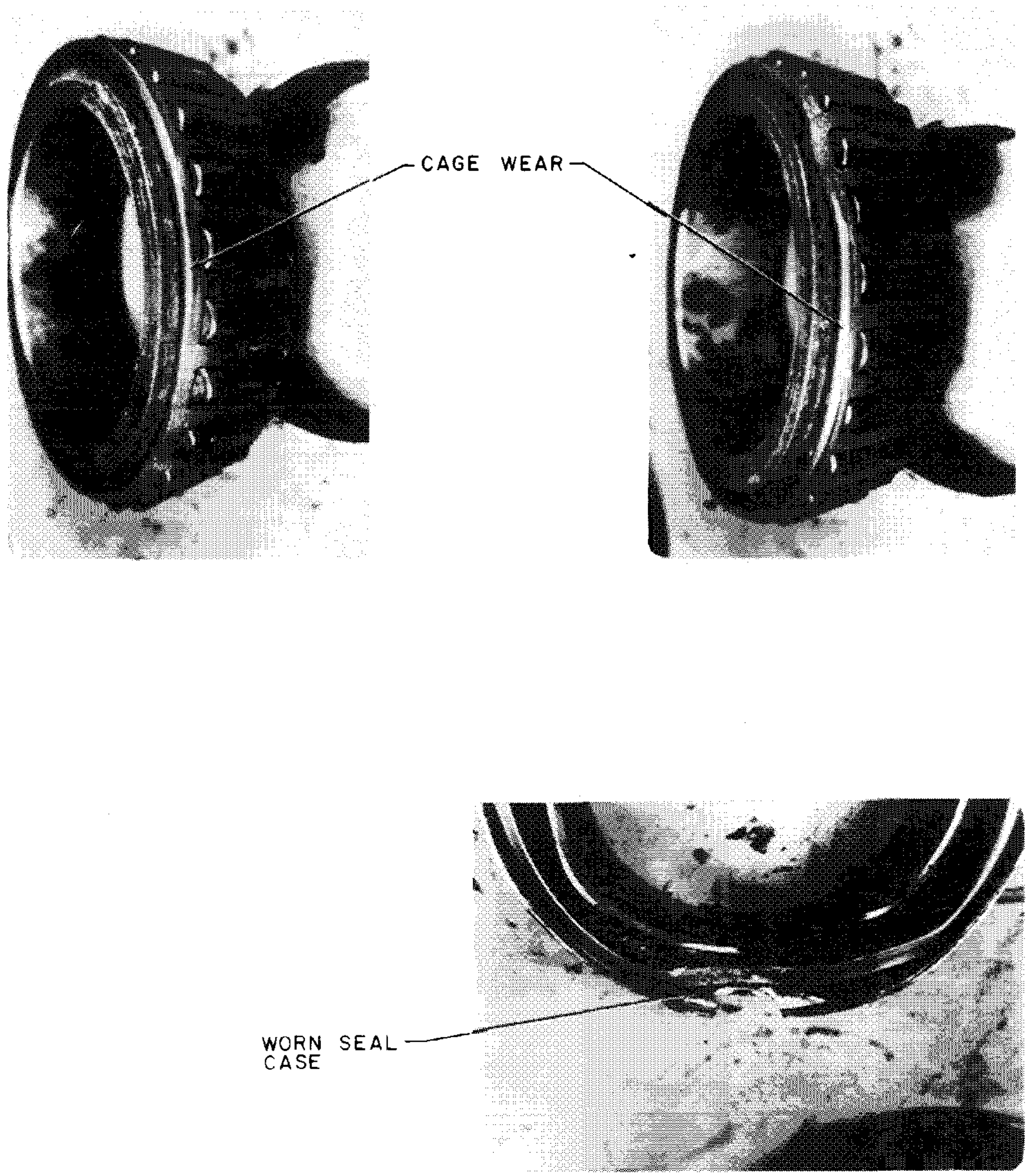


FIGURE 22. BEARING FP-6 AFTER 69.5 TEST HOURS SHOWING SEAL CASE AND CAGE WEAR WHICH CAUSED ECCENTRIC CAGE OPERATION

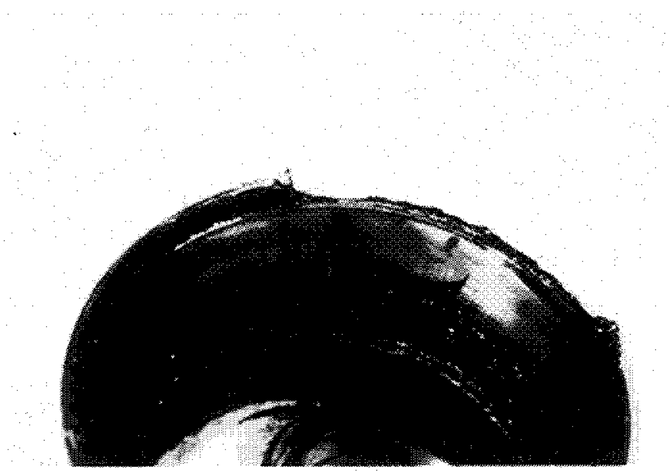


FIGURE 23. BEARING FP-6 AFTER 69.5 TEST HOURS SHOWING THAT INBOARD END WAS NOT SIGNIFICANTLY AFFECTED BY THE HIGH OUTBOARD END TEMPERATURE

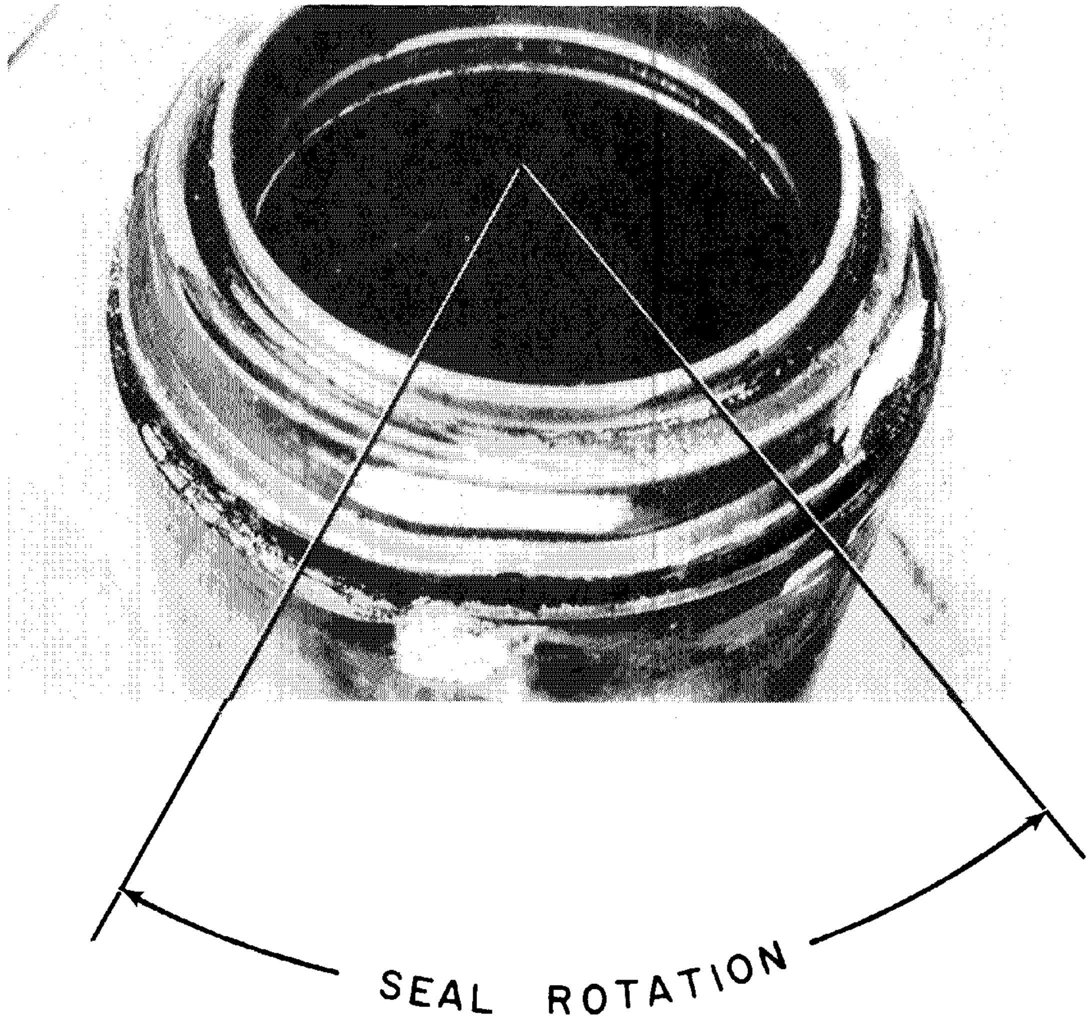


FIGURE 24. BEARING FP-6 AFTER 69.5 TEST HOURS SHOWING ROTATED AND COCKED SEAL

SUMMARY OF INITIAL CONDITION

- At least one spalled roller
- Seals loose in C' bore

SUMMARY OF EFFECT OF ADDITIONAL RUNNING

- Discovered total of seven spalled rollers on one cone. These were likely there before the test.
- Heavy particle indentation
- Seal case OD's smooth and sharp edged from rotating in cup
- Both cones spalled

QUALITATIVE EVALUATION OF DEGRADATION

- Slight, neglecting questionable spalls

OVERALL PARAMETRIC BEHAVIOR

· Temperature rise above ambient:	45 - 50^{o}F
· Vibration level:	120 - 140 g's
· Sound pressure level:	10 - 20 DYNE/cm^{2}

GENERAL DESCRIPTION OF DATA

- Temperature and sound very steady throughout
- Vibration mildly erratic throughout

FIGURE 25. BEARING FP-7, SUMMARY OF PERTINENT INFORMATION

The test of bearing FP-7 was uneventful except for the fact that both seal cases rotated at shaft speed throughout the entire 150 hours. Also overall sound and vibration levels were quite high with sizable amplitudes at all characteristic component frequencies. These high amplitudes may be attributable to the dried character and small quantity of grease which resulted in less attenuation than available from the other test bearings, as well as the excessive lateral clearance ($\approx$ 0.032 inches).

The most significant things noticed upon disassembly of bearing FP-7 after completion of the 150-hour test was the spalled rollers on the outboard cone (Figure 26) and the spalls on both inner races. Only one roller had been noted as spalled at the pretest inspection. Admittedly, this inspection tended to be cursory because of the desire to not disturb the grease and wear particles so it is likely that more than one roller was initially spalled. Furthermore, the inner race spalls could not be seen without a thorough cleaning (which was not accomplished until after the test). The metal chips between rollers (Figure 26) were noticeably smaller after the test than before. Some minor particle indentation and small brinell-like marks were noted in the outboard cup (Figure 27). The inboard cup became very heavily particle indented giving a "mottled" appearance (Figure 28). Figure 29 shows worn and sharp edged seal case resulting from rotation within the cup C' bore during the test.

4.1.4 <u>Bearing FP-10</u>

Pertinent information regarding bearing FP-10 is summarized in Figure 30. This bearing had a broken and a cracked cup. The cracked side was installed on the inboard side of the test rig and oriented angularly such that the bending stress in the cup would be near maximum at the location of the crack ($\approx 45^{\circ}$ from the direction of load).

During the course of the test, temperatures remained relatively constant. Overall sound and vibration levels were mildly erratic during periods of the test duration. Demodulated sound and vibration level variations at outer race frequency were relatively large for this bearing.

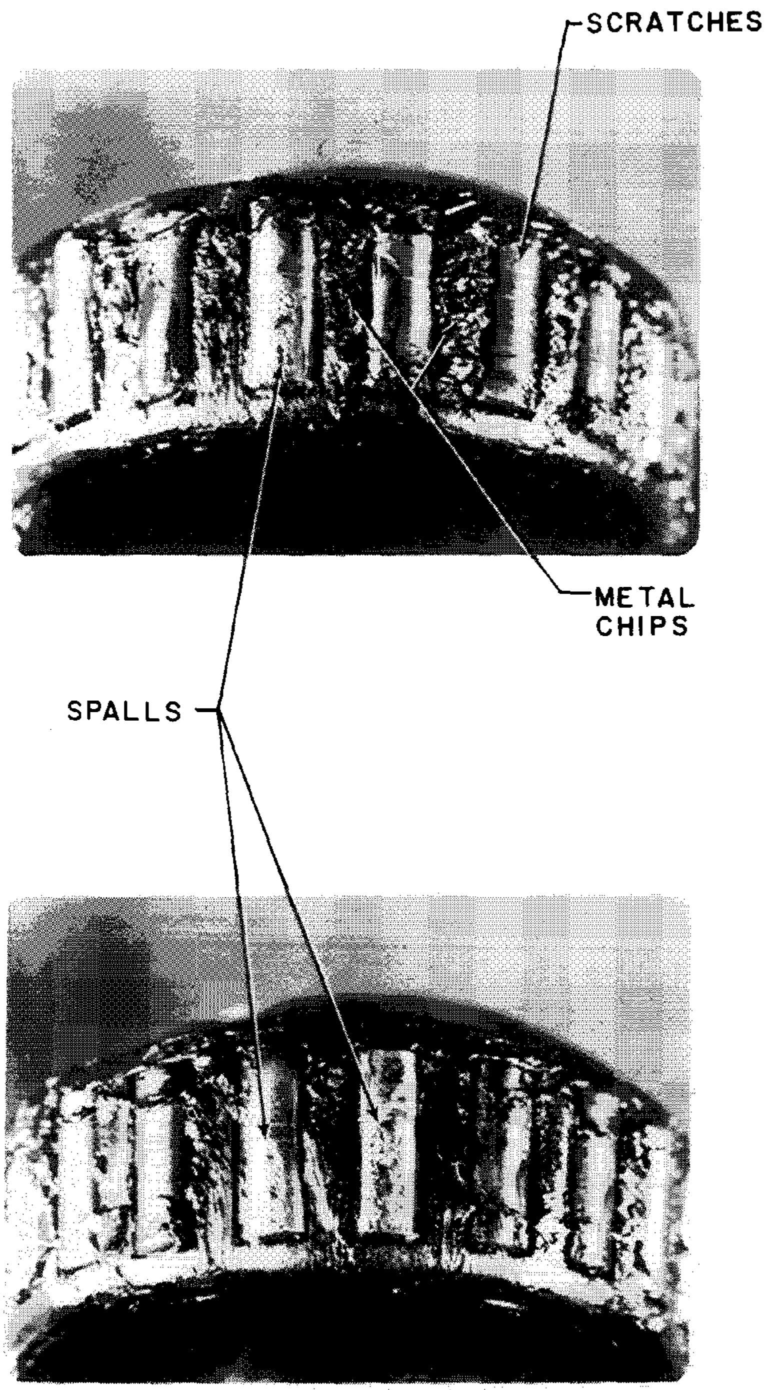


FIGURE 26. BEARING FP-7 AFTER 150.5 TEST HOURS SHOWING OUTBOARD CONE

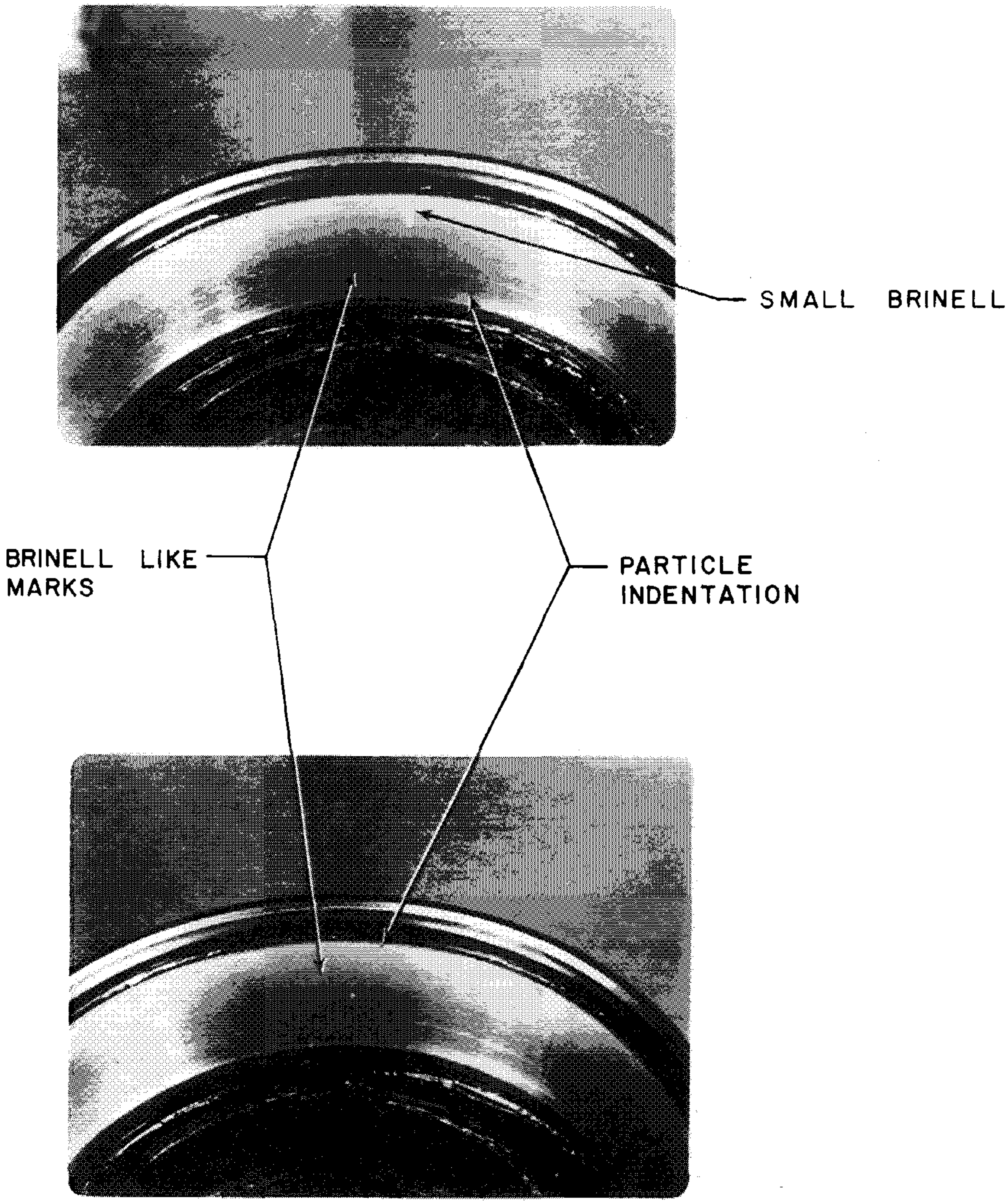
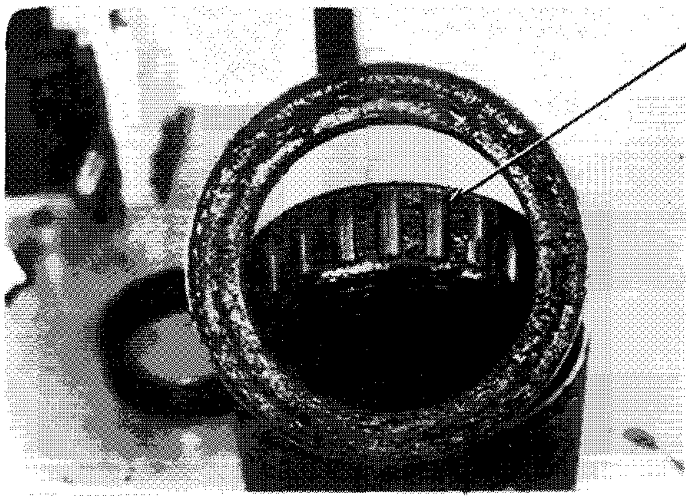


FIGURE 27. BEARING FP-7 AFTER 150.5 TEST HOURS SHOWING BRINELL LIKE MARKS AND PARTICLE INDENTATION ON OUTBOARD CUP



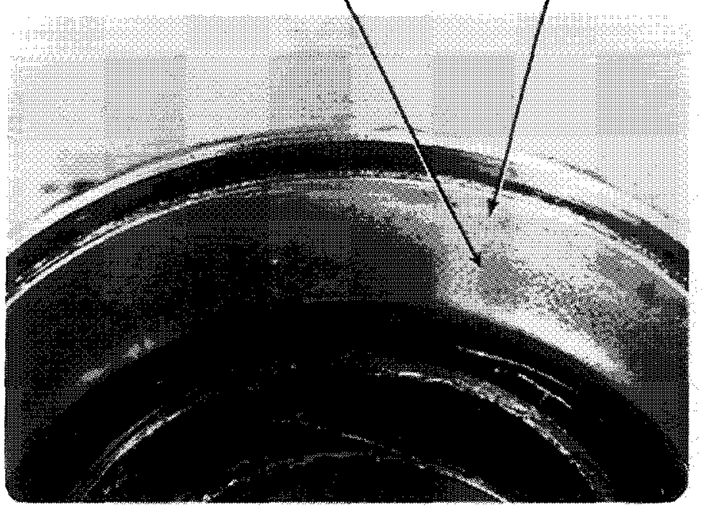
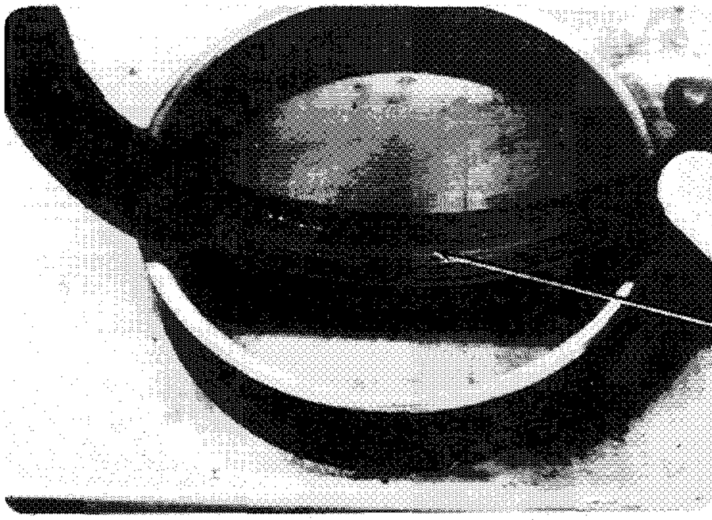


FIGURE 28. BEARING FP-7 AFTER 150.5 TEST HOURS SHOWING INBOARD SIDE CONDITION

310-8 23



310-8 24

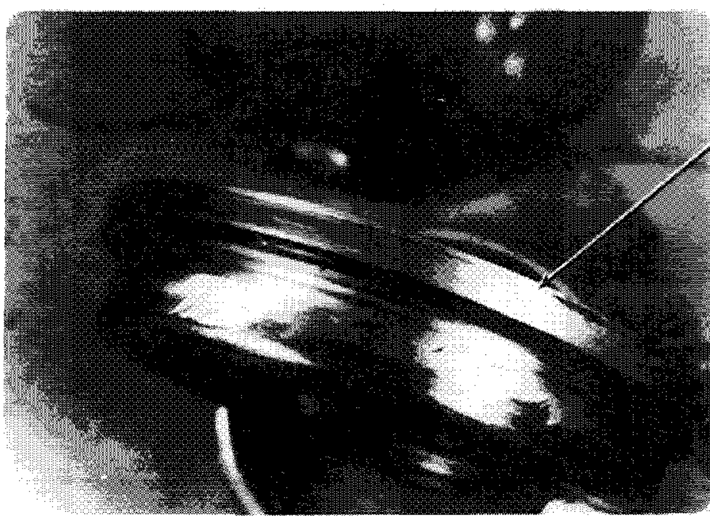


FIGURE 29. BEARING FP-7 AFTER 150.5 TEST HOURS SHOWING SEAL CONDITION

SUMMARY OF INITIAL CONDITION

- Cup cracked on one side
- Cup broken on other side

SUMMARY OF EFFECT OF ADDITIONAL RUNNING

- Crack extended in size and number
- Race surface in cracked area spalled
- Particle indentation on both races adjacent to damaged areas
- Roller wear

QUALITATIVE EVALUATION OF DEGRADATION

- Severe

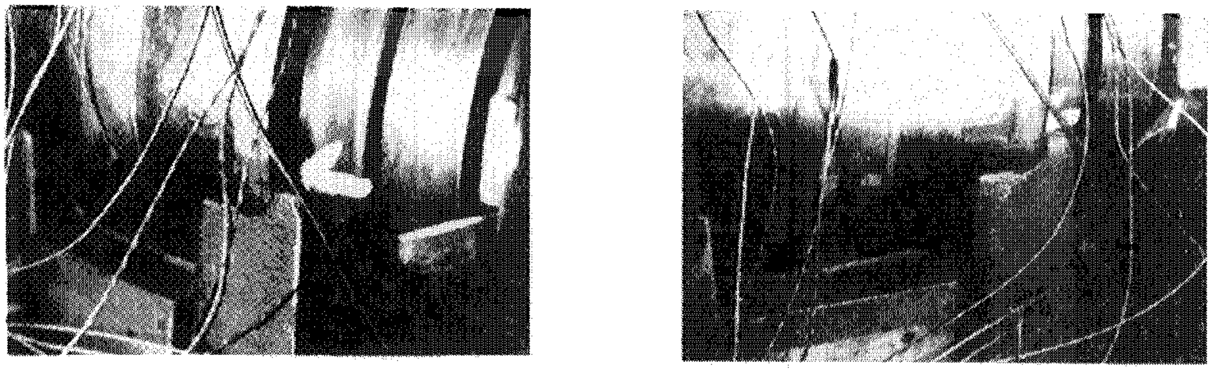
OVERALL PARAMETRIC BEHAVIOR

· Temperature rise above ambient:	50 - 60^{o}F
· Vibration level:	12 - 40 g's
· Sound level:	1.2 - 5.7 DYNE/cm^{2}

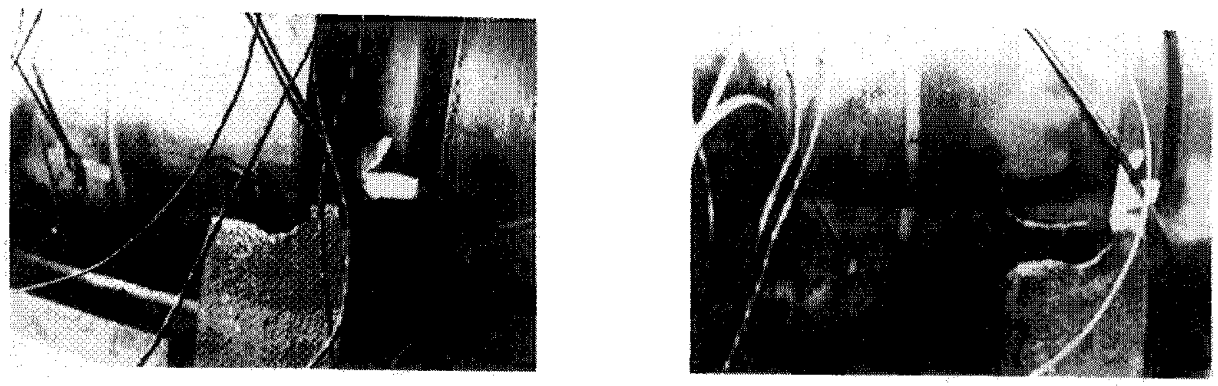
GENERAL DESCRIPTION OF DATA

- Temperature levels remained fairly steady for approximately 80 hours then became mildly erratic
- Sound and vibration increased in amplitude at the fifth start ($\approx$ 48 hours) and remained fairly constant until $\approx$ 113 hours and then began to gradually increase.
- Sound and vibration erratic and high between 65 and 72 hours

FIGURE 30. BEARING FP-10, SUMMARY OF PERTINENT INFORMATION

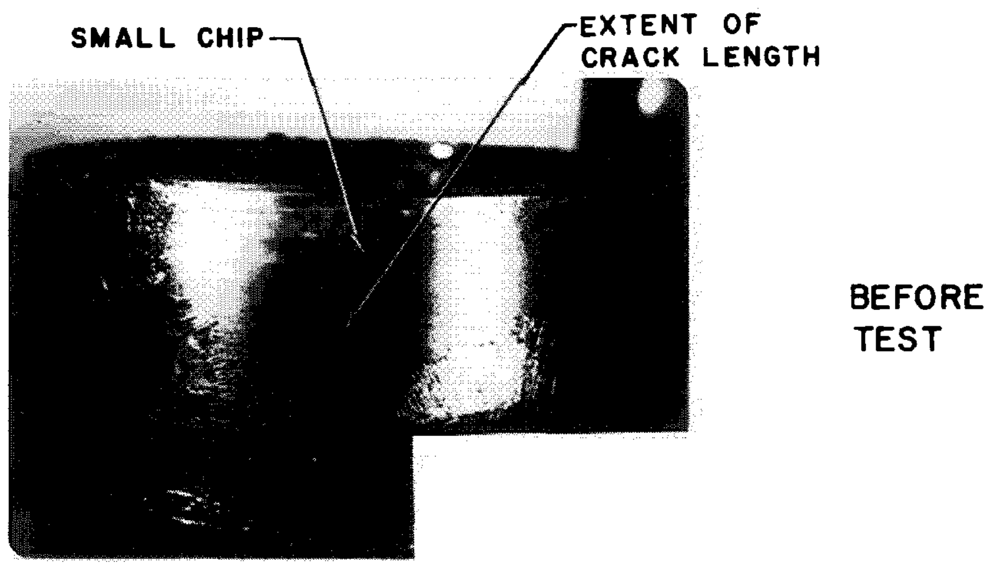


AFTER 31.2 HOURS



AFTER 47.9 HOURS

FIGURE 31. BEARING FP-10 CRACK GROWTH DURING TEST



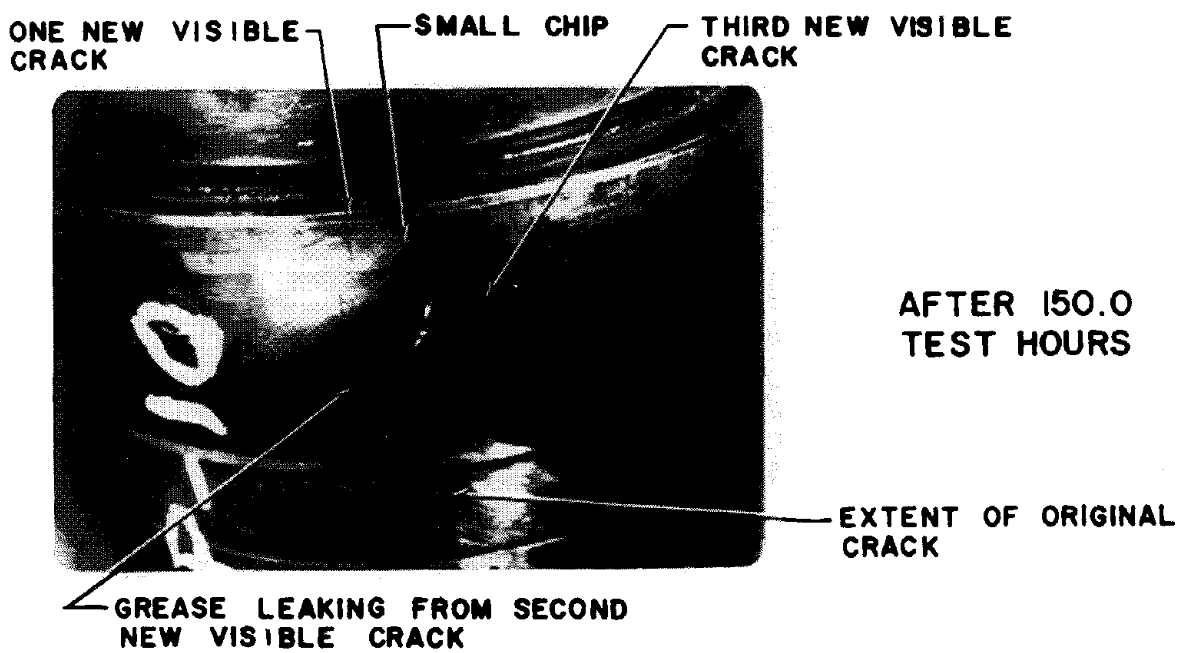


FIGURE 32. BEARING FP-10 SHOWING CRACK GROWTH ON INBOARD END

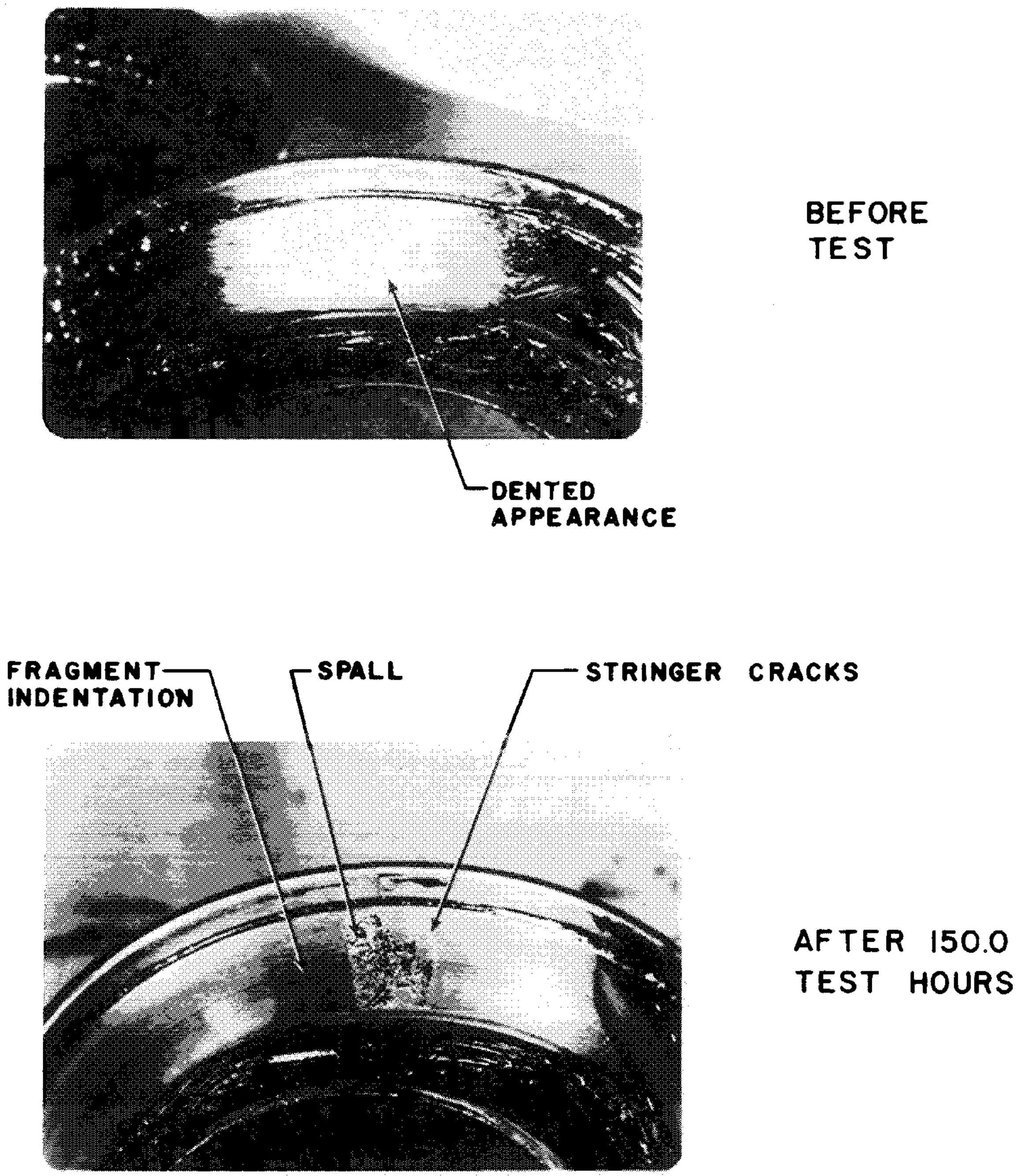
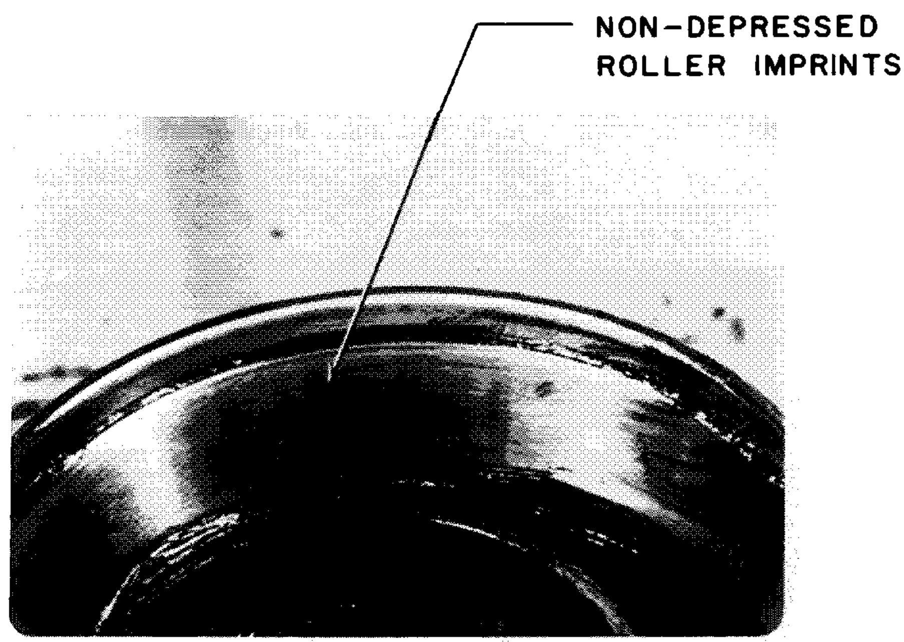


FIGURE 33. BEARING FP-10 SHOWING DEGRADATION OF INNER RACE ON INBOARD END



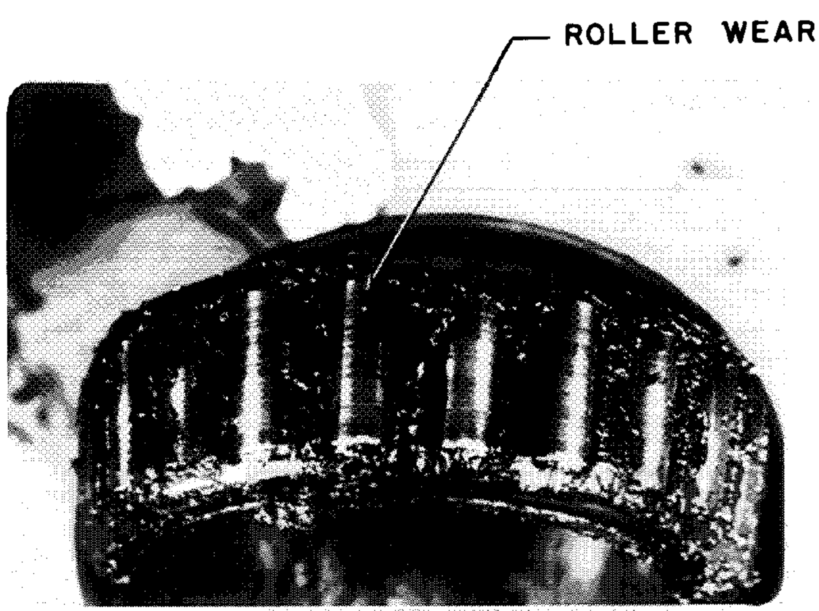


FIGURE 34. BEARING FP-10 AFTER 150.0 TEST HOURS SHOWING ROLLER WEAR AND "IMPRINTS" ON INBOARD END

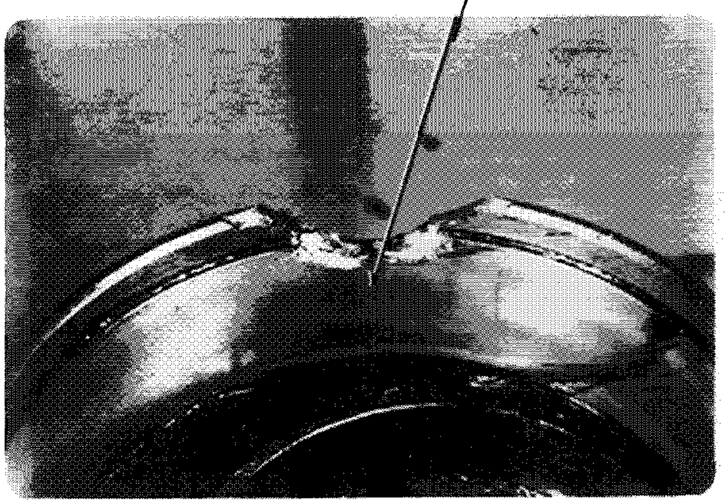
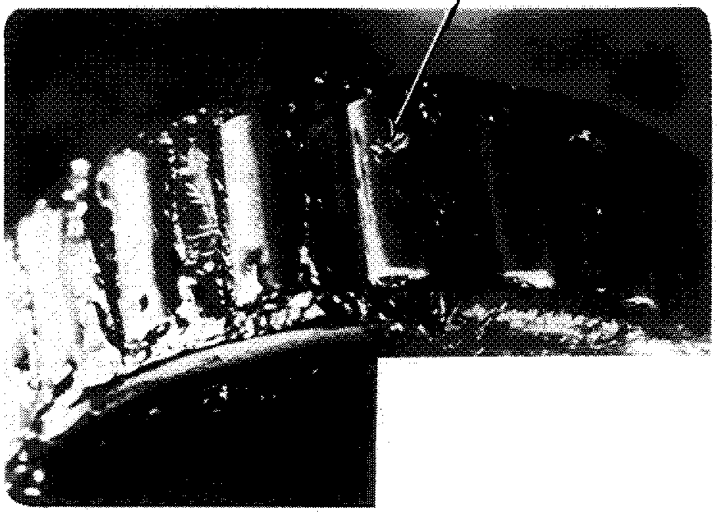


FIGURE 35. BEARING FP-10 SHOWING PARTICLE INDENTATION IN AREA ADJACENT TO BREAK ON OUTBOARD END



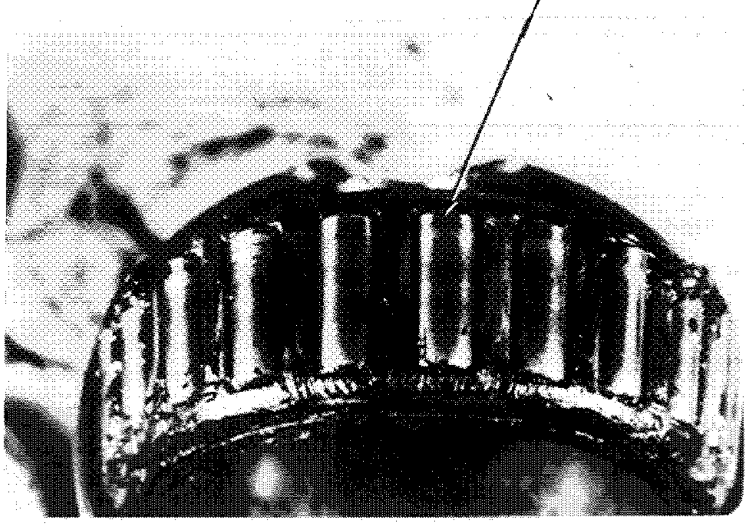


FIGURE 36. BEARING FP-10 SHOWING ROLLER WEAR IN AREA OF BROKEN CUP ON OUTBOARD END

The crack grew and new cracks developed relatively early in the test, as can be seen in Figures 31 and 32. The race surface in the area of the cracks became deeply spalled with several stringer-like cracks emanating from the leading edge of the spall (Figure 33). Fragment indentation is also evident on the surface trailing the spall in Figure 33. The cup surface 180° from the cracked and spalled area had impressions of roller imprints that were not depressed into the surface (Figure 34). This may be the result of a preferentially located microslip between the roller and race caused by opposite side rollers passing the deep spall. Figure 34 also shows heavy roller wear, most likely caused by the debris from the spall. Figure 35 shows the outboard end cup break that did not change in appearance throughout the test. Some particle indentation did appear however. Outboard end roller wear occurred over the roller length that traversed the cup break (Figure 36).

4.1.5 Bearing FP-11

Figure 37 summarizes the pertinent information regarding bearing FP-11. This bearing was similar to FP-10 in that it was cracked and broken, but not quite as severely. Also instead of the widely dented race appearance in the area of the crack, FP-11 was deeply and discernably brinelled. Bearing FP-11 ran throughout the test with temperature, vibration and sound levels remaining relatively steady.

The 150-hour test had no effect on the outboard (broken) cup end (Figure 38). The crack on the inboard end extended and a second crack became visible (Figure 39). The race surface in the area of the crack/brinell developed a multitude of small cracks emanating from the leading edge of the brinell (Figure 40) similar to the cracks around the spall in bearing FP-10 (Figure 33). Also, a very small chip broke out of the inner edge of the races at the trailing edge of the brinell.

4.1.6 Bearing FP-12

The pertinent facts pertaining to bearing FP-12 are summarized in Figure 41. It was initially water-etched on one side with the opposite

SUMMARY OF INITIAL CONDITION

- Cup broken one side
- Cup cracked and brinelled on opposite side

SUMMARY OF EFFECT OF ADDITIONAL RUNNING

- Original crack extended and second crack became visible
- Brinelled area has fine surface cracks

QUALITATIVE EVALUATION OF DEGRADATION

- Moderate

OVERALL PARAMETRIC BEHAVIOR

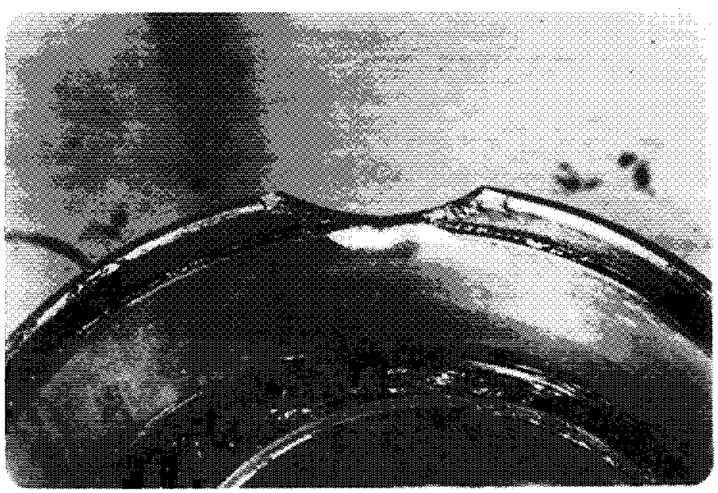
• Temperature rise above ambient:	40 - 55°F
• Vibration level:	25 - 30 g's
• Sound pressure level:	1.3 - 2.0 DYNE/cm^2

GENERAL DESCRIPTION OF DATA

- Temperature, sound, vibration relatively steady throughout test
- Few temperature spikes lasting an hour or two generally accompanied by a reduction in sound and vibration

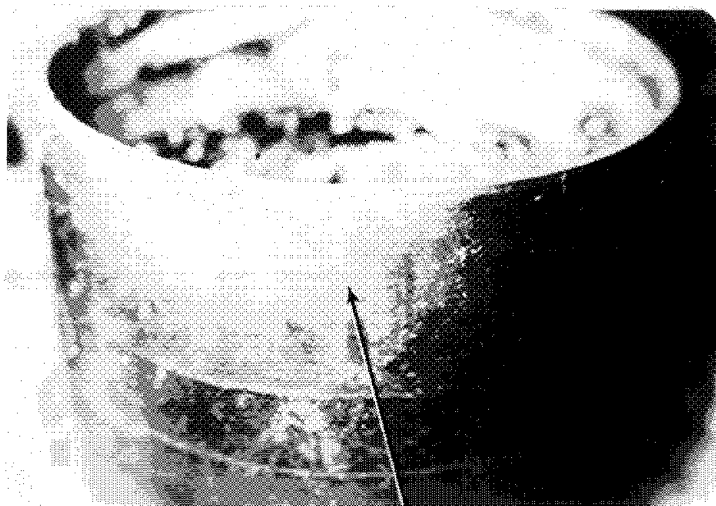
FIGURE 37. BEARING FP-11, SUMMARY OF PERTINENT INFORMATION

BEFORE
TEST



AFTER 151.6
TEST HOURS

FIGURE 38. BEARING FP-11 SHOWING OUTBOARD END BROKEN CUP



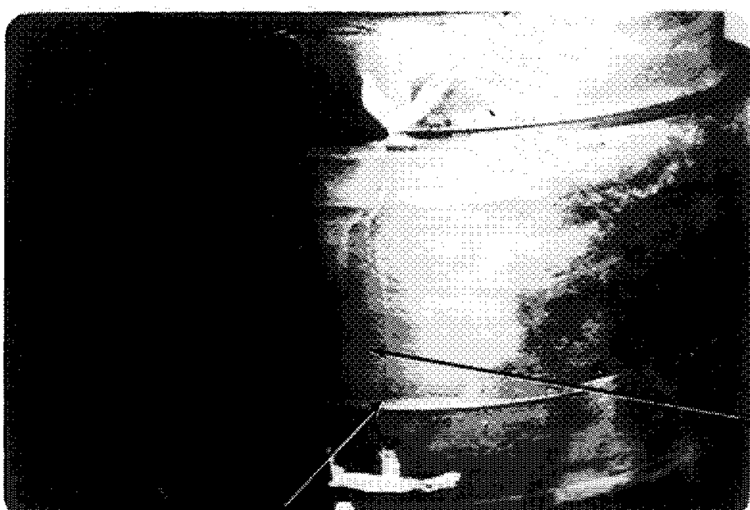
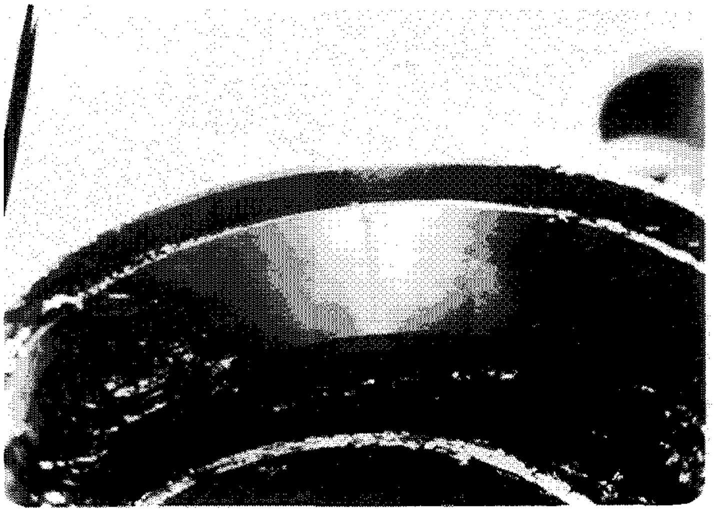


FIGURE 39. BEARING FP-11 SHOWING INBOARD END CRACK PROPAGATION

310-2
35

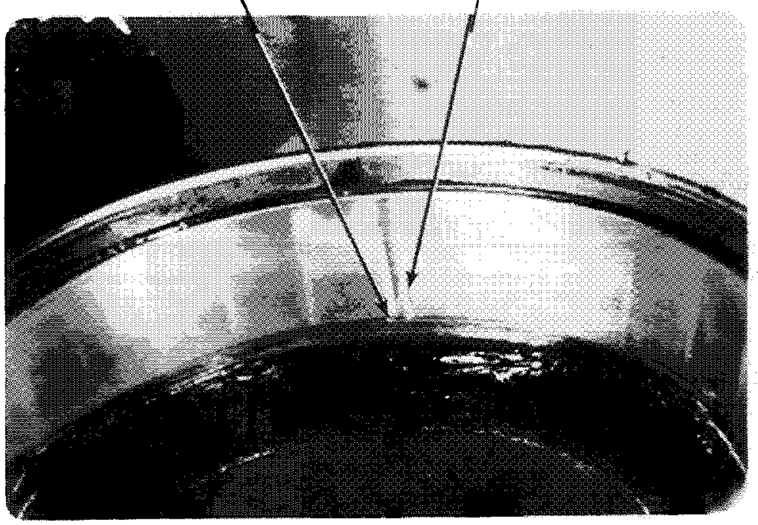


BEFORE
TEST

CHIP

CRACKS APPEARING
AFTER TEST

310-9
15



AFTER 151.6
TEST HOURS

FIGURE 40. BEARING FP-11 SHOWING INBOARD END BRINELL AND CRACKS

SUMMARY OF INITIAL CONDITION

- Water damage on one side
- Broken cup on opposite side

SUMMARY OF EFFECT OF ADDITIONAL RUNNING

- Small chip broke out of race adjacent to break
- Small wear track on race

QUALITATIVE EVALUATION OF DEGRADATION

- Slight

OVERALL PARAMETRIC BEHAVIOR

· Temperature rise above ambient:	60 - 70^{o}F
· Vibration level:	25 - 65 g's
· Sound pressure level:	3.3 - 6.0 DYNE/cm^{2}

GENERAL DESCRIPTION OF DATA

- Relatively steady operation for first 68 hours
- Temperature, sound, and vibration erratic during remainder of test

FIGURE 41. BEARING FP-12, SUMMARY OF PERTINENT INFORMATION

side having a broken cup. The first half of the test was steady and uneventful with the latter half experiencing relatively erratic temperature, sound, and vibration behavior.

The only significant effect of the 150-hour test on bearing condition was that a small chip broke out of the outer race adjacent to the original crack and that a wear track developed near the inner edge of the outer race (Figure 42).

4.1.7 Loader Bearings LB-1 and LB-2

Two different 6 x 11, center loader bearings were used during the course of the six failure progression plus initial checkout tests. These bearings were loaded to 52,500 pounds and were water-cooled so that they ran at about 110-125°F. Both were brand new and had been purchased directly from the manufacturer. The only instrumentation applied directly to the loader bearings were two thermocouples to monitor seal case temperature.

While removing bearing FP-10 from the test shaft, it was noted that the loader bearing (LB-1) felt rough upon being turned by hand, and accordingly it was removed for inspection. One end was found to be grossly spalled. Photographs of this bearing after 676.8 hours of running are shown in Figures 43 through 45. Its condition is also summarized in Table 6.

This severe spalling came as a surprise since we had run other center bearings at twice rated load on previous test programs for longer periods of time with only very small spalls developing. The only previous negative experience with such highly loaded bearings was excessive temperature and premature grease deterioration. The loader bearing cooling jacket used for the subject failure progression test program eliminated the grease thermal degradation problem as is evidenced in Figures 43 through 45.

Because of the positive experience with regard to spalling that we had had previously, we passed off the spalling situation with LB-1 as simply an improbable occurrence.

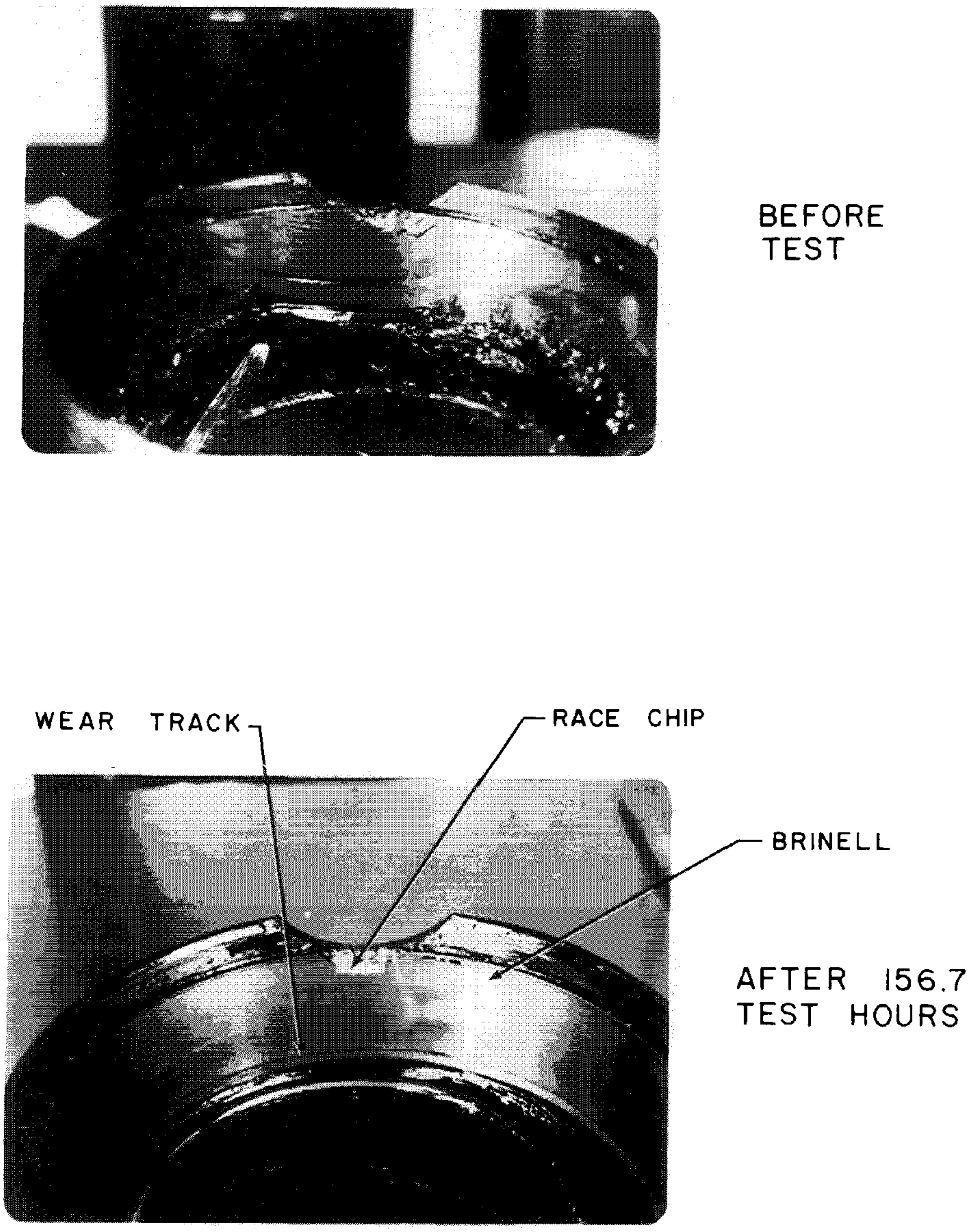


FIGURE 42. BEARING FP-12 SHOWING INCREASED SIZE OF RACE CHIP

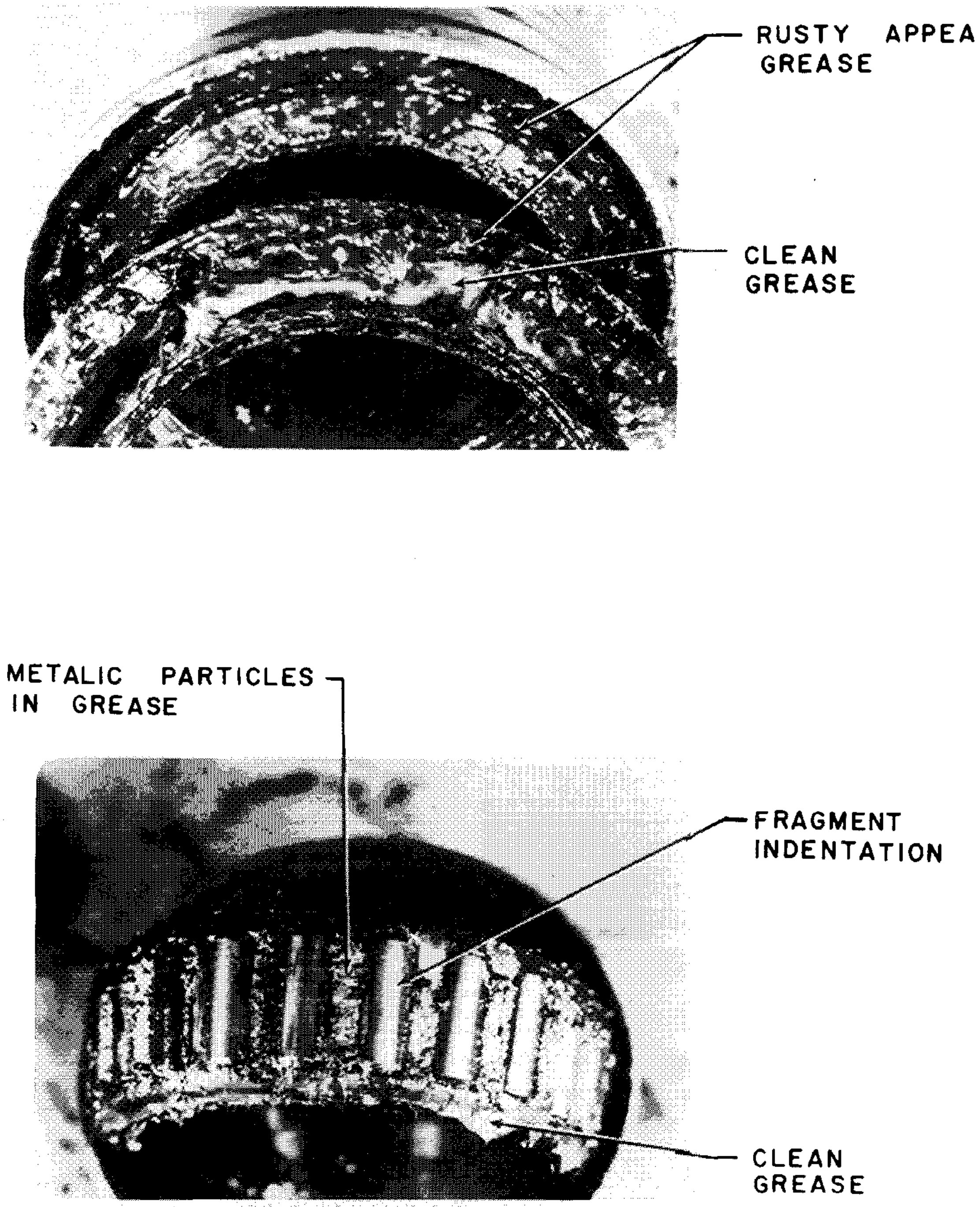
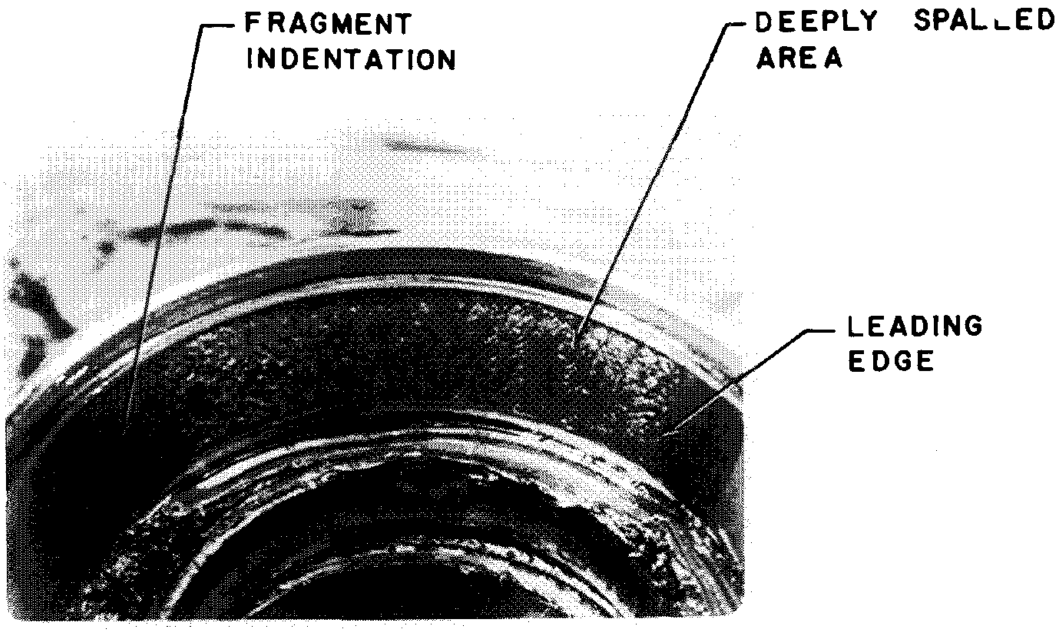


FIGURE 43. BEARING LB-1 AFTER 676.8 TEST HOURS AT TWICE RATED LOAD



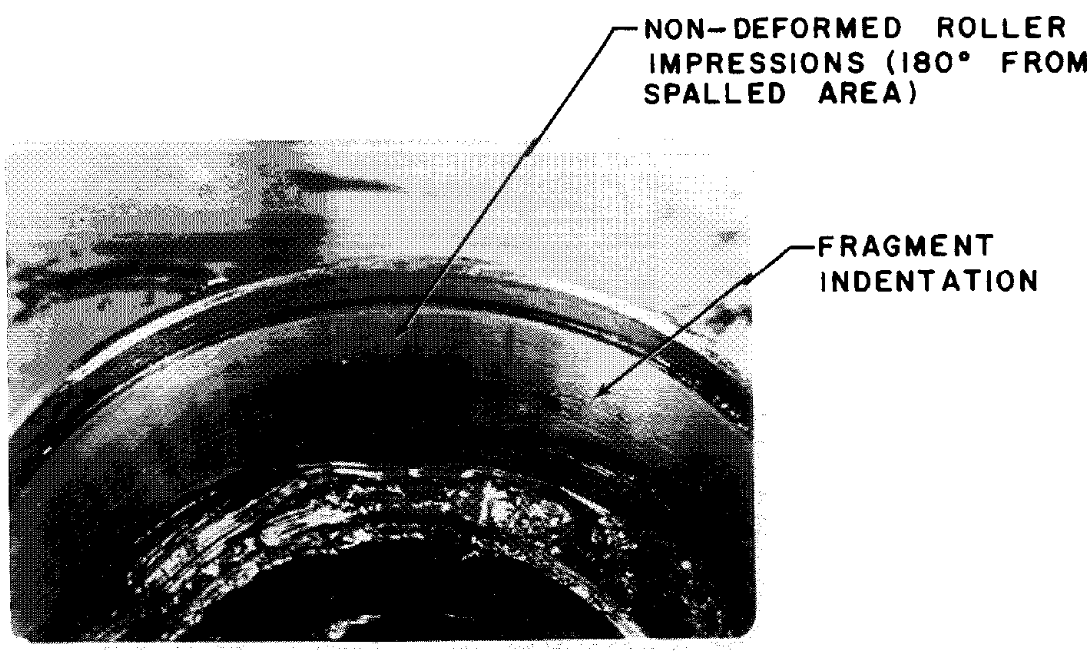


FIGURE 44. BEARING LB-1 (DRIVE END) AFTER 676.8 HOURS AT TWICE RATED LOAD

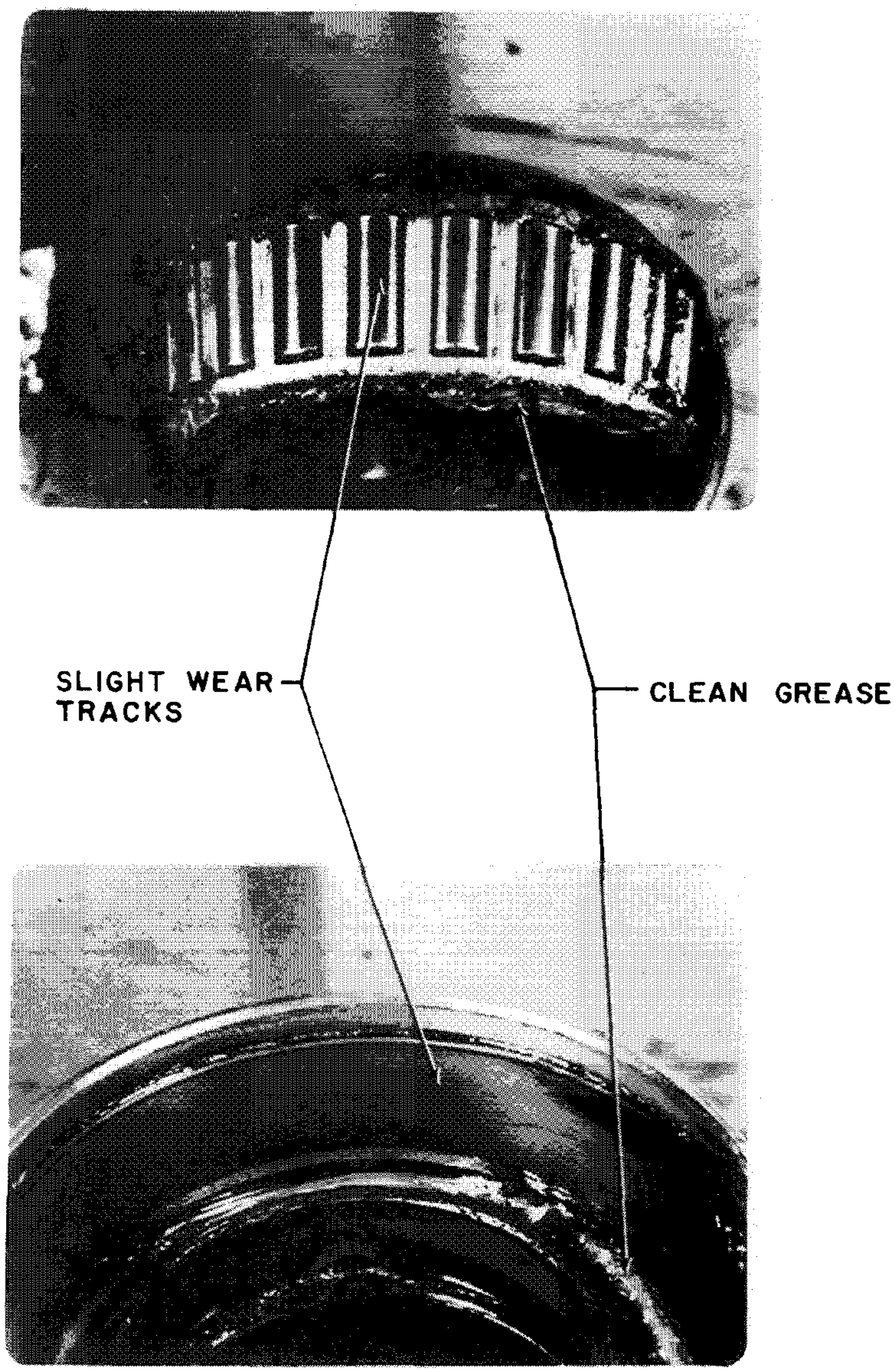


FIGURE 45. BEARING LB-1 (TEST BEARING END) AFTER 676.8 HOURS AT TWICE RATED LOAD

Failure progression tests of bearing FP-11 and FP-12 were conducted using new loader bearing LB-2. At the conclusion of the test of FP-12, LB-2 was disassembled and inspected (after 308.3 total running hours). Its condition was similar, although not as severe, as was LB-1 after 676 hours. (See Table 6 and Figures 46 through 48.)

Clearly, two bearings successively experiencing such severe fatigue damage were no fluke, especially in light of previous experience which indicated little if any spalling problems for several hundred hours.

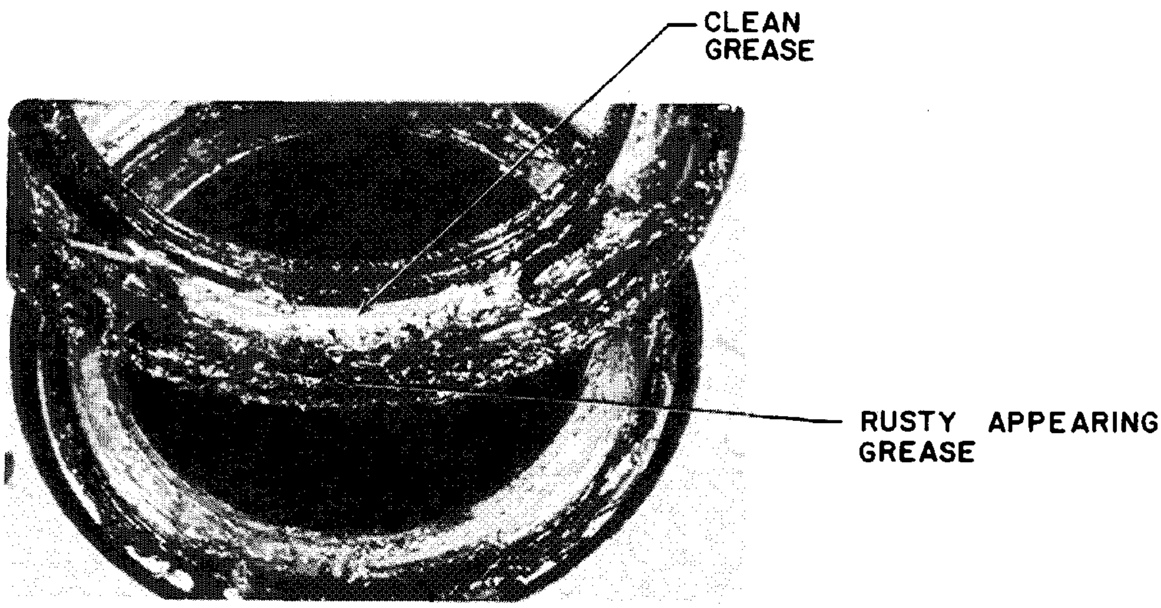
The two differences between the application of LB-1 and LB-2 on the subject test program and previous center loader bearings were:

a. LB-1 and LB-2 were water-cooled;

b. Load was applied to the bearing from a modified adapter through the water-cooling jacket rather than directly from a standard adapter.

Water cooling in itself is not considered to be the cause. The average water temperature was maintained generally between 100 and 110 ^{o}F, thus no significant clearance closing thermal gradients were present. Furthermore, the excellent condition of the grease indicated that local temperatures were not excessive, as would most likely be the case had internal clearance been lost (to the point of generating fatigue damage).

Instead, it is postulated that the method of applying load through the cooling jacket was the cause of the spalling problems. The standard adapter is designed so that the side frame load is picked up on two raised and crowned ribs and then distributed in a near uniform fashion over a relatively wide circumferential area of the cup as is shown pictorially in Figure 49a. When the cooling jacket was placed between the bearing and the adapter, the load was uniformly distributed over the contact area of the cooling jacket. However, the cooling jacket was too stiff to accommodate the clearance between the jacket bore and bearing O.D., as is shown in Figure 49b. This resulted in the load being concentrated along a line on the cup instead of being distributed over a wide area.



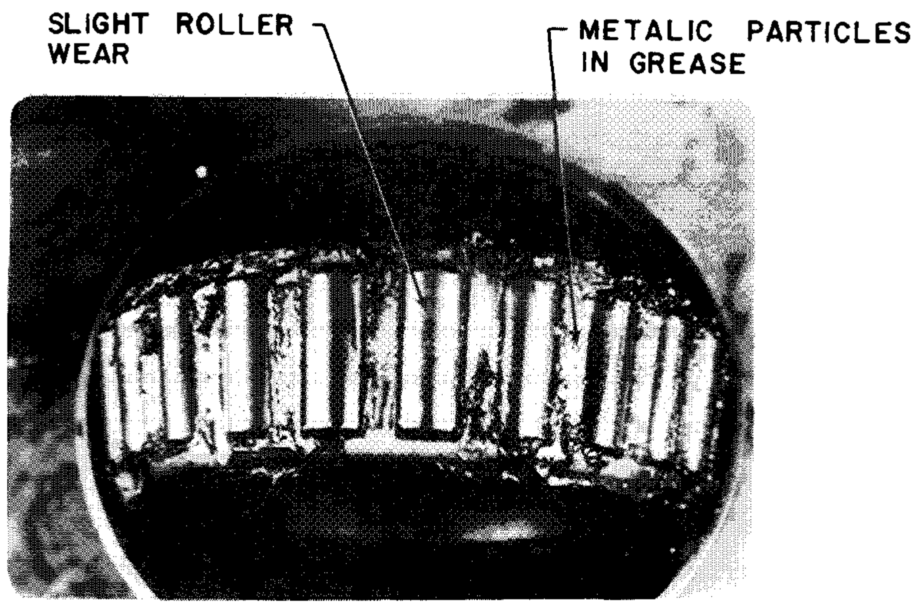


FIGURE 46. BEARING LB-2 (TEST BEARING END) AFTER 308.3 HOURS AT TWICE RATED LOAD

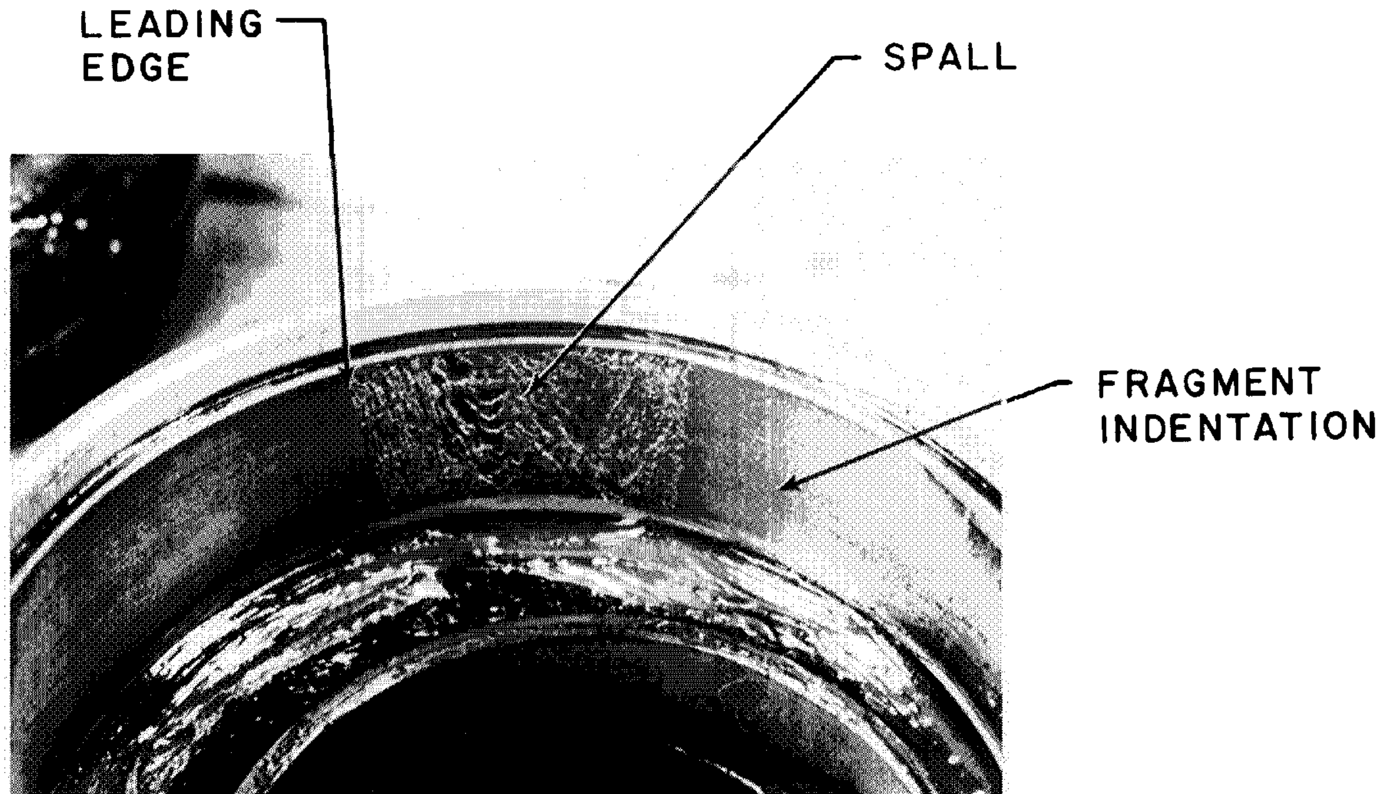
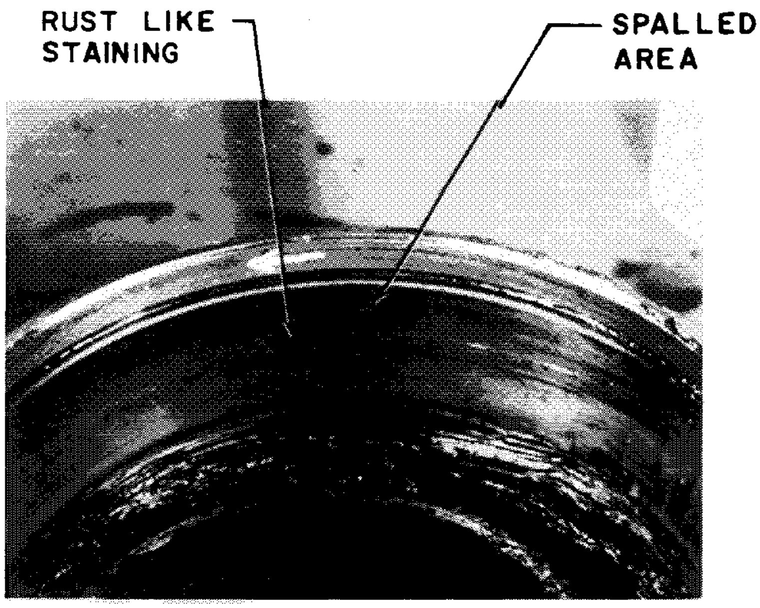


FIGURE 47. BEARING LB-2 (TEST BEARING END) AFTER 308.3 HOURS AT TWICE RATED LOAD



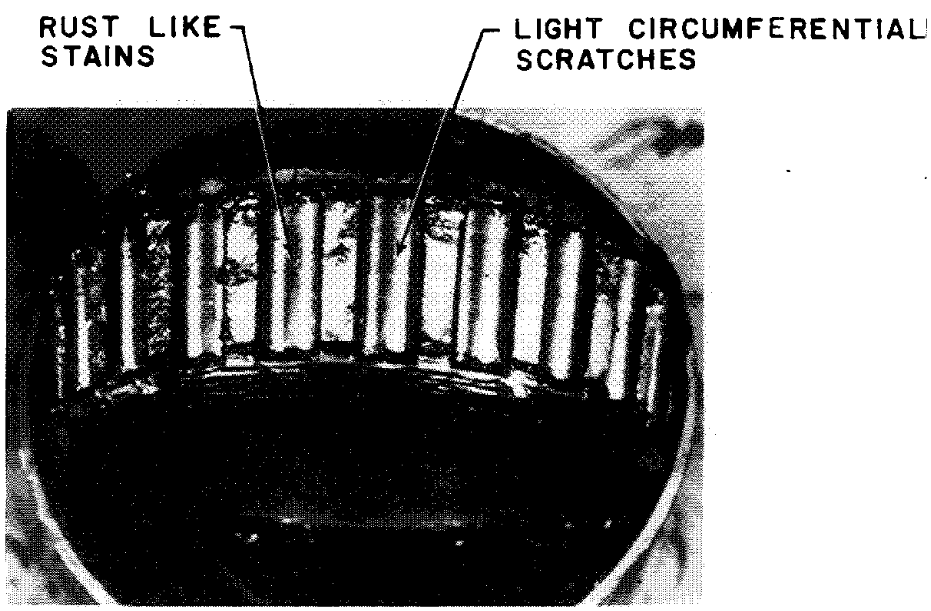
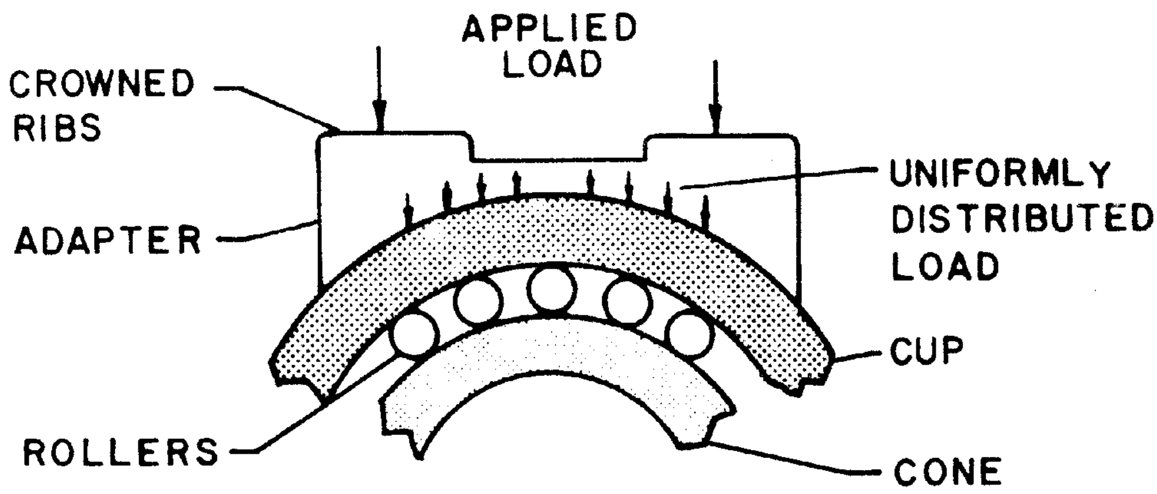
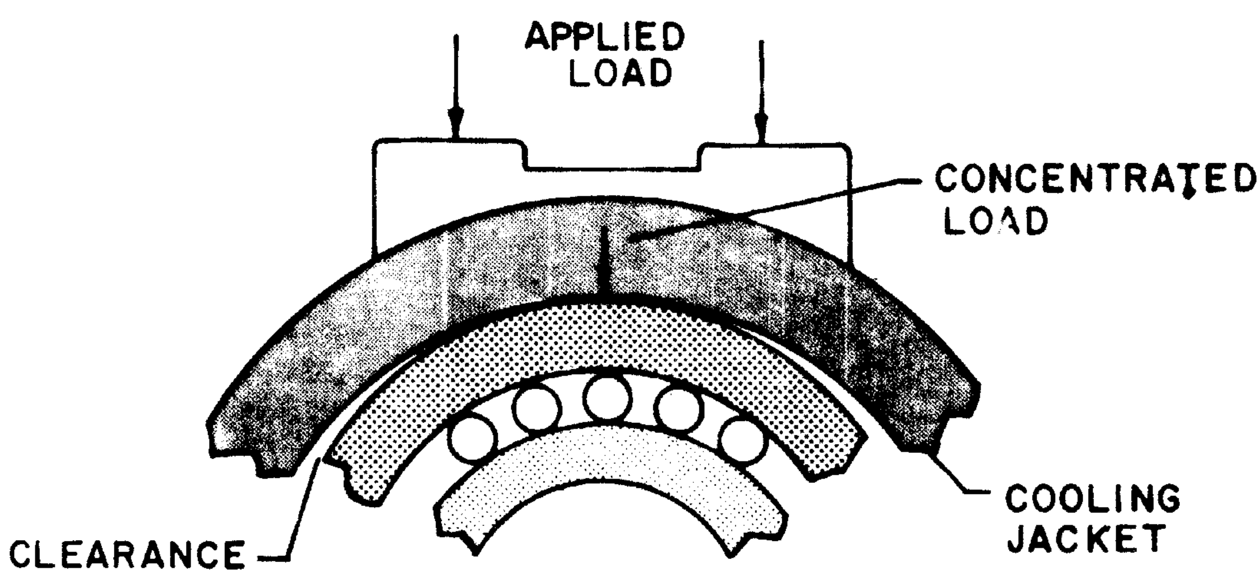


FIGURE 48. BEARING LB-2 (DRIVE END) AFTER 308.3 TEST HOURS AT TWICE RATED LOAD



a) WITH STANDARD ADAPTER



b) WITH COOLING JACKET

FIGURE 49. EFFECT OF COOLING JACKET ON BEARING LOAD DISTRIBUTION

Although the degraded loader bearings did not influence the progression of defects in the test bearings, they did materially influence the sound and vibration data. This problem is discussed further in Section 4.2.

4.2 SUMMARY OF MEASURED PARAMETERS

The instrumentation and data acquisition procedures used were described in Section 3.2. This section summarizes the thermal and vibrational behavior of the test bearings during the failure progression tests. In Figures 50 through 55, the test bearing's temperature, overall vibration level, and significant demodulated characteristic frequency amplitudes (normalized) are shown as a function of test time. Figure 56 summarizes all demodulated characteristic frequency amplitudes.

The points plotted in Figures 50 through 55 correspond to the times that demodulated spectral data were manually taken and transferred to the computer for analysis and storage. These points were then simply connected with straight lines. Thus, variations between demodulated data recording intervals are not shown.

In Figures 50 through 55, the temperature is the difference between the bearing temperature measured at the center of the cup and the temperature of the outside air that was directed on the bearings (thermocouple locations 3B and 3E - see Figure 4). It should be noted that thermocouple location 3B was not necessarily the hottest location, as was previously shown in Figure 15 with regard to bearing FP-6. Overall vibration data is that taken from the accelerometer mounted on the test bearing and is high passed at 10 KHz. The demodulated data shown in Figures 50 through 55 has been normalized by dividing the raw demodulated amplitude by the overall (undemodulated) vibration amplitude. Sound data, both high frequency and demodulated, correlated well with vibration data. Since there was essentially no difference in general character between the two, only vibration data has been presented in Figures 50 through 55.

It is seen in Figures 50 through 55 that except for bearing FP-6 (Figure 51), the temperature rise of each bearing was generally between 40 and 60°F and did not vary significantly throughout the test. The sharp valleys or low points in the temperature plot of FP-6 (Figure 51) were points taken shortly after restarts during the test before the bearing had reached "steady state." Thus, they are not indicative of condition or actual fluctuations. Had bearing FP-6 not been routinely stopped on four different occasions during the test, it would have shown a smoother temperature rise characteristic early in the test and may have failed somewhat sooner.

Overall vibration levels for all bearings were generally in the 10 to 60 g range except for FP-6 just prior to the final failure process and for FP-7 throughout its entire test.

Except for bearing FP-6, which experienced a very dramatic degradation process, neither temperature rise nor overall vibration (or sound) was a particularly good indicator of bearing condition or change in bearing condition. The high temperature rise shown by FP-6 resulting from the jamming rollers certainly was a precursor of the final failure process (loss of adequate lubrication) which showed up as a very high temperature rise during the final few hours. It should also be noted that a large increase in vibration level preceded the sharp increase in temperature by several hours.

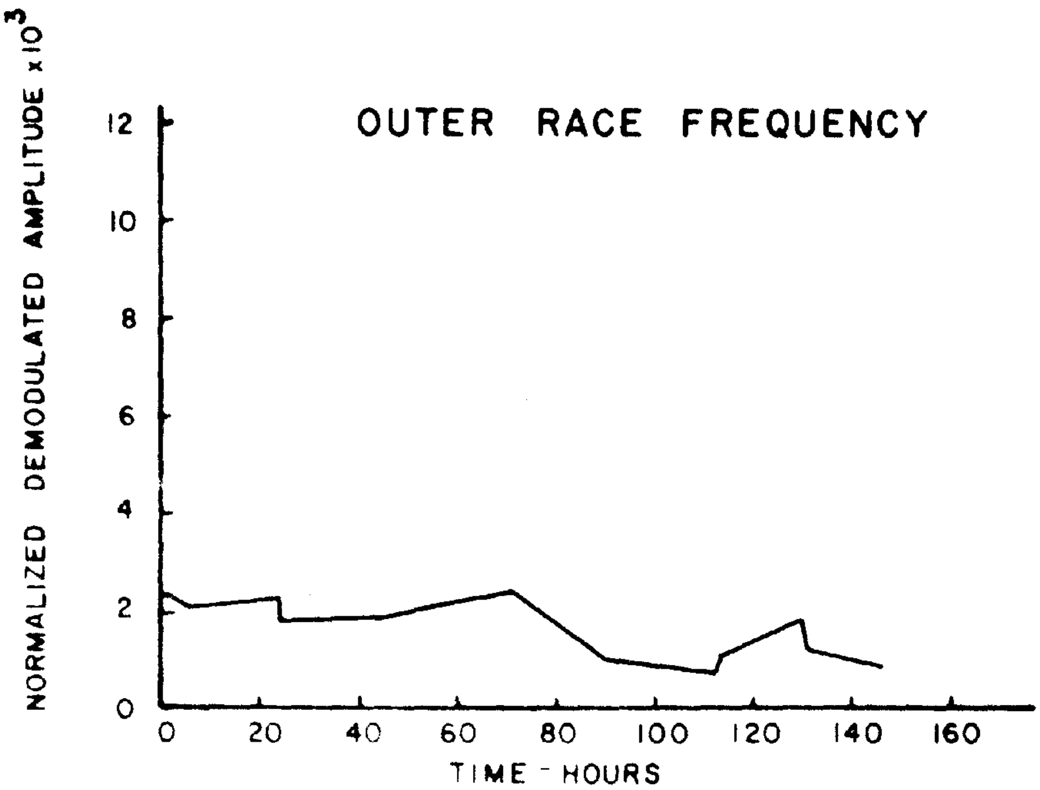
It was not expected that either temperature or overall vibration level would be a particularly discriminating indicator of anything but gross changes in bearing condition. However, the demodulated vibration and sound amplitudes were expected to show reliable degradation and failure trends. Raw demodulated data, for any given bearing, gave a reasonably good indication as to which component (outer race, inner race, roller, cage) was the most significantly damaged. However, in comparing one damaged bearing against another, the correlation with respect to degree of damage was poor. Thus, normalizing procedures were employed.

Figure 56 shows the average of all demodulated data points for each bearing tested, normalized with respect to the maximum average amplitude for that particular bearing. Here it is seen that the predominant frequency for bearings FP-8, 10, 11, and 12 was 96 HZ, which was the roller-outer race passing frequency and was indicative of the cup spalls or cracked and broken cups that these bearings possessed. Bearing FP-6 showed a predominance of cage frequency (4 HZ) which was indicative of the severely rubbing cage. Bearing FP-7 was uncharacteristic in that it showed relatively high amplitudes at all characteristic frequencies and had an abnormally high overall vibration level. (See Figure 52.) The relatively high amplitudes of 88 HZ and 116 HZ were to be expected because of the roller and inner race spalls. The small quantity and dried nature of the grease in the cones may have contributed to the high amplitude at cage frequency (4 HZ) as well as the high overall levels. The action of the seal cases rotating and rubbing in the cup also contributed to the high overall level. High outer race characteristic frequency (96 HZ) amplitudes in FP-7 can only be explained of the spalling degradation that was occurring in the center loader bearing (LB-1) at that time.

The demodulated amplitudes as a function of time shown in Figures 50 through 55 were normalized by dividing the demodulated amplitude (at the frequency of interest) by the overall undemodulated amplitude occurring at the same time (data point). The normalized 96 HZ (outer race) frequency amplitude is plotted for each bearing in Figures 50 through 55 as this frequency tended to dominate in all tests and because all but FP-8 had some form of outer race defect. Amplitudes at other frequencies are plotted for FP-6 and FP-7 because their amplitudes were of the same order as or higher than the 96 HZ amplitude.

Except for bearing FP-6, little can be concluded with regard to the signature at 96 HZ indicating an outer race defect because of the effects of the degrading loader bearings. With regard to FP-6, it should be noted that the outer race frequency decreased (relative in the overall vibration) as the temperature increased. This is probably a result of a reduction in internal clearance. It should also be noted that the major contributor to the high vibration level late in the test of FP-6 was cage frequency and that it preceded the dramatic temperature rise.

Cage rubbing had occurred throughout the test, as evidenced by its more than preceptible level from the beginning, and was probably the cause of the generally high temperature level throughout the test. It is postulated that somewhere between 40 and 58 hours (the point where cage frequency amplitude increased drastically), sufficient cage wear had occurred to permit cage eccentricity to increase to the point where the rollers could roll and "wedge" under the separators. When this occurred, cage frequency amplitudes increased markedly, rollers became skewed causing them to rub on the race flange and temperatures increased to the point where all the available oil evaporated from the grease. Once the oil was essentially gone, the wear and binding process accelerated and compounded the thermal runaway situation.



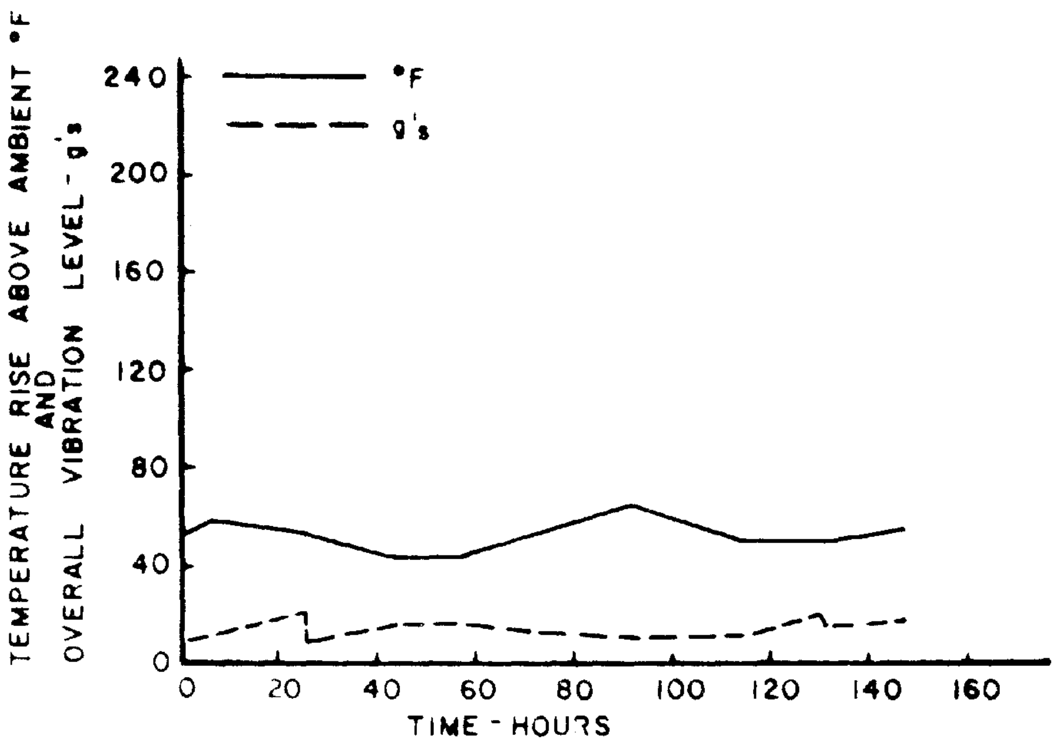


FIGURE 50. SUMMARY OF TEMPERATURE AND VIBRATION DATA, BEARING FP-8

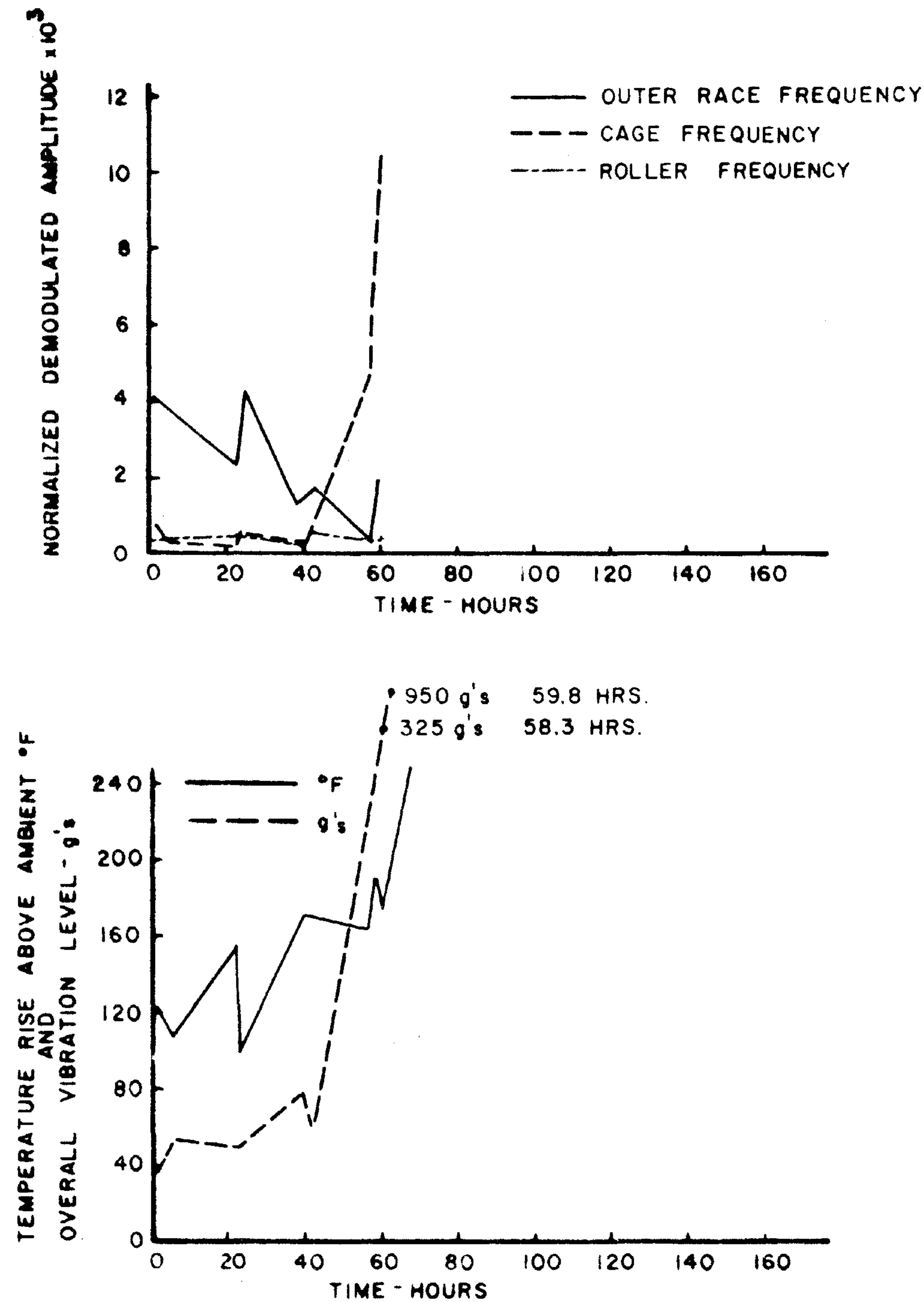


FIGURE 51. SUMMARY OF TEMPERATURE AND VIBRATION DATA, BEARING FP-6

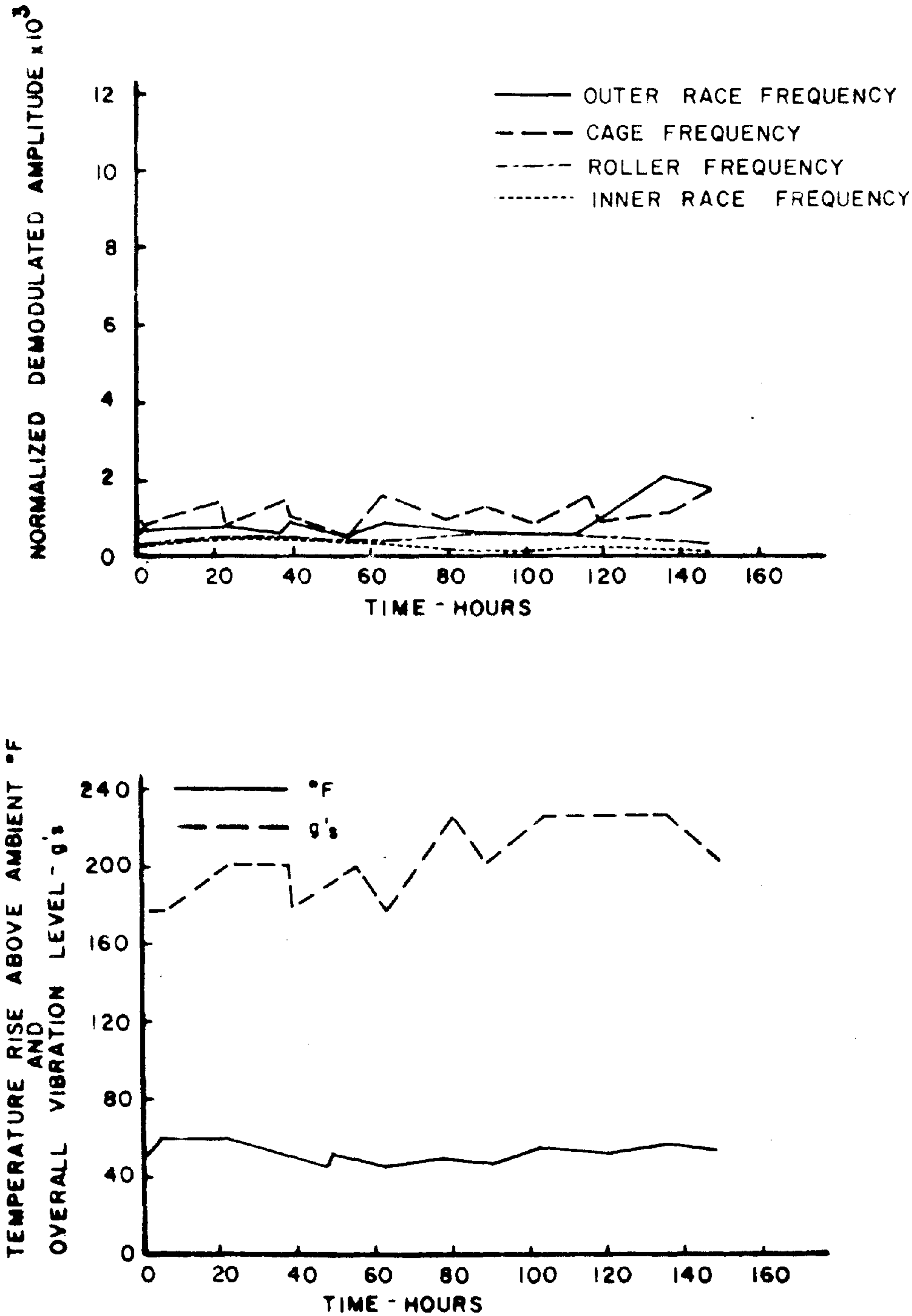


FIGURE 52. SUMMARY OF TEMPERATURE AND VIBRATION DATA, BEARING FP-7

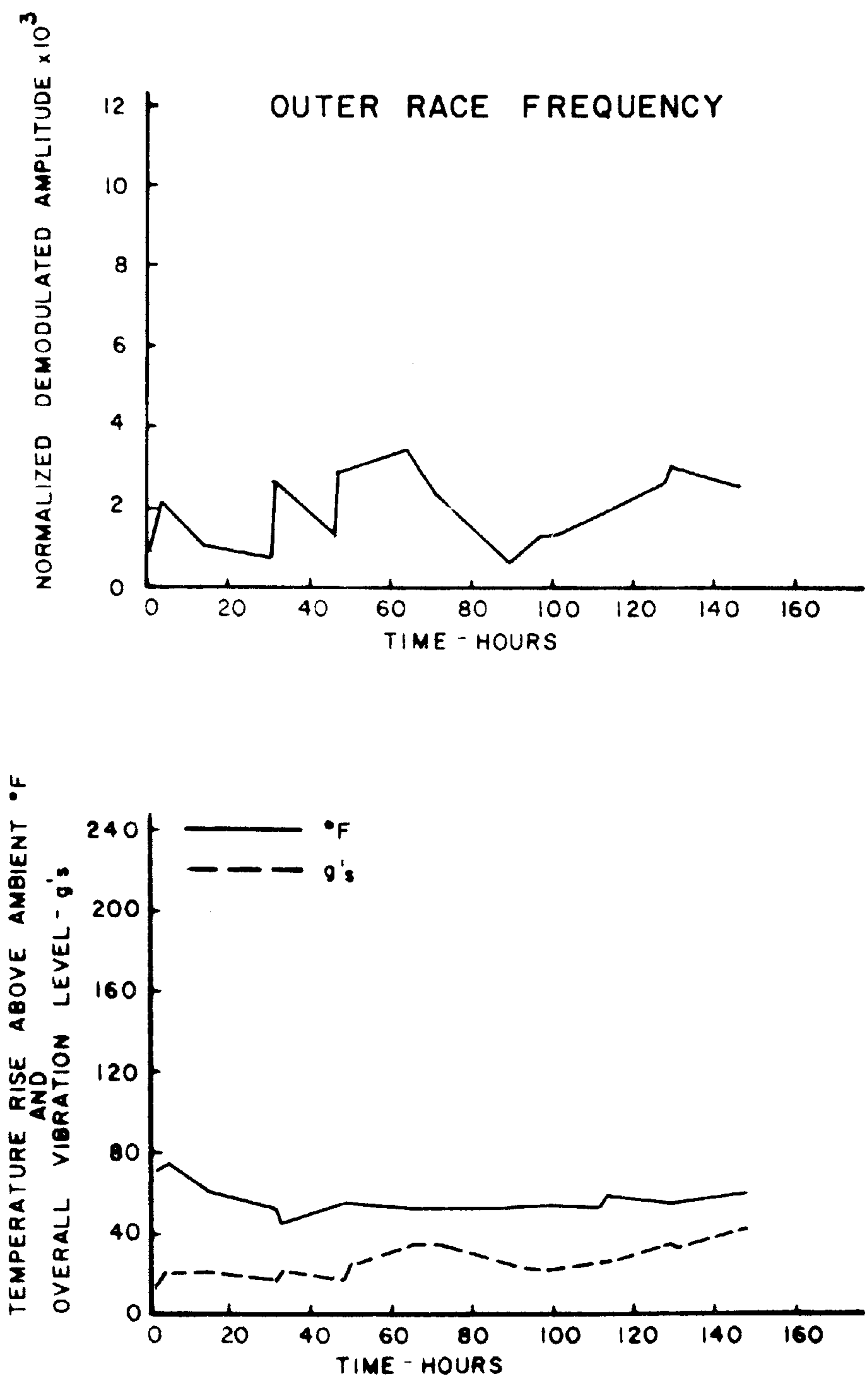
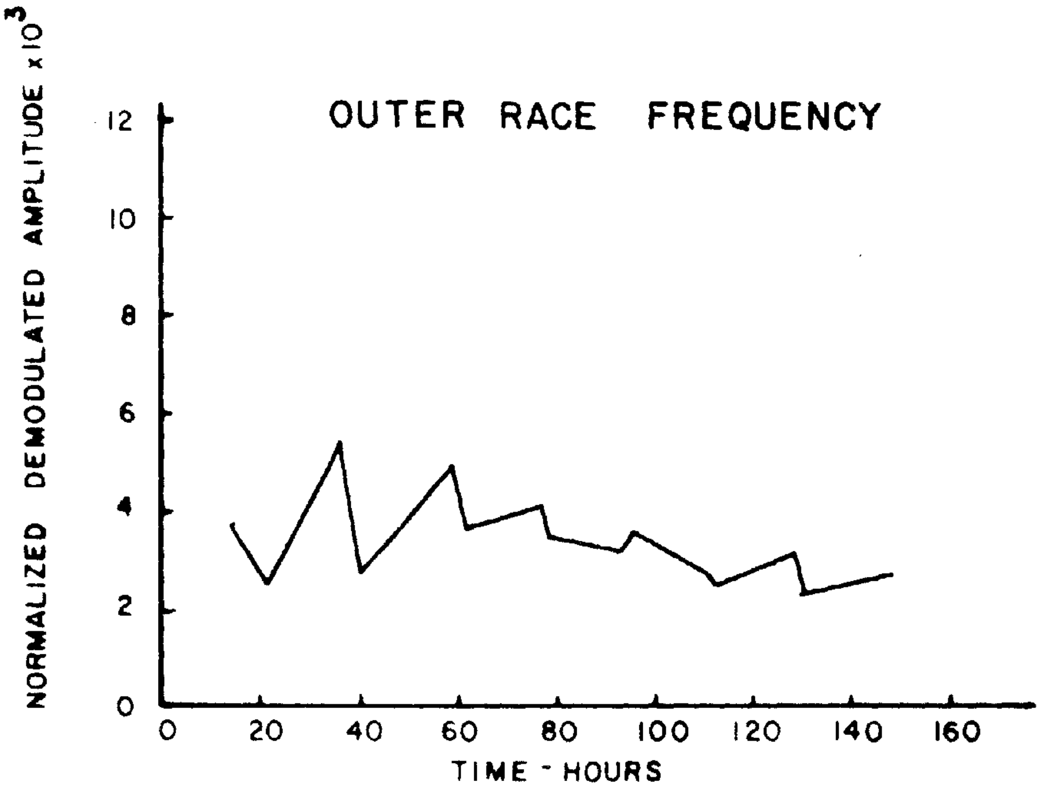


FIGURE 53. SUMMARY OF TEMPERATURE AND VIBRATION DATA, BEARING FP-10



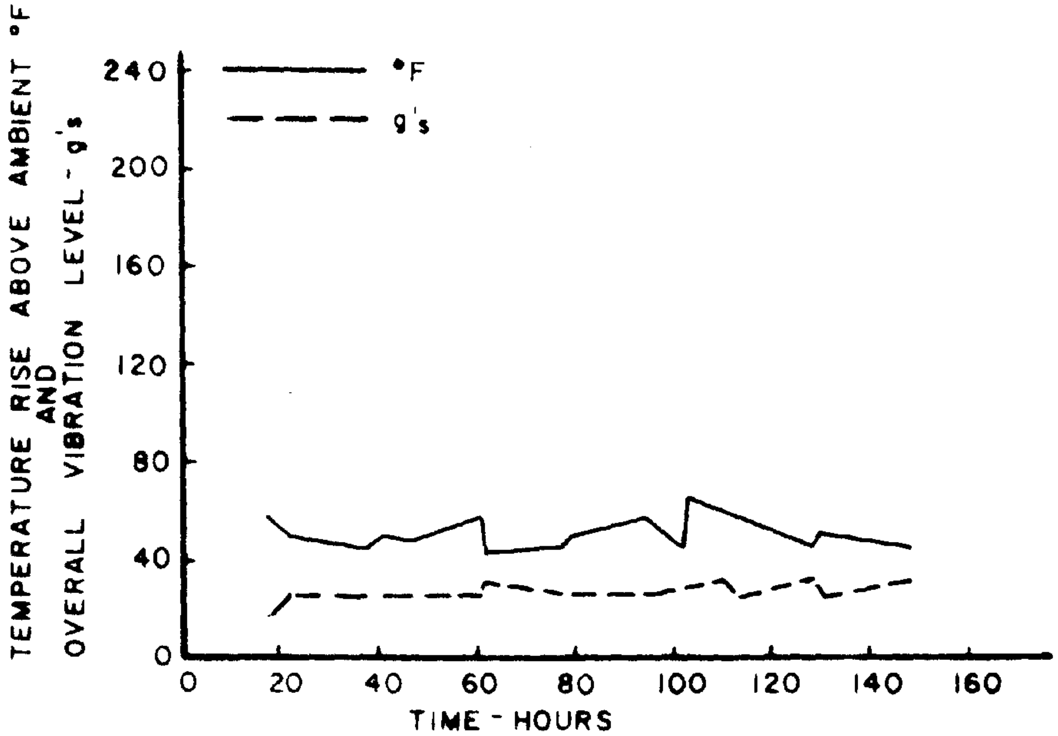
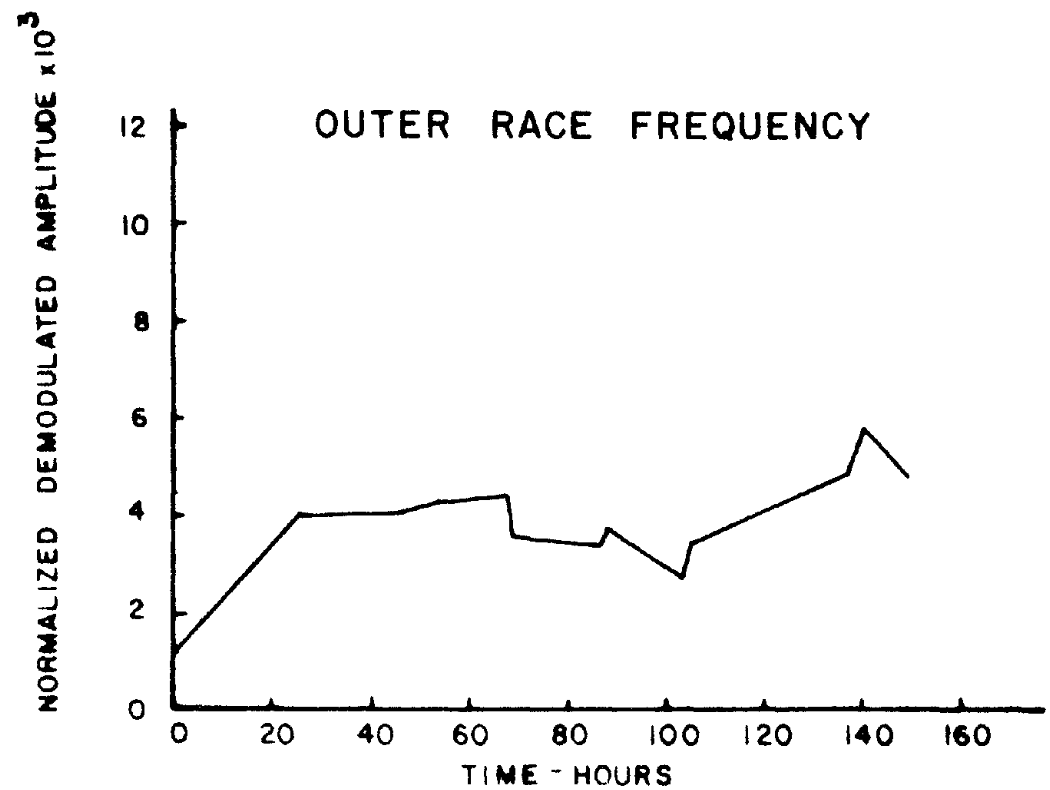


FIGURE 54. SUMMARY OF TEMPERATURE AND VIBRATION DATA, BEARING FP-11



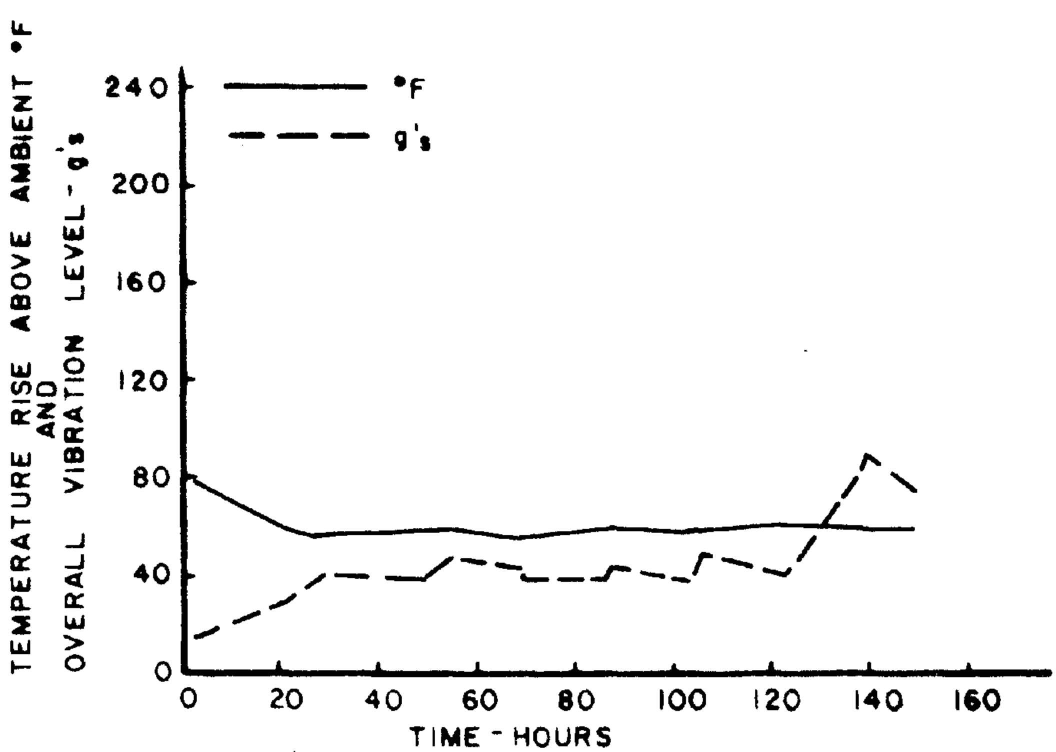


FIGURE 55. SUMMARY OF TEMPERATURE AND VIBRATION DATA, BEARING FP-12

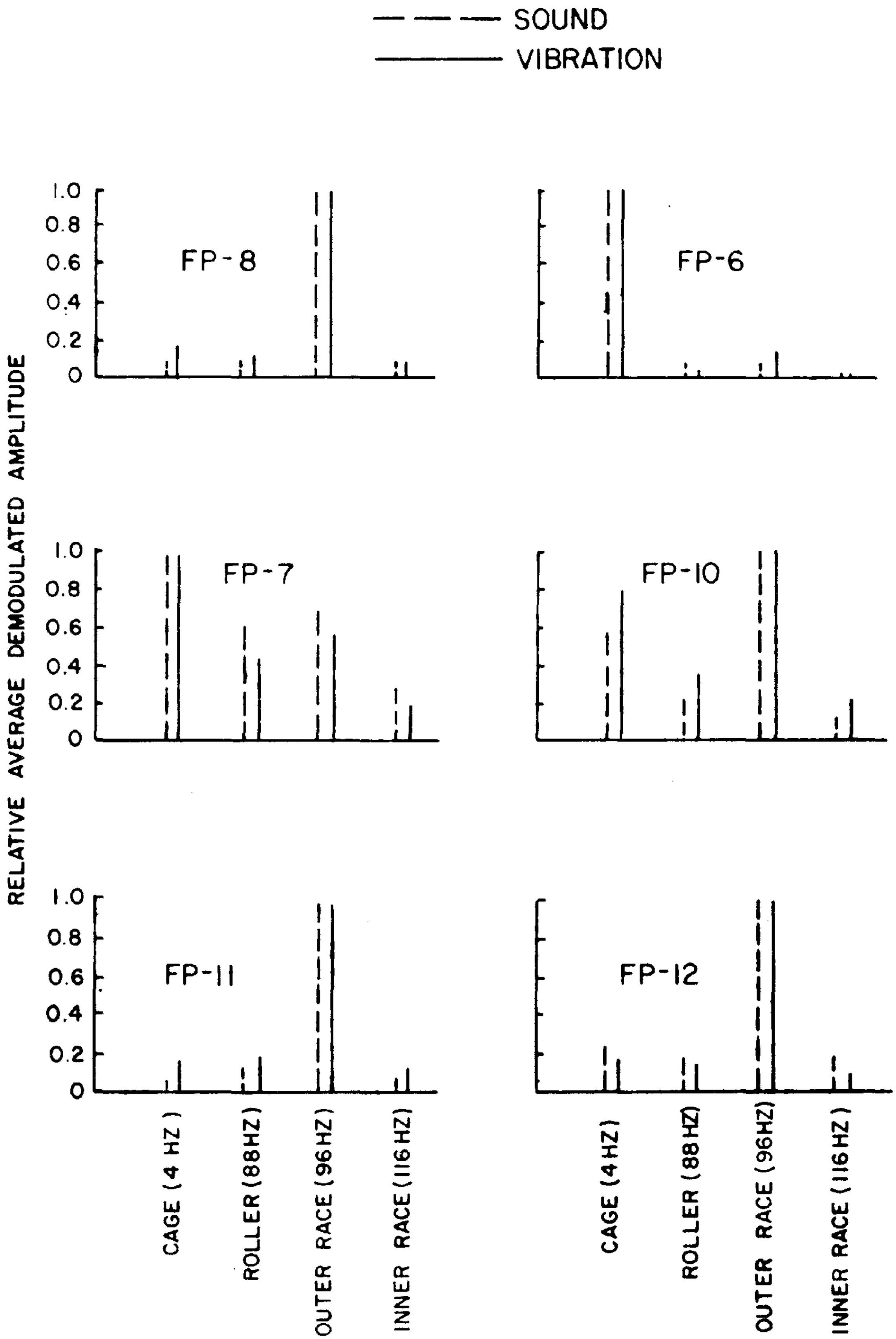


FIGURE 56. RELATIVE CHARACTERISTIC DEFECT FREQUENCY AMPLITUDES

4.3 EVALUATIONS OF GREASE SAMPLES

Small samples of grease were taken from each bearing before and after the planned 150-hour test. These were taken from one of the seal cases near the cone surface and were approximately a half of a cubic centimeter in volume. Each was subjected to an oil content test and the residue (after oil dissolution) from each was microscopically examined.

The results of the oil content test are summarized in Table 7. The most significant information gained from the oil content tests was the fact that the oil content was very near that of new grease, no matter how deteriorated the grease may have appeared. It is interesting to note that the oil in the grease taken from the seal case next to the cone that failed on FP-6 still had a reasonably high oil content. Thus, it can be concluded that grease adjacent to a highly distressed component cannot provide sufficient lubrication to prevent catastrophic failure once the oil in the cone is depleted.

The microscopic examination revealed red or rust-like particles in the samples from FP-8 and FP-12. Both of these bearings had been water etched. The microscopic examination revealed no other new information.

4.4 SUMMARY OF DEFECT PROGRESSION PROCESS

A laboratory failure progression scenario was suggested in Section 2.0 (Figure 1) as part of the rationale for the selection of the particular bearings that were tested during the course of the subject program. Figures 57 through 62 summarize the course of events that actually transpired during the tests for each bearing. These figures are in the same format as Figure 1.

From these figures, as well as the preceding discussion, the following observations with regard to the failure or degradation process can be made.

1) Spalls grew in size and their debris created fragment indentation (FP-8, 7, 10, LB-1, 2).

2) Water etching did not lead to spalls (FP-8, 12).

3) Brinells did not become spalled, although FP-11 might have with additional running since small fissure cracks had developed at the edge of the brinell. Also FP-10 developed a large spall

TABLE 7

OIL CONTENT FROM
FAILURE PROGRESSION TEST BEARING GREASES

Bearing Number Bearing Number	Oil Content (% by Weight)	Macroscopic Examination Comments
FP-8 Before	80.2	
After	71.5 (1)	Some magnetic particles
FP-6 Before	84.7	Soft and Soupy
After	71.4 (1)	Contained metal slug and large number of magnetic particles
FP-7 Before	79.3	
After	63.9 (1)	Large number of magnetic particles
FP-10 Before	84.4	A few magnetic particles
After	82.5	A few magnetic particles
FP-11 Before	83.3	Some magnetic particles
After	82.5	" " "
FP-12 Before	79.1	" " "
After	79.4	" " "
New Grease	81.4	

(1) Abnormally low oil content reading probably due to large relative weight of metallic debris.

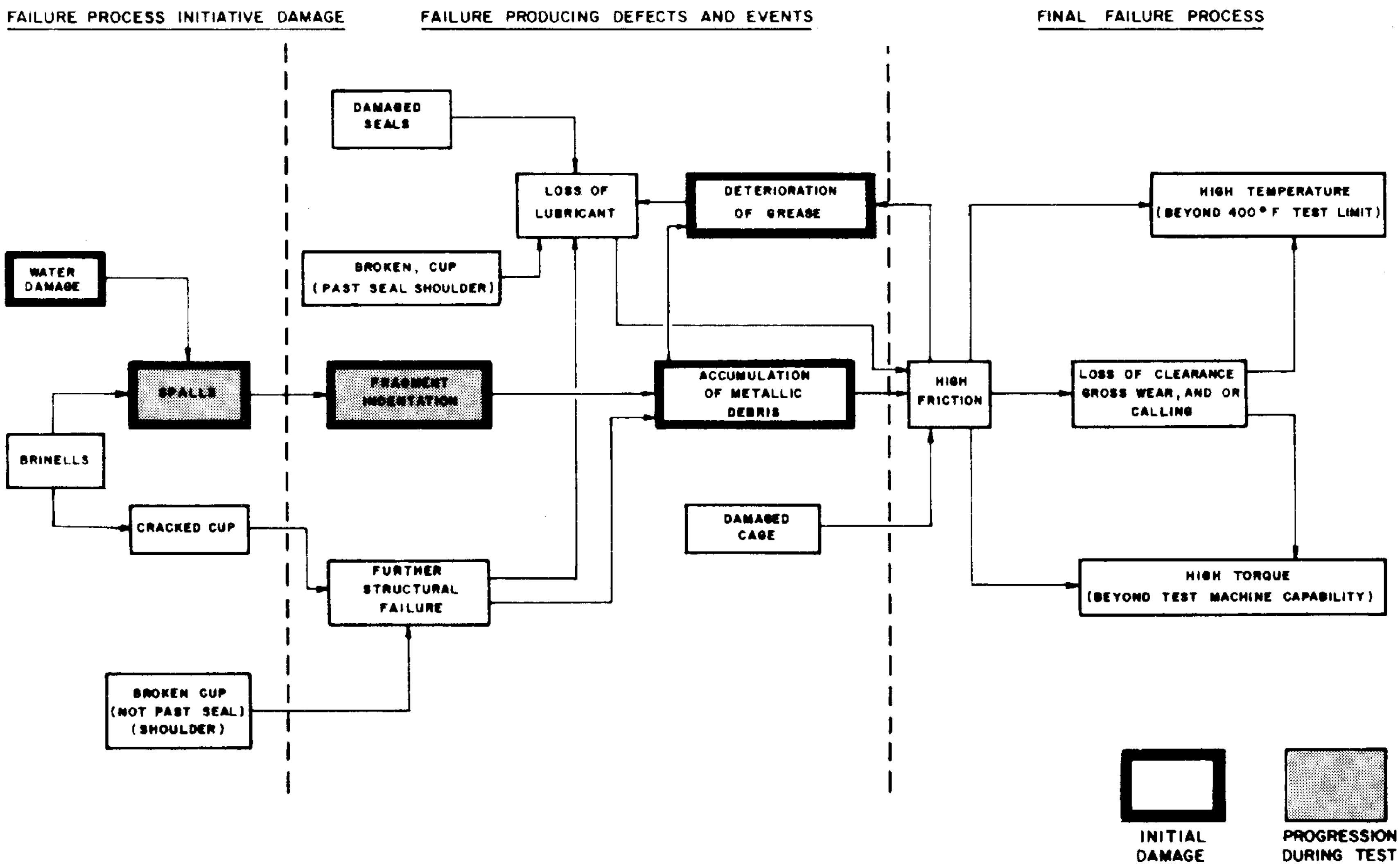


FIGURE 57. DEFECT PROGRESSION, BEARING FP-8

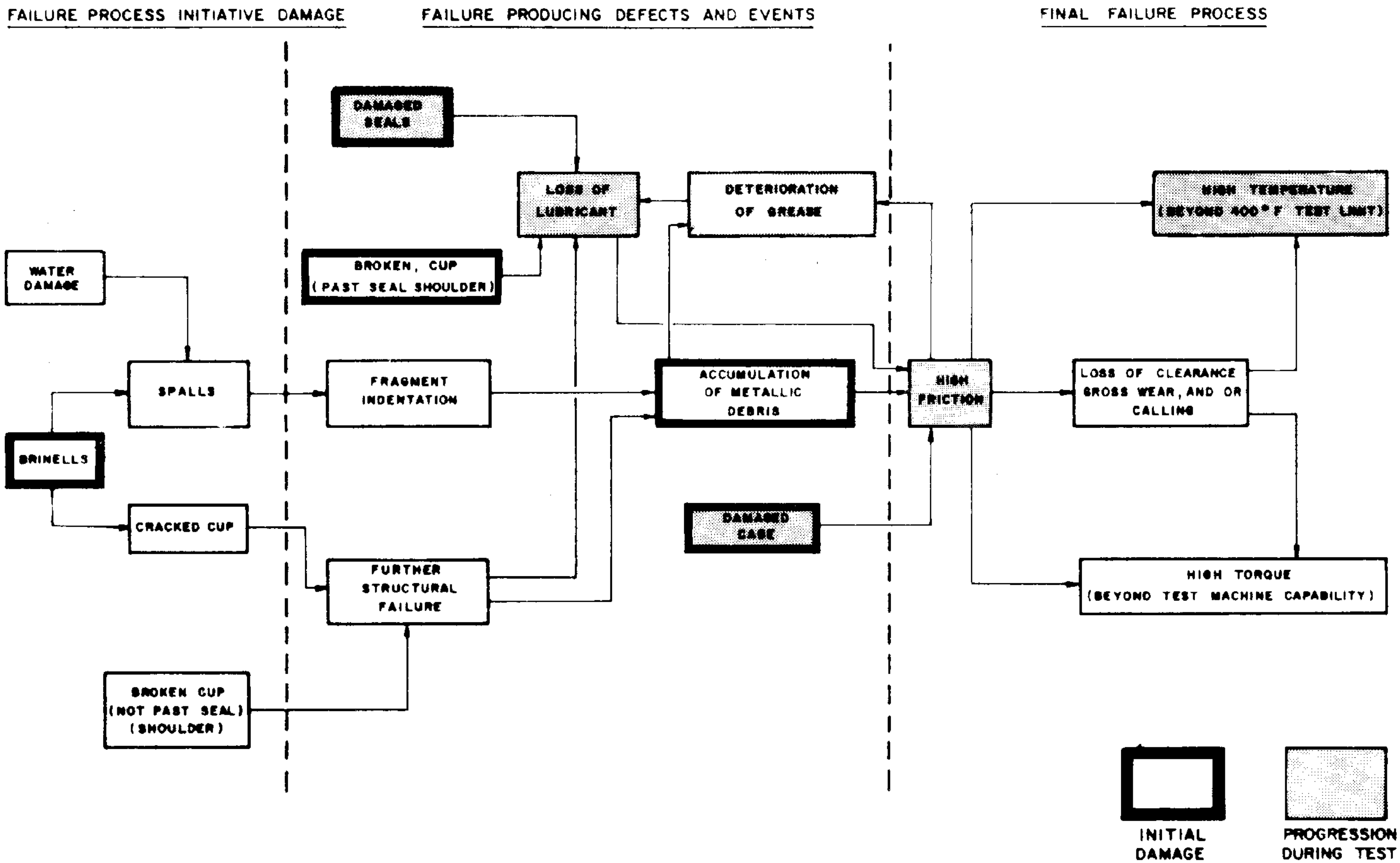


FIGURE 58. DEFECT PROGRESSION, BEARING FP-6

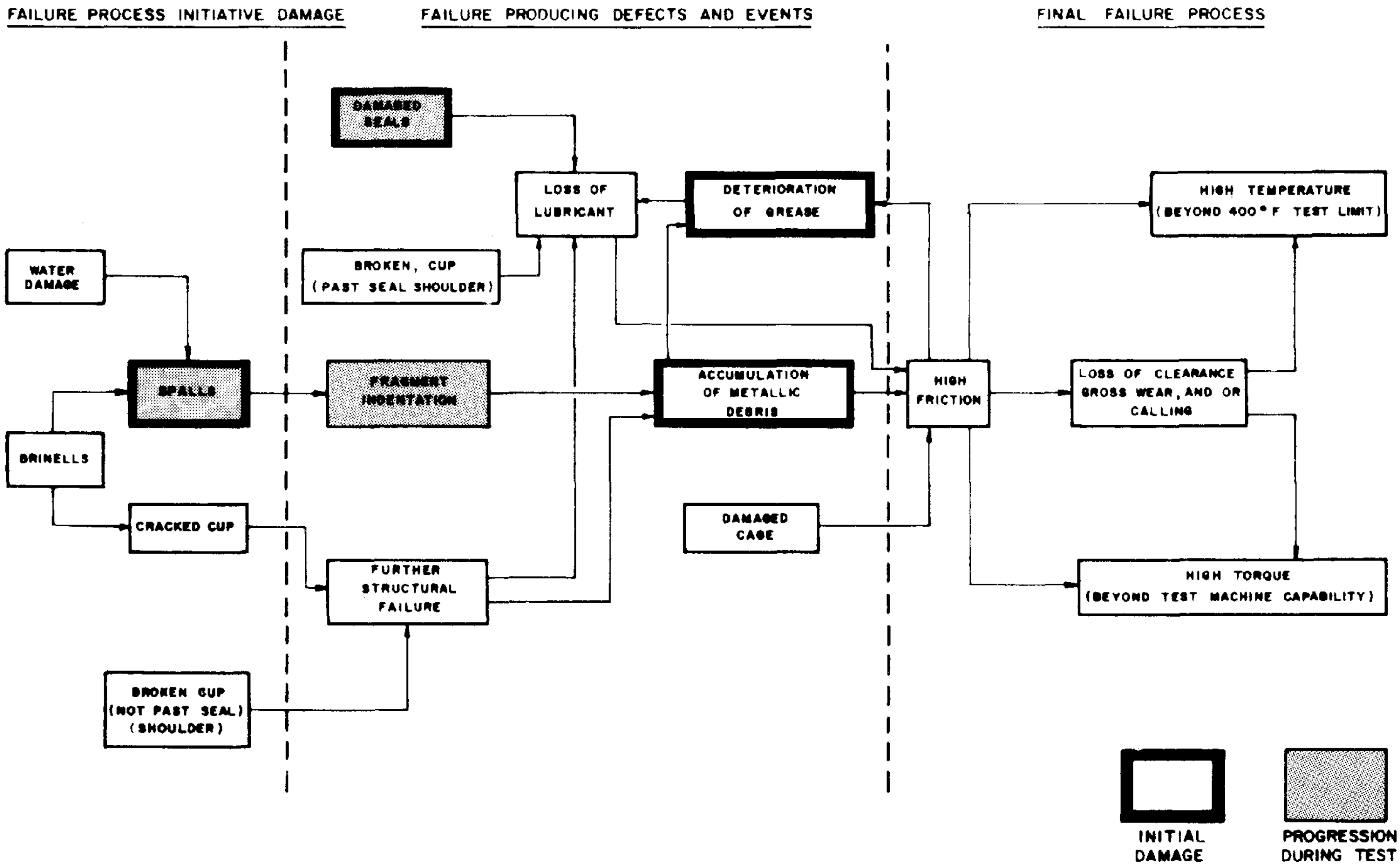


FIGURE 59. DEFECT PROGRESSION, BEARING FP-7

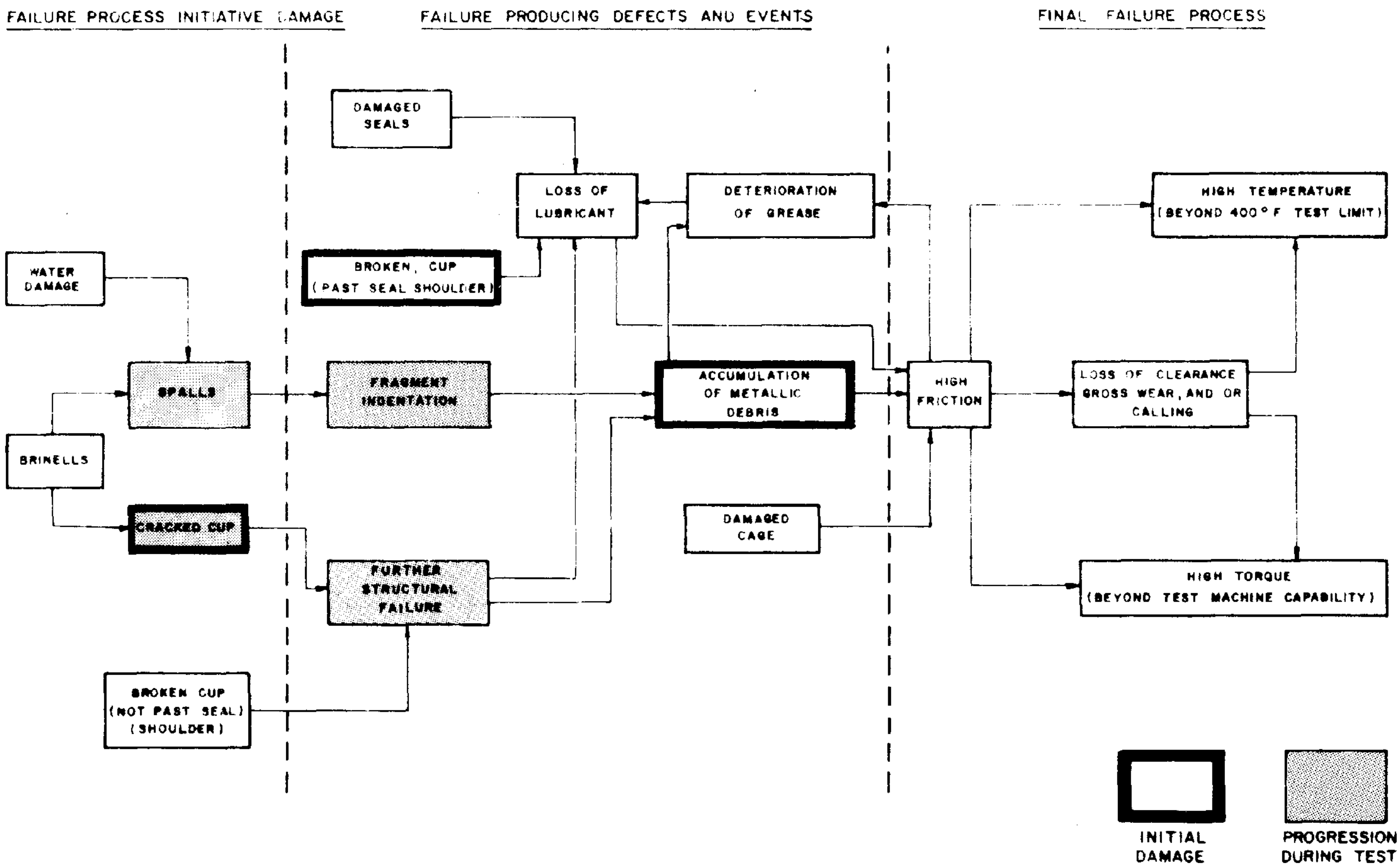


FIGURE 60. DEFECT PROGRESSION, BEARING FP-10

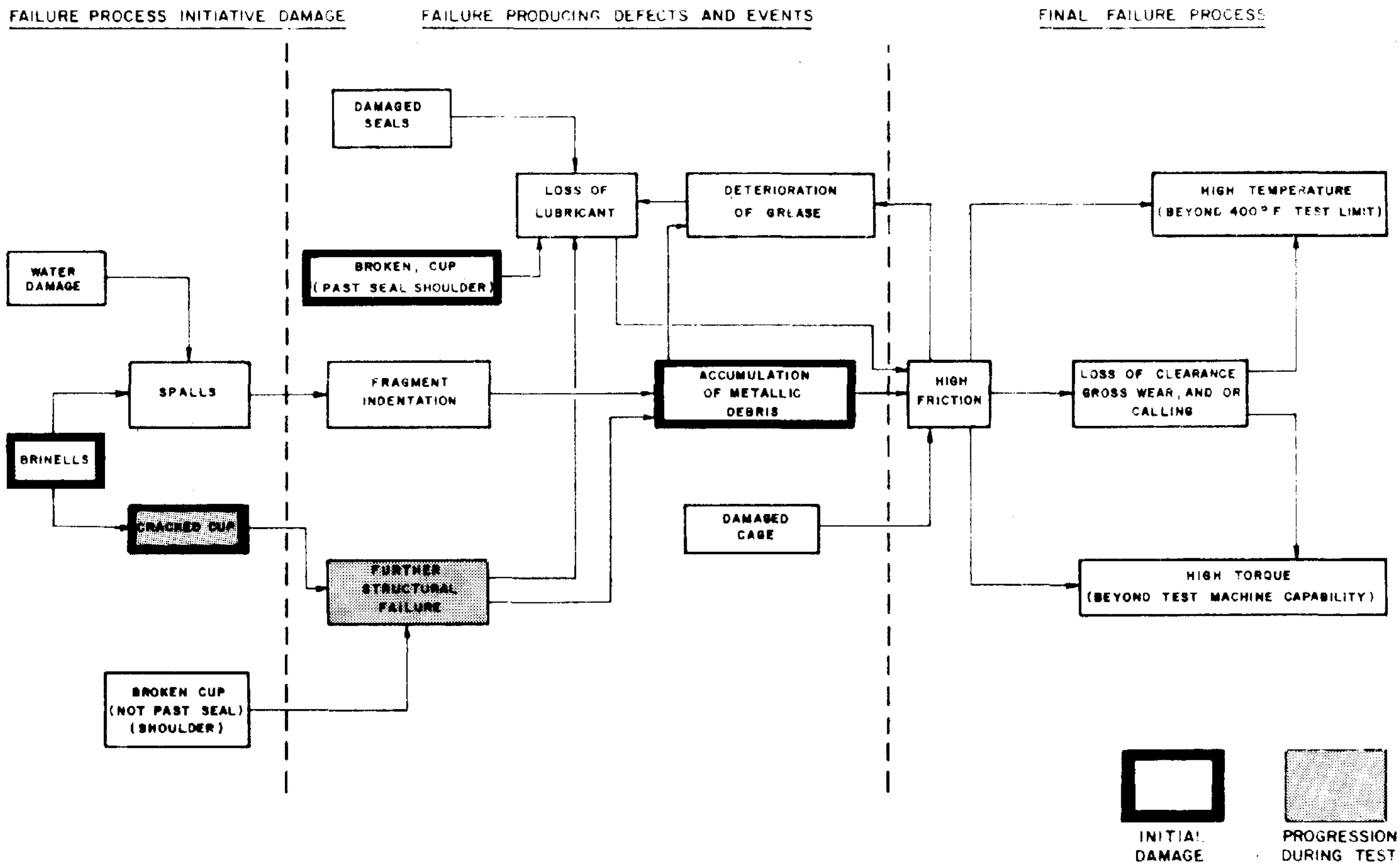


FIGURE 61. DEFECT PROGRESSION, BEARING FP-11

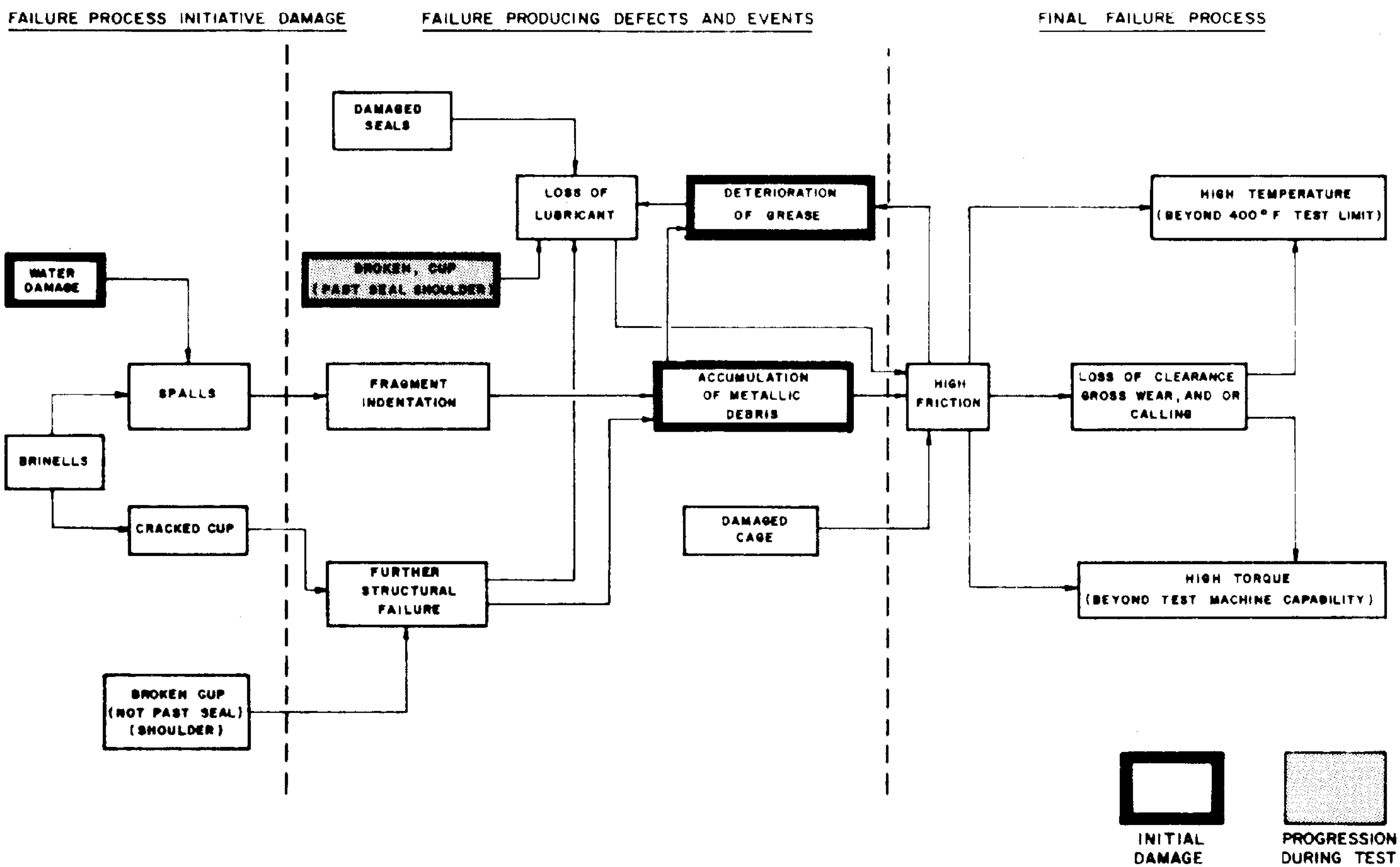


FIGURE 62. DEFECT PROGRESSION, BEARING FP-12

in an originally dented and cracked race area. This dent, however, was not what would be classically termed as a brinell.

4) Cracks in cups did enlarge and new cracks developed (FP-11). Both cracked bearings had an appearance that would lead one to think that additional running would result in further structural damage--possibly to the point where a sizeable piece might break completely out of the raceway.

5) In only one case did damaged seals lead to loss of lubricant (FP-6). However, this loss was small and did not contribute significantly to the failure process.

6) Likewise, only FP-6, of the four bearings tested with cups broken past the seal shoulder, leaked grease. It should be noted, however, that all bearings with damaged seals or broken cups were initially low on grease so that the pumping action to expell the grease was minimal. The grease remaining, however, was entirely adequate to perform the required lubrication process (providing there was no gross heat generating defect as in the case of FP-6).

7) Deteriorated grease in terms of dried or watery appearance (FP-7, 8, 12) had little effect, except to possibly produce noise (FP-7).

8) Debris in the grease appeared to have no significant effect on the bearing. In fact, the size of the metallic particles was noticeably smaller after the test than before.

9) Of the defects encountered, the damaged cage (FP-6) was the only one that was severe enough to produce bearing failure within the allotted test period. It should be noted, however, that it was probably the combination of the damaged cage <u>and</u> damaged seal that produced the events which ultimately lead to failure. (See discussion in Section 4.1.2.)

4.5 EFFECTIVENESS OF ONBOARD WARNING DEVICES

It was originally intended that three different onboard warning devices be functionally evaluated on the subject program. These were:

1) An adapter temperature sensing device (the DOT-STAR* device) designed to activate when the sensing element reaches 240°F.

2) An axle temperature sensing device designed to activate when the end of the axle reaches 240°F. This device emits ultrasonic energy upon actuation.

3) A temperature sensing end cap bolt designed to activate when the end of the bolt reaches 200°F. This device emits ultrahigh frequency RF energy upon activation.

The latter device, which is being developed on a separate contract under DOT sponsorship, was not available in time to be evaluated during the subject test program, and therefore will not be discussed further here.

Several of the DOT-STAR devices were made available to the subject program for evaluation. These included the side-frame mounted containment casings, the NITINOL based firing pin assemblies, pyrotectic thermal batteries, alternate normally open electrical switches which could be used in place of the thermal batteries for the subject test program, and sensor housings which house the firing pin assembly and battery (or switch).

The operation of the DOT-STAR sensor is described briefly below. The tip or end of the sensor housing is held in contact with the adapter by means of a spring load. Bearing heat is conducted through the adapter to the sensor surface and thence to the firing pin assembly. When the NITINOL clamping ring reaches 240°F, it releases the spring loaded firing pin which in turn activates the thermal battery. The electrical output of the battery can then be used

---

*System for Train Accident Reduction developed for the U.S. Department of Transportation by the Naval Systems Weapons Center.

to activate another device to ultimately warn the train crew of the hot bearing situation.

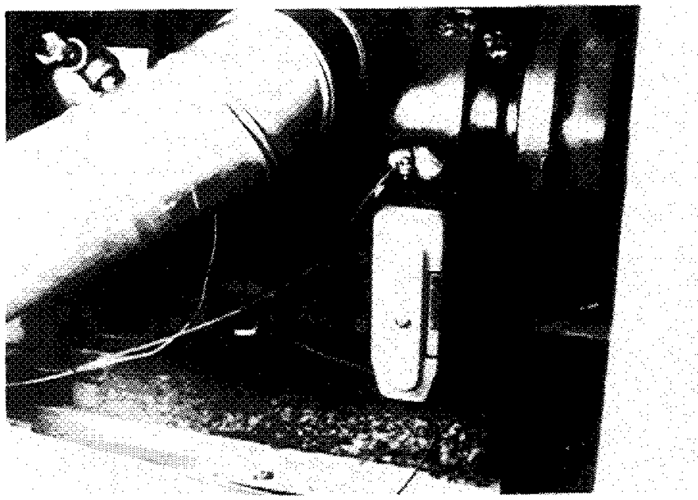
For the subject tests, the battery was not used. Instead the device was set up so that the firing pin actuated the electrical switch and continuity was sensed. (See Figure 3.) The frame mounted containment housing was mounted to an angle which in turn was mounted to the test rig base to simulate the actual railcar installation. (See Figure 63.) The same hardware was used during the tests of all six bearings.

One axle temperature sensing, ultrasonic sound emitting, device was made available by its inventor in time to be installed for the second test series (bearing FP-6). This device attaches to the end cap by means of the center hole which normally accommodates the grease fitting. The device consists of the following major functional components:

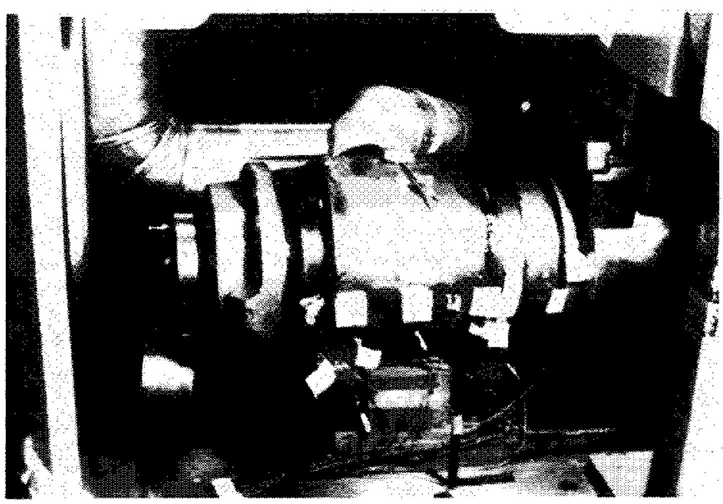
1) A hollow central shaft which is screwed into the end cap center hole and rotates with the axle. This central shaft contains the temperature sensing element which is spring loaded against the axle, a low temperature melting alloy, and a spring loaded central pin which actuates a latch mechanism when excessive temperatures are reached (when the alloy melts).

2) A two-piece housing assembly mounted to the central shaft on ball bearings which protrudes from the end of the axle. Under normal operating conditions, the housing assembly is keyed to the central shaft and rotates with it.

3) A latch assembly within the two-piece housing assembly which keeps the two pieces together and keys the housing to the central shaft. When the latch is activated by the central pin in the shaft, it allows the outer portion of the housing to swing away from the inner portion on a hinge between the two halves; and it removes the pin (key) connecting the housing assembly to the shaft. With the outer half of the housing swung away (and down) from the inner half (Figure 63), a restraining movement is imposed upon the inner portion of the housing which allows the central shaft to rotate



DOT-STAR SENSOR



ULTRASONIC EMITTING DEVICE
(SHOWN AFTER ACTUATING ON TEST OF FP-6)

FIGURE 63. ONBOARD SENSORS

relative to the housing assemblies on the internal ball bearings. In other words the housing stops rotating with the axle.

4) A cam-actuated hammer/anvil assembly which is activated by the relative motion between the housing and central shaft. The anvil is tuned to be resonant in the 40 to 50 KHz range. It is this ultrasonic sound that could be sensed by a trackside ultrasonic detector.

For the laboratory setup, a transceiver and RF receiver were employed to sense the state of the ultrasonic sound emitting device. (See Figure 3.)

Only FP-6, of the six bearings tested, reached temperatures that should have triggered the alarm devices. The DOT-STAR device did not trigger at any time during the elevated temperature excursions experienced by FP-6. This fact is not surprising however in that the adapter only reached a temperature of 220°F at the time that the maximum bearing cup temperature exceeded 390°F.

At the conclusion of the test series (following the test of FP-12), the DOT-STAR device was checked to determine if it was functional. This was done by clamping the device in a manner such that the sensing tip was held against a 1/4-inch-thick steel plate and by heating the plate with a propane torch. Both the plate temperature adjacent to the sensing tip and the sensing tip temperature itself were measured. The device activated when the sensing tip reached 245°F; thus, it functioned as designed (design point is 240°F at the NITINOL restraining ring which is inside the sensing tip). The actual plate temperature, however, exceeded the limit (400°F) of the measuring equipment being used. It is estimated that plate temperature at the sensor surface was in excess of 500°F at the time of sensor actuation.

The ultrasonic detector did actuate some time during the last 10 hours of the unattended test as is evidenced by the fact that the housing had become unlatched (Figure 63) and the temperature sensing alloy had melted. However, the device did not emit ultrasonic energy to trigger the receiver and timing system shown in Figure 3. Thus, it is not known exactly when the device actuated. The cause of the alarming device failure was a small screw that had vibrated loose. This screw retained part of the "hammering" mechanism making the alarm system inoperative.

The ultrasonic detector was returned to its manufacturer for repair and was not returned in time to be applied to any of the subsequent bearings for retesting.

Although the testing was indeed limited, and only one of the bearings (FP-6) became hot enough to warrent an over-temperature alarm, it would appear that neither device is suitable, in its present form, to be utilized as an onboard hot box detector.

The DOT-STAR device is rugged, relatively non-vulnerable, and well made (quality of workmanship). However, the point of temperature measurement (adapter surface) is just too far removed from the source of heat for it to be an accurate and repeatable indicator of bearing temperature. The ultrasonic sensor, on the other hand, senses temperature in a much more indicative location (end of axle). However, it suffers from an inadequate degree of product engineering and quality control which has resulted in poor reliability, excessive size and vulnerability, and difficult application problems.

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

Based upon the test program described in this report, the following specific conclusions have been drawn:

1) At least one initial defect in each of the bearings tested progressed during the course of the 150-hour planned test period.

2) In general, spalls increased in size and produced noticeable particle indentation. Spalling damage in itself, however, did not result in failure producing events. Bearing FP-8 gave the impression that spalling might become quite severe if subjected to further running, and FP-11 gave the impression that the microcracks that developed around the brinell would soon develop into a spall.

3) Cups with cracks into the race area enlarged in size and in number. Although race damage occurred during the test in the cracked areas, failure-producing events did not occur. However, they too gave the impression that subsequent running would produce sufficient race damage to cause possible failure-producing events. However, one must also ask what might have happened had these bearings also been subjected to normal railroad environment shock loadings during the course of the test.

4) The one bearing that did fail (FP-6), or at least reached the preset test shutdown temperature of 390°F, appeared to require a combination of the "right" defects to cause "failure." In this case it was the combination of a bent cage and a damaged seal which allowed the rollers to become jammed, causing excessive frictional heating.

5) The method of loading the bearings, i.e., use of an unworn well-fitting adapter, is important in assuring that the bearing reaches its full fatigue life potential -- as evidenced by the premature fatigue failures of both loader bearings which had their load applied in an inappropriate manner.

6) Only one of the bearings (FP-6) lost any appreciable quantity of grease through damaged cups or seals. Yet all but FP-6 (because of its other failure processes) ran exceptionally cool throughout the test. This would imply that those bearings, which were initially somewhat depleted of grease, had already reached (during actual railroad service) an equilibrium grease quantity level as far as grease "pumping" was concerned. The smaller quantity of grease remaining resulted in low losses (thus low heat generation), yet was sufficient to perform the necessary lubrication process.

7) Although some of the grease appeared to be dried or otherwise degenerated, the percentage by weight of oil in the grease was near that of new grease (both before and after the tests) with only bearing FP-8 showing an inexplicable decrease in oil content following its test.

8) Large particles that were in the cage area prior to testing became noticeably smaller as a result of the test running. Although considerable indentation and wear was noted, there was no evidence that the particles and other debris caused any perceptable jamming of the bearing components.

9) Neither of the two prototype onboard thermal sensors proved to be adequate in that neither gave warning when bearing FP-6 reached temperatures up to 390°F. The DOT-STAR sensing element was too far removed from the source of heat to adequately sense excessive bearing temperature. The ultrasonic sound emitting sensor sensed the high temperature; however, failure of a secondary component prevented it from completing its alarming function.

10) Neither temperature nor overall vibration and sound level was a discriminating indicator of bearing condition. High temperature was, however, an indicator of the final failure process in bearing FP-6.

11) Demodulated sound and vibration amplitudes gave accurate indications as to which of the bearing components were damaged and gave valuable assistance in postulating the the failure process in bearing FP-6. The damaged loader bearings, however, sufficiently clouded the dynamic data to the point where the degradation process could not be adequately tracked during the progress of the other bearing tests.

In general, the tests were rewarding in that a measurable progression of damage was experienced with each test bearing and that some light was shed on the railcar roller bearing failure process. It was generally known that the railcar roller bearing is an extremely rugged device. These tests, in a sense, fortified that knowledge. If nothing else, their tolerance to grease leaking avenues and low grease quantity was demonstrated -- fortifying the industry's contention that repacking after short intervals is unwarranted. The bearings were also extremely tolerant of the presence of large metallic particles within the moving components, and cracks in the raceway -- at least in the absence of a shock environment. It appears, based upon the one "failure" experienced, that some high friction causing mechanism, i.e., jamming of one component into or against another, is ultimately required to cause failure. Furthermore, the time to failure is probably a function of the oil evaporation rate of the grease within the affected cone. The grease between cones and in the seal case probably has little effect upon the final failure process.

On the other hand, the tests were disappointing in that only one "failure" occurred and failure conclusions based upon one failure are shaky at best. It is likely that most of the bearings that were available for testing were far from the final failure process (with the exception of FP-6). It would appear that those that had cup and seal damage had been removed from service because of a derailment and thus had little service running after they were damaged. None gave evidence of having been removed because of excessive temperature.

5.2 RECOMMENDATIONS

Based upon the conclusions made as a result of this program and upon other observations made during the course of the program, it is recommended that additional failure progression tests be conducted. Specifically, it is recommended that:

1) Bearing FP-8 be subjected to an additional 150-hour test to determine whether the peculiar surface appearance surrounding many of the spalls is indicative of gross subsurface damage.

2) Bearings FP-10 and FP-11 be subjected to an additional 150-hour test to determine whether the cracks in the raceway will eventually cause large pieces of the race to break out and perpetuate a failure process.

3) Consideration be given to shock-loading bearings FP-10 and FP-11, either before or during subsequent testing, to further promote or accelerate the existing crack damage.

4) Additional candidate bearings be secured from actual railroads (as opposed to contract bearing rebuilders). These candidate bearings should have either experienced heat damage in the field or, in the judgment of experienced railroad personnel, be "on the verge" of failure.

5) The method of loading the center loader bearing be revised to improve the load distribution on its cup.

APPENDIX A

BEARING INSPECTION LOGS

Bearing Number FP-1

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-1	
NEGATIVE NUMBERS	14-A thru 16-A	

DESCRIPTION OF BEARING CONDITION

Before Test

Timken F*70
- Broken cup at seal C'bore
- Good looking grease

BEARING INSPECTION LOG

Bearing Number FP-2

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-1	
NEGATIVE NUMBERS	17-A thru 18-A	

DESCRIPTION OF BEARING CONDITION

Before Test

Timken 68
- Water damage
- Very black grease

BEARING INSPECTION LOG

Bearing Number FP-3

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-1	
NEGATIVE NUMBERS	19-A	

DESCRIPTION OF BEARING CONDITION

Before Test

Timken 66
- Water damage
- Black grease with slimy appearance

BEARING INSPECTION LOG

Bearing Number FP-4

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-1	
NEGATIVE NUMBERS	20-A	

DESCRIPTION OF BEARING CONDITION

Before Test

Timken 77
- Broken cup at seal C'bore.
- Grease appears nearly new

BEARING INSPECTION LOG

Bearing Number FP-5

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-1	
NEGATIVE NUMBERS	21-A	

DESCRIPTION OF BEARING CONDITION

Before Test

Timken 68

- Spalled both sides 360° around
- Photo typical of entire bearing

BEARING INSPECTION LOG

Bearing Number FP-6

INSPECTION DATE	8/16/78	1/19/79
TEST HOURS	0	69.5
CONE BORE SIZE (inches)	5.639/5.639	5.639/5.638
LATERAL CLEARANCE (inches)	.008	.015
FILM ROLL NUMBER	310-1 and 310-2	310-5 and 310-6
NEGATIVE NUMBERS	22-A thru 24A	2A-10A
	4 thru 13	2A-24

DESCRIPTION OF BEARING CONDITION

Before Test

Timken K (no date)

Cup broken both ends 90% apart (22A and 24A)

One end has badly dented seal case and cage with a double dent (22A, 23A, 7, 10, 11). Moderate brinells adjacent to break (8/9) and no obvious particles or frozen rollers.

Other end has moderately dented seal case (24A, 12, 13), a dented cage with a large piece (1/8" square x 1-1/4" long) of the cup jammed between cage and cone (4,5) plus smaller pieces elsewhere, one roller at counterclockwise extreme of dent as viewed in 4 and 5 seems frozen in place, and a moderately heavy brinell adjacent to broken cup (6).

After Test

Outboard side:

- Badly dented seal end (had been outboard test position end). Grease oozing from seal OD (2A, 3A) grease caked and hard in seal case (4A, 8A). Cage area dry of grease with ends of rollers blue from heat and worn (5A, 6A, 7A). Five roller separators were bent outward and had rubbed on the cup race (13A - 15A and 18A). Rollers appear to have been forced under separators causing them to bend outward. This was caused by cage rubbing on bent portion of seal, 180° away, forcing cage into an eccentric running condition. Approximately 180° of cage is rolled down at large OD end (14A-16A and 18A). Grease is dried and hard in a very thin layer over entire circumference of cup near outboard end (9A-12A). Layer can be removed with solvent. Wear track from bent roller separators is smooth (17A). Wear ring cracked

Inboard Side:

- Seal rotated 45° in direction of rotation and became cocked and jammed in place (19A). All rollers free, found another brinell 180° from original (24A). No evidence of large chip or other pieces. Grease feels of metallic debris. Grease in general shows little or no evidence of overheating.

BEARING INSPECTION LOG

Bearing Number FP-7

INSPECTION DATE	8/16/78	2/2/79
TEST HOURS	0	150.5
CONE BORE SIZE (inches)	5.690/5.690	5.689
LATERAL CLEARANCE (inches)	.032	.027
FILM ROLL NUMBER	310-2	310-8
NEGATIVE NUMBERS	14 thru 18	3 thru 25

DESCRIPTION OF BEARING CONDITION

Before Test

Hyatt 64

- Low on grease, dried between rollers
- Metal chips in grease between rollers (17,18)
- No significant cup damage (source of particles not obvious)
- Both seals very loose in cup (can turn by hand)
- One roller spalled per B. Geren at original disassembly (could not find at 8/16/78 inspection)

After Test

Test position outboard end

- Metal chips between roller but much finer in size
- All rollers have circumferential scratches near their mid-plane (7,8)
- Seven rollers are spalled (13-15); one (15) worse than others
- Peculiar brinell type mark on outer race (9,10); some fairly deep particle marks (10-12)
- One large and a few small inner race spalls

Test position inboard end

- Outboard end of rollers show a brownish discoloration (17,18)
- Cup shows severe fragment indentation, is mottled in appearance (19-22)
- Seal lip worn (23)
- Seal case O.D. smooth with sharp edge from rotating in cup (24)
- Two moderate size inner race spalls

BEARING INSPECTION LOG

Bearing Number FP-8

INSPECTION DATE	8/16/78	1/19/79
TEST HOURS	0	150
CONE BORE SIZE (inches)	5.689/5.689	5.689/5.689
LATERAL CLEARANCE (inches)	.018	.021
FILM ROLL NUMBER	310-2	310-7
NEGATIVE NUMBERS	19 thru 26	3A thru 12

DESCRIPTION OF BEARING CONDITION

Before Test

Timken 69

- Rusty appearing grease - caked between rollers (19,20)
- Metal chips between rollers
- Both sides moderately spalled (every spall photographed - 21 thru 26)
- Water damage on side photographed with quarter (24 thru 26)

After Test

- Grease seems to have lost rusty appearance (7A,8A)
- Metal particles between rollers not significantly different from before test condition
- New spall on "penny" side (6A)
- All spalls slightly larger than before
- Many spalled areas give the illusion that either severe subsurface damage has occurred adjacent to the actual spalls or that the surfaces had been brinelled prior to spalling (3A,4A,6A,9A,10A). These indications were not apparent prior to testing (21,22,24,26).
- Fragment indentation on quarter side appears to have increased in size and number (11A/25)
- Rust-like appearance of water etches has disappeared (11A/25 and 9A/24)
- Both outer races had heavy rust-like stain near seal counter bore. This stain could be removed with heavy wiping pressure.

BEARING INSPECTION LOG

Bearing Number FP-9

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-2	
NEGATIVE NUMBERS	27	

DESCRIPTION OF BEARING CONDITION

Before Test

Timken 63

- Grease appears relatively fresh (probably recently reworked)
- Much particle indentation
- 2 moderate spalls one side (27)
- 3-4 very small spalls on other race (not photographed)

BEARING INSPECTION LOG

Bearing Number FP-10

INSPECTION DATE	8/16/78	3/30/79
TEST HOURS	0	150.0
CONE BORE SIZE (inches)	5.687	5.687-5.689 out of round
LATERAL CLEARANCE (inches)	.012	.012
FILM ROLL NUMBER	310-2	310-9 and 310-10, 310-12
NEGATIVE NUMBERS	28 thru 32	16 thru 25 and 2 thru 4 and 1 thru 8

DESCRIPTION OF BEARING CONDITION

Before Test

NDH (date unknown)

- Cup broken one side (28,29)
- Metal chips between rollers on broken side (30 - chip rolled from edge onto roller for photograph)
- Cup cracked (31, 32) on side opposite break ($\approx 90^{\circ}$ from break)

After Test

Inboard end:

- Crack has propagated into center portion and chipped out (16, 17 also see 310-12, 1-8).
- On the inside, cracked area has become a large spall with several stringer cracks (18-20).
- Fragment indentation adjacent to spall in direction of rotation (CCW from spall) (18-20).
- Cup race 180° from spall/crack shows roller imprint but no discernible indentation (21).
- Rollers have circumferential scratches easily felt with fingernail. Grease between rollers is gritty, looks and feels of metallic debris (22, 23).

Outboard end:

- Particle indentation in area adjacent to break (25) which did not appear to be there before test.
- Cone depleted of grease, rollers have fine circumferential scratches -- heaviest at large end (3,4).

BEARING INSPECTION LOG

Bearing Number FP-11

INSPECTION DATE	8/16/78	3/30/79
TEST HOURS	0	151.6
CONE BORE SIZE (inches)	5.687/5.689*	5.688
LATERAL CLEARANCE (inches)	.025	.028
FILM ROLL NUMBER	310-2	310-9
NEGATIVE NUMBERS	33 thru 36	9 thru 15

DESCRIPTION OF BEARING CONDITION

Before Test

NDH (date unknown)

- Cup broken one side (33,34) - chips in grease
- Crack in line with break on opposite side (35,36)
- Crack goes through edge brinell (35) - both sides of brinell cracked

After Test

Outboard end

- Very little grease leaked at break (9)
- Break in same condition as before test (10)
- Race in good condition (10)

Inboard end

- Crack propagated along entire width of race (11-13)
- A second crack on outside (which was not obvious before) appears to have developed (11-13)
- Brinell appears deeper near center of bearing and seems to have a few cracks. Small chip broken out of edge of brinell (14,15)
- Cone shows no undue distress. Grease feels gritty.

* Out of round - both ends

BEARING INSPECTION LOG

Bearing Number FP-12

INSPECTION DATE	8/16/78	3/30/79
TEST HOURS	0	156.7
CONE BORE SIZE (inches)		5.688
LATERAL CLEARANCE (inches)		.015
FILM ROLL NUMBER	310-2 and 310-3	310-9
NEGATIVE NUMBERS	37(310-2) and 4A thru 8A (310-3)	2 thru 8

DESCRIPTION OF BEARING CONDITION

Before Test

Brenco 74

- Water damage one side (4A,5A)
- Other side cup broken (6A,7A) with metal particles in cage area (6A,8A)

After Test

Outboard end

- Chip at edge of break increased in size (3)
- Wear track on small diameter of cup (3)
- Brinell mark clockwise from break (3) which is not noticeable in before test picture (7A)
- Some water etching opposite break (4)
- Rollers water etched (8) and grease feels of metal particles

Inboard end

- Water etching (5,6) similar to before test (4A,5A); one etched area may be soon to develop into spall (5)
- Rollers not etched (7)

BEARING INSPECTION LOG

Bearing Number FP-13

INSPECTION DATE	8/16/78	
TEST HOURS	0	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-3	
NEGATIVE NUMBERS	10A thru 11A	

DESCRIPTION OF BEARING CONDITION

Before Test

Brenco 63
- Several brinells around one side (10A,11A)
- One large spall (11A) on same side as brinells
- One small brinell on opposite side (not photographed)

BEARING INSPECTION LOG

Bearing Number LB-1

INSPECTION DATE	3/30/79	
TEST HOURS	676.8	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-10	
NEGATIVE NUMBERS	5A/6 thru 18A/19	

DESCRIPTION OF BEARING CONDITION

After Test

Drive end

- Surface of grease in seal case appears "rusty". Underneath is fresh looking (6A/7,7A/8)
- Metal particles in grease between rollers giving a glittered appearance (8A/9-10A/11); rollers show severe fragment indentation
- Very wide spalled area in load zone (11A/12-14A/15); spall appears to end more abruptly and sharply at leading edge (rotation is counterclockwise in pictures - not as shown in 14A/15). Fragment indentation on trailing edge.
- Outer race 180° from spall shows roller imprints with no discernable indentation (15A/16).

Test bearing end

- Grease in seal case gives same appearance as other end (16A/17)
- Outer race in good condition - somewhat brownish in appearance 18A/19.
- Slight circumferential scratches on rollers (17A/18) - note good (new) grease condition on small end of cone.

BEARING INSPECTION LOG

Bearing Number LB-2

INSPECTION DATE	3/30/79	
TEST HOURS	308.3	
CONE BORE SIZE (inches)		
LATERAL CLEARANCE (inches)		
FILM ROLL NUMBER	310-10	
NEGATIVE NUMBERS	19A/20 thru 24A	

DESCRIPTION OF BEARING CONDITION

After Test

Test bearing end

- Grease in seal case blackish where it contacts cone, but like new under surface (20A/21).
- Outer race spalled (21A/22); spall sharp and well defined at leading edge with fragment indentation at trailing edge. Note clean grease in cup center portion.
- Metallic particles in grease between rollers (22A/23); some particle indentation and circumferential scratching.

Drive end

- Grease in seal case has same appearance as other end (20A/21)
- Small spall in outer race (24A) with rust-like staining on trailing edge
- Metallic particles in grease between rollers (23A/24); rust-like staining similar to race, slight circumferential scratching.

APPENDIX B

REPORT OF NEW TECHNOLOGY

Laboratory endurance tests of six railcar roller bearings that had previously suffered physical damage or were otherwise degraded as a result of actual railroad service were conducted. The primary objective of the test was to obtain a better understanding of the railcar roller bearing failure process(es) and the manner in which bearing defects progress. A 150-hour test with 26,250 pounds radial load (equivalent full car load) at 528 rpm (equivalent to approximately 52 mph) was planned for each bearing. Only one bearing actually failed to complete the 150-hour or 7,800-mile test. All bearings exhibited further measurable degradation or defect progression during the course of the tests.

After a diligent review of the test procedures and results described in this report, it is believed that no patentable innovation, improvement or new invention was made.

31 Copies