FRA/ORD-81/28, I
DOT-TSC-FRA-81-15, I

37

Improvement of Railroad Roller Bearing Test Procedures and Development of Roller Bearing Diagnostic Techniques

Volume I - Acceptance Test

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March 1982
Final Report

U.S. Department of Transportation
Federal Railroad Administration

Office of Research and Development
Washington DC 20590

03 - Rail Vehicles &
Components

Technical Report Documentation Page

1. Report No. FRA/ORD-81/28,I	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle IMPROVEMENT OF RAILROAD ROLLER BEARING TEST PROCEDURES AND DEVELOPMENT OF ROLLER BEARING DIAGNOSTIC TECHNIQUES Volume I - Acceptance Test		5. Report Date March 1982 6. Performing Organization Code
7. Author(s) W.D. Waldron, J.M. McGrew, A.I. Krauter		8. Performing Organization Report No. DOT-TSC-FRA-81-15,I
9. Performing Organization Name and Address *Shaker Research Corporation Northway 10 Executive Park Ballston Lake, New York 12019		10. Work Unit No. (TRAIS) RR028/R1309 11. Contract or Grant No. DOT-TSC-917
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 October 1974-April 1977 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

Bearing defect data from 8,000 railroad roller bearings are analyzed to determine their defect modes and defect rate distributions. Cone bore growth, brinelling, and fatigue are identified as the predominant defect modes during the first 12 years of the aging process. The results of the study show that after approximately two years of service, 10 percent of all railroad roller bearings exhibit a defect of one type or another for which at least one component would be condemned if it were in a rework shop. The present AFBMA* method of calculating fatigue spalling, modified to account for lubricant film thickness effects, correlates reasonably well with the observed incidence of spalling (10 percent fatigue life of about 11 years). The problem lies in the fact that AFBMA calculation procedure does not consider the other competing defect modes which contribute far more to the overall defect rate than spalling. The relationship between "defect rate" and "failure rate" is not direct, of course, and an examination of "condemning limit" definitions relative to the progression of bearing failure in service is needed.

An accelerated life acceptance test procedure consisting of a laboratory test for fatigue and brinelling resistance, followed by a field test to certify for other defect modes, was designed; and an example of its implementation is described.

*Anti-Friction Bearing Manufacturers' Association

17. Key Words Tapered Roller Bearings, Railroad Roller Bearings, Bearing Defects, Certification	18. Distribution Statement DOCUMENT IS AVAILABLE TO THE U.S. PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

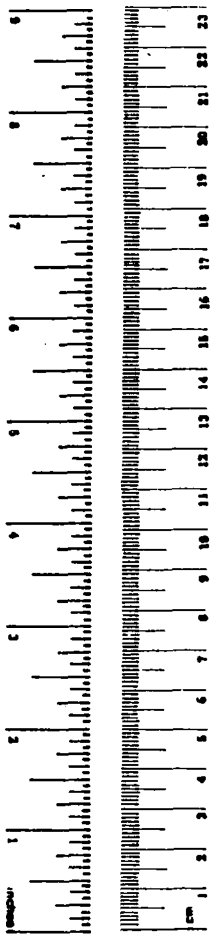
19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
UNCLASSIFIED	UNCLASSIFIED	248	

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

METRIC CONVERSION FACTORS

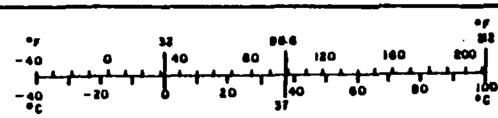
Approximate Conversions to Metric Measures

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



PREFACE

This study was conducted for the Federal Railroad Administration through the Transportation Systems Center (TSC) in Cambridge, Massachusetts. Mr. Roger K. Steele was the initial technical monitor. He was succeeded by Mr. James M. Morris.

New York State derailment data was provided 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 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 reviewed a draft of this report and made 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, conclusions or 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 is made to Messrs. Bibel, Hayes, and Trollinger, whose tolerance of our intrusion allowed us to complete the tests successfully.

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.

Mr. O. W. Knoblock of the Association of American Railroads made available all relevant certification data and procedures.

The Southern Railway System, Norfolk & Western Railway, Delaware and Hudson and New Departure Hyatt also supplied valuable information and guidance. Messrs. R. F. Tuve and C. M. Scott of the Southern, Mr. H. Miller of the N & W, and Mr. J. T. Colpoys of the D & H were particularly helpful.

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NOMENCLATURE

A	Contact area	in.^{2}
AAR	Association of American Railroads	
AFBMA	Anti-Friction Bearing Manufacturers Association	
b	Width of contact zone	in.
B_n	Life for which (1-n)% of all bearings will survive	years
c	Coefficient in plastic stress equation	
C	Dynamic load capacity	lb
C_o	Basic static load rating per AFBMA	lb
C_s	Static load capacity	lb
d	Roller diameter	in.
d_m	Mean pitch diameter	in.
D	Bore diameter	in.
DN	Bearing speed parameter bore x rotational speed	mm-rpm
D(t)	Number of defective bearings of age, t	
D_1	Diameter of plastic-elastic interface	in.
D_2	Outer diameter of cone	in.
ΔD	Cone bore growth	in.
$D^*(t)$	Number of bearings removed from population in year, t	
E	Elastic modulus	psi
E'	$1/2\left[\frac{1-\nu_1^{\,2}}{E_1}+\frac{1-\nu_2^{\,2}}{E_2}\right]$	in.^{2}/lb
f	Ratio of total good bearings to "O.K." bearings	
$f_1(P_B)$	Normal probability distribution of brinelling limit	
$f_2(P)$	Normal probability distribution of load	
$f_3(P_B-P)$	Normal probability distribution of brinelling strength-stress difference	
$f_4(M)$	Standardized normal probability density function	

f_c	Factor tabulated in the AFBMA standard for roller bearings at values of the parameter, γ	
f_i	Fraction of time spent at ith operating condition	
$f(t)$	Probability density function	
$F(t)$	Cumulative probability density function	
$F^*(t)$	Cumulative probability density function for all railroad bearings in year, t	
$G(t)$	Number of good bearings of age, t	
$G^*(t)$	Failure rate	
h	Lubricant film thickness	in.
$h(t)$	Hazard rate or instantaneous failure rate	
$h^*(t)$	Instantaneous failure rate	
$H(t)$	Cumulative hazard	
i	Number of roller rows in bearing	
I	Function of bearings reworked each year	
k	Constant in plastic deformation equation	
K	Kilo	
ℓ	Effective roller length	in.
L	Component life	hours, years, miles
L_n	Life for which (1-n)% of all bearings will survive	revolution, miles
L_t	Test time or distance	hours, miles
$\overline{L}_n$	Composite life for which (1-n)% of all bearings will survive	revolutions, miles
LCL	Lower confidence level	
m	Normalized brinelling stress-strength difference $= \dfrac{\overline{P}_B - \overline{P}}{\sqrt{\sigma_S^2 - \sigma_P^2}}$	

n	Number of bearings	
$n(P)$	Number of load cycles	
n_a	Number of bearing failures to accept lot	
n_r	Number of bearing failures to reject lot	
N	Shaft rotational speed	revolutions per minute
NFL	No field lubrication	
$O(t)$	Number of oversize bearings of age, t	
$O_1(t)$	Number of oversize and out-of-round bearings of age, t	
p_m	Mean pressure (Section 6.4.1)	psi
P	Bearing load	lb
P_D	Number of defective bearings in population	
P_B	Brinelling load	lb
$P^*(t)$	Railroad roller bearing population in year, t	
$\Delta P^*(t)$	Increase in railroad roller bearing population in year t	
$Pr(t)$	Probability	
$Pr(A)$	Probability of acceptance	
q	Roller load per unit length	lb/in.
Q	Roller load	lb
r_c	Consumer risk	
r_p	Producer's risk	
R	Roller radius	in.
R_a	Reliability deemed to be acceptable	
R_u	Reliability deemed to be unacceptable	
$R(t)$	Reliability	
Rc	Hardness on Rockwell C Scale	

S	Factor in grease life equation	
S_c	Grease factor in grease life equation	
S_N	Speed factor in grease life equation	
S_P	Load factor in grease life equation	
$S(t)$	Number of railroad roller bearings added to population in year t	
t	Time	hours, years
t_F	Time at 100 F percentile	hours, years
Δt	An increment of time	hours, years
T	Temperature	^oF, ^oC
$u(F)$	$\ln\{\ln(1-F)\}$	
U	Rolling speed	in./sec.
V	Variable in inverse power law model	
V_o	Reference variable in inverse power law model	
Z	Number of rollers	
α	Weibull slope parameter	-
α_c	One-half included cup angle	degrees
α_o	Pressure-viscosity coefficient	$in.^2/lb$
β	Weibull slope parameter	
γ	$d \cos\alpha_c/d_m$	
γ_o	Weibull location parameters	
δ	Depth of contact deformation	in.
δ_B	Depth of condemnable brinell	in.
δ_o	Depth of contact deformation under basic static load rating, C_o	

ε_p	Circumferential plastic strain	in./in.
ζ	P_B-P	lb
η	Weibull characteristic life	years
$\bar{\eta}$	Lubricant viscosity	$\frac{\text{lb-sec}}{\text{in.}^2}$
Λ	Specific film thickness = h/δ	
μ	Mean bearing life	hours, years
ν	Poisson's ratio	
ν_o^+	Number of zero crossings of fluctuating load	
σ	Surface roughness	in.
σ_p	Standard deviation of load distribution	lb
σ_s	Standard deviation of brinelling strength distribution	lb
σ_y	Elastic limit	psi
σ_{ye}	Elastic limit at edge of contact	psi
σ_δ	Standard deviation of deformation distribution	in.

1. INTRODUCTION, CONCLUSIONS, AND SUMMARY

1.1 BACKGROUND

The roller bearing was introduced into freight car journal use in the U.S. in 1954, and its numbers have risen so that today approximately 60 percent of the freight car fleet of U.S. railroads rides upon roller bearings. The change to roller bearings has led to a marked decrease in the number of setouts for overheated axles (1). However, in the period from 1965 through 1971, there was an increase in the rate at which roller-bearing-equipped cars were reported to have hot bearings (2). The number of train accidents attributable to overheated roller bearings has remained a small portion of the total (near 1%). The observation of increased roller bearing setouts prompted the Federal Railroad Administration to caution "that the roller bearings become less effective with age" (2). At the request of the Federal Railroad Administration, the Transportation Systems Center (TSC) initiated a program to explore the possibility of achieving improved railroad roller bearing test and diagnostic procedures.

In an effort to examine roller bearing failure and defect behavior in a most detailed fashion and to ascertain whether improved test procedures and diagnostic techniques could have a substantial cost-beneficial impact on bearing serviceability, a three-phase program was devised. The first phase, of four months duration, had as its objectives: (a) the analysis of railroad roller bearing failure and defect behavior and the identification of failure modes, and (b) conception of improved cost-beneficial approach(es) to certification and diagnostics based on the results of (a). In phases II and III (one year each): (a) improvements in the test procedure process were to be proposed and tested leading to establishment of guidelines for improved cost-beneficial testing practices and (b) potentially useful diagnostic approaches were to be developed and tested leading to the definition of performance specifications for one or more methods.

The results of the work performed by Shaker Research under Contract DOT-TSC-917, "Improvement of Railroad Roller Bearing Test Procedures and Development of Roller Bearing Diagnostic Techniques--Phase I and II" are contained in two volumes. Volume I presents the test procedure aspects of the work, and Volume II the diagnostic aspects. This report (Volume I) describes the work directed at improving the railroad roller bearing test procedures.

1.2 CONCLUSIONS

With respect to bearing acceptance aspects, a statistical analysis of bearing population characteristics indicates that the age of the roller bearing population will increase from its current median of approximately eight years to some asymptotic limit. As the age increases, bearing failures can be expected to increase. Although the current catastrophic failure rate (as measured by roller-bearing-caused derailments) is not large in terms of the total roller bearing population (approximately 86 roller-bearing-caused derailments per year out of a total roller bearing population of approximately eight million bearings), its cost in terms of potential loss of life and private property loss must be considered in light of the increasing potential for failure.

Analysis of bearing defect (as defined by the AAR*) data with respect to spalling indicates that the bearing equivalent design life of 250,000 miles at full load (equivalent to 500,000 miles at 80 percent load) is not met. This is equivalent to an L_{10}** life of approximately 19 years under average service. Furthermore, when all defects are considered, the L_{10} life is reduced to approximately two years. The major defects that contribute to this reduction are cone bore growth, followed by brinelling, and seal problems. If cone bore growth is neglected as a defect, the L_{10} defect life is approximately three years.

Although the age at which a railcar roller bearing develops a condemnable defect is strikingly young, it is known (and has been experimentally demonstrated in the subject program) that bearings possessing condemnable

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* Association of American Railroads.

** Life which 90% of all bearings will survive.

defects can operate for many thousands of miles without affecting the safety of operation. However, the analyses described in this report indicate that the number of failures (confirmed setouts and catastrophic failures) will increase in the future due both to the projected increase in size of the roller bearing population and the increase in age of the population.

To reduce the number of projected future failures, the proportion of bearings in the population that possess defects will have to be reduced. This can be accomplished by improving the quality of the initial product and/or by improving the method(s) by which defective bearings are removed from the population.

Quality improvement should be emphasized in the areas of fatigue and brinelling resistance as these defect modes, along with cone bore growth, were the most significant modes leading to the short L_{10} defect life. Since the brinelling resistance tests conducted on this program indicated a relationship between susceptibility to brinelling and cone bore growth, it has been tentatively concluded that improvements in brinelling resistance will yield corresponding improvements in cone bore growth resistance. This line of reasoning, along with the fact that no relationship between cone bore growth and bearing failure has been established, has led us to place priority on defects other than cone bore growth in spite of the fact that it was the most prevalent cone defect found at rework.

An analysis of the current acceptance test reveals that greater protection against the fatigue, brinelling, and cone bore defects can be achieved by making the certification procedure more stringent. Indeed, bearings purposely made from materials known to have very poor fatigue and wear resistance showed no signs of distress after having run much longer than the current test requires. To this end, an accelerated life test procedure consisting of a laboratory test for fatigue and brinelling resistance, followed by a field test to certify for other modes, was designed.

The concept of the laboratory test was experimentally verified during the subject program; and suggestions for, and examples of, its implementation were made. The sequential sampling plan type test recommended would minimize the user's (railroad's) risk at the laboratory stage while minimizing the manufacturer's risk during the field test stage--which is the most user protective plan.

1.3 SUMMARY OF WORK AND REPORT ORGANIZATION

The work was divided into two basic tasks: an acceptance task and a diagnostics task. This report (Volume 1) covers the acceptance task.

Section 2 of the report describes the concept of acceptance and examines the current test procecure. It is shown that in the present acceptance procedure there is a high probability of accepting a poor quality bearing and that significant improvements in consumer (i.e., the American railroads) protection can be achieved by making the procedure more stringent.

Before making recommendations to improve the test procedure, it was first necessary to define the general railcar problem in terms of relative numbers of failures and/or faults and to categorize the failures and/or faults in such a way that the important factors to be considered during later phases could be defined on a rational basis. This was accomplished by collecting, sorting, and analyzing data from a variety of sources on defective and failed bearings.

These data included: bearing rework shop inspection reports covering approximately 8,000 bearings (of which approximately 20 percent were found to be defective) provided by Brenco, Inc.; joint inspection reports for approximately 400 "set out" bearings; reduced AAR failure data for approximately 775 failed roller bearings provided by the Southern Railway System; and published AAR Journal Failure Reports.

The data, the manner in which they were reduced, and an overview of this analysis are discussed in Section 3 of this report.

Statistical analysis of the data, where appropriate, is further discussed in Section 4. Section 4 concentrates on the reliability aspects of the bearing system (bearing components, seals, and lubricant) as well as the prediction of failure rate versus time for all significant failure modes and the total bearing as an integrated entity.

The gains to be achieved by accelerated testing are indicated in Section 5.

Section 6 is directed toward a demonstration of accelerated bearing tests, tests for defective metallurgy, and bearing brinelling tests. The major conclusion is that bearings with known metallurgical defects can easily pass the current acceptance test.

In Section 7, a model for an improved acceptance procedure is outlined. This procedure is based upon both analytical and experimental work conducted during the course of the program. The proposed procedure offers significant gains in consumer protection while not unduly raising the bearing producer's risk of having an acceptable bearing rejected erroneously.

2. BEARING ACCEPTANCE TESTS

The purpose of the certification test is to provide evidence that the railroad roller bearing has the desired reliability in service.

All demonstrations of reliability are governed by the principles of statistical hypothesis testing. Such tests lead to a decision either to:

a) Accept the bearing as having acceptable reliability.

or b) Reject the bearing as having unacceptable reliability.

Regardless of the conclusion, the experimenter cannot avoid the possibility of an incorrect decision. The consumer's risk, or r_c, is the probability that a bearing with unacceptable reliability will be accepted. The producer's risk, or r_p, is the probability that a bearing with acceptable reliability will be rejected.

There are six characteristics of any acceptance test that must be specified:

1. The reliability deemed to be acceptable, R_a.
2. A value of reliability deemed to be unacceptable, R_u.
3. Producer's risk, or r_p.
4. Consumer's risk, or r_c.
5. The probability distribution to be used for number of failures or for time to failure.
6. The sampling scheme.

The first four of the above characteristics must be set by the appropriate industry and/or government bodies that are charged with the economic and safety responsibility of the railroad industry. Section 4 will discuss the probability distribution of the bearing assembly. In this section, the concept of producer's and consumer's risk is defined and the current test scheme is examined.

2.1 CONSUMER'S AND PRODUCER'S RISK

The probability of acceptance of a "lot," Pr(A) will be a function of the quality of the "lot" and will have a form as pictured in Figure 1.

The magnitude of r_c, r_p, R_a, R_u, and the sample size n are all intertwined. The closer R_a is to R_u, the smaller r_c is specified or the smaller r_p is specified, the larger the sample size n must be. If the experimenter fixes r_c, r_p, R_a, and R_u, then n is determined. Because of test facility limitations, it is often customary to fix n. When this is done, only three of the above four quantities can be chosen independently. A common practice when n must be fixed is to specify the acceptable reliability R_a and the producer's risk r_p and then to plot Pr(A) and to observe its magnitude as a function of various possible values for R_u, the unacceptable reliability. If the discriminating power is unacceptable, then R_u, r_p, or n must be altered. Such a plan places the control emphasis on the producer's risk. To place the emphasis on the consumer's risk for a fixed sample size n, one would specify the unacceptable reliability R_u and the consumer's risk r_c and then plot Pr(A) as a function of possible values for R_a, the acceptable reliability. If the discriminating power is unacceptable, then R_a, r_c, or n must be altered.

2.2 ATTRIBUTES AND VARIABLES TESTS

If each bearing is merely classified as acceptable or unacceptable, then the test is an attributes test. If the service life of the bearings under test is assumed to have a specific probability distribution, such as the Weibull, then the test is a variables test. Attributes tests may be performed even if a probability distribution is assumed by dichotomizing the life distribution into acceptable and unacceptable distance to failure.

If no probability distribution for life is assumed and an attributes test is performed whereby n components are placed on test for t miles and the number of failures observed, then all that can be estimated for the "lot"

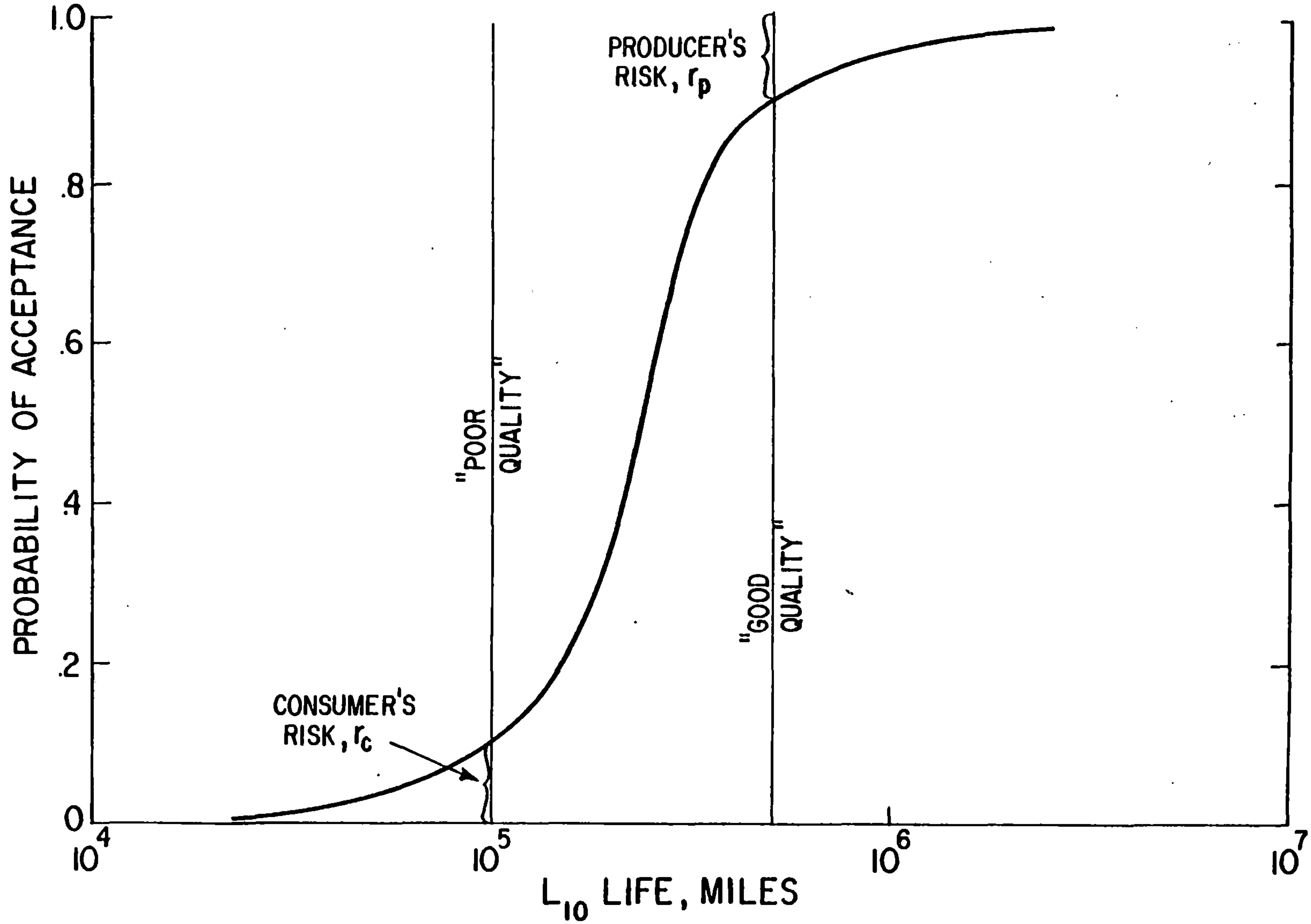


FIGURE 1. OPERATING CHARACTERISTIC CURVE

is its reliability at the distance t. To make statements regarding any other distance is not possible without knowledge of the failure distribution. For example, if two bearings were placed on test for 2,050 miles and neither bearing failed, then with 50 percent chance of being right we can say that reliability at 2,050 miles is greater than 0.794. With 90 percent chance of being right we can say that the reliability is greater than 0.464. (See Appendix Table I of Reference 3.) Here, reliability is the probability that a bearing will last longer than $t = 2,050$ miles. Nothing can be said about the probability distribution for the bearing. If for the same test described above one assumed the service life distribution to be Weibull with shape parameter $\beta = 2$, then with 50 percent confidence in being right we can say that the L_{10} life is greater than 1,386 miles (or with 90 percent confidence, that the L_{10} life is greater than 760 miles).

For attributes plans with sample size n and acceptance of lot when no failures are observed, the 50 percent and 90 percent lower confidence limits for the reliability at the test distance are plotted in Figure 2. For the same attributes plans and with the additional assumption that the failure distribution is Weibull with shape parameter β, the 50 percent and 90 percent lower confidence limits for L_{10}/t as a function of the sample size n are plotted for two values of β (1, 2) in Figure 3. The properties of many other plans allowing for some failures to be observed and still having the lot accepted can be obtained from Appendix Table I of Reference 3.

Attributes tests are simple because they only involve counting the failures. In this sense they are cheaper to administer, although the price can be excessive in that larger sample size n is needed to achieve the same producer's and consumer's risk than for a plan where one actually measures the distances to failure. Attributes tests are convenient in that one does not need to know the defect distribution to estimate the reliability at the test time. Also, even when a distributional assumption is made, they can be used to obtain Pr(A) and lower confidence limits on L_{10} for sampling plans that allow zero failures for acceptance. However, if a distributional form

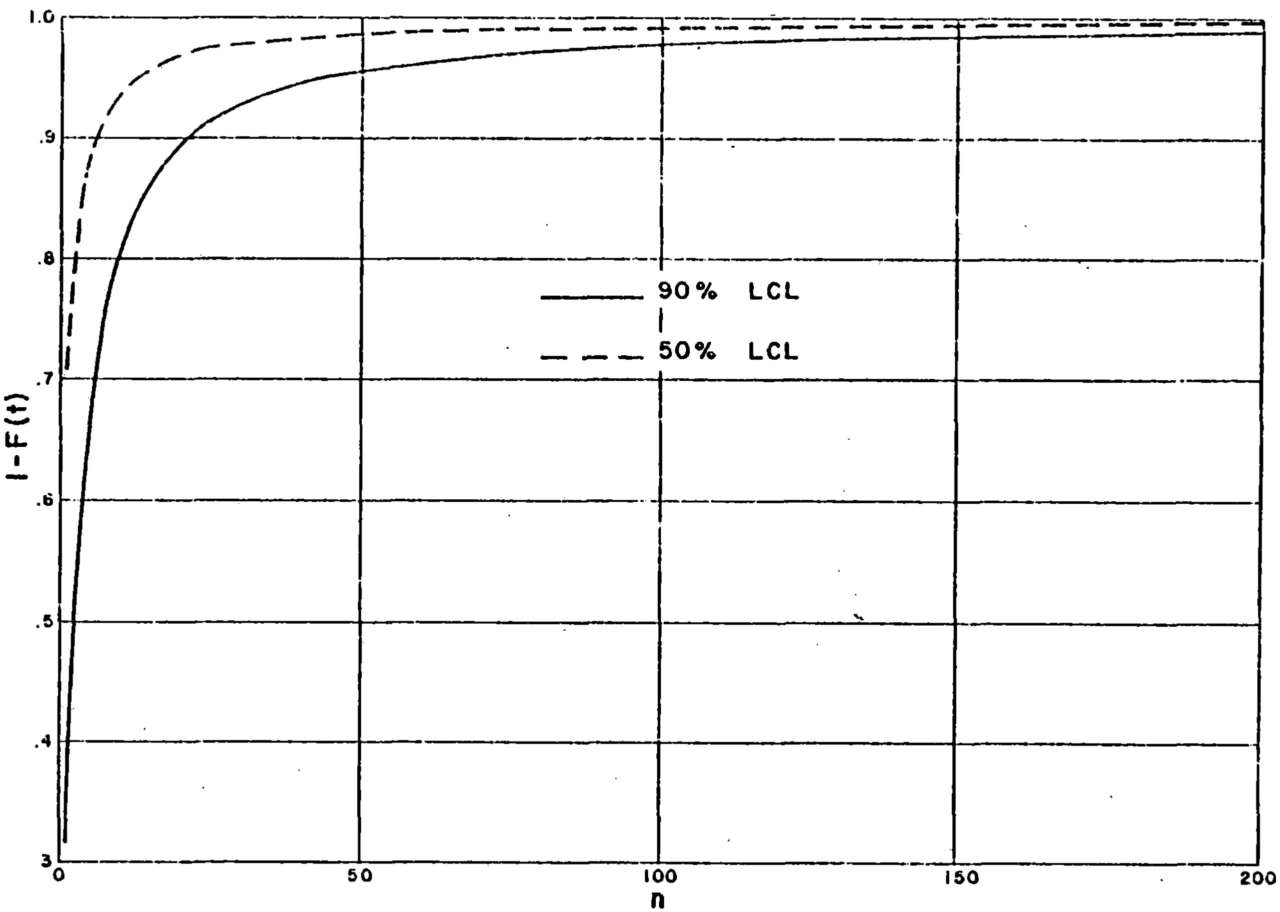


FIGURE 2. LOWER CONFIDENCE LIMIT (LCL) ON RELIABILITY AT TIME t AS A FUNCTION OF n FOR TEST BASED ON n BEARINGS WITH 0 FAILURES BY TIME t

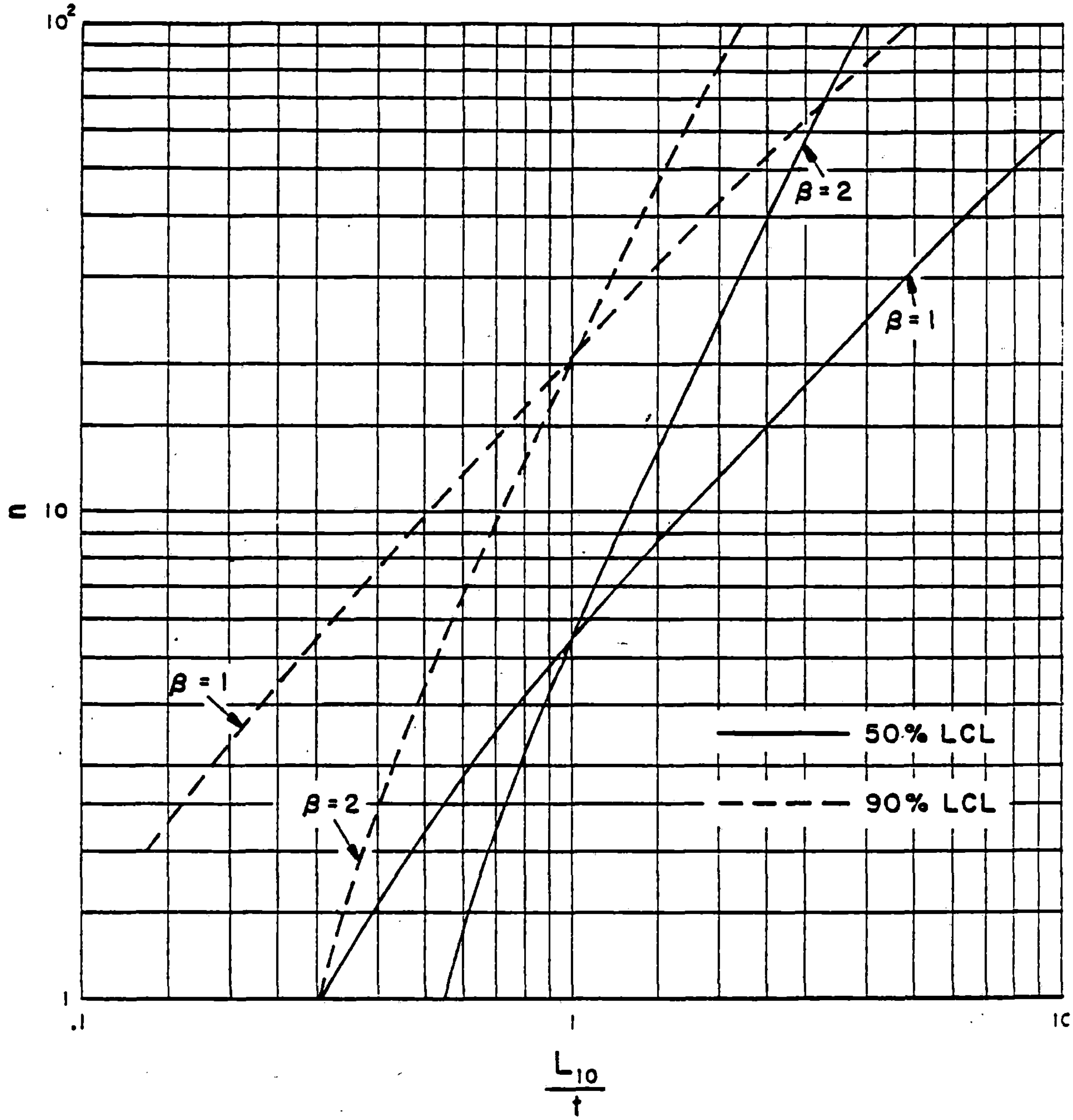


FIGURE 3. LOWER CONFIDENCE LIMIT (LCL) FOR L_{10}/t AS A FUNCTION OF n BASED ON A WEIBULL MODEL WITH SHAPE PARAMETER β. (PLACE n BEARINGS ON TEST FOR t MILES. ACCEPT LOT IF THERE ARE NO FAILURES)

can be assumed as in the case of roller bearings and if one can test long enough to observe failures, then the information contained in these measured times to failure can be used to give greater discriminating power. For variables sampling plans, it is essential that some failures be observed. Hence, accelerated testing is desirable. However, this creates little difficulty if a model is available for relating failure times under accelerated conditions to failure times under normal conditions.

For variables sampling plans, several variations can be considered:

1. Time Truncated Test Plans. As with attributes plans, n items could be placed on test for a given time duration and the actual failure times observed. Such plans are easy to administer since the total testing duration is known. Their disadvantage is that insufficient failure data may accrue if the items are longer lived than anticipated.

2. Failure Truncated Test Plans. Here n items are placed on test, and testing is continued until r have failed. The failure times for each of the r failures are recorded. Although good information on failure data is accrued, the test may go on for a long time if the component is of better quality than anticipated. However, a fringe benefit is that of all the plans, this one is most likely to have some information to judge the suitability of the distributional assumption for the defect occurrence.

3. Sequential Test Plans. Here, items are placed on test either singly or in small groups in sequence. Such sampling plans can be used for attributes or variables plans. In general, the total testing time on the average will be shorter. However, for individual lots, the testing time could be excessive and unknown. For bearings of quality midway between acceptable and unacceptable, the testing time would be greatest. For items of very low quality, the number of bearings needing to be tested would be much smaller than with a fixed sample size test procedure.

2.3 CURRENT ACCEPTANCE PROCEDURES

The AAR currently certifies journal roller bearings, grease, and seals using an attributes sampling plan. A limited sample of each component is tested for a given period of time. To pass the acceptance test, the test components must perform without failure.

2.3.1 Laboratory Tests

Table 1 is a summary of the test procedures currently used by the AAR to certify bearings, seals, and grease.

In the case of roller bearings which have a specification fatigue life of 500,000 miles at 80 percent load, the acceptance test takes one week and is equivalent to 2,050 miles of operation under simulated load, speed, and temperature conditions. Only one bearing is tested.

For grease acceptance, sixteen samples are tested for eight weeks, equivalent to 39,392 miles. Eight samples are tested alone, and eight are mixed with a combination of all other certified greases to evaluate compatibility.

The number of seals tested depends on the batch size which the manufacturer desires to certify. As Table 1 shows, this sample size is always less than two out of one thousand. The test seals are run for an equivalent of 12,600 miles under simulated conditions.

The question is whether the limited sample size and test times of current test procedures provide significant information about the performance of the general population and at what confidence level. Using the observed defect rate distribution of railroad roller bearings, we now examine the efficacy of the current procedure in describing the characteristics of the general population.

TABLE 1

SUMMARY OF CURRENT ACCEPTANCE PROCEDURES

Component	Roller Bearing	Grease	Seals
Design Specification	500,000 miles service with a load factor of 80%	3 year relubrication interval	250,000 miles
Sample Size	2	8 under simulated conditions. 8 under compatibility conditions.	≤ 1,000 2 1,001 - 5,000 4 5,001 - 7,500 6 7,501 - 10,000 8
Test Duration	2050 mi. { 5 1/2 x 10 6 x 11 2235 mi. 6 1/2 x 12	49,392 miles	12,600 miles
Elapsed Time	1 week	8 weeks	1 week, 3 days
Test Conditions	Temperature: 1/4 time @ 90°F 1/4 time @ 130°F 1/4 time @ 15°F 1/4 time @ -45°F Speed: 80 mph 18 start-stop cycles Load: 20,000 lbs. for 5 1/2 x 10 26,250 lbs. for 6 x 11 30,000 lbs. for 6 1/2 x 12	Temperature: room temperature Speed: 63 mph Load: axle weight only	Temperature: 8,240 miles @ 75°F 2,480 miles @ -45°F 1,880 miles @ 130°F Speed: 80 mph Pressure: 0 psig
Cost	$2,000	$2,500	$500/Set of Four (4)
Reference	AAR Standard, D53-1971	AAR M-917-64	RB-6 Letter. 6/5/72

A Pr(A) curve is given in Figure 4 for the present acceptance test. Under this test, n = 1 bearings are placed on test for 2,050 miles with acceptance of lot if no failures are observed. For the curve, the failure distribution is assumed to have Weibull form with shape parameter $\beta=2$. It is seen from this curve that the present laboratory test has a 98 percent chance of accepting a bearing with only a 10,000 mile L_{10} life and an 80 percent chance of accepting a bearing with an L_{10} life equal to the test time of 2,050 miles.

2.3.2 Field Tests

Once the bearing passes the initial acceptance test, the bearing receives a conditional approval. This approval carries a stipulation limiting the number of applications to 2,000 car sets that may be applied to interchange cars the first year. After 300 car sets of the 2,000 have been in service for a period of not less than one year, the manufacturer may request a joint inspection of four car sets of bearings each having a minimum of 25,000 miles of operation. Should the joint inspection indicate that the bearings have been performing satisfactorily, the manufacturer is authorized an additional sales allotment of 2,000 car sets. In not less than one year after the sale of the first 300 of the additional 2,000 car sets, the manufacturer may request a further inspection for two car sets that have had at least two years service and 50,000 miles of operation; and two car sets of the second group which have had not less than one year's service and 25,000 miles of operation. Should these inspections indicate satisfactory bearing performance, the manufacturer is permitted unlimited sales of such bearings.

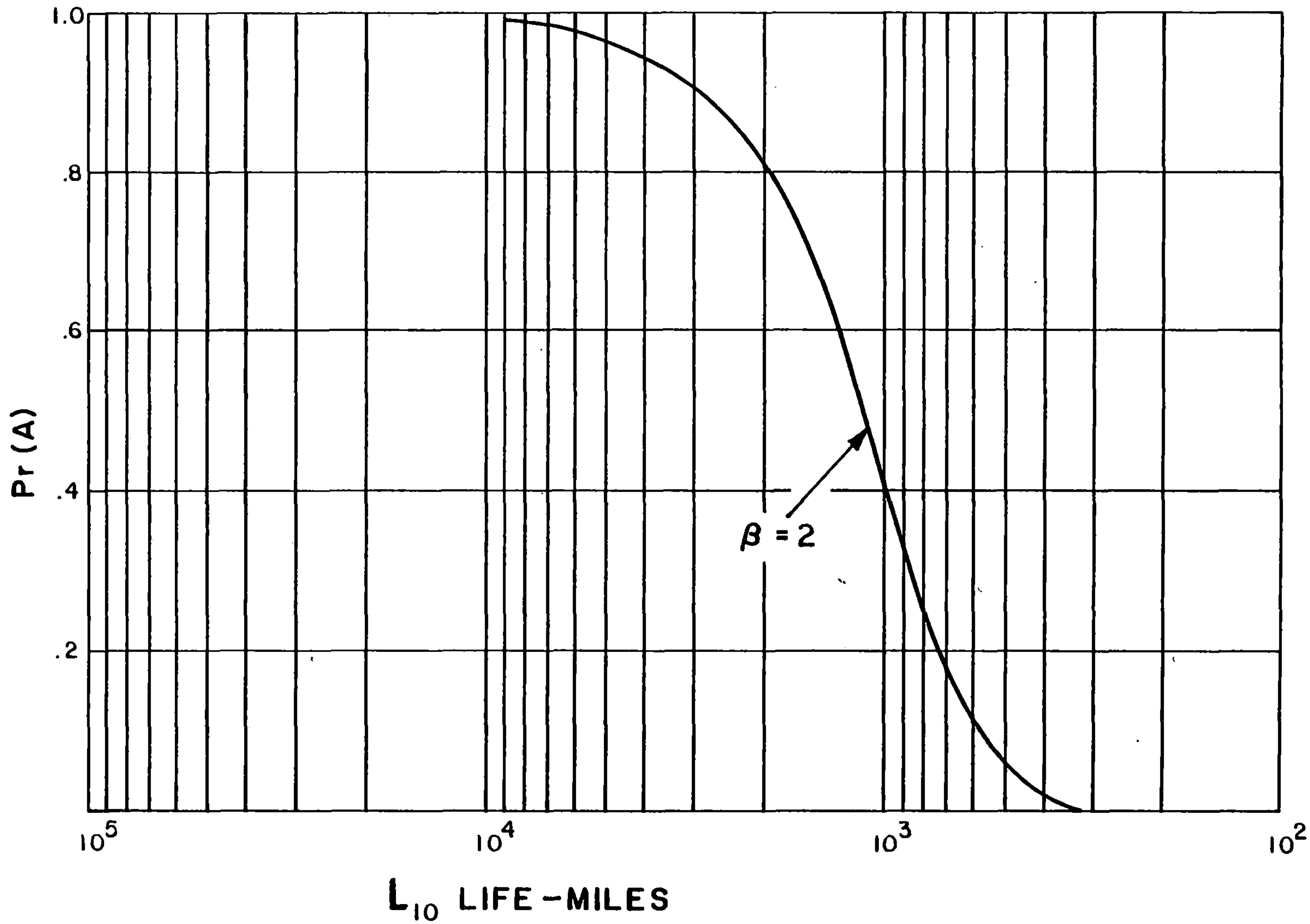


FIGURE 4. PROBABILITY OF ACCEPTANCE OF LOT AS A FUNCTION OF L_{10} USING WEIBULL MODEL WITH SHAPE PARAMETER β AND TEST PLAN. 1 BEARING ON TEST. ACCEPT LOT IF NO FAILURES BEFORE 2050 MILES

The current acceptance procedure for roller bearings is a three stage sampling plan and is illustrated in Figure 5. We now determine the characteristics of this procedure; i.e., the probability of accepting a "good quality" and a "poor quality" bearing. For the purpose of this analysis, a "good quality" bearing is taken as one whose L_{10} life is 500,000 miles while a "poor quality" bearing is taken as one whose L_{10} life is 100,000 miles.

Under the current procedure, a sample of one bearing is laboratory tested. The test bearing is tested for 2,050 miles under rated load. At the end of the 2,050 miles, the bearing is inspected. If the sample bearing fails, the bearing lot is rejected. If the sample bearing does not fail, a second stage of testing (actual use testing) is initiated.

In the second stage, 32 bearings are taken from field use after 25,000 miles of service. The number of defective bearings is recorded. If this number of bearings is greater than 0, the bearing lot is rejected. If the number of bearings so recorded is 0, a third stage of testing is initiated.

In the third stage, 16 bearings are taken from field use after 50,000 miles of service. The number of defective bearings is recorded. The bearing is accepted (approval granted) if 0 defective bearings are found. The bearing lot is rejected if more than 0 defective bearings are found.

It should be noted that in the present acceptance procedure precise numbers have not been adopted for the rejection numbers in Stages 2 and 3.

The current laboratory test has a probability of accepting a poor quality bearing of more than 0.98.

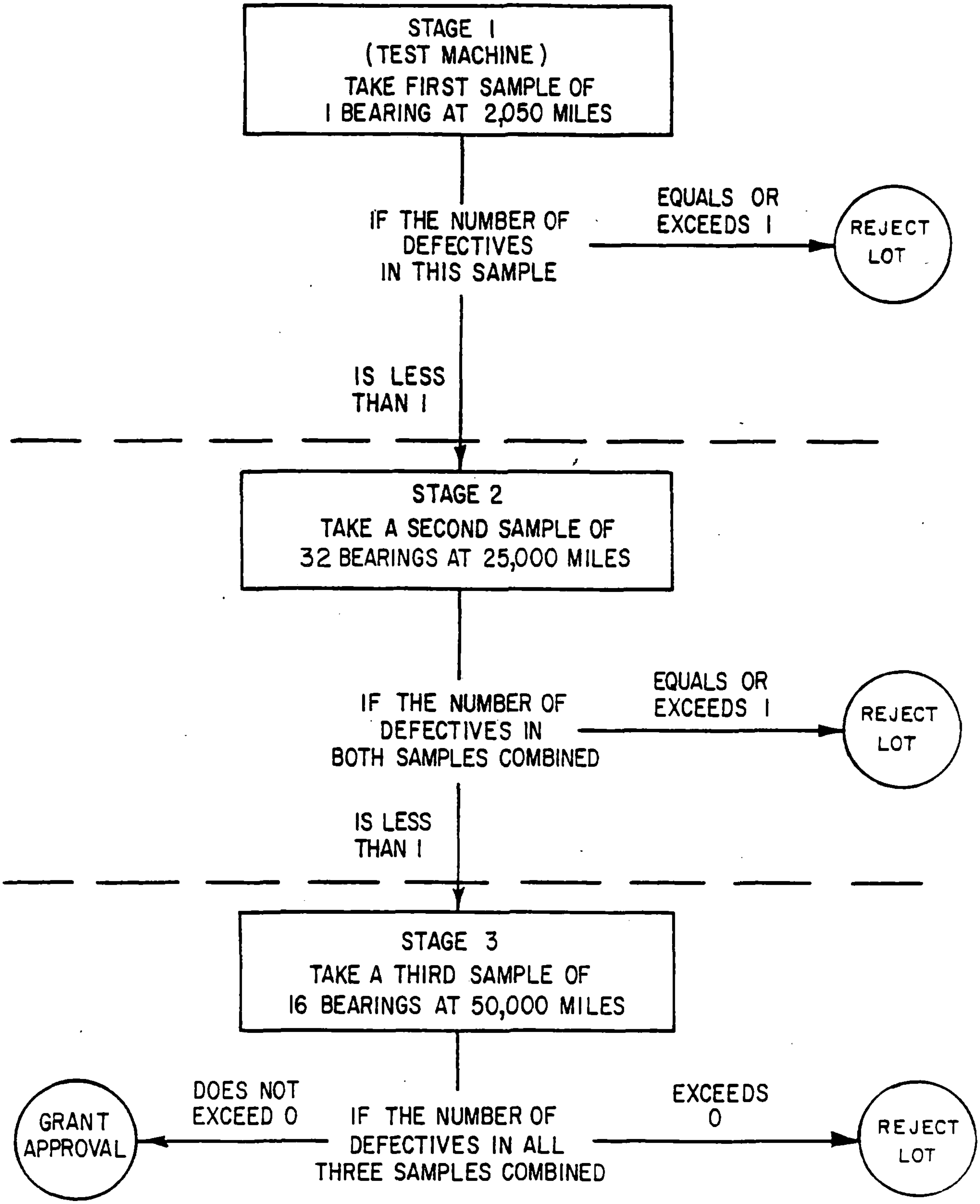


FIGURE 5. "PRESENT" CERTIFICATION PROCEDURE

For the subject analysis the rejection numbers in all three stages were taken to be the most consumer protective. In this way, the characteristics of the procedure in protecting the consumer could be explored. In addition, information would be obtained on how the characteristics would vary as n and the test time, L_t, were changed.

The results of the analysis are shown in Figures 6 and 7. Figure 6 is a plot of the probability of accepting a good* quality bearing (top three lines) and of accepting a poor quality bearing (bottom three lines) as a function of the number of bearings (n). The dependence of the probability of acceptance on the test miles (L_t) is also given.

From the plot, it can be seen that with enough bearings (say 8) in the first stage and enough test miles (say 100,000) in the first stage, the consumer protective policy employed can produce a reasonably low probability of acceptance for a poor quality bearing; i.e., the probability of acceptance of the poor quality bearing is about 0.1. Also, it can be seen that as the number of bearings or the number of test miles in the first stage increases, the probability of accepting either a good or poor quality bearing decreases.

The plot indicates that while the probability of accepting a poor quality bearing can be made suitably low by a suitable combination of n and L_t, the probability of accepting a good quality bearing is not very high. Consequently, this consumer protective policy unduly penalizes the producer. The consequences of making the policy slightly less consumer protective can be ascertained from the graph in Figure 7. This graph is identical to the first graph except that the bearing is not rejected in the first stage if 0 <u>or</u> 1 defects are recorded during the inspection.

*For illustrative purposes a "good" quality bearing is defined as one with a L_{10} life of 500,000 miles.

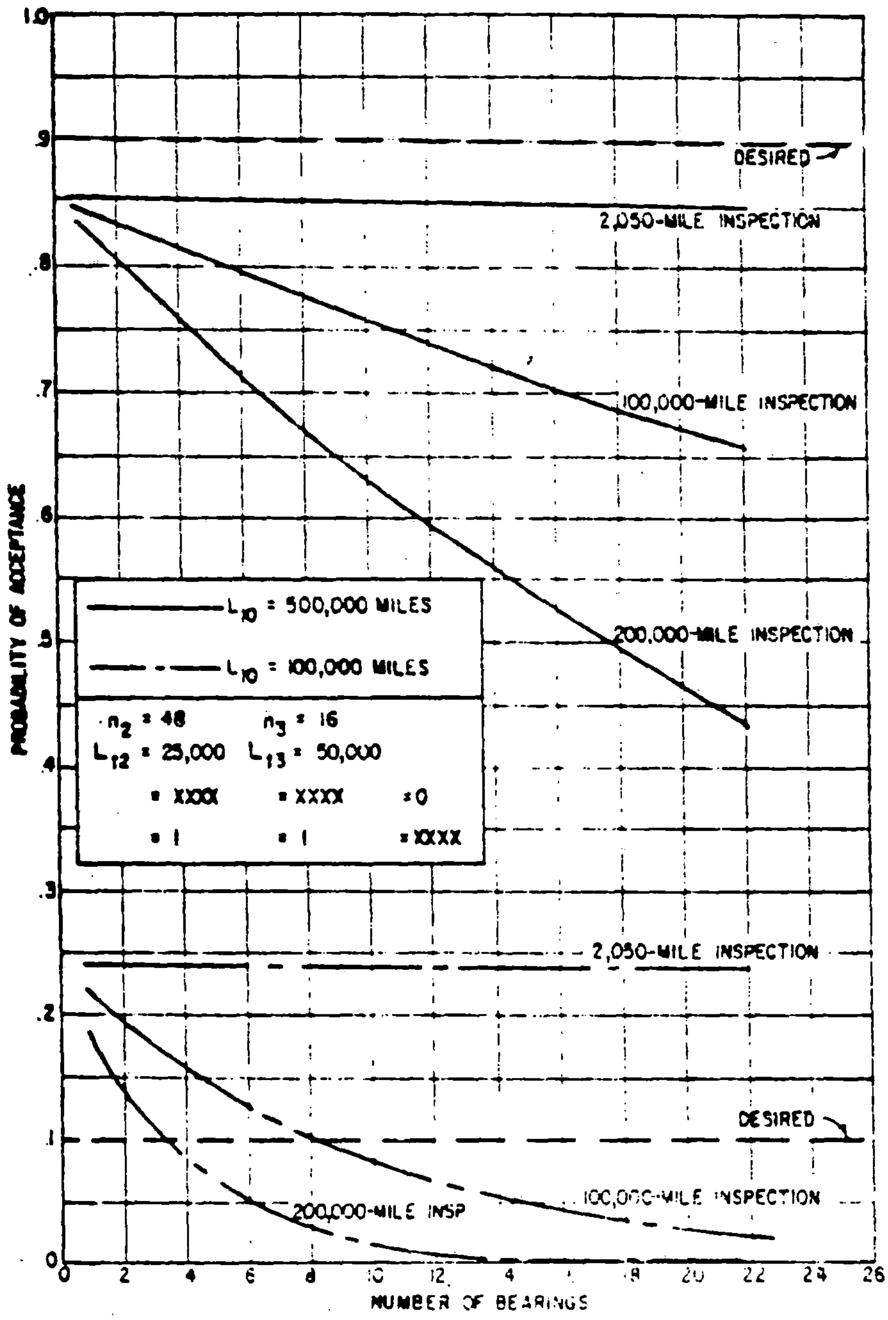


FIGURE 6. PROBABILITY OF ACCEPTANCE VERSUS NUMBER OF BEARINGS. NO ALLOWABLE DEFECTIVE BEARINGS IN THE THREE STAGES

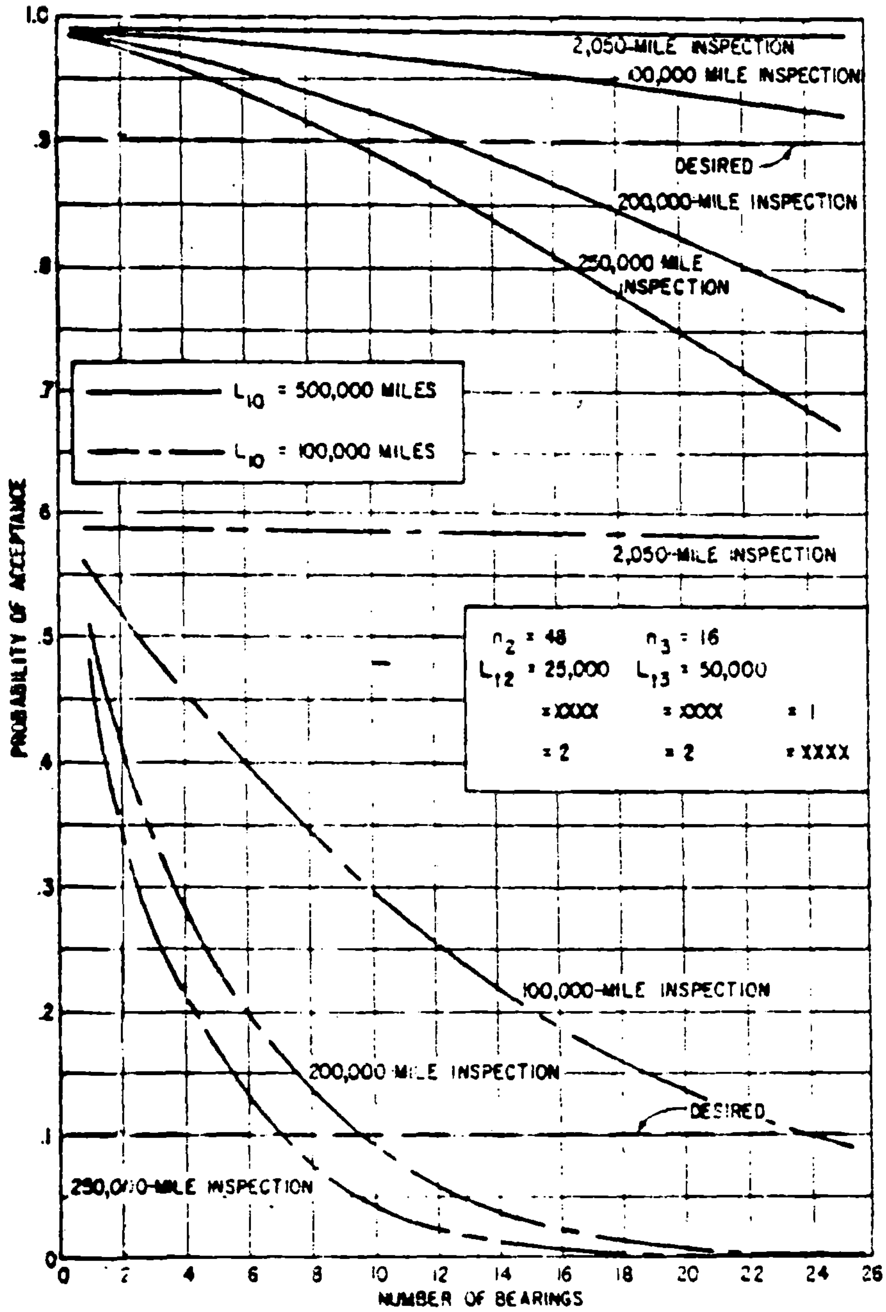


FIGURE 7. PROBABILITY OF ACCEPTANCE VERSUS NUMBER OF BEARINGS. ONE ALLOWABLE DEFECTIVE BEARING IN THE THREE STAGES

Figure 7 shows that all curves in the figure have been raised (when compared to those in Figure 6). In addition, for suitable combinations of n and L_t, it is possible to attain reasonable acceptance probability for both good and poor quality bearings. For example, if $n = 10$ and $L_t = 200{,}000$ miles, the probability of accepting a good quality bearing is greater than 0.9 and the probability of accepting a poor quality bearing is less than 0.1.

The previous discussion has shown that the present acceptance procedure has a high probability of accepting a poor quality bearing and that significant improvements in consumer protection can be achieved by making the procedure more stringent. However, before specific recommendations for an improved procedure can be made, we must first determine the probability distribution to be used for the number of failures. This will be done in Sections 3 and 4.

3. ROLLER BEARING DEFECT AND FAILURE DATA

The first step in determining the probability distribution for time to failure involved obtaining and categorizing bearing defect and failure data such that the dominant modes of failure could be identified. Five sources of data were utilized:

1. Bearing defect data from three bearing rework shops operated by a bearing manufacturer.
2. Joint inspection records of set-outs provided by a bearing manufacturer.
3. Journal performance report provided by the AAR.
4. Confirmed "hot-box" or "burn-off" data compiled by a railroad from AAR records.
5. Seal defect data provided by a bearing manufacturer.

3.1 BEARING REWORK SHOP DATA

Data was collected at three bearing rework shops over a period of approximately two months on every cup and cone which had condemnable defects per Reference (4). For each defective component, the nature of the defect and the date of manufacture were recorded. In addition, for nearly every defective component, the date of manufacture of the next good bearing was noted so as to obtain an estimate of the age distribution of both good and defective bearings. A sample of the raw defect data is shown in Figure 8. The information shown in Figure 8 was then transcribed, keypunched, and finally stored in computer memory. The final data, in the format for computer input, are shown in Figure 9. The information contained for each bearing component (one data line) includes manufacturer, size, date of manufacture, order number, bearing condition, defect, and number of times bearing was previously reworked. The data were then analyzed with a computer to separate defects, age distribution, etc. The total sample size and overall defect data are summarized in Table 2.

BBS CONE DATA

Location Knoxville Customer Southern Total Number of Cones Inspected 150 Date 8-21-74 rev 11/7/74 line 2631-2774(1) 47(2)

S.O. NO. 136 Bearing Size 6x11 BRENCO 60 Timken 193 Hyatt 42 Other Sheet of

Manufacturer	Bore Oversize	Bore Out Of Round	Spalled Race or Roller	Cage Bent, Broken or Worn	Water Etched	Roller Broken or Jammed	Other, Specify	GOOD CONE DATA	
Brenco	2631-39		65 2681				2689-91	2704-16	9
Timken	2640-68	64 2679	66 66 64 64 2682-85		65 65 2686-87		2692-99	2717-58	67
Hyatt	2669-78	64 2680			67 2688		2700-03	2759-74	24
Other									

FIGURE 8. TYPICAL BEARING SHOP DEFECT RECORD

```
3601 T,11,64,K135,G,ØK,0
3602 T,11,62,K135,G,ØK,0
3603 T,11,62,K135,G,ØK,0
3604 T,11,65,K135,G,ØK,0
3605 T,11,67,K135,G,ØK,0
3606 T,11,60,K135,G,ØK,0
3607 T,11,56,K135,G,ØK,0
3608 T,11,63,K135,G,ØK,0
3609 T,11,62,K135,G,ØK,0
3610 T,11,68,K135,G,ØK,0
3611 B,12,69,K135,S,ØS,0
3612 T,12,61,K135,S,ØS,0
3613 T,12,64,K135,B,ØS,0
3614 T,12,61,K135,S,ØS,0
3615 T,12,61,K135,S,ØS,0
3616 T,12,61,K135,S,ØS,0
3617 T,12,64,K135,B,SR,0
3618 T,12,64,K135,B,SR,0
3619 B,12,63,K135,S,ØB,0
3620 T,12,63,K135,S,ØB,0
3621 T,12,63,K135,G,ØK,0
3622 T,12,63,K135,G,ØK,0
3623 T,12,70,K135,G,ØK,0
3624 T,12,70,K135,G,ØK,0
3625 T,12,70,K135,G,ØK,0
3626 T,12,70,K135,G,ØK,0
3627 T,12,61,K135,G,ØK,0
3628 T,12,70,K135,G,ØK,0
3629 T,12,68,K135,G,ØK,0
3630 T,12,67,K135,G,ØK,0
3631 B,11,69,K136,B,ØS,0
3632 B,11,69,K136,B,ØS,0
3633 B,11,69,K136,S,ØS,0
3634 B,11,69,K136,B,ØS,0
3635 B,11,67,K136,B,ØS,0
3636 B,11,67,K136,S,ØS,0
3637 B,11,65,K136,S,ØS,0
3638 B,11,65,K136,B,ØS,0
3639 B,11,65,K136,S,ØS,0
3640 T,11,63,K136,S,ØS,0
3641 T,11,64,K136,S,ØS,0
3642 T,11,60,K136,B,ØS,0
3643 T,11,65,K136,S,ØS,0
3644 T,11,64,K136,B,ØS,0
3645 T,11,64,K136,S,ØS,0
3646 T,11,68,K136,B,ØS,0
3647 T,11,68,K136,S,ØS,0
3648 T,11,66,K136,S,ØS,0
3649 T,11,70,K136,S,ØS,0
3650 T,11,64,K136,B,ØS,0
3651 T,11,68,K136,S,ØS,0
3652 T,11,68,K136,S,ØS,0
3653 T,11,64,K136,S,ØS,0
3654 T,11,59,K136,S,ØS,0
3655 T,11,59,K136,S,ØS,0
3656 T,11,64,K136,S,ØS,0
```

FIGURE 9. TYPICAL BEARING DEFECT DATA INPUT FORMAT

TABLE 2

SUMMARY OF BEARINGS SAMPLED AT REWORK SHOPS

	Cups	Cones	Total
Total Number in Sample	8,090	14,122	22,212
Total Number Defective	1,875	2,069	3,944
Percent Defective	23	15	18

Table 3 shows the number and percentages of defects by defect category, while Table 4 summarizes the data in terms of the total number of defective bearings that did not show bore defects (first two items in Table 3).

This differentation was made for two reasons., The first step in the inspection process was to inspect the bearing bores. Those bearings which had experienced bore defects were removed from the sample at that point, and were thus not inspected for any other defects. Bore size defects were found to be the dominant defect.

Since bore defects dominate, their presence tends to diminish the importance of other defects. One must assume that had the inspection process continued on bearings possessing bore defects, other defects would have been found. Thus, removing the bearings with bore size and shape defects from the sample gives a better indication of the relative importance of other bearing defects.

The rework shop data were also used to construct Weibull defect rate plots for a number of defects. These are included in Appendix A and are discussed in Section 4.

In reviewing the data, it appeared that there was a difference in the defect rate and character by both user railroad and manufacturer. (Each rework shop represented a different railroad). To see whether the differences observed were statistically significant, several contingency table X^2 tests were run. Such tests allow one to determine whether variations in data which are tabulated according to two criteria could have arisen by chance. The conclusion is drawn with respect to a level of confidence; e.g., the probability that the two criteria are not associated is, say, 0.95.

TABLE 3

SUMMARY OF BEARING DEFECTS FOUND AT REWORK SHOPS

	Cups		Cones	
	Number	% Total	Number	% Total
Oversize Seal or Cone Bore	391	20.9	1,227	59.3
Out-of-Round Seal or Cone Bore	94	5.0	11	0.5
Spalled	309	16.5	190	9.2
Brinelled	617	32.9	421	20.3
Broken Cup, Roller, or Cage	239	12.7	63	3.1
Water-Etched	206	11.0	144	7.0
Other	19	1.0	13	0.6
Total	1,875	100.0	2,069	100.0

TABLE 4

SUMMARY OF BEARING DEFECTS FOUND AT REWORK SHOPS

(Bearings with Bore Defects Removed from Data)

	Cups		Cones	
	Number	% Total	Number	% Total
Spalled	309	22.2	190	22.9
Brinelled	617	44.4	421	50.7
Broken Cup, Roller, or Cage	239	17.2	63	7.5
Water Etched	206	14.8	144	17.3
Other	19	1.4	13	1.6
Total	1,390	100.0	831	100.0

From the results of these tests, the following observations were made:

1. With a probability greater than 0.999:
   a) The distribution of defective bearings by defect type varies with manufacturer and with user.
   b) The distribution of oversize bearings, brinelled bearings and spalled cups varies with manufacturer and with user.

2. With a probability greater than 0.995, one railroad had an abnormally high incidence of broken cups.

3. With a probability greater than 0.950, one manufacturer had:
   a) An abnormally high incidence of spalled cones.
   b) An abnormally low incidence of brinelled cups and cones.

4. With a probability of greater than 0.90, one railroad had:
   a) A high incidence of spalling.
   b) A low incidence of brinelling.
   c) A low incidence of oversize cones.

It is not surprising that there is a difference in defect characteristic between different railroads because of the differences in usage. However, the differences between manufacturers would indicate that there may be room for improvement and that bearing specification and/or certification requirement changes may have the effect of reducing the overall bearing defect rate. This aspect is discussed further in Section 4.

3.2 RAILROAD BEARING MANUFACTURER JOINT INSPECTION REPORTS

A sample of 389 joint inspection failure reports covering the period of 1970 to 1974 were also reviewed. The bearings involved were all of one bearing manufacturer and were from two different railroads.

The causes for the set-out and the disposition of each of the 389 bearings is summarized in Table 5. It is interesting to note that more than half of the bearings were set-out due to being overgreased, and that nearly two-thirds were returned to use with the only service, if any, being cleaning, lubrication, and installation of new seals.

TABLE 5

SUMMARY OF SAMPLED JOINT INSPECTION REPORTS

	BEARING DISPOSITION						
SET-OUT CAUSE	RETURN TO SERVICE	REPAIR	SCRAP	UNKNOWN	TOTAL	% TOTAL	% REAL ④
Overgreased	211 ①	5 ②	3 ③	1	220	57	④
Seal Worn, Bent, or Loose	1	3	4	4	12	3	9
Bolts Loose or Missing	0	1	18	2	21	5	16
Adapter Worn or Misplaced ⑦	7	5	23	3	38	10	29
Journal Undersize	0	1	6	1	8	2	6
Spalled	0	1	2	0	3	1	2
Brakes or Wheels Hot	5	0	0	1	6	2	4
Rework Improper	0	0	2	1	3	1	2
Derailment	0	0	0	2	2	-	1
Water in Grease	0	8	3	0	11	3	8
Cone Oversize ⑥	0	5	6	0	11	3	8
Broken Roller	0	0	1	0	1	-	0.6
Brinnel	0	1	0	0	1	-	0.6
Broken Cup ⑤	0	1	1	0	2	-	1.2
Unknown ⑧	0	0	11	0	11	3	8
False Setout 9	33	0	0	6	39	10	④
Total	257	31	80	21	389	100	
% Total	66	8	21	5	100		

① Most Probably Had Seals Replaced as a Matter of Course Prior to Being Returned to Service
② Repair Required for Defects Unrelated to Set-Out
③ Scrapped Due to Heat Related Damage
④ Overgreased and False Setout Removed from Total
⑤ Causing Loss of Lubricant
⑥ Causing Motion Between Cone Bore and Journal
⑦ Usually Causing Adapter to Rub on End Cap
⑧ Either a Burnoff or Bearing Destroyed to Point Where Causative Examination Is Impossible
⑨ No Evidence of High Temperature or Internal Defect

Since many of the railroads have recognized the operational problems caused by overgreasing, the AAR interchange rules were recently changed to permit a longer relubrication interval (from 36 to 48 months) or to eliminate relubrication altogether in the case of the NFL (no field lubrication) bearing. As these changes become implemented, the overgreased bearing problem should be significantly reduced, if not virtually eliminated. However, new emphasis will have to be placed upon seal and lubricant life.

To obtain a better understanding of the causes of bearing problems not associated with overgreasing, the number of overgreased bearings and false set-outs was removed from the sample and the "real" percentage calculated based upon this reduced population. Here it is seen that more than half of the bearing set-out causes are non-bearing-related.

3.3 AAR JOURNAL PERFORMANCE REPORT

Since it can be argued that the sample of 389 bearings may not be representative of all set-outs, the results shown in Table 5 were compared to the AAR bearing performance report for the first half of 1974, which includes reports from all railroads engaged in interchange service. This comparison is shown in Table 6.

Considering the wide variance in inspection techniques and criteria, and the wide variance in maintenance procedures (especially with regard to lubrication) between various railroad shops and yards, the correlation between the AAR bearing performance report and the Table 5 summary is reasonably good.

3.4 CONFIRMED "HOT-BOX" OR "BURN-OFF" DATA

Bearing failure data reduced from AAR work sheets were provided by a major railroad. These data included all reported failed bearings of the three major U.S. manufacturers that occurred during the second half of 1971 and all of 1972. Only bearings built since 1965 were included. "Failure" was determined as a result of either an inspection subsequent to removal because of a hot box set-out, or because of a burn-off. Bearings removed for other reasons, i.e., wheel removal, high water, other defects, were not included. Thus, each bearing included in this subset of data was a potential cause for a catastrophic failure.

TABLE 6

COMPARISON OF FAILURE REPORTS DATA

Set-Out Cause	AAR Journal Performance Report (Percent)	Joint Inspection Reports (Percent)	Confirmed Hot-Box or Burn-Off Data
Adapter Related	6.1	10	4.8
Wheel Related	0.5	2	0
Cap Screw or End Cap Related	7.3	5	10.0
Seal or Lubricant Related	36.7	63	57.8
Backing Ring Related	0.6	0	0
Broken, Fused, Missing Bearing	4.4	3	6.4
Unknown -- Sent to Bearing Shop	15.5	10	21.0
Other	29	7	

The reported causes of failure are summarized in Table 7 and are compared with the joint inspection data and AAR journal performance report data in Table 6. Again, the correlation is reasonably good.

TABLE 7

SUMMARY OF CONFIRMED HOT-BOX OR BURN-OFF DATA

	Number	% Total
Installation Problems		
Loose Fit	16	2.1
Lubrication/Seal		
Defective/Loose Seal	288	37.1
Excessive Lubrication	35	4.5
Leaking Grease	94	12.1
Non-Specification Grease	14	1.8
Dry or Low Grease	15	1.9
Water Contamination	3	0.4
Components		
Adapter Worn/Cocked	37	4.8
Bearing Brinelled	7	0.9
End Cap Missing/Defective	32	4.1
Cap Screw Missing/Loose	46	5.9
Cage Broken/Defective	1	0.1
Spalled Raceways	9	1.2
Visible Mechanical Damage	5	0.6
Other		
Other Cause Determined	36	4.6
Advanced Failure/Destroyed	50	6.4
Undetermined	88	11.3
Total	776	100.0

3.5 SEAL DEFECT STUDY

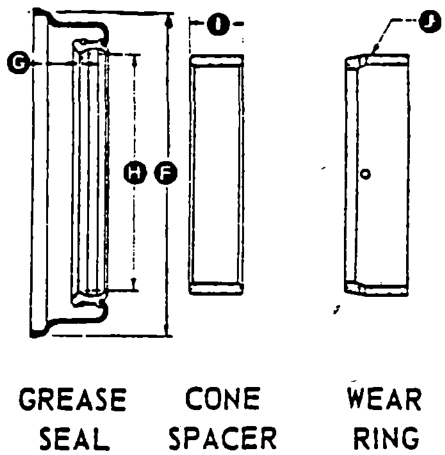
Since the AAR journal performance reports, the joint inspection reports, and the confirmed hot-box of burn-off data previously discussed (Table 6) indicated a high incidence of seal-related problems, a sample of 500 seals was taken for defect analysis during the rework process.

A seal defect was defined on the basis of the old Roller Bearing Manufacturers (RBM) rule number 5.18 (Figure 10). As seal replacement is now mandatory at rework, this rule is obsolete but it still is a valid criterion for a defective seal. In addition to the RBM 5.18 rule, one additional criterion -- defined as "other" -- included such things as "blistered," "cracked," "split," etc.

Two groups of seals were taken. In the first group, data were gathered on 450 seals (6-1/2 x 12 and 6 x 11 sizes). These data included seal manufacturer and date, bearing type, average seal case outer dimension ().D.), pass or fail notation on undersize wear ring test, width of primary lip wear path, and any relevant comments. Figure 11 shows a typical sample of the data after they had been prepared for computer processing. Table 8 explains the code used.

TABLE 8 DATA FORMAT CODE

Seal No.	Seal Date/Type	Make
1	Ex. 7-60 T = Timken B = Brenco H = Hyatt	N = National MLP = Mich. Precis. B = Brenco C/R = Chicago Rawhide U.S. = Sealastomer



Bearing Class. (Size)	D (5½ x 10)	E (6 x 11)	F (6½ x 12)	G (7 x 12)
MAX. ROUNDNESS LIMITS	−.004" TO + .011" FROM F MIN. BUT MUST AVERAGE WITHIN F SIZE LIMITS.			
F MAJOR DIA. MAX.	7.763"	8.263"	9.388"	10.288"
F MAJOR DIA. MIN.	7.757"	8.257"	9.382"	10.282"
G SEAL LIP WEAR	WEAR PATH MUST BE LESS THAN 1/8" WIDE.			
H SEAL LIP FIT	MUST PICK UP WEAR RING WHEN FITTED ON .010" UNDERSIZE DIA. RING			

Reference: Roller Bearing Manufacturers rule number 5.18.

FIGURE 10. RAILROAD ROLLER BEARING SEAL CONDEMNING LIMITS

Date of Mfg. | Type | Make | Diameter | Fit | Wear

Disc Sector

Data No.

```
 519
0731 63BN   9.334Y1/4
0747 60TN   9.380M3/32
0751073TN   9.363Y1/32
0765 68TC/R9.379Y3/32

 520
0771073TN   9.378Y1/32
0783 74TN   9.381Y1/32
0796 67TC/R9.380M1/8
0801073TN   9.380Y1/16

 521
0811073TN   9.381Y1/32
0821267TC/R9.380M1/8
0833 63TUS 9.380Y1/16
0845 68TUS 9.382Y1/16

 522
0852 73TN   9.380Y1/32
0862 73TN   9.388Y1/16
0877 67TUS 9.381Y1/16
0883 68TUS 9.379Y1/16

 523
0891073TN   9.381Y1/32
0901073TN   9.378Y1/32
0911073TN   9.378Y1/16
0925 74TN   9.382Y1/32

 524
0935 68TC/R9.379Y1/32
0942 73HN   9.381Y1/16
0954 65TMLP9.333Y3/32
0963 67TN   9.383M3/32

 525
0971 66BN   9.380Y1/8
0983 65TN   9.381M1/8
0993 65TN   9.381M1/8
1002 62BN   9.383M1/4

 526
1016 68BN   9.382M1/4
1027 67TMLP9.379Y1/16
1036 63BN   9.358M1/4
1049 65TMLP9.353Y1/16
```

FIGURE 11. TYPICAL SEAL DEFECT DATA

In the second group, detailed data on 50 seals were collected which provided dates on seals as well as the associated bearing manufacturer and rework date. Also included were detailed mechanical test data (seal lip micro-hardness and seal removal/installation force), and average inner dimension (I.D.) of the seal primary lip.

An overview of the number and nature of the defects found in the first group of 450 seals is summarized in Table 9.

TABLE 9 DISTRIBUTION OF DEFECTS FOR FIRST GROUP OF 450 SEALS

Defect Mode	Number of Defects
Wear	128 (1/8 in = 80; > 1/8 in = 48)
Diameter	229
Fit	96
Other	13
Total Defects	466
Total Defective Seals	294

It is seen that the greatest defect found was "diameter out of specification," followed by "wear," "fit" and "other." The total number of defects is larger than the total number of defective seals due to the fact that some seals exhibited more than one defect.

These data were used to perform three types of analysis: distributuon of defect mode as a function of time, hazard plot of each defect mode, and a Weibull distribution of each defect mode. These analyses are described in Section 4.4 and Appendixes B and C.

3.6 SUMMARY OF SIGNIFICANT DEFECTS AND FAILURE MODES

All of the significant bearing defects and causes of setouts (excluding over-greased and false) from Tables 3 through 7 are summarized in Table 10. The defects initially considered for improved certification procedures are shown in the right-hand column of Table 10. Those not checked are either clearly non-bearing-related or seem to be of minimal importance.

Although some defects appear to have a low incidence of occurrence, (i.e., broken roller) they can readily lead to catastrophic failure and are thus probably high-cost problems. Others (e.g., brinelling) appear to have a low incidence relative to setouts, yet are the cause for a large number of scrapped bearings -- again a likely high-cost problem.

Furthermore, all of the checked defects (with the possible exception of water etching) can lead to catastrophic failure. Although spalling and brinelling are not usually thought of as catastrophic failures, spalling can lead to excessive debris generation and eventual jamming or breakage, while brinelling can lead directly to breakage or spalling. Breakage can clearly lead to catastrophic failure. Similar paths to catastrophic failure can be constructed for the other checked defects. Water etching, per se, is probably not as serious a defect as some of the others. However, it is an indication that excessive water has entered the bearing and that the lubricant has been seriously degraded. Thus, water etching was maintained as an initial area of concern for the diagnostics portion of the work (Volume II) but was not considered for improved acceptance testing.

Each of the defects that were checked on Table 10 was considered a potential subject of the following:

1. Improvement by design, manufacturing technique, or quality control,
2. Improvement in certification procedure,
3. Improvement in railroad maintenance practices, and
4. ~~Improvement of diagnostic devices.~~

These are summarized in Table 11.

TABLE 10

SUMMARY OF BEARING AND BEARING-RELATED PROBLEMS

	Rework Shop Percent of All Defective (1)	Rework Shop Percent of All Defective, Less Bore Defects (1)	Joint Inspection Reports Percent of All, Less Overgreased and False Set-Out	Confirmed Hot-Box and Burn-Off Data	Defects to Be Considered for Improved Certification
Spalled	16.5	22.9	2	1.2	X
Brinelled	32.9	50.7	0.6	0.9	X
Water Etched	11.0	17.3	0	0	
Broken Cup	12.7	17.2	1.2	0.6	X
Broken Roller/Cage	3.1	7.5	0.6	0.1	X
Oversize Cone	59.3	--	8	2.1(2)	X
Out of Round	5.0	--	0	0	
Other	1.0	1.6	0	15.9	
Seal Worn, Loose, or Bent			9	37.1	X
Bolts Loose or Missing			16	10.0	
Adapter Worn, Misplaced, or Broken			29	4.8	
Journal Undersize			6	(2)	
Brakes or Wheels Hot			4	0	
Improper Rework			2	0	
Derailment			1	0	
Water in Grease			8	0.4	
Unknown (Bearing Destroyed)			8	6.4	
Loose or Broken Backing Ring			0	0	
Improper Lateral			0	(2)	

(1) Largest of Cups and Cones
(2) Total of "Loose Fit"

TABLE 11

AREAS OF POSSIBLE IMPROVEMENT TO REDUCE SPECIFIC DEFECTS

Defect	Possible Subject of Improved Design, Mfg. or Q.C.	Possible Subject of Improved Certification Technique	Possible Subject of Improved R.R. Main. Practice	Possible Subject of Improved Diagnostic Device
Spalled	Yes	Yes	No	Yes
Brinelled	Yes	Yes	No	Yes
Water Etched	Yes(1)	Yes(1)	Yes	Yes
Broken Cup	Yes	Yes	Yes	Yes
Broken Roller/Cage	Yes	Yes	No	Yes
Oversize	Yes	Yes	Yes	Yes
Seal Worn, Loose, or Bent	Yes	Yes	Yes	Yes
Bolts Loose or Missing	No	No	Yes	Yes
Adapter Worn, Misplaced, or Broken	Yes	No	Yes	Yes
Journal Undersize	No	No	Yes	Yes
Improper Rework	Yes	No	Yes	Yes
Water in Grease	Yes(1)	Yes(1)	Yes	Yes
Loose or Broken Backing Ring	Yes	No	Yes	Yes
Improper Lateral	No	No	Yes	Yes

(1) As related to the seal

The treatment of diagnostic devices is covered in Volume II of the final report (5) prepared under this contract.

3.7 DESIGN ASPECTS

Although the bearing-related design aspects are not specifically a part of this program's work scope, they do have a signficant influence on the overall bearing failure problem as evidenced by the conclusions made earlier that there is a difference in the rate of occurrence of different defects among different manufacturers and that there is a large occurrence of certain defects, e.g., oversize cones. Thus, the design, manufacturing, and quality control aspects that lead to these differences and occurrences have an influence on the acceptance test aspects of the program.

It is also evident that the adapter and the backing ring are significant factors in the overall bearing problem. These two components and their mating parts (side frame and axle) are certainly candidates for redesign in any new suspension system development effort.

4. ROLLER BEARING RELIABILITY CHARACTERISTICS

The railroad roller bearing assembly is made up of three components:

1. The bearing itself (cup, cones, spacer).
2. The seals.
3. The grease.

The survival of a roller bearing assembly requires survival of all its components. Utilizing the theorem that the probability of the joint occurrence of independent events is given by the product of their respective probabilities, the probability of survival (or reliability) $R(t)$, of the bearing assembly is the product of the reliabilities $R_1(t)$, $R_2(t)$, $R_3(t)$ of the bearing, grease, and seal. This product can be written as

$$R(t) = R_1(t) \cdot R_2(t) \cdot R_3(t) \tag{1}$$

where t is the life in hours and the failure distribution function (percent failed for the assembly), $F(t)$, is

$$F(t) = 1 - R(t). \tag{2}$$

The assembly life is then described by the equation*

$$\left[\frac{t}{B_{10}}\right]^{\beta} = \sum_{i=1}^{3}\left[\frac{t}{B_{10_i}}\right]^{\beta_i} \tag{3}$$

where

$$\beta = \frac{\ln \sum_{i=1}^{3}\left[\frac{t}{B_{10_i}}\right]^{\beta_i}}{\ln \left[\frac{t}{B_{10}}\right]} . \tag{4}$$

*Weibull distributions are assumed for $R(t)$, $R_1(t)$, $R_2(t)$, $R_3(t)$. See Appendix D.

Thus, the bearing assembly can be described by a simple expression and the contribution of each component to the reliability of the system can be evaluated.

This section presents a study of the failure and defect behavior observed for the components of the tapered roller bearings in railroad service. The observed data have been fitted with a Weibull failure distribution and estimates have been made of the Weibull parameters. A comparison is also made between the incidence of fatigue spalls and estimates of fatigue life based on the method of the Anti-Friction Bearing Manufacturers' Association (AFBMA).

4.1 BEARING DESCRIPTION

The most common railroad roller bearing is a double row tapered configuration as shown in Figure 12. The outer race is generally referred to as the cup. The two inner races or cones are separated by a spacer and held on to the axle by an end cap. Table 12 represents the pertinent dimensions for the two sizes considered in this report.

4.2 BEARING FAILURES AND DEFECTS

The incidence of bearing failures and defects is of great importance from the point of view of both safety and economics. Further, the variation of defect incidence with age is an important factor in deciding on maintenance and/or replacement policy. For the purposes of this report, a bearing failure is defined as a verified hot box or derailment. A defect is a condemnable defect as defined by Reference (4). See Table 13 for brief definitions. Note that a "defective" part is not necessarily nonfunctional from a service standpoint.

4.2.1 Bearing Failure Rate

To assess the seriousness of the bearing problem to the railroad industry and to the public, bearing failures (verified hot boxes and derailments) reported by the Association of American Railroads have been tabulated for the period from July 1, 1971 through December, 1972. These are summarized in Table 14.

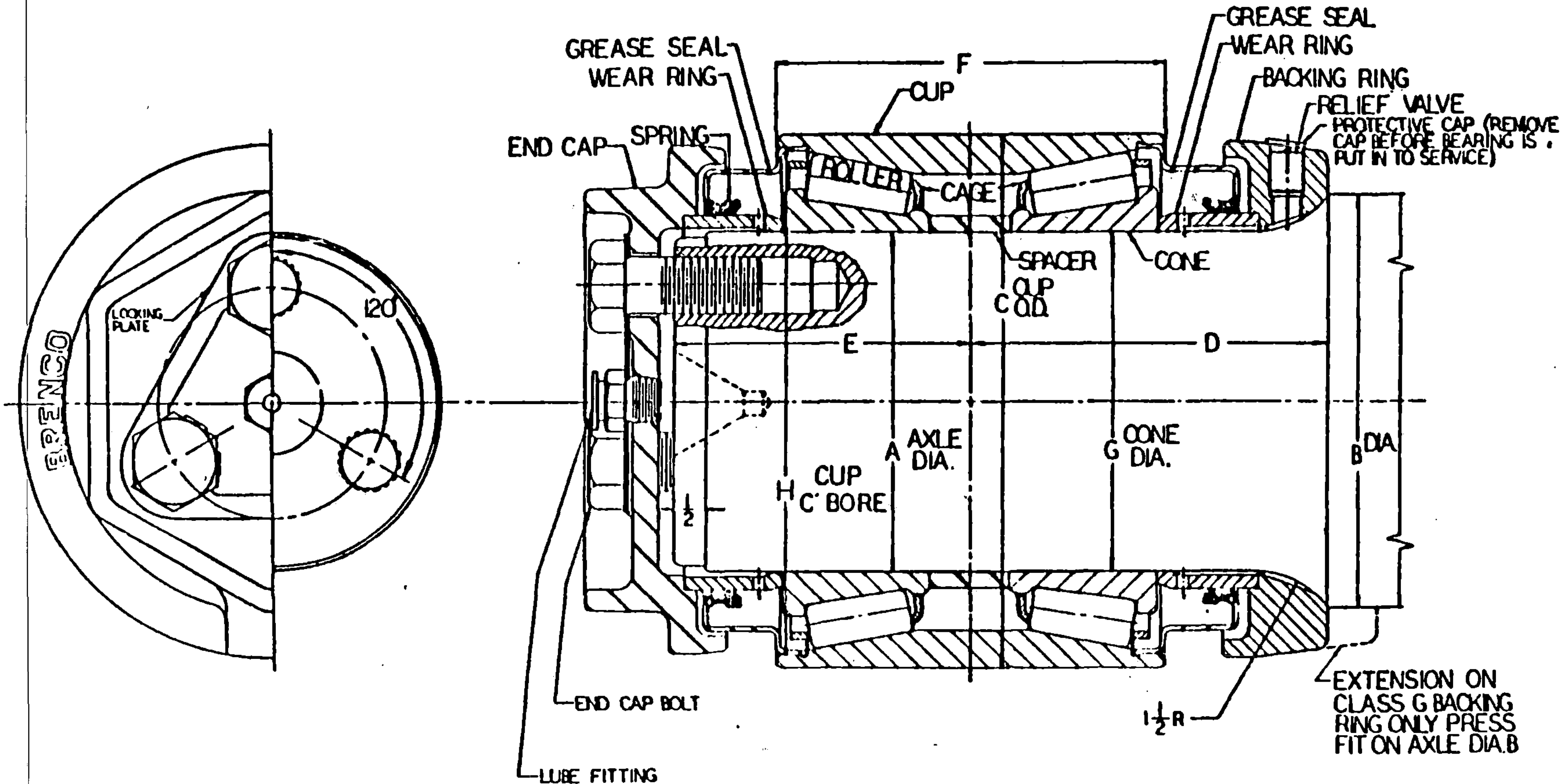


FIGURE 12. CROSS SECTION OF TYPICAL RAILROAD ROLLER BEARING

TABLE 12

NOMINAL DIMENSIONS OF SAMPLE BEARINGS

Class Size	A Min. (New) in. (mm)	B Nom. in. (mm)	C Min. Avg. (Used) in. (mm)	D Nom. in. (mm)	E Nom. in. (mm)	F Nom. in. (mm)	G Max. Avg. (Used) in. (mm)	H Max. Avg. (Used) in. (mm)
E (6x11)	5.6905 (144.54)	7.0 (178)	8.6775 (220.41)	5 15/16 (151)	5 (127)	6.437 (163.5)	5.6890 (144.50)	8.254 (209.65)
F (6 1/2x12)	6.1905 (157.24)	7.5 (191)	9.9275 (252.16)	6 7/16 (164)	5 5/16 (135)	7.250 (184.2)	6.1890 (157.20)	9.379 (238.23)

TABLE 13

SERVICE DEFECT DEFINITIONS FOR RAILROAD ROLLER BEARING

OVERSIZED:	An inner ring or cone is rejected if the bore exceeds an average diameter which would only provide a 0.0015 inch (0.038 mm) tight fit with the minimum AAR journal diameter (new cones provide a 0.002 inch (0.51 mm) to 0.004 inch (0.102 mm) tight fit depending on cone and journal intolerance size variation). An outer race or cup is rejected if the average seal counterbore I.D. would not provide at least 0.004 inch (0.102 mm) tight fit with the minimum diameter AAR approved seal case O.D. (normal fits usually range from 0.006 inch (0.152 mm) to 0.015 inch (0.381 mm) tight).
SPALLED:	Fatigue pits or spalls on a cup race which would result in a spall greater than 3/8 inch (9.525 mm) in any surface direction are cause for rejection. For all practical purposes, any spall on a cone greater than "pin head" size (about 1/32 inch (.787 mm) was the criterion used when data were collected) would cause rejection of a cone.
BRINELLED:	A "heavy" roller indentation in a cup that exceeds one-half raceway or a moderately severe brinell that extends across the entire raceway will cause cup rejection. Criteria used for "heavy" and "moderate" brinells were more critical than current limits of 5/32 inch (3.962 mm) and 3/32 inch (2.362 mm) wide, respectively, now defined in current revision of Reference (3). All cones mating with condemned cup race were rejected as brinelled also.
WATER-ETCHED:	Corrosion or etching caused by water or acidity which results in corrosion lines and pit marks is cause for rejection if it cannot be removed from rollers or raceways by polishing. Current interpretation of guidelines in Reference (3) allows some residual etching after polishing.

TABLE 14

FREIGHT CAR ROLLER BEARING

FAILURE RATES BY YEAR*

CONFIRMED HOT BOXES AND DERAILMENTS

FOR PERIOD JULY 1, 1971 THROUGH DECEMBER, 1972

Year of Manufacture	Sales** Thousands of Car Sets	Average Age (Years)	Number of** Failures	Failures per Thousand Car Sets
1965	27.8	7	55	1.978
1966	36.6	6	75	2.049
1967	32.8	5	51	1.555
1968	25.5	4	38	1.49
1969	35.8	3	26	0.726
1970	40.9	2	37	0.9046
1971	36.2	1	17	0.4696
1972	12.5***	0.5	3	0.24
Total	248.1		302	1.217

* Data courtesy of the Association of American Railroads, Semi-Annual Report.

** Data for representative subset of the roller bearing population.

*** One vendor only.

The failure rate or hazard was represented by the Weibull model. The hazard equation for the instantaneous failure rate at age t is:

$$h(t) = \frac{\beta}{\eta\beta} (t - \gamma_o)^{\beta-1}. \tag{5}$$

The equation of best fit (yielding values η and β) can be obtained by logarithmetically transforming both the instantaneous hazard rate and the component age and then performing a simple linear least-squares regression on the transformed data. The result of the regression is then:

$$\ell n\ h(t) = (\beta-1)\ \ell n(t) + \ell n(\beta/\eta^{\beta}) \ , \tag{6}$$

where:

$(\beta-1)$ = the regression coefficient.
$\ell n(\beta/\eta^{\beta})$ = the intercept obtained from the regression.

The statistical significance of the linear regression can be tested by: 1) an F test from an analysis of variance of the regression or, 2) reference to a table of critical values of the correlation coefficient. For the roller bearing data, the regression is statistically significant.

Figure 13 shows the result of the regression on failures per thousand car sets versus age in years for a representative subset of roller bearings for which sales data were available. The curved lines on the figure are the 95 percent confidence bands on the regression line.

The total number of confirmed bearing failures is relatively small when compared to the total population (roughly 0.9×10^6 car sets in 1974). The important factor is that the instantaneous failure rate increases with bearing age (i.e., $\beta > 1.0$) and the incidence of failures can only increase as the average age of the railroad bearing population rises.

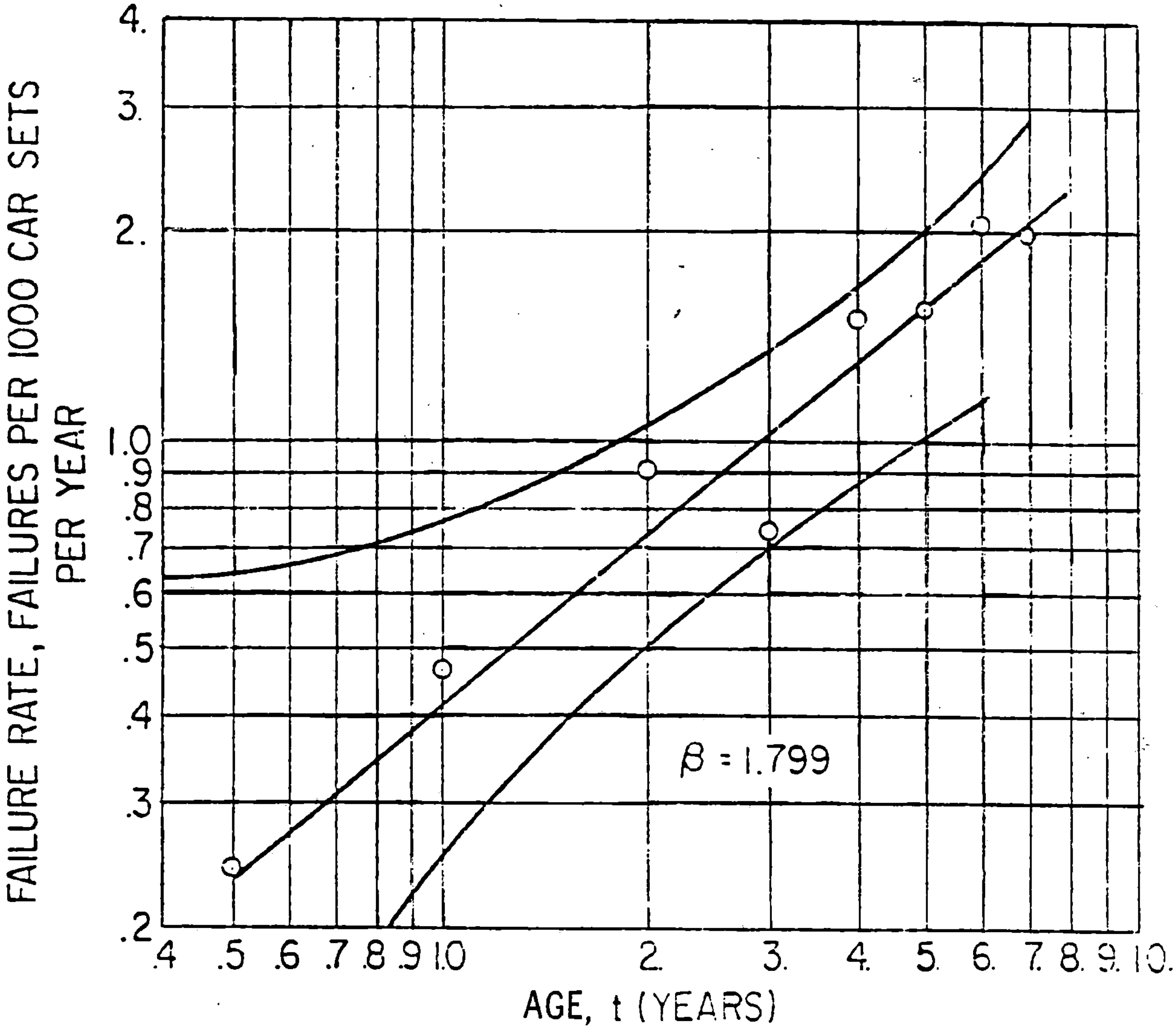


FIGURE 13. FAILURE RATE CURVE FOR RAILROAD ROLLER BEARINGS

It should be noted that the curve fit of Figure 13 assumes $R(t) \rightarrow 1$. For example, in calculating the hazard it was assumed that all of the bearings sold in 1965 were still in the population in 1972. This assumption tends to underestimate the value of β, and therefore, the estimated hazard rate will be a lower bound on the actual hazard rate.

4.2.2 Acquisition of Data on Defective Bearings

Data were collected at three bearing rework shops over a period of approximately two months on every cup and cone which had condemnable defects per Reference (4). For each defective component, the nature of the defect and the date of manufacture were recorded. In addition, for nearly every defective component, the date of manufacture of the next good bearing in the inspection line was noted to obtain an estimate of the age distribution of both good and defective bearings. The information contained for each bearing component (one data line) included manufacturer, size, date of manufacture, order number, bearing condition, defect, and number of times the bearing was previously reworked. The total sample size and overall defect data were summarized previously in Table 2.

The data obtained from the rework shops were analyzed to yield statistically useful information. This included age and age distribution information for the various defect modes as well as confidence limits for many of the values determined.

First determined for the sample was the defective proportion of the bearings manufactured in each year. For this determination, the number of defective bearings for each defect type and for each year was obtained from the data file. An estimate of the number of good bearings made in the year under consideration was added to the number of defective bearings made in that year. This estimation was necessary because the age distribution of only a fraction (called "O.K." bearings) of the good bearings was known. The estimate assumed that the age distribution for this "O.K." fraction of good bearings was representative of that for all the good bearings in the sample. Consequently, the number of bearings manufactured in each year could be written as $D(t) + G(t)$; where $D(t)$ and $G(t)$ are, respectively, the numbers of defective and "O.K." bearings made in year t and where f is the ratio of the total number of good bearings in the sample to the total number of "O.K." bearings in the sample.

For the oversize bearing defect, the defective bearing proportion in the yearly population was taken to be $O(t)/(D(t) + G(t)f)$, where $O(t)$ is the number of oversize bearings made in year t. For the remainder of the defects, a modified version of this ratio was deemed to be necessary because of the bearing inspection procedure used in the rework shops. This procedure involved first checking the bearings for oversize and for out-of-round. If these defects were found, the bearings were appropriately categorized; however, checks for additional defects in those bearings were not made. Consequently, the defective bearing population undoubtedly contained many oversize and out-of-round bearings which were also spalled, brinelled, etc. The modified version of the above ratio tended to correct for the manner in which the bearings were inspected. For spalling, brinelling, etc., the modified version of the ratio is $B(t)/(D(t) - O_1(t) + G(t)f)$ where $B(t)$ is the number of spalled (or brinelled, etc.) bearings made in year t and where $O_1(t)$ is the number of oversize and out-of-round bearings made in year t. In effect, the proportion given by this ratio is that for the bearings which remained after the oversize and out-of-round bearings had been removed.

The results obtained by use of the above procedure are given in Tables 15 and 16. Only bearings having an age of 12 or less are listed. Older bearings were also considered but not included because the confidence limits* on the proportions obtained were excessive.

*The confidence limits calculated were those on the proportion of defective bearings in the general field bearing population. For this, it was assumed that the sample obtained was representative of the bearings in the field. While bearings removed because of car derailments (AAR rule) constituted a significant percentage of the total received, this assumption seemed appropriate since the vast majority of the bearings in the sample arrived there because of events not associated with bearing performance.

TABLE 15

PERCENT OF DEFECTIVE CONES --
VARIATION WITH DEFECT TYPE AND WITH AGE

	Age, Years											
Mode	1	2	3	4	5	6	7	8	9	10	11	12
Total	4.24	10.43	5.84	8.29	12.76	13.40	9.71	12.80	15.26	16.62	19.42	19.48
Spalled	1.43	0.59	0.43	1.27	0.53	0.86	1.51	1.39	1.65	1.93	1.78	2.43
Oversize	1.41	4.66	2.50	4.25	7.06	8.27	5.24	6.29	8.04	9.75	13.61	12.40
Brinelled	0	2.75	0.43	1.06	2.53	2.76	1.51	2.70	4.64	4.35	3.76	4.10
Other*	1.40	2.43	2.48	1.71	3.06	1.51	1.45	2.42	0.93	0.59	0.27	0.64
Oversize, Spalled, & Brinelled	2.84	8.0	3.36	6.58	9.96	11.89	8.26	10.38	14.29	16.03	18.89	18.31

* Other includes "bore out-of-round," "cage bent, broken or worn," "water-etched," "roller broken," and "other miscellaneous."

TABLE 16

PERCENT OF DEFECTIVE CUPS —

VARIATION WITH DEFECT TYPE AND AGE

Mode	Age, Years											
	1	2	3	4	5	6	7	8	9	10	11	12
Total	11.65	13.39	9.34	11.71	16.71	21.65	16.85	27.03	25.51	24.66	26.65	30.62
Spalled	0	0.24	1.11	2.03	1.35	2.93	3.20	4.45	4.37	6.85	4.55	8.63
Oversize	4.37	2.50	1.30	1.00	3.43	2.67	2.81	4.76	4.32	3.00	7.05	7.12
Brinelled	4.64	2.62	2.00	3.05	4.26	8.64	4.37	8.01	9.82	8.84	10.39	10.52
Other*	2.64	8.04	4.93	5.63	7.66	7.41	6.47	9.81	7.00	5.97	4.66	4.35
Oversize, Spalled, & Brinelled	9.01	5.36	4.41	6.08	9.05	14.24	10.38	17.22	18.51	18.69	21.99	26.27

* Other includes "broken," "water-etched," "out-of-round," and "miscellaneous."

4.2.3 Bearing Defect Rate

The sample of bearings taken in three rework shops was used to determine the extent of defects in the railroad roller bearing population and the way that the incidence of these defects varies with age. Figures 14 and 15 show the percentage of bearings found defective as a function of bearing age for both cones and cups. The percentage of bearings found defective increases with bearing age and three defect modes become dominant with increasing age -- spalling, brinelling, and oversize.

These data now permit us to determine a probability distribution for predicting time to defect. Since the bearing industry has traditionally used a Weibull failure distribution, the analyses performed here also make the same assumption.

Accordingly, the data collected in the rework shops were fitted with a Weibull distribution as described in Appendix D. The cumulative defects are shown in Figure 16.* The data for both cups and cones for all defect modes exhibit a Weibull slope of approximately 1, which means that the defect distribution is approximately exponential; i.e., the defect rate is approximately constant. Although the defect rate is approximately constant, the absolute number of defective bearings in the population will increase with time because of the accumulation of defective bearings in the population and because the population is increasing.

The number of defective bearings in the population of bearings made in a given year, is, essentially, the number which has accumulated since the bearings were new. Consequently, as the bearings age many will develop more than one defect.** Neverthless, the defects associated with the highest Weibull slope will eventually be present most frequently in the bearings made in a particular year. Such defect modes, called

*This calculation assumes that defects in cones are not independent events.

**Only one defect was associated with a defective bearing by the rework shops. Some bearings, of course, had more than one defect. Consequently, the defect proportion results discussed here can be regarded as lower bounds on the actual defect proportions.

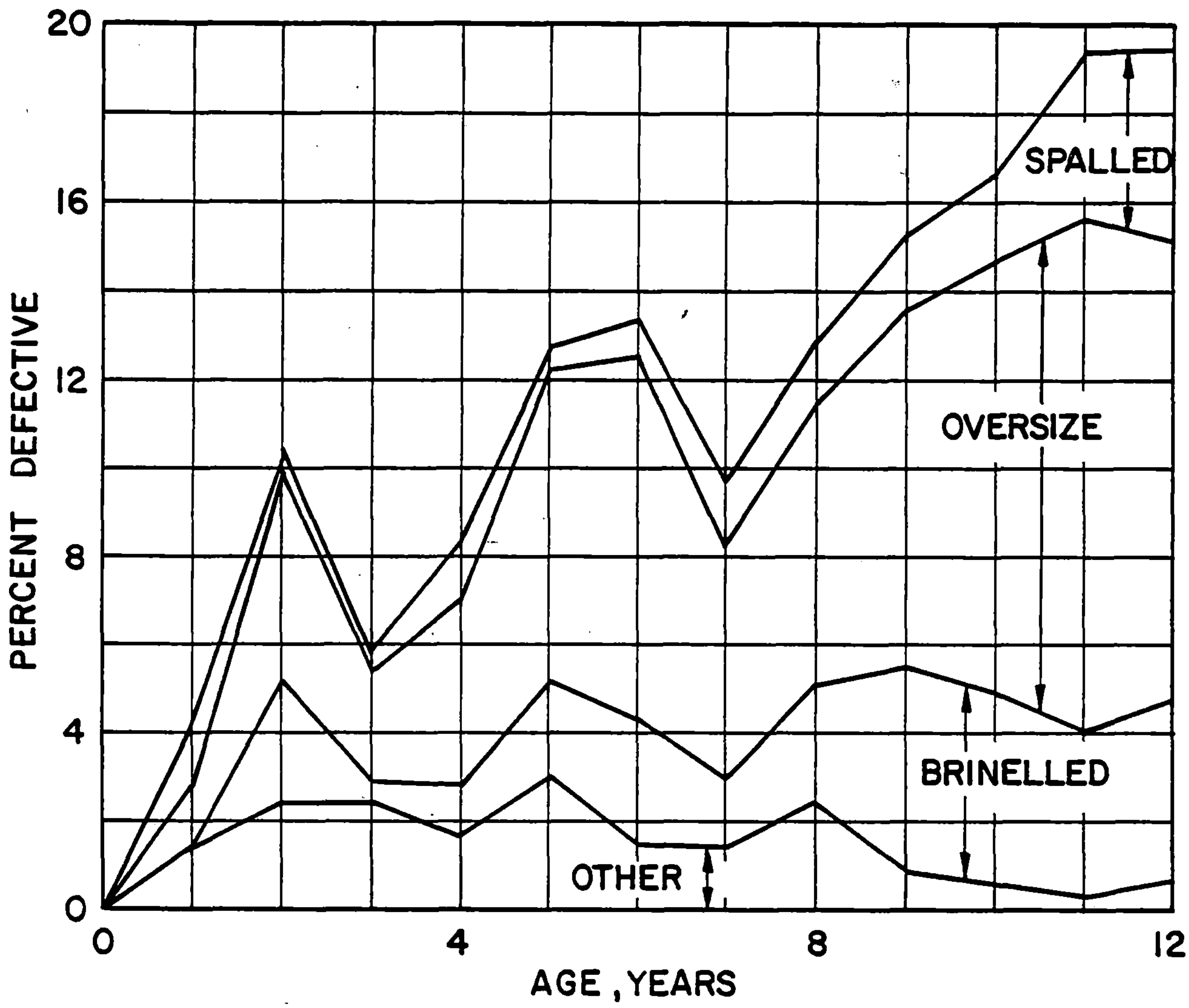


FIGURE 14. PERCENT DEFECTIVE CONES VERSUS AGE FOR VARIOUS DEFECT MODES

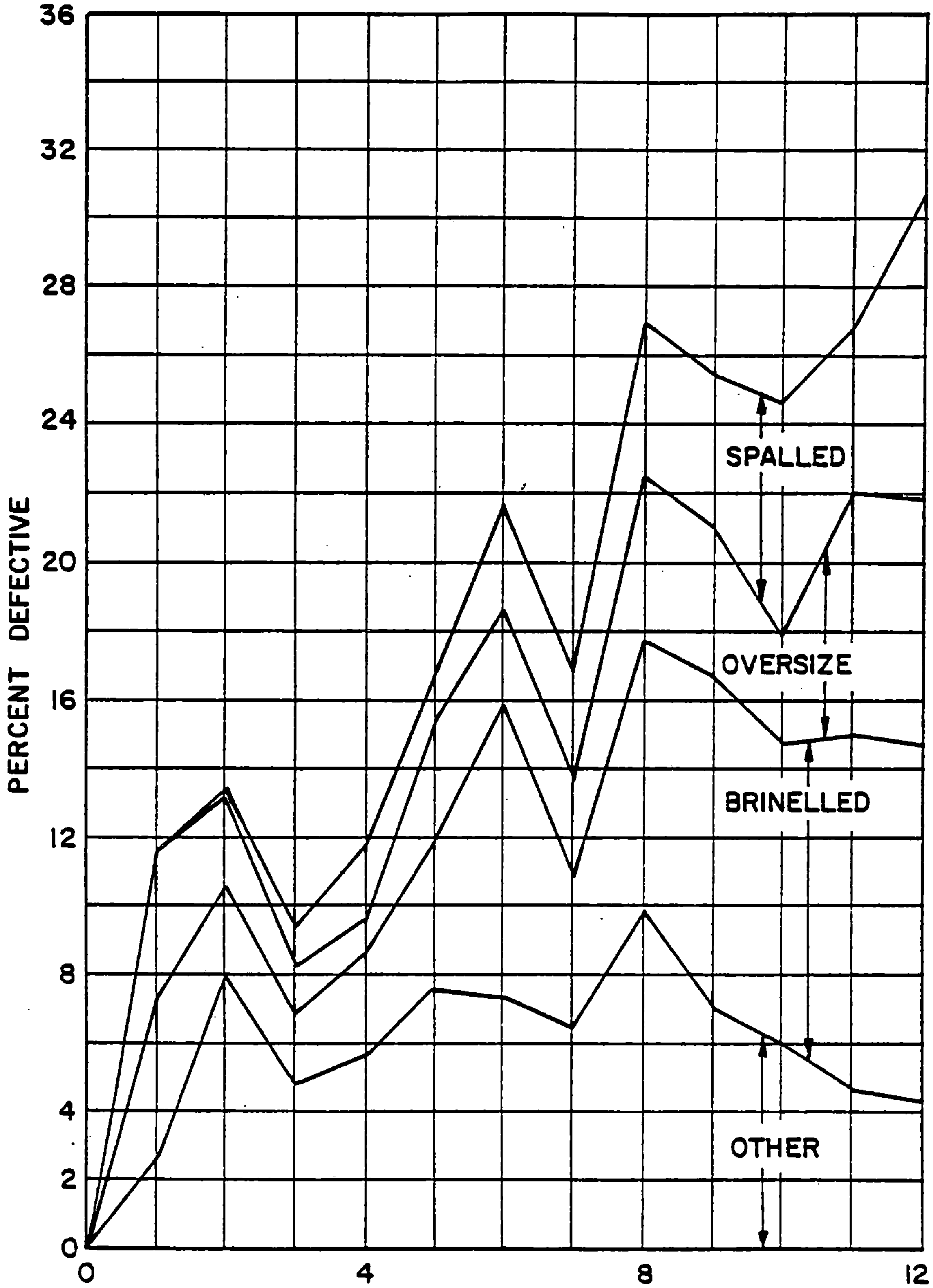


FIGURE 15. PERCENT DEFECTIVE CUPS FOR VARIOUS DEFECT MODES

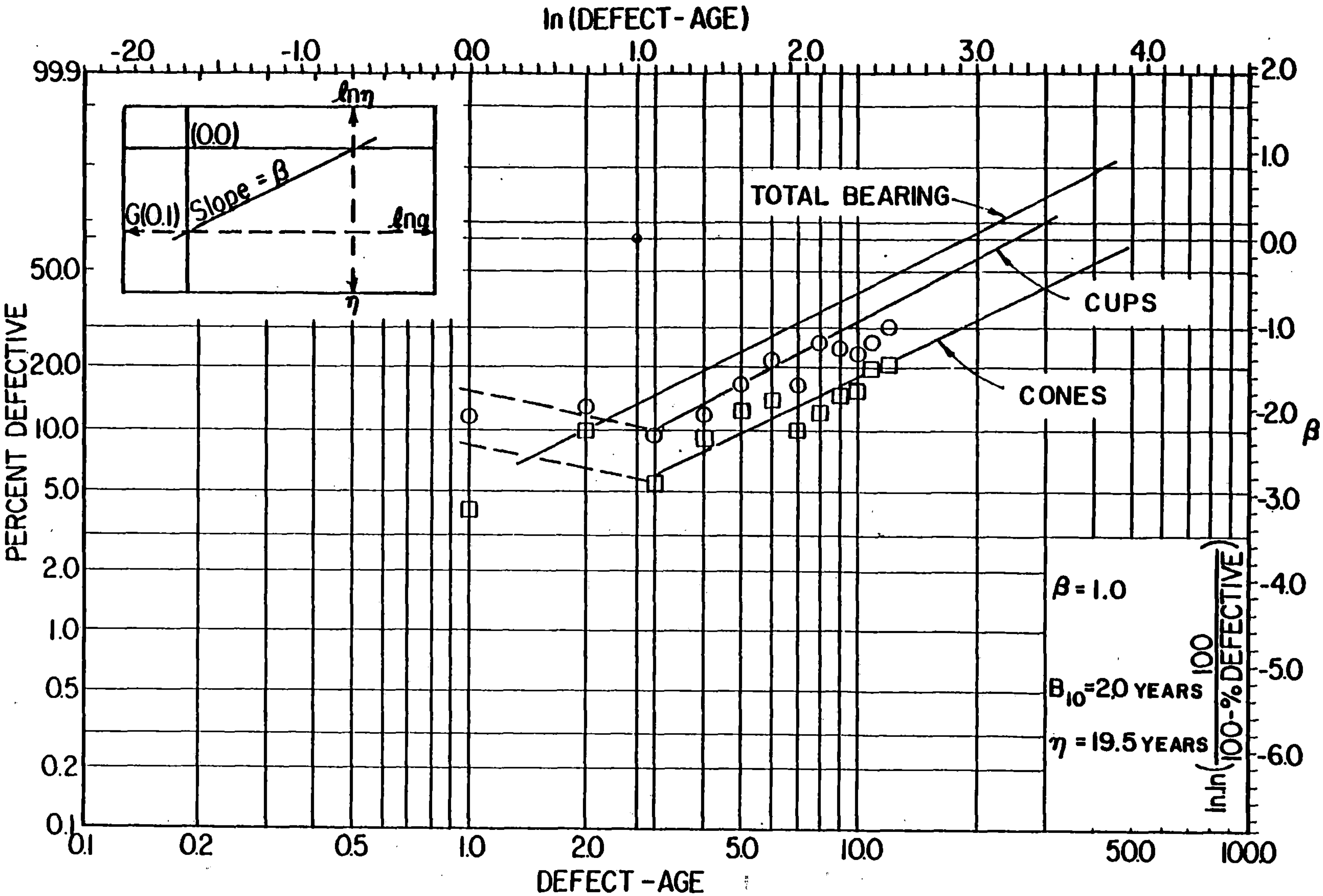


FIGURE 16. CUMULATIVE PERCENT DEFECT VERSUS AGE -- ALL DEFECT MODES

limiting modes, are summarized in Table 17. It is likely that the combined Weibull slope and, consequently, defect rate will increase in the future as the limiting modes of oversize, spalling, and brinelling become dominant.

The results of the study are a potential concern in that they indicate that after approximately two years of service 10 percent of the railroad roller bearings examined exhibited a defect for which they would be condemned if they were in a rework shop. As will be shown in the next section, if the average bearing met the AFBMA calculated fatigue life, the B_{10} life would be approximately 11 years.

4.3 SIGNIFICANT COMPETING DEFECTS

The survey described in the previous discussion identified three defect modes which became dominant with increasing age — spalling, brinelling, and oversize. We now discuss these three modes in greater depth.

4.3.1 Fatigue Defect Mode

Fatigue defects, or spalling, are the classical defects and are the limiting defect mode. The fatigue/life relationship for bearings has traditionally been expressed in terms of a two-parameter Weibull distribution. For application to bearings it is more convenient to represent the Weibull distribution as follows:

$$R(t) = e^{-0.10536(t/L_{10})^{\beta}} \quad (7)$$

Values have been determined experimentally for the Weibull slope, β. There is some variation in the value as reported by several authors:

β = 1.125 . . . . Reference (6).
β = 2.0 . . . . Reference (7) - the most commonly used value.
β = 1.5 - 3.0 . . . . . Reference (8) - varies depending on bearing size.

In the railroad roller bearing industry the L_{10} life is calculated using either the AFBMA (9) or Timken methods (10). The AFBMA method uses the equation:

TABLE 17

LIMITING DEFECT MODES:

SUMMARY OF WEIBULL SLOPES AND CHARACTERISTIC LIVES

	CUPS		
Failure Mode	Characteristic Life Years	B_{10} Life Years	Weibull Slope
Spalling	42.5	12.8	1.88
Oversize	55.6	14.1	1.64
Brinelling	47.0	9.8	1.44
Combined Spalling, Oversize, and Brinelling	28.2	5.6	1.40

	CONES		
Failure Mode	Characteristic Life Years	B_{10} Life Years	Weibull Slope
Spalling	145.1	30.6	1.45
Oversize	52.6	9.2	1.29
Brinelling	50.6	15.4	1.88
Combined Spalling, Oversize, and Brinelling	36.4	6.5	1.31

$$L_{10} = (\frac{C}{P})^{3.3} \times 10^6 \text{ revolutions.} \tag{8}$$

Values of C are tabulated in Table 18 for the most common bearing types in service today.

We next need to determine the load, P. Table 19 summarizes the vertical loads for both loaded and unloaded cars as a function of bearing size. Using Equation (8), the L_{10} life has been calculated and is tabulated in Table 18 for both the loaded and unloaded condition. The calculation is further complicated by the fact that the bearing does not operate under constant load during its entire life. Table 20 illustrates this by showing that the average freight car travels 20,600 miles (33,152 kilometers) per year and that it is loaded for 57 percent of these miles. The composite life then becomes:

$$\bar{L}_{10} = \frac{1}{f_1/L_{10_1} + f_2/L_{10_2}} \tag{9}$$

where f_i is the fraction of the time corresponding to the L_{10_i} life. This composite $\bar{L}_{10}$ life is also shown in Table 18. This predicted life is much higher than achieved in practice. The reason for this is that the procedure assumes adequate lubrication which does not exist in a grease-lubricated bearing running most of its life at low speeds. (See Figure 17).

AAR Grease Properties — The development of the elastohydrodynamic lubrication theory shows that lubricant films of thicknesses on the order of of microinches (0.0254 micrometers) and tens of microinches (0.254 micrometers) occur in rolling contact. Since surface finishes are of the same order or magnitude as the lubricant film thickness, the significance of rolling-element bearing surface roughnesses to bearing performance become apparent.

The AAR roller bearing grease specification (M-917-64) calls for a high quality grease containing a petroleum oil as described in Table 21.

TABLE 18

CALCULATED L_{10} ROLLER BEARING FATIGUE LIFE

Bearing Size	Basic Dynamic Capacity, C		Loaded			Unloaded			$\bar{L}_{10}$ Millions of Revs.	Years of Ave. Service	
			Load		L_{10} (Rev)(10^{-6})	Load		L_{10} (Rev)(10^{-6})		Uncor. for Film Thick.	Cor. for Film Thick.
	Pounds	Newtons	Pounds	Newtons		Pounds	Newtons				
5½ x 10	127,534	567 261	21,000	93 408	384.806	8,375	37 252	7993.111	651.439	51.74	15.52
6 x 11	133,633	594 400	26,250	116 760	214.975	8,250	36 696	9799.800	371.009	29.47	8.84
6½ x 12	181,735	808 357	31,500	140 112	324.879	7,875	35 028	31515.227	565.564	44.92	13.47
7 x 14	209,083	930 001	38,000	169 024	277.821	9,375	41 700	28154.419	565.044	44.88	13.46
6 x 11	144,201*	641 406	26,250	116 760	276.355	8,250	36 696	12597.870	484.833	38.51	11.55
5½ x 10	135,773*	603 918	21,000	93 408	473.108	8,375	37 252	9827.300	830.014	65.93	19.78

* Timken method

TABLE 19

BEARING CLASSES, SIZES, AND VERTICAL LOADS

Class	Size (Axle Journal Diameter and Length, in.)	Nominal Car Capacity, Tons (Current AAR Designation)	Gross Rail Load,* (On Four Axles)		Vertical Load on Each Bearing, Approx. Loaded		Vertical Load on Each Bearing, Approx. Unloaded	
			Pounds	Newtons	Pounds	Newtons	Pounds	Newtons
B	4½ x 8	30	103,000	165 727	12,000	19 308	5,375	8 648
C	5 x 9	44	142,000	228 478	16,750	26 951	6,750	10 861
D	5½ x 10	55	177,000	284 793	21,000	33 789	8,375	13 475
E	6½ x 11	77	220,000	353 980	26,250	42 236	8,250	13 274
F	7 x 12	100	263,000	423 167	31,500	50 684	7,875	12 671
G	7 x 12	125	315,000	506 835	38,000	61 142	9,375	15 084

* Gross rail load equals light weight of car plus loading.

TABLE 20

AVERAGE MILES PER YEAR BY CAR TYPE

(REPRESENTATIVE SAMPLE)

Car Type	Average Distance Traveled per Year		Percent Loaded Distance
	Miles	Kilometers	
Box - General Service	21,000	33,795	68.3
Box - Equipped	27,000	43,451	54.0
Box - Refrigerated	29,000	46,670	54.1
Gondola	19,000	30,577	57.7
Pulpwood	8,700	14,001	48.0
Hopper - Open Top	13,000	20,921	52.2
Hopper - Covered	23,000	37,014	49.9
Tank	19,000	30,577	49.0
Flat - TOFC/COFC	57,000	91,730	74.4
Flat - Autorack	42,000	67,591	50.4
Flat - Other	21,000	33,795	51.9
Average, All Cars	20,600	33,152	57.0
	(56.6 mi/day)	(91.1 km/day)	

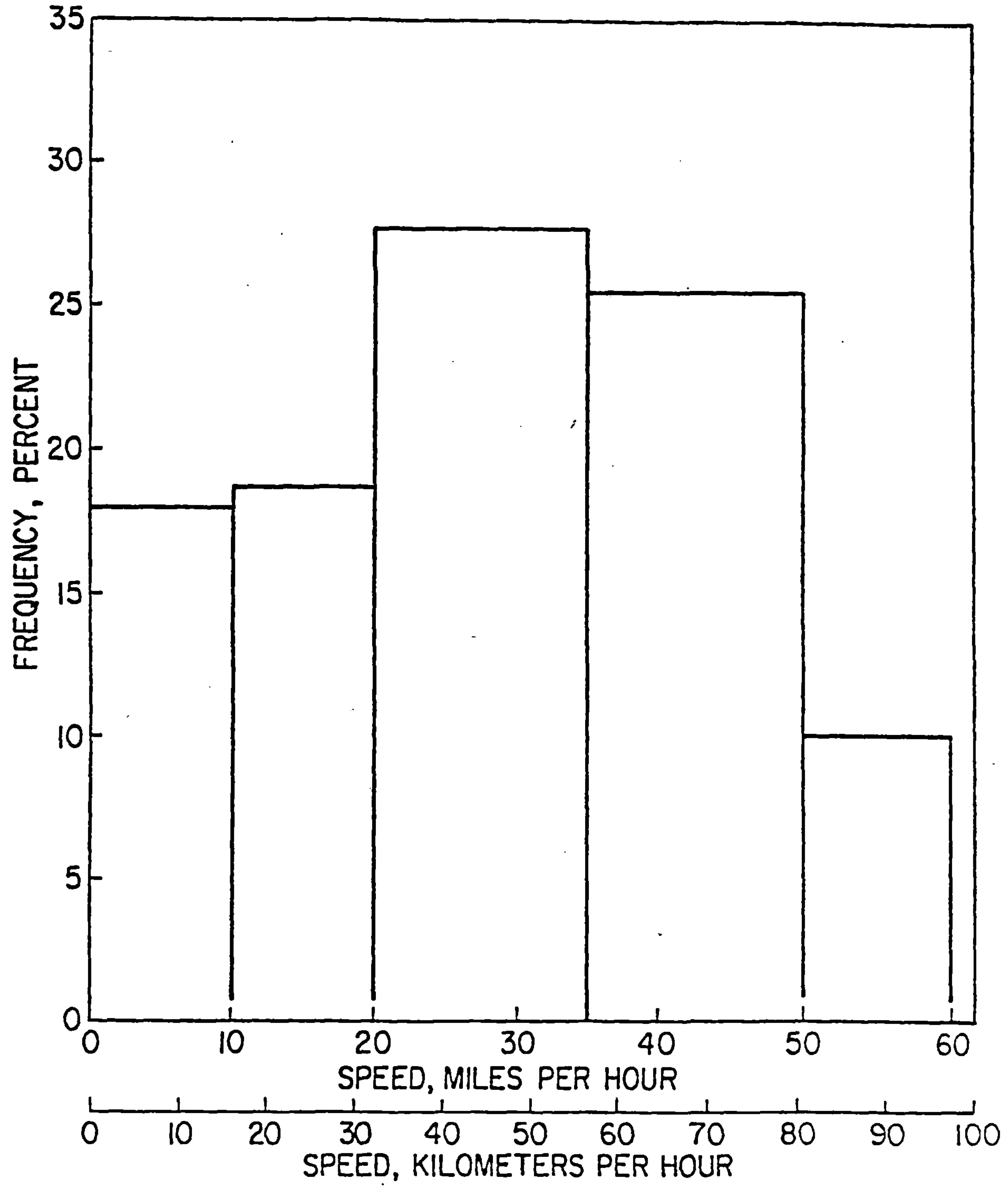


FIGURE 17. THROUGH FREIGHT TRAIN SPEED PROFILE (REPRESENTATIVE SAMPLE)

TABLE 21

PROPERTIES OF BASE OIL OF AAR ROLLER BEARING GREASE

	Requirements	Method of Analysis
Flash (Open Cup), Minimum	$340^{o}F$ ($171^{o}C$)	ASTM D-92
Pour Point, Upper, Maximum	$0^{o}F$ ($-18^{o}C$)	ASTM D-97
Saybolt Univ. Viscosity at $100^{o}F$ ($38^{o}C$)	450-550 sec.	ASTM D-445
Saybolt Univ. Viscosity at $210^{o}F$ ($99^{o}C$), Minimum	59 sec.	ASTM D-445

Using the Jones analysis (10) modified for elastohydrodynamic effects (11), the film thickness in a railroad roller bearing has been calculated as a function of speed and temperature and is shown in Figure 18. A range of values is given which covers all bearing sizes and loads (loaded and unloaded cases). An average composite surface roughness has been assumed, as shown in Table 22.

Referring to Figure 18, it is seen that the specific film thickness, $\Lambda = \frac{h}{\sigma}$, is less than 0.6 for most of the bearing's life. The lubrication factor for this condition (page 137) is 0.3. The life adjusted for film thickness is also shown in Table 18.

Since most bearings in the railroad industry are selected on the basis of the AFBMA or Timken methods, it is instructive to compare this prediction with the incidence of spalling fatigue found in this study. This comparison is shown in Figure 19 where the cumulative incidence of spalling for cups and cones and their combined incidence has been plotted as a function of age. Since the survey includes both 6 x 11 and 6-1/2 x 12 bearings, a range of expected B_{10} life is indicated using both the Timken and AFBMA methods. The AFBMA method gives a fairly accurate estimate of B_{10} life, while the Timken method is a little more optimistic.

The present specification calls for 500,000 miles (804,500 kilometers) before 10 percent bearing replacement at 80 percent load or full load for 250,000 miles (402,250 kilometers). The conditions of Table 20 are equivalent to about 85 percent load or 400,000 miles (643,600 kilometers). At 20,600 miles (33,145 kilometers) per year this represents a specification B_{10} life of 19.4 years. Actually, as Figure 19 shows, 10 percent replacement of at least one component due to condemnable fatigue defect would be expected in about 11 years (based on the inspection of the subject 8,000 bearings).

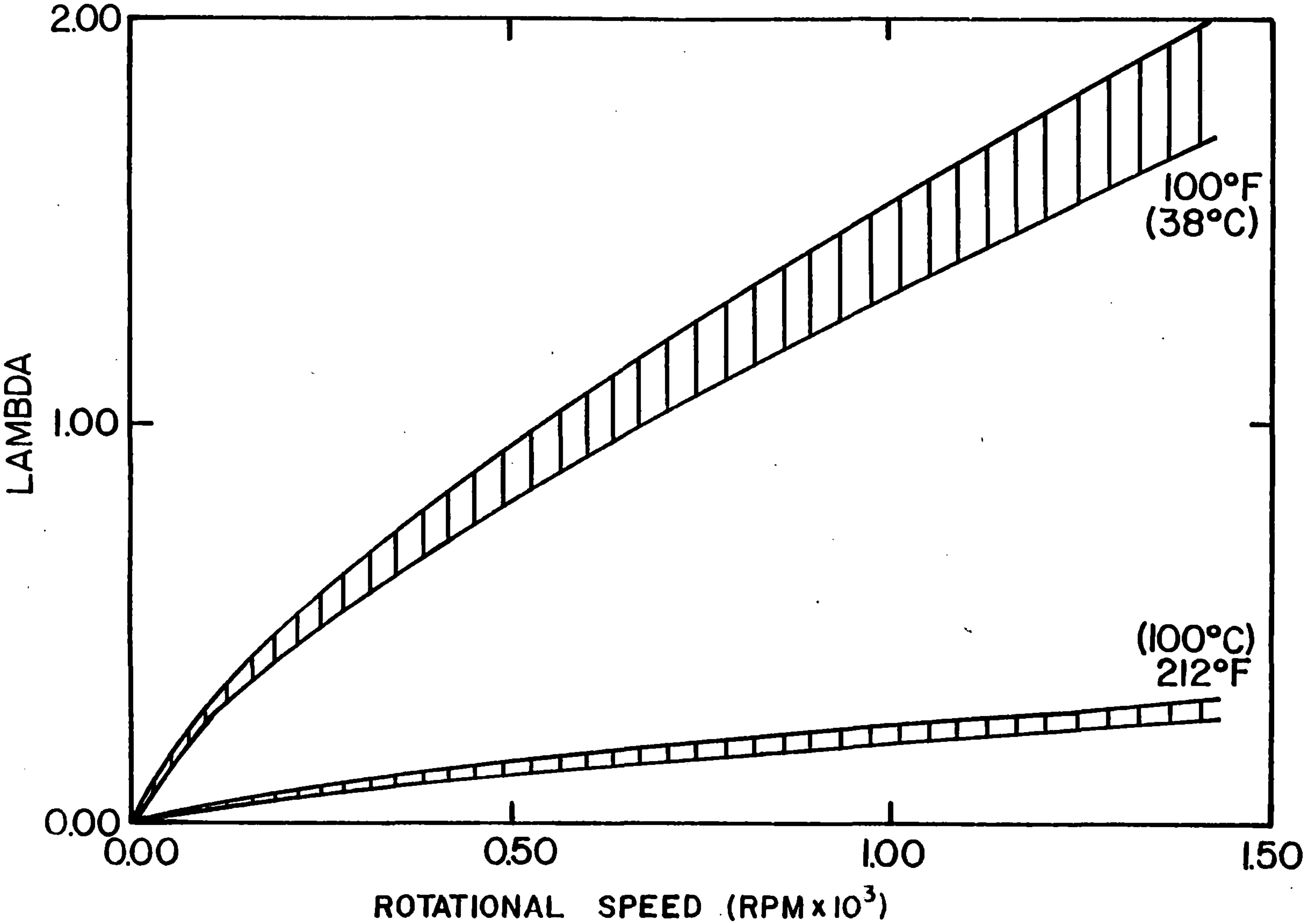


FIGURE 18. RAILCAR ROLLER BEARING LUBRICANT FILM THICKNESS

TABLE 22

TYPICAL RAILROAD ROLLER BEARING SURFACE FINISHES

Component	Range		Average	
	Inches rms	Micrometers rms	Inches rms	Micrometers rms
Cup ID	$15 - 35 \times 10^{-6}$	0.38 - 0.89	25×10^{-6}	0.64
Cone OD	$15 - 35 \times 10^{-6}$	0.38 - 0.89	25×10^{-6}	0.64
Roller	$10 - 20 \times 10^{-6}$	0.25 - 0.50	18×10^{-6}	0.46

Composite $\sigma = \sqrt{\sigma_1^2 + \sigma_2^2} = 30.81 \times 10^{-6}$ inches (0.78 micrometer).

σ = surface finish.

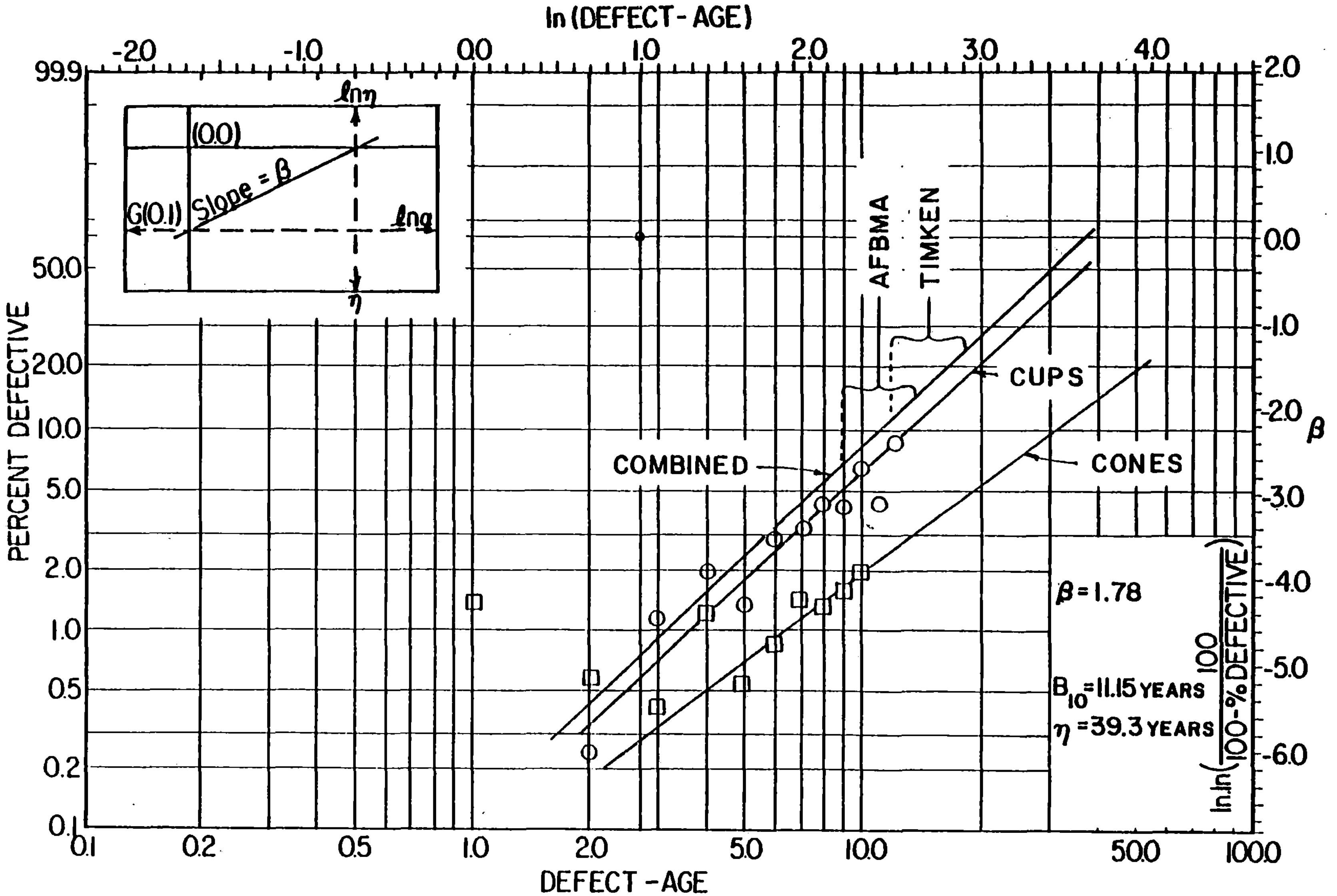


FIGURE 19. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE -- SPALLING DEFECT MODE

4.3.2 The Brinelling Defect Mode

The AAR specifies the size of an allowable permanent indentation, termed brinelling, as an indentation 5/32-inch wide over one-half the race length or an indentation 3/32-inch wide over the entire length of the race.

It is shown in References 13 and 14 that even very small loads will produce indentations; i.e., deflection curves appear to go through the origin on a deformation-versus-load plot. Experience has shown that permanent deformations have little effect on the operation of the bearing if their magnitude at any given contact point is limited to a maximum of 0.0001d. If the deformations become much larger, the cavities formed in the raceways cause the bearing to vibrate and become noisier, although bearing friction does not appear to increase significantly and bearing operation is essentially not impaired in any other manner.

The basic static load rating, C_o, of a roller bearing is defined by the AFBMA as that bearing load which will cause a permanent deformation at the maximum loaded element and at the weaker of the inner or outer raceway contacts of 0.0001d. Hence, $\delta_o/d = 0.0001$.

4.3.2.1 Analytical Model _ Based on empirical data for bearing quality steel hardened between 63.5 and 65.5 Rockwell C, Lundberg, Palmgren and Brutt (13, 15) developed the following formula to describe permanent deformation for line contact between roller and raceway:

$$C_s = \frac{1.97 \times 10^{-17}}{d^2} \left[\frac{Q}{\ell} \left(\frac{1}{1 \mp \gamma} \right)^{1/2} \right]^3, \qquad (10)$$

where,

C_s = static load capacity (lb)

d = roller diameter (in.)

d_m = mean pitch diameter (in.)

Q = roller load (lb)

ℓ = effective roller length (in.)

$$\gamma = \frac{d \cos\alpha_c}{d_m}.$$

The upper sign refers to inner race contact and the lower sign refers to outer race contact.

For most roller bearing applications the maximum roller load can be approximated by:

$$Q_{max} = \frac{5P}{iZ \cos\alpha_c}. \tag{11}$$

Setting $P = C_s$ yields:

$$C_s = 0.2\ i\ Z\ Q_{max} \cos\alpha_c. \tag{12}$$

Substituting $\delta/d = 0.0001$ in Equation (12) gives:

$$C_s = 3420\ i\ Z\ \ell\ d \cos\alpha_c\ (1 \mp \gamma)^{1/2}. \tag{13}$$

In Equation (13), the smallest C_s is taken by the AFBMA (9) to give static load capacity of a roller bearing as:

$$C_o = 3130\ i\ Z\ \ell\ d \cos\alpha_c\ , \tag{14}$$

where,

C_o - static load rating, lb

i - number of rows of rollers

Z - number of rollers per row

ℓ - effective length of roller, in.

d - roller diameter, in.

α_c - one-half included cup angle, degrees.

The basic static capacities and allowable depth of permanent deformation (based on the AFBMA criterion of the most common railroad roller bearing sizes) are summarized in Table 23.

Figure 20 is a plot of depth of deformation at the most heavily loaded roller versus bearing load. This calculation is based on the AFBMA brinnelling criterion, Equation (14).

TABLE 23

DESIGN DATA FOR TYPICAL RAILCAR ROLLER BEARINGS

Factor Name	$5\frac{1}{2}$ x 10	6 x 11	$6\frac{1}{2}$ x 12	7 x 12
d, Mean Roll Diameter	0.6874"	0.7047"	0.8424"	0.9094"
ℓ_{eff}, Effective Roll Length	1.520"	1.540"	1.860"	2.015"
d_m, Mean Roll Pitch Diameter	6.5845"	7.0946"	7.9906"	8.7964"
α_c, 1/2 Incl. Cup Angle	10^o-0'	10^o-0'	10^o-0'	10^o-0'
Z, No. Rolls per Row	23	24	23	23
i, No. Rows per Bearing	2	2	2	2
C, Basic Dynamic Capacity, lbs.	117,082	124,621	170,663	196,634
C_o, Basic Static Capacity, lbs.	150,437	163,046	225,597	263,834
Allowable Depth of Permanent Deformation, δ = .0001d μin	68.74	70.47	84.24	90.94

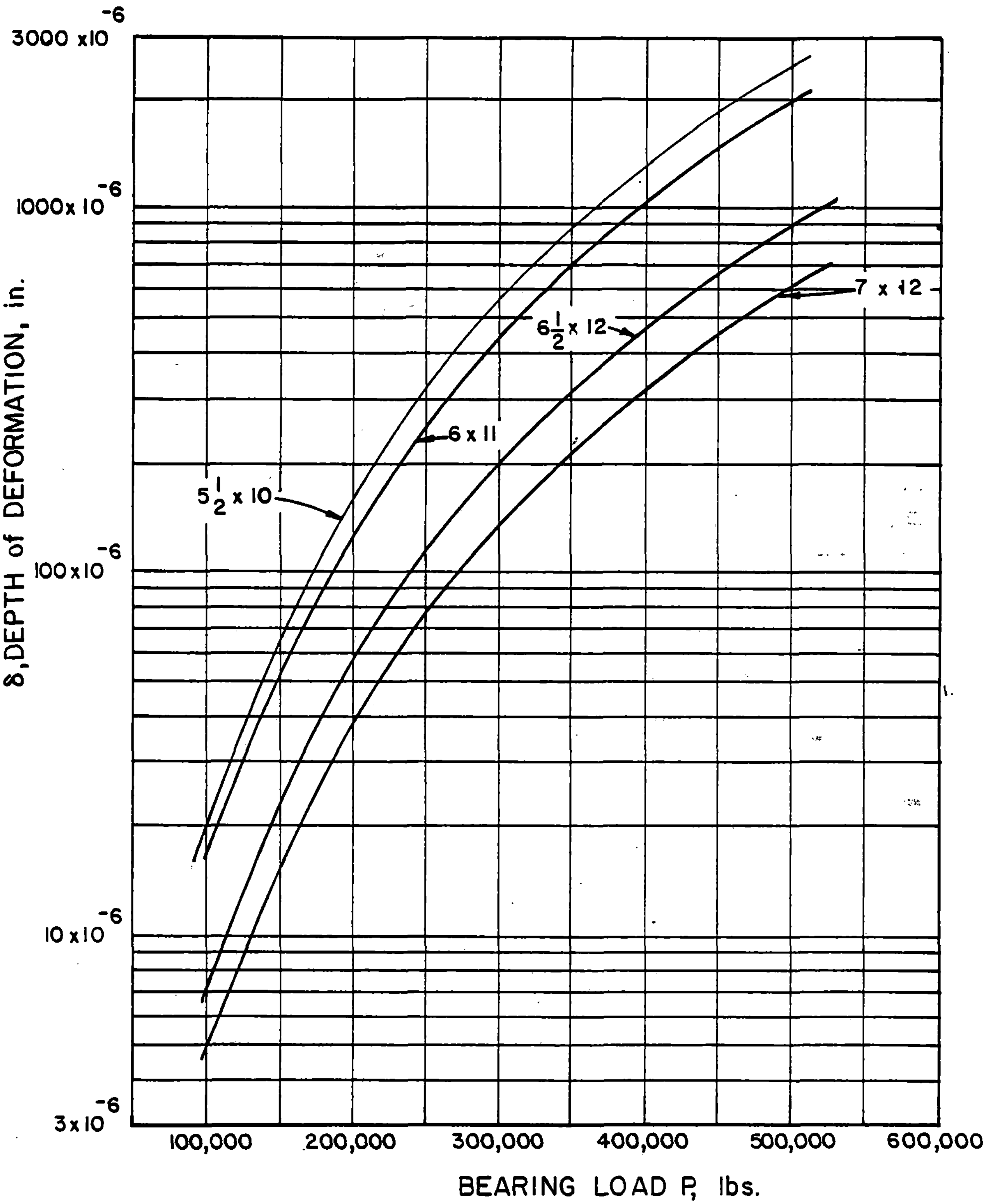


FIGURE 20. DEPTH OF DEFORMATION AT MOST HEAVILY LOADED CONTACT VERSUS BEARING LOAD (USING AFBMA CRITERIA)

4.3.2.2 Distribution of Brinelling Resistance — Contact stress and deflection within the elastic limit are well known due to the work of Hertz. The plastic deflection which occurs when the elastic limit is exceeded has not been studied extensively. The principal reports on plastic deflection on rolling element bearings are the reports by Palmgren and others (13) and (14).

Wickstrand (16) later measured the depth of permanent indentations in 52100 steel plates due to cylindrical rollers. He found that for steel tracks the depth of deformation could be expressed by:

$$\ln \delta = 3.43 + (3 + 0.19d) \ln q - (2.03 + 0.63d) \ln d - 0.19 \text{ Rc}, \tag{15}$$

where:

δ = Depth of indentation, in.
q = Q/ℓ = Lineal load, lb /in.
d = Roller diameter, in.
Rc = Hardness on Rockwell C scale.

Statistically, it can be shown that the probable error for the ln δ for this derived equation is about $\pm$ 0.26. In other words, we can assume that the probable value of δ varies from 77 to 130 percent of the calculated value.

When one examines Equation (15), one can see that scatter is not surprising. From the coefficient "ln q" we see that the effect of any small error in "ln q" is about tripled in deformation. The effect of hardness is also very strong. One point of hardness changes the depth about 20 percent. It is very difficult to measure hardness much closer than a full point on the Rockwell C scale. Hence, from a consideration of hardness alone, accuracy much closer than plus or minus 20 percent cannot be expected.

Unfortunately, Equation (15) is for through hardened steel and not case hardened as is used in tapered roller bearings. For this reason, the brinelling tests described later in this report were undertaken to evaluate the brinelling resistance of railroad roller bearings.

The work of Wickstrand shows that there is an inherent distribution of resistance to brinelling (i.e., strength) even in supposedly identical steel and geometry. At the same time, there is a distribution of load (i.e., stress) to which the bearings will be subjected in service.

From the failure governing stress and strength point of view, reliability R is given by "all probabilities that the failure governing strength exceeds the failure governing stress" or:

$$R = Pr(P_B > P) . \tag{16}$$

By transferring P to the left-hand side of the inequality, we get:

$$R = Pr(P_B - P > 0). \tag{17}$$

Equation (17) says that reliability is given by all probabilities that the difference between strength and stress is positive. This corresponds to the positive area under the difference distribution $f_3(P_B-P)$ as shown in Figure 21. If we denote (P_B-P) by ζ, then Equation (17) may be written as:

$$R = \int_0^d f_3(\zeta)d , \tag{18}$$

where d is the upper limit of ζ.

When $f_1(P_B)$ and $f_2(2)$ (the probability distribution for brinelling limit and load, respectively) are both normal distributions, then $f_3(\zeta)$ is also normal; hence, in terms of the parameters of the distributions, Equation (17) becomes:

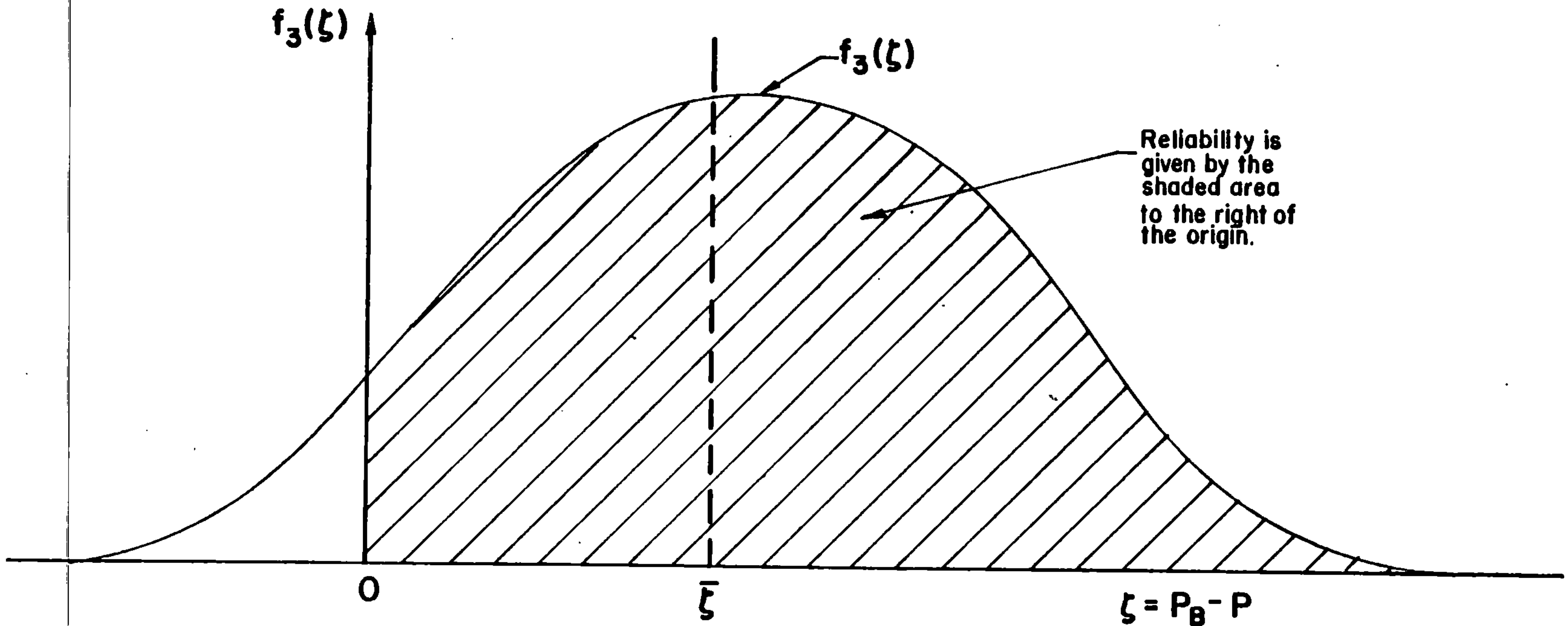


FIGURE 21. STRENGTH-STRESS DIFFERENCE DISTRIBUTION

$$R = \int_{m}^{\infty} \phi(z)dz \tag{19}$$

The function $\phi(z)$ is the standardized normal probability density function, and values of the integral can be obtained by entering normal distribution area tables with the value of:

$$m = - \frac{\overline{P}_B - \overline{P}}{\sqrt{\sigma_s^2 - \sigma_P^2}} = - \overline{\zeta}/\sigma\zeta \ , \tag{20}$$

where:

$\overline{P}_B$ = mean of the brinelling strength distribution

$\overline{P}$ = mean of the load distribution

σ_s = standard deviation of the strength distribution

σ_P = standard deviation of the load distribution.

Therefore,

$$R = f_4(m) = F_4(- \overline{\zeta}/\sigma\zeta). \tag{21}$$

It may be seen that with normally distributed $f_1(P_B)$ and $f_2(P)$, reliability can be calculated once m is known. Given in Figure 22 is the reliability plotted versus m on probability paper. Thus, given P, P_B, σ_s, and σ_P, m can be calculated and the reliability can be obtained from Figure 22.

4.3.2.3 Distribution of Brinelling Loads – The determination of railroad roller bearing failure modes requires an adequate understanding of the freight car truck operational load environment and the influence of truck, car, and operational parameters on this environment. The term "load environment" in this context refers to a description of forces

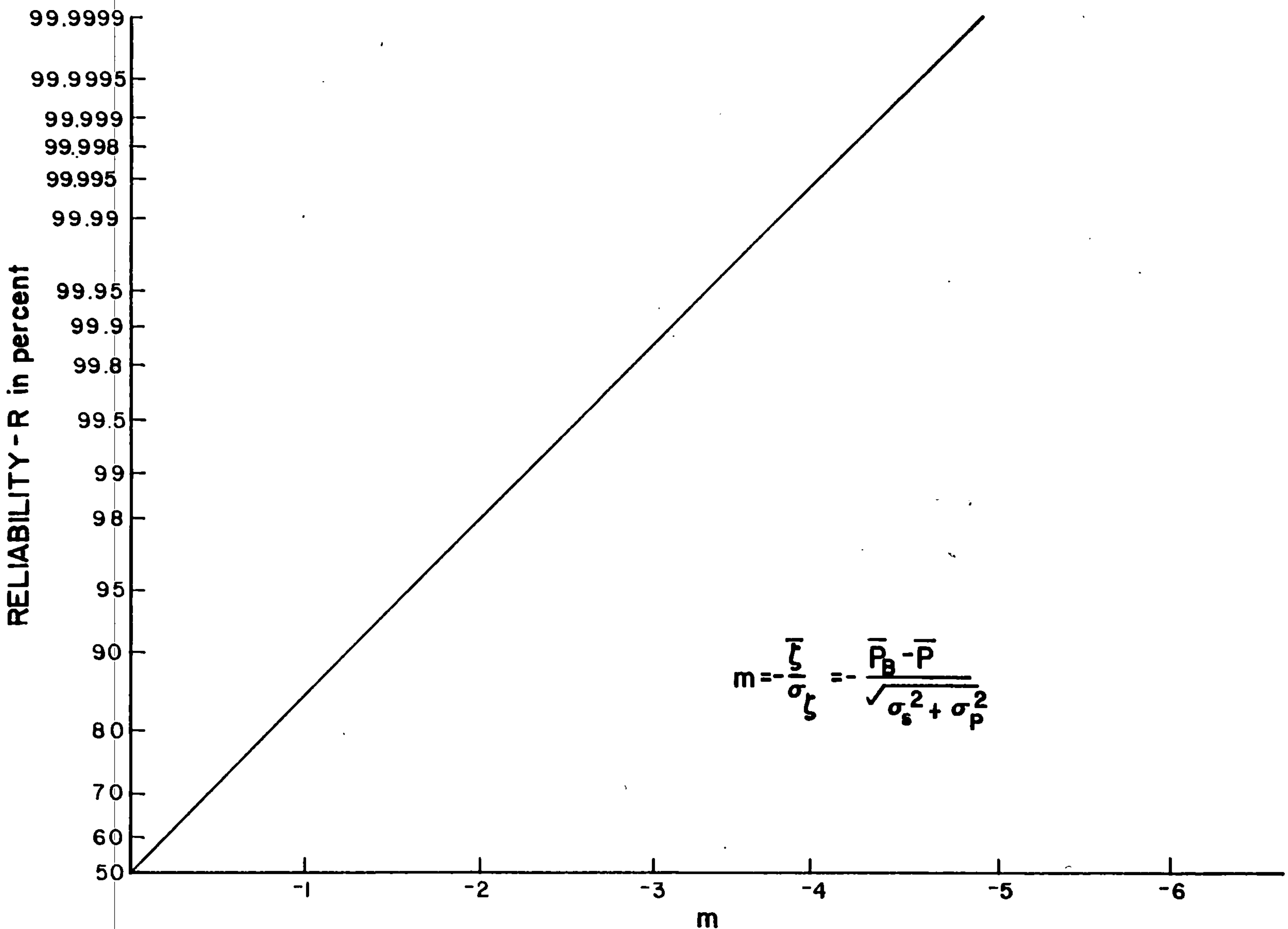


FIGURE 22. RELATIONSHIP BETWEEN RELIABILITY AND FACTOR m

on the bearing which are significant with respect to its life and reliability. This includes the magnitude of the cyclic fluctuations of the load level and the number of cycles at various load levels anticipated during the lifetime of the bearing. Truck and car parameters which influence the environment include the spring travel of the suspension system, the type of damping mechanism, the degree of wheel wear, car truck center distance, height of car center of gravity, etc. Operational parameters such as the weight of the car, train speed, track conditions, etc., will also influence the load environment.

Vertical loads on the structural elements of the truck are characterized by an average value representing the car weight and fluctuations about this level due to the dynamic interaction of the suspension system with rail deflections resulting from the nonuniform resilient response of the track substructure, the passage of the truck over track irregularities (such as at crossings and turnouts), and wheel defects (such as a wheel flat spot).

Johnson (17-19) obtained load records from load cells at opposite sides of an axle at the roller bearing adapter interface. His data showed a nominal static load at this interface of slightly over 30,000 pounds for a loaded car with a nominal capacity of 100 tons. The predominant frequency of the alternating component of load is slightly over 1 Hz representing a rocking motion of the car, which is evident by the 180 degree out-of-phase character of these oscillations. Passage of the car over the turnout initiates a higher frequency in-phase bouncing motion of the truck which occurs at the frequency of 3 to 5 Hz.

Johnson's (17-19) data are based on results from the analysis of B&LE test records and can be used to formulate load spectra for performance testing of freight car truck components. All data are presented with reference to a loaded 100-ton-capacity car (263,000-pound rail load). Load data for cars of lower capacity may be estimated by assuming that the forces are proportional to the rail load.

Roller Bearing Adapter Interface Vertical Load Data. When dealing with the rotating parts on the truck (the wheels axles and bearings), one must consider cyclic stresses that are developed in these components during their rotation even though there is only a steady load acting on the component. Thus, for these components the loads should be defined by a spectrum showing the number of revolutions by which given load levels are exceeded. A vertical load spectrum with reference to revolutions per mile is used in Figure 23. The spectrum represents the variation in the vertical load acting on the bearing.

Lateral Wheel Load Spectrum. The lateral wheel load data can be utilized to develope a load spectrum with reference to bearing life recognizing that a steady lateral load will produce one cycle of moment load in the bearing per wheel revolution. The spectra presented in this section were developed by Johnson (17) from an analysis of selected lead-axle test run data over the B&LE test track and therefore represent the effects of the distribution of tangent and curved track segments found in this section of track. Figure 24 shows the average spectrum from 10 test runs where there were no special conditions influencing the lateral wheel load.

The spread of the data is illustrated by showing plus and minus one standard deviation.

4.3.3 Cone Bore Defect Mode

One of the surprising results of the bearing defect study was the large number of oversize bores which were observed. This defect mode is not well known or fully understood.

4.3.3.1 Postulated Mechanisms — It is postulated that the growth of cone bores can be produce by several mechanisms. These mechanisms include:

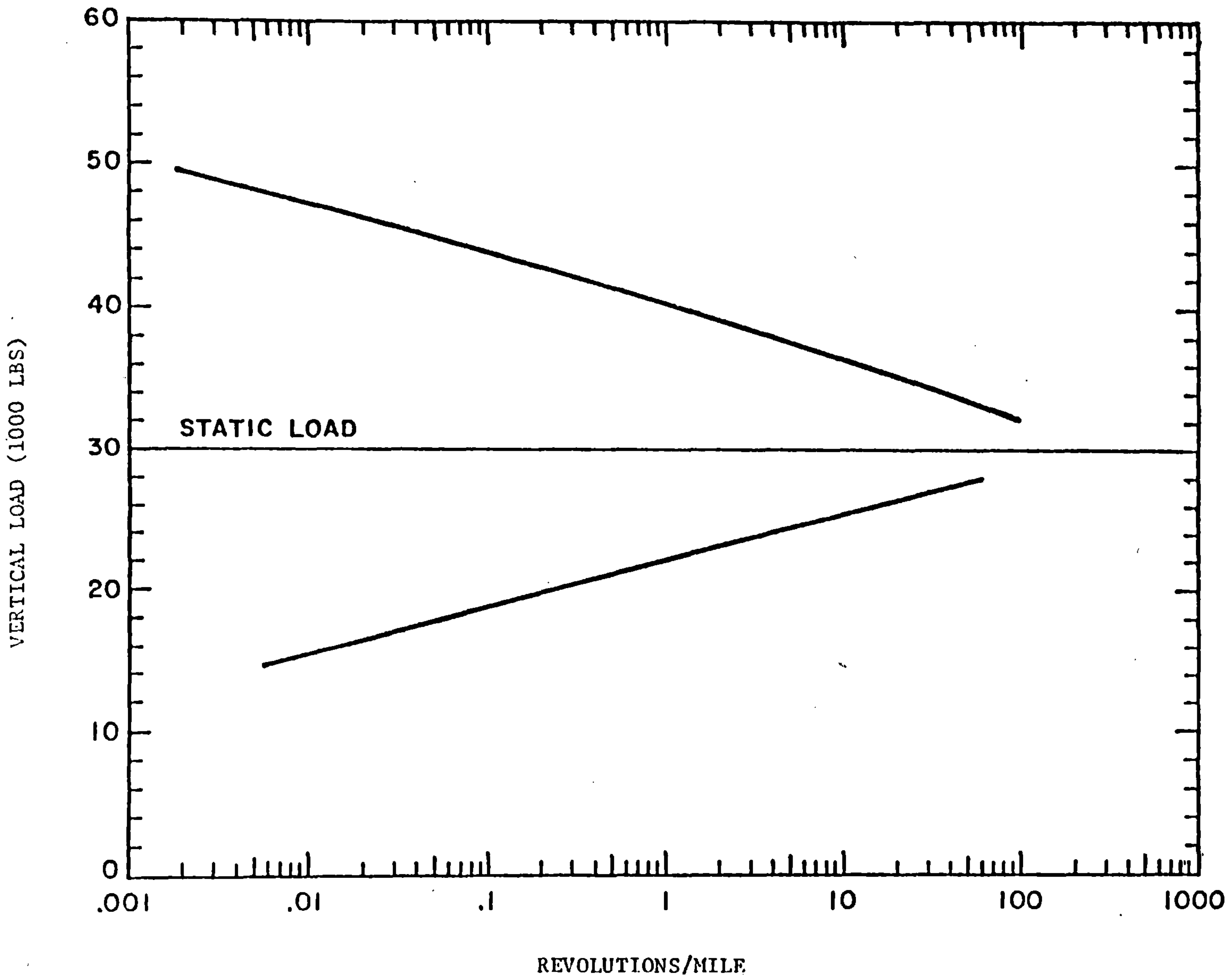


FIGURE 23. AVERAGE VERTICAL BEARING ADAPTER LOAD SPECTRUM FOR ROTATING COMPONENTS WITH REFERENCE TO NUMBER OF REVOLUTIONS BY WHICH LOAD LEVEL IS EXCEEDED (AFTER REFERENCE 17)

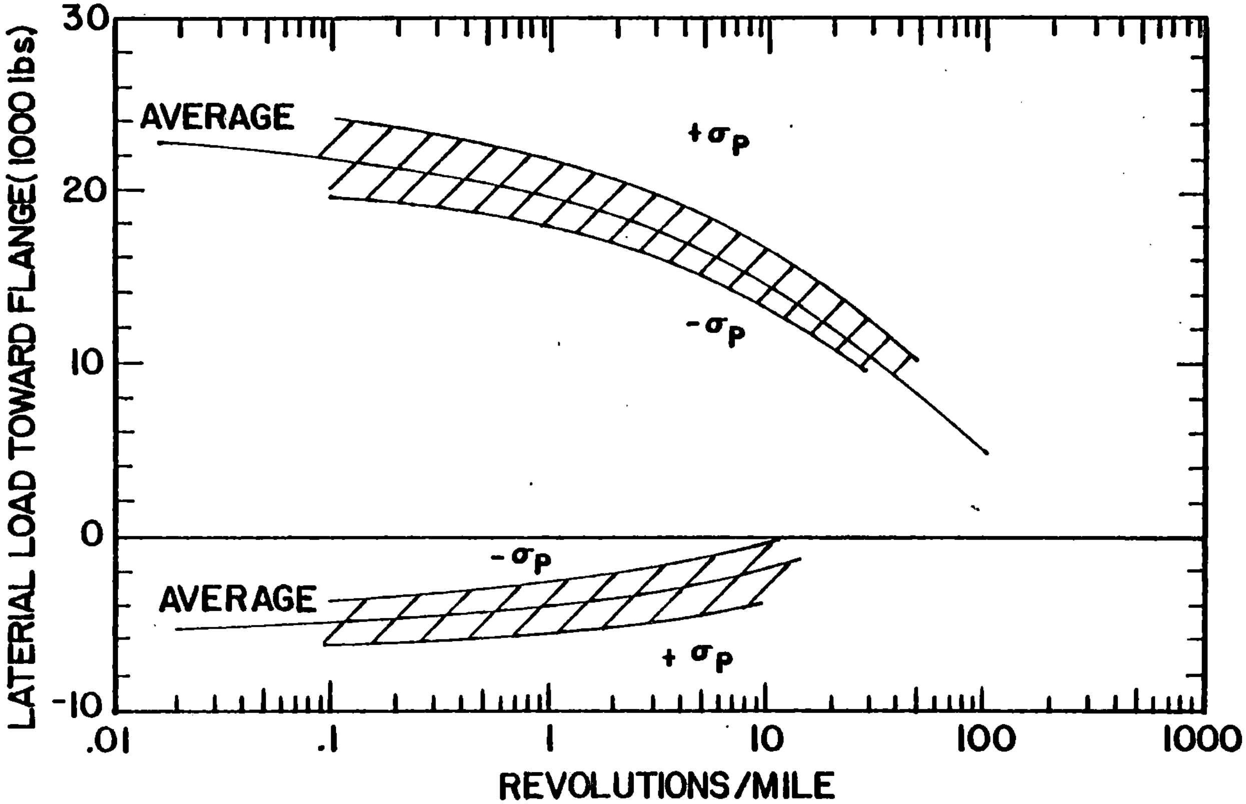


FIGURE 24. AVERAGE LEAD AXLE LATERAL WHEEL LOAD SPECTRUM AT 35 MPH (AFTER REFERENCE 17)

1. Relative cone-axle motion which could cause wear.
2. Creep (stress relaxation).
3. Volume changes in steel during service.
4. Plastic deformation during service.

The first three mechanisms probably are not significant contributions to the phenomenon. Mechanism 1 is unlikely because no large torques exist in the axle-bearing combination which tend to cause relative rotational motion. In addition, no significant abrasion of axles or bores is known to have been found on axles or cone bores which have oversize bores. Mechanism 2 is very unlikely since creep in steels is practically nonexistent at the temperatures under which the bearings operate. Mechanism 3 would include the volume change produced when retained austenite transforms to martensite. This volume change in service, however, is small and is restricted to a very thin surface layer. In addition, no significant operating changes in the austenite-martensite proportions have been observed in railroad bearings.

Mechanism 4 is a likely candidate for an explanation of the cone bore growth phenomenon. The mechanism is composed of two parts -- plastic deformation and the associated increase in the cone bore as a result of this deformation. Qualitatively, such a mechanism could consist in the following series of events.

The loading of the bearing and/or the occasional excessive loading which all bearings experience cause some plastic deformation of the cone at its roller surface. After a period of time, the entire surface of the cone has experienced such deformation. This deformation tends to enlarge the roller surface circumferentially. Since this surface is restrained by the remainder of the cone, the surface is put in compression while the remainder of the cone is in tension. The stresses associated with this compression-tension field then produce cone enlargement, i.e., cone bore growth.

The plastic deformation aspect of the above cone bore growth mechanism can have at least two origins: yielding due to stresses in excess of the elastic limit and cyclic softening of a material which is initially hard. Yielding due to stresses in excess of the elastic limit need not require very high stress levels, i.e., the elastic limit can be considerably lower than the 0.2 percent offset stress commonly taken as the engineering yield stress. This is especially likely in steels heat—treated to high hardness levels (20). In such steels, imperfections in the crystal lattice of the martensite can move at relatively low stress levels. In addition, there is some evidence that the deformation associated with yielding can be progressive, i.e., additional plastic deformation occurs at the roller surface each time loading is applied. This progressive yielding can occur when the maximum Hertzian pressure is greater than two times the elastic limit (21). This is equivalent to a load of about 70 percent in excess of that necessary to cause initial yielding.

Cyclic softening of a material which is initially hard can occur in quenched and tempered steels as well as in mild steels (22-24). Such softening can be substantial -- about a 50 percent decrease in strength has been observed for 4340 steel (21). The extent of the softening appears to be associated with the initial hardness of the steel. For 4142 steel, the cyclic softening is small at 670 BHN but increases as the hardness is decreased to 380 BHN (23). The softening process is affected by the magnitude of the applied stress -- cyclic stresses in mild steel below the yield stress tend to produce cyclic softening while the same stresses above the yield stress tend to produce cyclic hardening.

The increase in the cone bore which results from the deformation of the roller surface can be determined with a suitable model of the cone. Such a model (25), in simplified form, consists of two concentric cylinders, Figure 25. The outer cylinder represents that portion of the cone in which the deformations are plastic. The inner cylinder represents that portion of the cone in which the deformations are elastic.

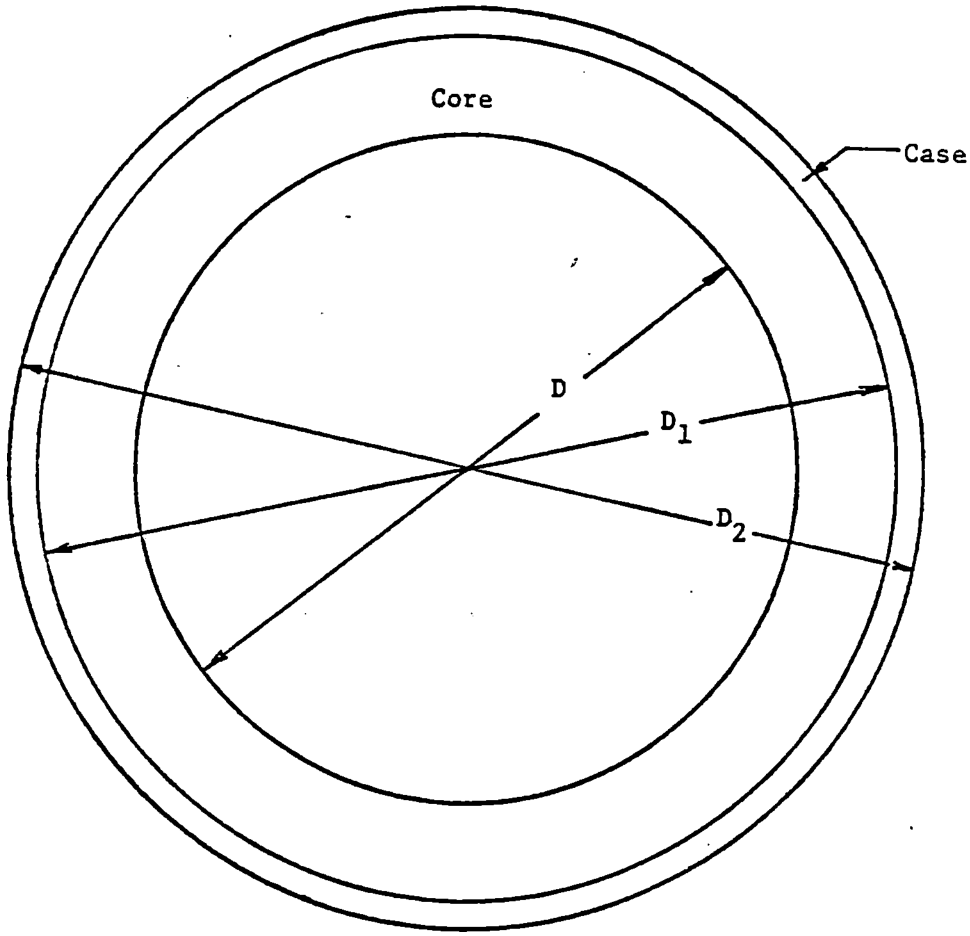


FIGURE 25. SCHEMATIC DIAGRAM FOR CONE BORE GROWTH MODEL

After plastic deformation of the outer cylinder occurs, the combination of both cylinders expands. The change, ΔD, in the diameter of the inner cylinder -- the cone bore growth -- can be calculated from the well known formula for compound cylinders. From (26), ΔD can be derived as:

$$\Delta D = \frac{(D_2^2 - D_1^2) D \varepsilon_p}{D_2^2 - D^2} , \qquad (22)$$

where D is the undeformed cone bore, D_1 is the diameter of the plastic-elastic interface, D_2 is the outer diameter of the cone, and ε_p is the circumferential-plastic strain. For a 6 x 11 cone, the dimensions D and D_2 are, respectively, 5.688 in. and 6.390 in. If the plastic strain in the plastic case is 0.002 in./in. (the engineering yield point) and if the plastic case is 0.015 in. deep, ΔD becomes 0.00054 in. This growth is in the range of that observed. Consequently, the model indicates cone bore growth sufficient to describe the growth actually obtained from cones in use.

4.3.3.2 Impact of Cone Bore Growth on Overall Defect Rate - Table 15 and Figure 14 shows that the oversize cone bore defect accounts for a majority of the cones rejected in rework. Since there is little if any evidence to indicate that oversize cones are a safety hazard, the overall bearing defect distribution is recalculated in Figure 26 with the oversize bearing defect removed. Removing this defect mode increases the bearing B_{10} life from two years to 2.4 years. This small change can be explained by the fact that the cup exhibits a higher defect rate (Table 16) than the cones and is therefore the limiting component.

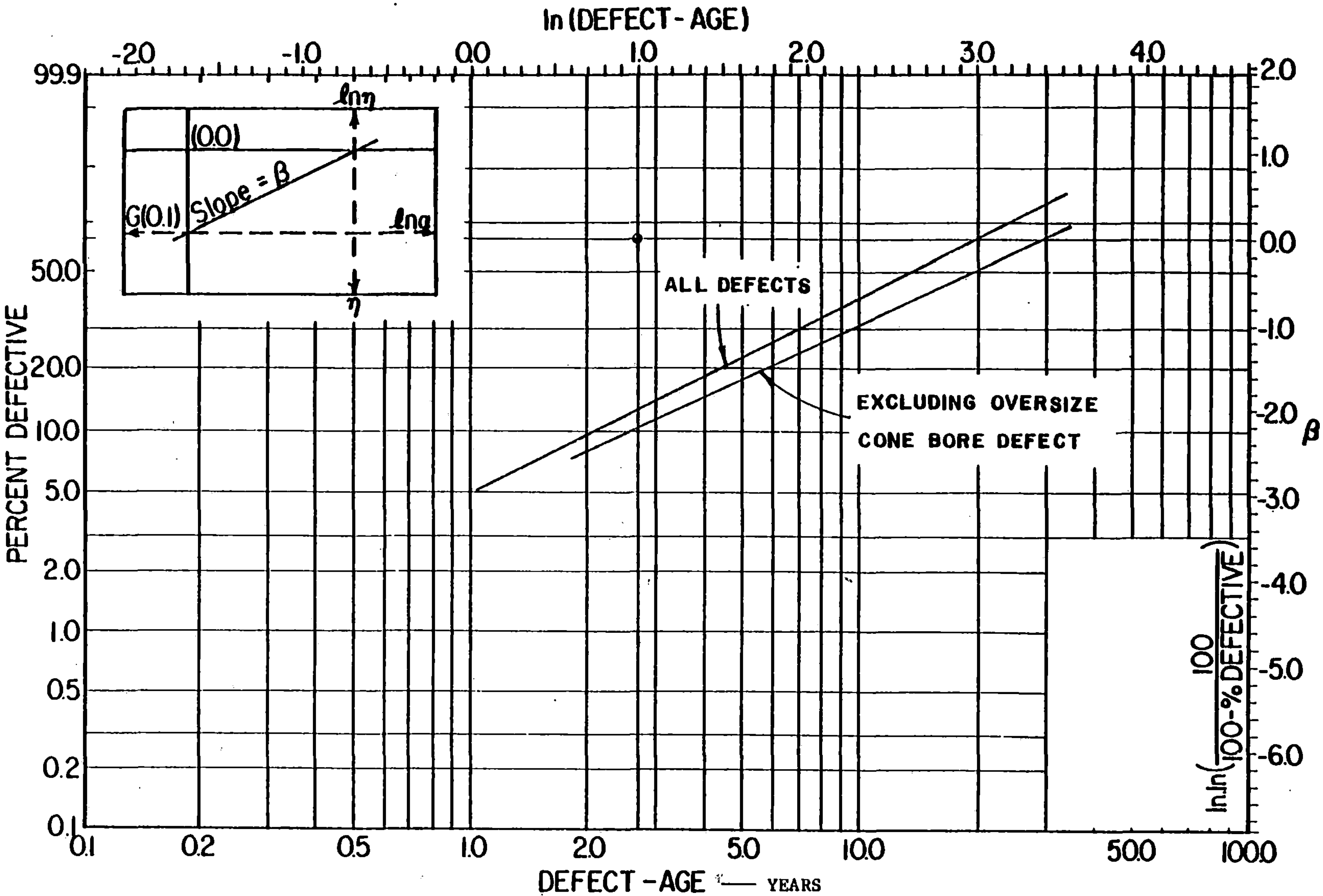


FIGURE 26. COMPARISON BETWEEN DEFECT DISTRIBUTIONS WITH AND WITHOUT OVERSIZE BORES

4.4 SEAL LIFE

The lip seal is almost universally used in the railroad industry for retaining grease in the bearing cavity of railroad roller bearings. The survey of railroad roller failure reports data described in Section 3 indicates that seal-related causes account for about half of all hotbox setouts and burnoffs. This is illustrated in Table 5 where data from three different sources confirm the significant proportion of seal-related failures.

Despite the obvious importance of seals to the railroad reliability problem, there is surprisingly little data on either the failure or defect rate characteristics of lip seals.

4.4.1 Test Experience

The performance of lip seals (even those of the same general design and material classification) depends on the particular compound formulation, type, and degree of cure and details of lip geometry, i.e., there can be wide variations in performance among manufacturers. To illustrate this, the results of an 89-hour test measurement by Brenco (26) on a range of different makes of seal are shown in Table 24.

TABLE 24

SEAL PERFORMANCE COMPARISON

Bearing Mfg. Used By	Seal Type	Average Residual Shaft Interference	Wear Ring Path Width (inches)
X	Reference Conventional - Case 1	0.038"	3/64 to 5/64
X	Conventional - Case 2 (Used Prior to 1970)	0.020	1/16 to 1/4
Y	Hydrodynamic - Case 1	0.025	3/64 to 1/8
Y	Hydrodynamic - Case 2	0.028	1/16 to 3/32
Y	Conventional - Case 3	0.023	3/64 to 1/16
Y	Conventional - Case 4	0.020	3/32 to 1/4
Z	Conventional	0.023	1/16 to 3/32

*Grease-lubricated bearing tested at 80 mph equivalent speed with programmed internal pressure up to 25 psi and bulk grease temperature controlled to 150°F.

It is apparent that the reference conventional case (X Case 1) has more shaft interference and hence greater performance capability to accommodate a higher degree of runout with longer life potential than even the hydrodynamic designs. In addition, the change in mechanical properties has been observed to be less with the reference case seal compound. Comparable results have been obtained in service-simulated, high-speed, fully loaded test bearings subjected to a normal outside environment. For example, the average residual shaft interference values for the reference conventional Case 1 (Mfr. X) and Case 3 (Mfr. Y) are:

(service-simulated test, 480,000 miles)

Mfg.	Case	Residual Shaft Interference
X	1	0.033
Y	3	0.011

Actual field service observations have been made that substantiate these general test results. After nearly 500,000 miles of high-speed unit-train operation, seals from Mfr. X Case 2 were observed to be excessively worn, and although still retaining grease, they were near the end of useful service life -- acting primarily as labyrinth type seals at this state. However, similar train operation with Mfr. X Case 1 seals showed relatively minor lip wear (less than 3/32-inch width) with significant residual shaft interference. Yet both of these seals pass the current certification procedure.

This experience shows the wide variation in seal quality that presently exists today. It is also important to observe that over half of the bearing failures reported by the AAR (Table 6) are classified as being seal-related.

4.4.2 Seal Defect Characteristics

Because of the lack of information on railroad roller bearing seal life, it was decided to take a sample of seals during the rework process. As described in Section 3.5, the seals were measured and the data recorded. The data were then coded, sorted by defect mode, and analyzed to determine seal defect characteristics with age. Basically, three types of analyses were performed. The first consisted of obtaining the distribution table for the parameter of interest as a function of time. The second involved obtaining a hazard plot for each failure mode; and the third consisted of fitting a Weibull defect distribution to each defect mode.

4.4.2.1 Distribution Curve — Figure 27 is a typical distribution table for the wear of a 6 x 11 seal. For example, Figure 27 tells us that there were 5 seals 11 months old. Three of these exhibited wear of 1/64 inch and two exhibited wear of 1/32 inch. For each defect distribution, tables were generated through 200 months.

4.4.2.2 Hazard Data - Plotting and analysis of hazard data (28) must take into account the form of the data. Defect data can be complete or incomplete. If defect data contain the defect times of all units in a sample, the data are complete. If defect data consist of defect times of defective units and running times of good units, the data are incomplete and are called censored and the running times are called censoring times. If the good units all have the same censoring time, which is greater than the defect time, the data are singly censored. If good units have different censoring times, the data are multiply censored.

Complete data result when all units become defective. Singly censored data result in life testing when testing is terminated before all units become defective. Multiply censored data result from removal of units from use before defect occurrence, from loss of units due to extraneous causes, and from collection of data while units are still operating.

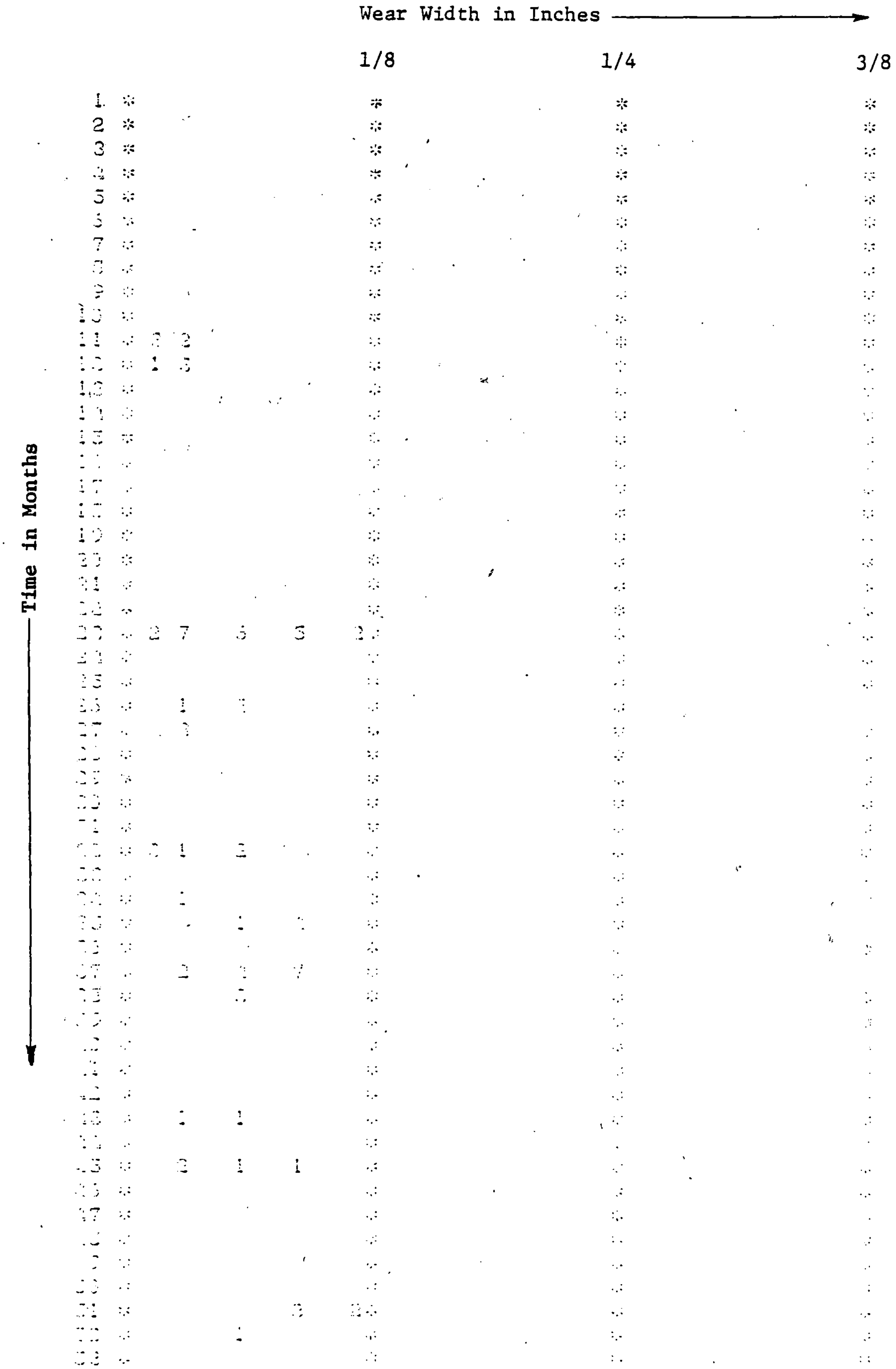


FIGURE 27. DISTRIBUTION CURVE FOR WEAR OF 6 x 11 BEARING SEAL

To obtain the hazard plot, we must invoke the defect criteria of wear 1/8 inch or greater. If the wear is less than 1/8 inch, it is counted is considered a discontinued test. If equal to 1/8 inch, it is counted as a failure. If greater, it is not counted since we have no way of knowing when it became defective -- i.e., reached 1/8-inch wear. If we treat good seals (wear < 1/8 inch) which are removed at rework as removal of units before defect occurrence, then our data populations are censored. We will further take the censoring time as the age of the seal at the time of removal at rework. Figure 28 is a typical example of hazard data.

The data have been ordered from youngest to oldest without regard to whether they are censoring times or failure times. The hazard value, h(t), for a failure time is 100 divided by the number K of units with a failure or censoring time greater than or equal to that failure time. The K value is given in parentheses next to the unit number. The cumulative hazard, H(t), is the cumulative sum of all the hazard values up to and including h(t). Using linear regression analysis, the equation

$$\log(t) = \frac{1}{\beta}\log H(t) + \log\eta \qquad (23)$$

was fitted to the data.

Once the hazard is known, the cumulative distribution function (percent defective) can be calculated from:

$$F(t) = 1 - e^{-H(t)} . \qquad (24)$$

This relationship is used later to derive Figure 29 and the plots in Appendix C.

4.4.2.3 Defect Rate Distribution — Using the defect life definitions described in Figure 10, the cumulative percent defective of all seals examined have been plotted in Figure 29. A Weibull slope of 2.34 was estimated from the data, which compares favorably with other life data

FOR 1/8" THRESHOLD, POPULATION SIZE IS . 131

AGE (MONTHS)	CUM. HAZARD	REV. RANK	HAZARD
23	.8196721311475	122	.8196721311475
33	1.6461184121389	121	.8264462809917
51	2.9448197108[illegible]	77	1.2987012987013
51	4.2606091845[illegible]	76	1.3157894736842
74	5.7098845468[illegible]	69	1.4492753623188
7[illegible]	7.1804727821[illegible]	68	1.4705882352941
7[illegible]	8.67301009[illegible]	67	1.4925373134328
7[illegible]	10.188161610[illegible]	66	1.5151515151515
7[illegible]	11.72662314[illegible]	65	1.538461538[illegible]
77	13.28912314[illegible]	64	1.5625
[illegible]	[illegible]	62	[illegible]
[illegible]	17.17[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	2
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
117	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	10	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	2	[illegible]
[illegible]	[illegible]	1	100

FIGURE 28. HAZARD DATA FOR WEAR OF 6 x 11 SEAL

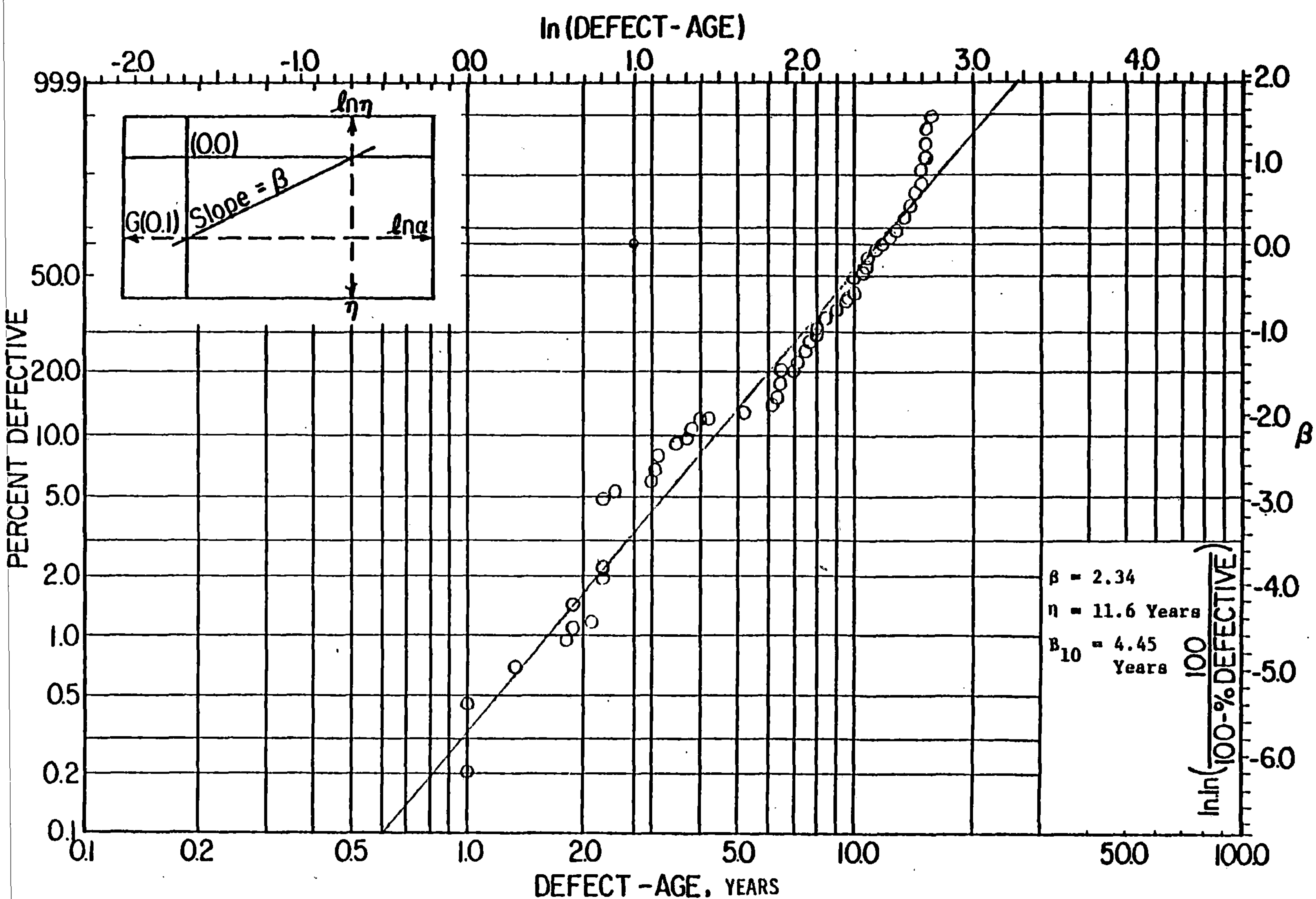


FIGURE 29. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE, ALL SEAL DEFECTS

reported for lip seals (28). Similar plots for individual defect modes are presented in Appendix C. The Weibull parameters for all defect modes are summarized in Table 25.

The seal B_{10} life was found to be 4.45 years. Since all seals are replaced at rework, this life is significant only if the mean time between bearing rework is greater than 4.45 years. If so, then defective seals can accumulate in the total seal population and may account for the high seal-related hotbox and burnoff incidents.

4.5 GREASE LIFE

In Reference (29) grease life is shown to vary according to a Weibull distribution given by:

$$\text{Log } L = -2.30 + \frac{2450}{273 + T} - .301S + \frac{1}{\beta}\log\left[\ln\frac{1}{1-F(t)}\right], \qquad (25)$$

where:

$S = S_G + S_N + S_W$

T = Temperature

β = Weibull slope = 3

$F(t)$ = Cumulative failure distribution

S_G = Grease life factor = 0 for railroad roller bearing grease

S_N = $0.86\ DN/DN_L$

S_P = $0.61\ DNP/C^2$.

Figure 30 shows a plot of grease life versus temperature for a fully loaded, 6 x 11 railroad roller bearing. Curves for other bearing sizes will be almost identical. The operating temperature is the dominant determinant of grease life. Speed is the next most important parameter. The load has even less of an impact on life than speed.

TABLE 25

WEIBULL PARAMETERS FOR
VARIOUS SEAL DEFECT MODES

Defect Mode	Weibull Slope β	Characteristic Life, η, Years	B_{10} Life, Years
All Defects	2.34	11.6	4.45
Wear, All Seals	3.54	15.7	8.46
Wear, 6 x 11 Seals	2.46	15.1	6.02
Wear, 6½ x 12 Seals	6.18	15	10
Diameter	2.04	16.6	5.5
Fit	4.71	15.2	9.45
Other (Blistered, Cracked, Etc.)	4.49	22.9	13.9

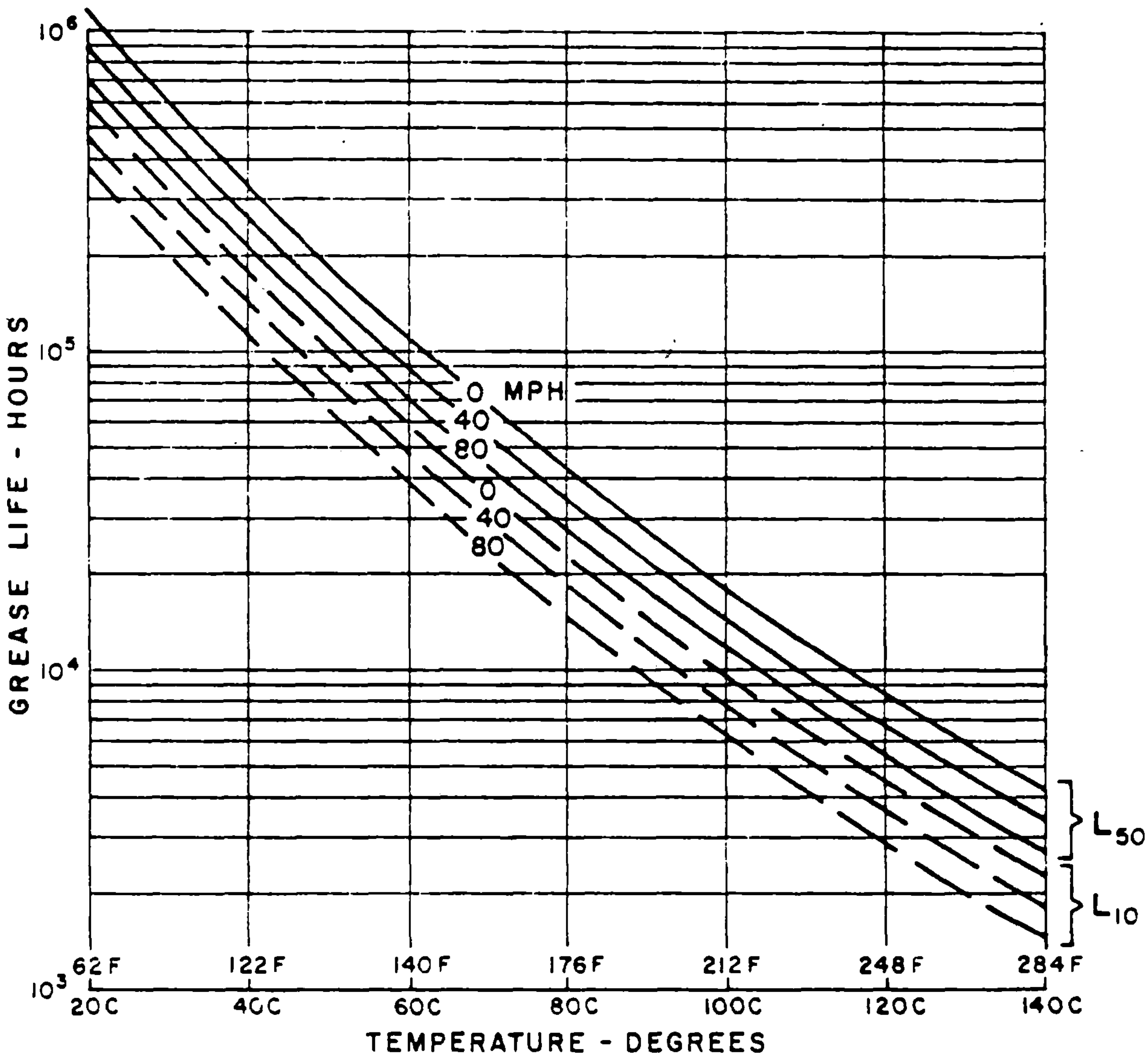


FIGURE 30. GREASE LIFE VERSUS BEARING OPERATING TEMPERATURE

Since the grease is subjected to temperatures in excess of 100°F (37°C) over a very short portion of the average bearing life, the grease L_{10} life will be in excess of 155,000 hours at a constant 20 mph (∿18 years). This life is beyond the lubrication interval and agrees with the failure study of Section 3 which showed no grease failures. For high speed or unit train applications, this life would be reduced significantly.

4.6 ROLLER BEARING ASSEMBLY DEFECT LIFE

As seen in Figure 16 the B_{10} defect life of a railroad roller bearing is 2.0 years with Weibull slope of 1.0. The grease life is 18.3 years with a Weibull slope of 3.0. (from Section 4.5). The seal, B_{10} life is 4.5 years with a slope of 2.34. Combining these reliability curves, using Equations (1) and (4) gives the bearing assembly defect life shown in Figure 31. The assembly defect life of 1.8 years is relatively short. However, it should be kept in mind that bearings containing condemnable defects can operate for many thousands of miles without affecting the safety of operation. (See Reference 5.)

4.7 ROLLER BEARING POPULATION CHARACTERISTICS

Experience has shown that the roller bearing is vastly superior to the plain journal bearing that it is replacing. Despite the fact that the roller bearing penetration of the fleet was over 60 percent by the end of 1975, the overall hotbox mileage statistics have remained unaltered at about 2,000,000 miles per setout for the last several years.

It is this levelling off of the hotbox mileage statistics with gives rise to questions about roller bearings becoming more likely to suffer failure with age. This section addresses this question and its effect on the freight car bearing population in the future.

4.7.1 Population Statistics

Table 26 summarizes the freight car ownership and fraction of freight cars equipped with roller bearings through 1975. Using these data, estimates can be made of the roller bearing population in the United

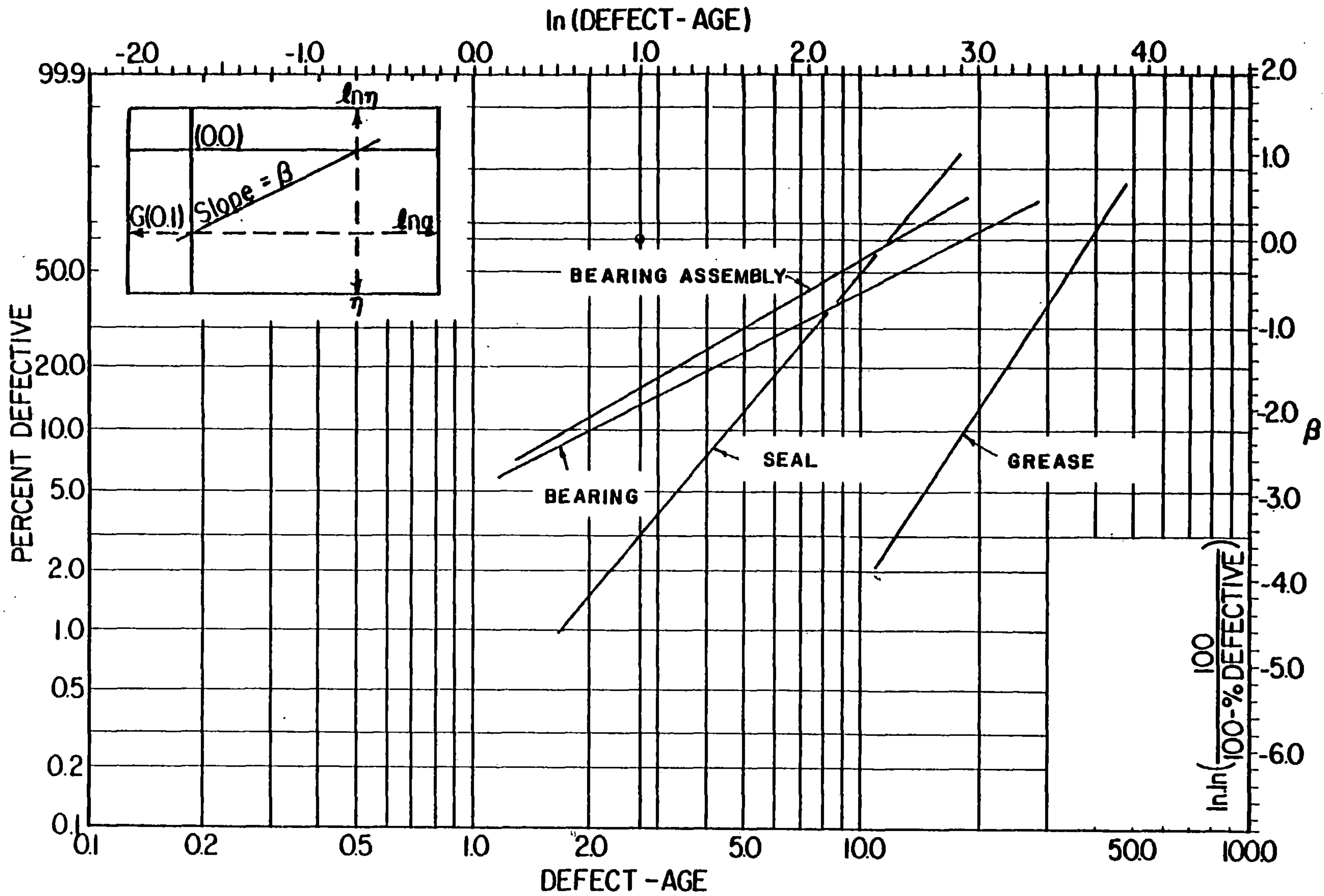


FIGURE 31. BEARING ASSEMBLY DEFECT LIFE

TABLE 26

FREIGHT CAR POPULATION FIGURES

Year	Total 1/ Freight Car Ownership	Freight Cars Equipped with Roller Bearings	% of Freight Cars 2/ Equipped with Roller Bearings
1975	1,723,605	1,068,635	62.0%
1974	1,720,573	980,727	57.0%
1973	1,710,659	919,308	53.74%
1972	1,716,937	885,939	51.6%
1971	1,762,135	731,990	41.54%
1970	1,784,181	656,911	36.45%
1969	1,791,736	580,385	32.34%
1968	1,800,375	505,740	28.09%
1967	1,822,381	450,714	24.73%
1966	1,826,499	367,464	20.12%
1965	1,800,662	273,455	15.19%
1964	1,796,264	209,007	11.64%
1963	1,814,193	156,721	8.64%
1962	1,850,688	121,280	6.55%
1961	1,905,268	97,114	5.10%
1960	1,965,486	76,674	3.90%
1959	1,980,531	47,286	2.39%
1958	2,031,181	38,420	1.89%
1957	2,054,311	34,661	1.69%
1956	2,009,764	27,352	1.36%

1/Reference:AAR Yearbook of Railroad Facts, 1976 Edition.

2/Reference:AAR Semi-Annual Summation of Performance Reports on Journal Roller Bearings.

States. This is shown in Table 27 and in Figure 32. Since 1964, the bearing population has been growing almost linearly. It is expected to continue to do so for the next ten years and eventually level off at a 2,250,000 car fleet in the 1990's.

The population increase in any one year is the difference between the addition rate and the removal rate, i.e.,:

$$\Delta P_n^* = S(t) - G^*(t). \tag{26}$$

Figure 33 illustrates this relationship for the year 1974. Of the 536,664 bearings removed in 1974, 45,616 were returned directly to service. Another 6,400 bearings were scrapped prior to reaching a rework shop. The remaining 484,608 bearings entered a rework shop where, based on the data of Table 2, approximately 19 percent (or 91,106 bearings), were scrapped. New components amounting to 11,679 bearings were added to rework to provide 405,181 reconditioned bearings which were returned to service. A similar flow diagram for 1975 is shown in Figure 34.

4.7.2 Bearing Rework as an Inspection Process

The majority of bearings sent to the rework shop for reconditioning are there by reason of wheel work and derailment rather than because they were pulled from service due to defects. Thus, the rework process is a form of inspection and provides the only significant means of removing defective bearings from the population. The inspection is only a partial process in that in 1974, for example, only 6.2 percent (491,048/7,894,266) of the bearing population was reworked, i.e., inspected.

If we assume that for bearings less than ten years old the method by which bearings are selected for rework is unbiased -- i.e., the chance of a bearing being selected for rework is independent of its age -- then the number of defective bearings introduced at time 0 remaining in the population at year t is:

TABLE 27

RAILROAD ROLLER BEARING

POPULATION HISTORY

Year	Estimated Roller Bearing Population $\overset{*}{P}(t)$	Estimated Population Increase $\Delta\overset{*}{P}(t)$	Estimated Total Bearing Population $P^{*}(t)_T$	Estimated New Roller Bearing Sales S(t)
1954	0	0	0	0
1956	218,816	218,816	16,078,112	218,816
1957	277,288	58,472	16,434,488	62,075
1958	307,360	30,072	16,249,448	34,068
1959	378,288	70,928	15,844,248	75,846
1960	613,397	235,104	15,723,888	243,078
1961	776,912	163,520	15,242,144	173,620
1962	970,240	193,328	14,805,504	205,905
1963	1,253,768	283,528	14,513,544	299,827
1964	1,672,056	418,288	14,370,112	440,025
1965	2,187,640	515,584	14,405,296	544,023
1966	2,939,712	752,072	14,611,992	790,288
1967	3,605,712	666,000	14,579,048	712,874
1968	4,045,920	440,208	14,403,000	492,805
1969	4,643,080	597,160	14,333,888	657,520
1970	5,255,288	612,208	14,273,448	680,527
1971	5,855,920	600,632	14,073,784	676,759
1972	7,087,512	1,231,592	13,735,496	1,323,730
1973	7,354,464	266,952	13,685,272	362,560
1974	7,894,266	539,802	13,764,584	637,353
1975	8,486,002	591,736	13,788,840	710,386

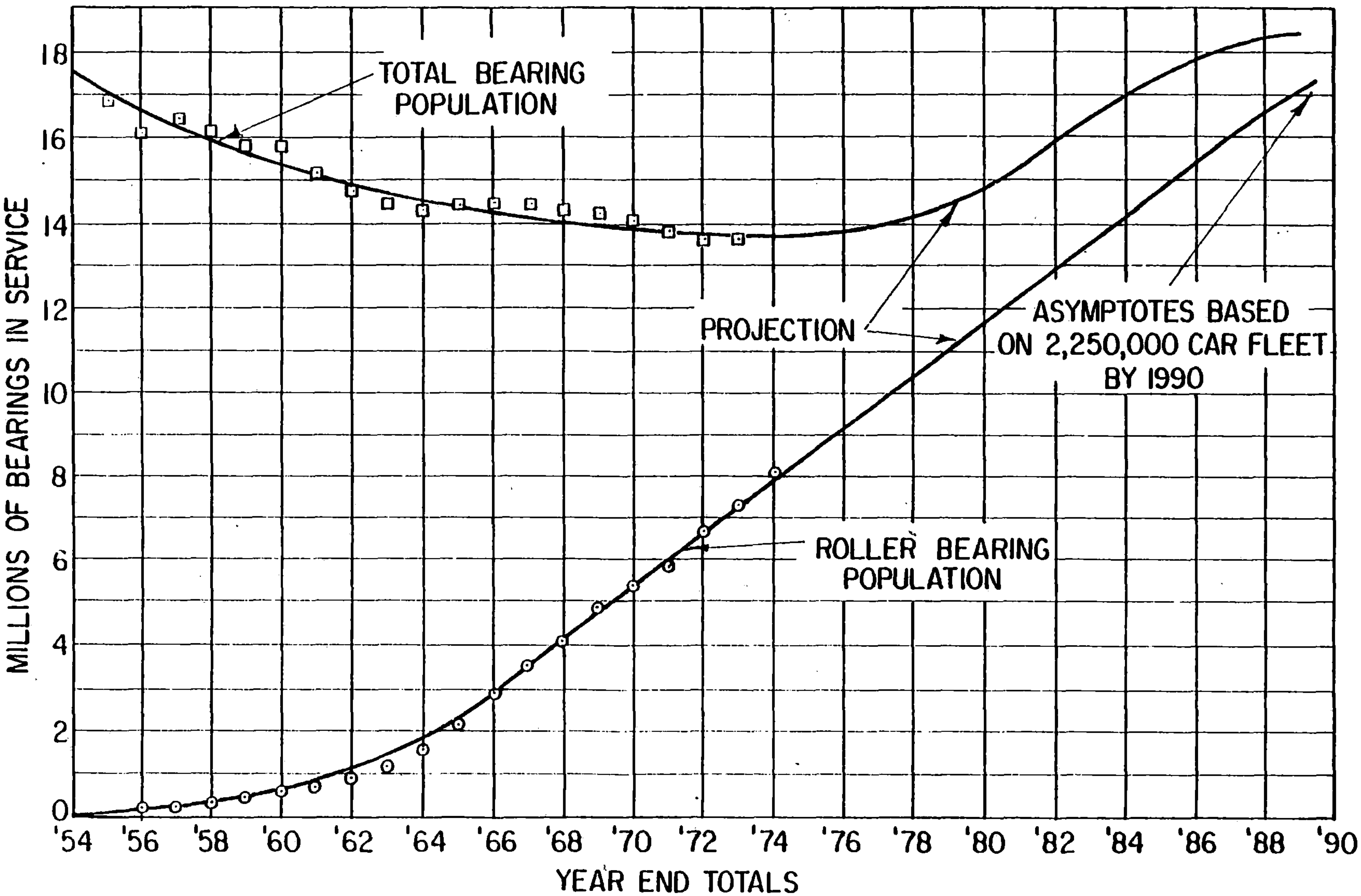
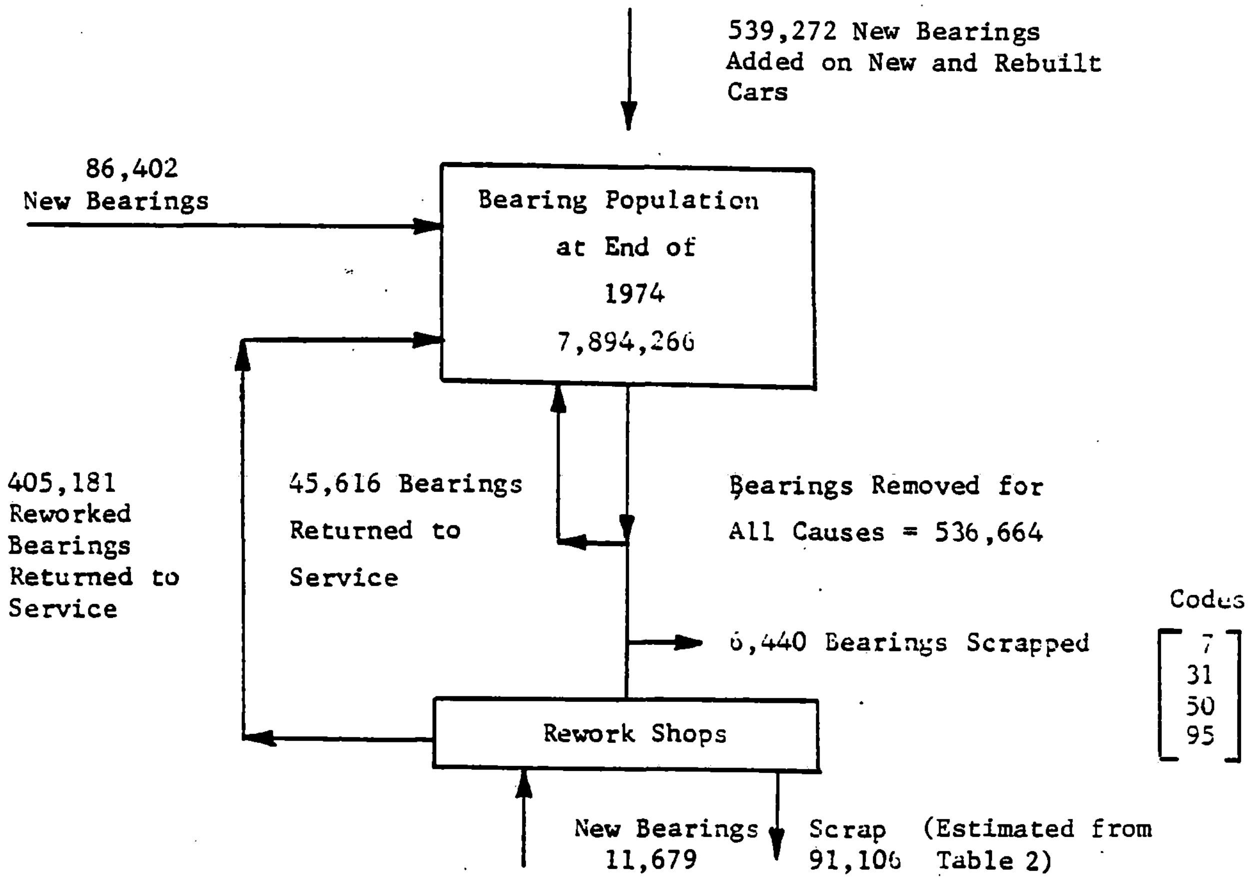


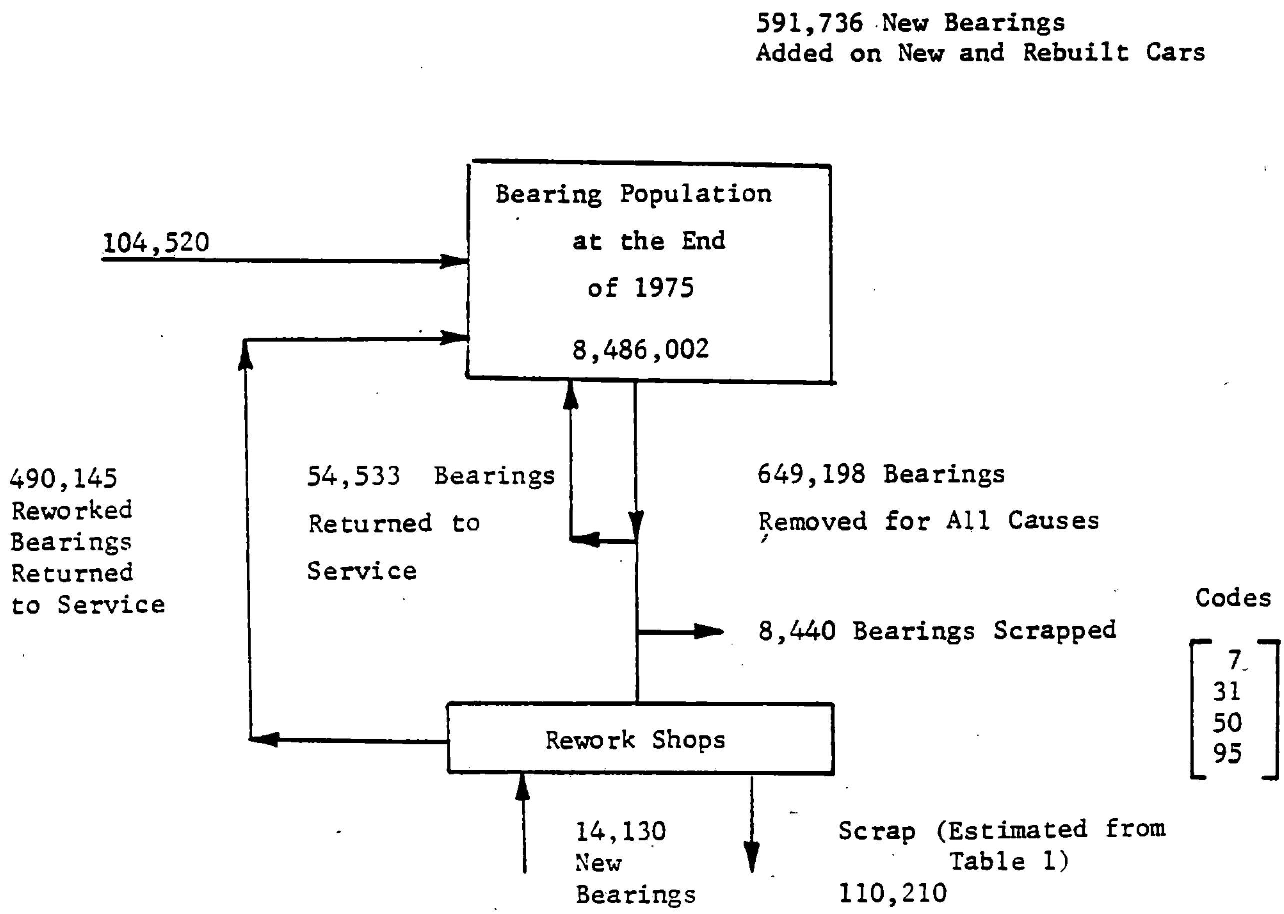
FIGURE 32. FREIGHT CAR BEARING POPULATIONS IN THE UNITED STATES



Total Number* of Bearings Added:	
New Bearings on New and Rebuilt Cars	539,272
Reconditioned Bearings	405,181
Second-hand Bearings Returned to Service	45,616
New Bearings	86,402
Total	1,076,471
Less Bearings Removed for All Causes	536,664
Population Increase	539,807

*Numbers based upon CRB data supplied by AAR and "The Yearbook of Railroad Facts."

FIGURE 33. BEARING FLOW DIAGRAM FOR 1974



Total Bearings* Added:

New Bearings on New and Rebuilt Cars	591,736
Reconditioned Bearings	490,145
Second-hand Bearings Returned to Service	54,533
New Bearings	104,520
Total	1,240,934
Less Bearings Removed for All Causes	649,198
Population Increase	591,736

*Numbers based upon CRB data supplied by AAR and "The Yearbook of Railroad Facts."

FIGURE 34. BEARING FLOW DIAGRAM FOR 1975

$$F^*(t) = F(t) - I \; F(t) - D^* \; (t-1) \; + D^* \; (t-1). \qquad (27)$$

Thus, for I = 1.0 (i.e., 100 percent inspection each year), $F^*(n) = 0$ and all bearings which have experienced a condemnable defect will have been removed from the population. This is illustrated in Table 28.

The effect of the percent inspection on the number of defective bearings remaining in the population is shown in Figure 39 (page 114). Thus, with the present level of rework, the number of defective bearings accumulating in the population could be significant.

Figure 35 applies to bearings at year n introduced into the system at time 0. If we add to this those bearings introduced in year 1, 2, 3, n, etc., then the population at year n is simply:

$$P^*(t) = \sum_{j=0}^{t} S(t) \; R^*(t-j). \qquad (28)$$

The number of bearings exhibiting defects in the population is:

$$P^*_D(t) = \sum_{j=0}^{t} S(t) \; F^*(t-j). \qquad (29)$$

It is anticipated that the roller bearing population will continue to grow at almost a linear rate over the next ten years. If the present level of rework (I = .062) and the rate of defect occurrence observed in the rework shop survey continue, then the number of defective bearings accumulating in the population can be calculated from Equation (29). This is illustrated in Figure 36, where the fraction of defective bearings (per AAR rule) accumulating in the population is shown as a function of time. This fraction will become substantial in the future unless the level of rework and/or inspection increases in order to cull out defective bearings as they occur.

4.7.3 Bearing Age Distribution

It was shown earlier in Figure 13 that there is an increasing bearing failure rate with bearing age. If the average bearing age remains

TABLE 28

CUMULATIVE NUMBER OF DEFECTIVE BEARINGS REMOVED FROM POPULATION

Year	Number of Defective Bearings Remaining in Population	Number of Bearings Remaining in Population	Cumulative Number of Defective Bearings Removed from Population
0	$F^*(0) = F(0)$	$R^*(0) = 1-D^*(0)$	$D^*(0) = IF(0)$
1	$F^*(1) = F(1) - D^*(1)$	$R^*(1) = 1-D^*(1)$	$D^*(1) = I\{F(1) - D^*(0)\} + D^*(0)$
2	$F^*(2) = F(2) - D^*(2)$	$R^*(2) = 1-D^*(2)$	$D^*(2) = I\{F(2) - D^*(1)\} + D^*(1)$
.			
.			
.			
.			
t	$F^*(t) = F(t) - D^*\ (t-1)$	$R^*(t) = 1-D^*(t)$	$D^*(t) = I\{F(t) - D^*(t-1)\} + D^*(t-1)$

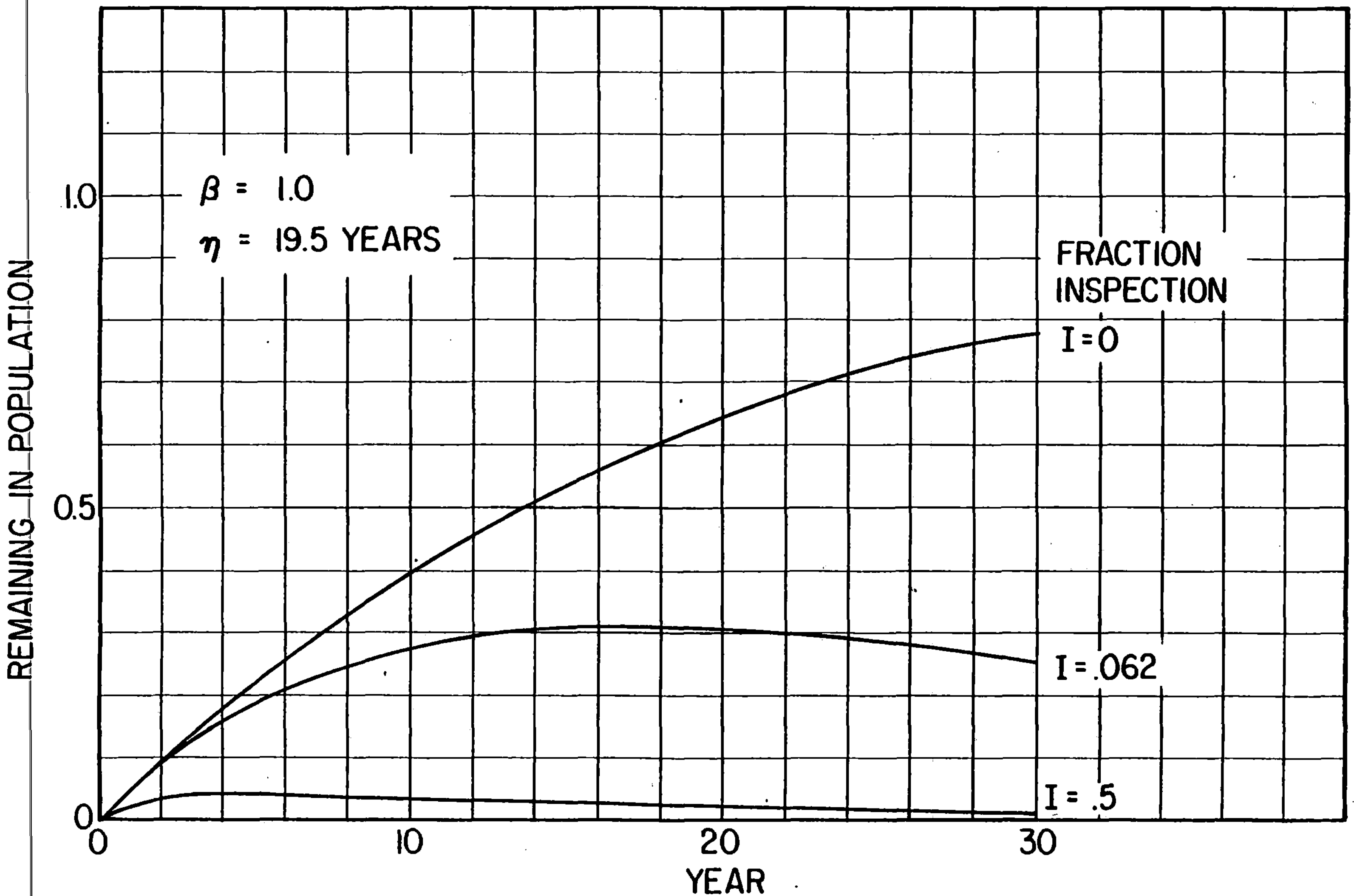


FIGURE 35. FRACTION OF DEFECTIVE BEARINGS REMAINING IN POPULATION AS A FUNCTION OF TIME

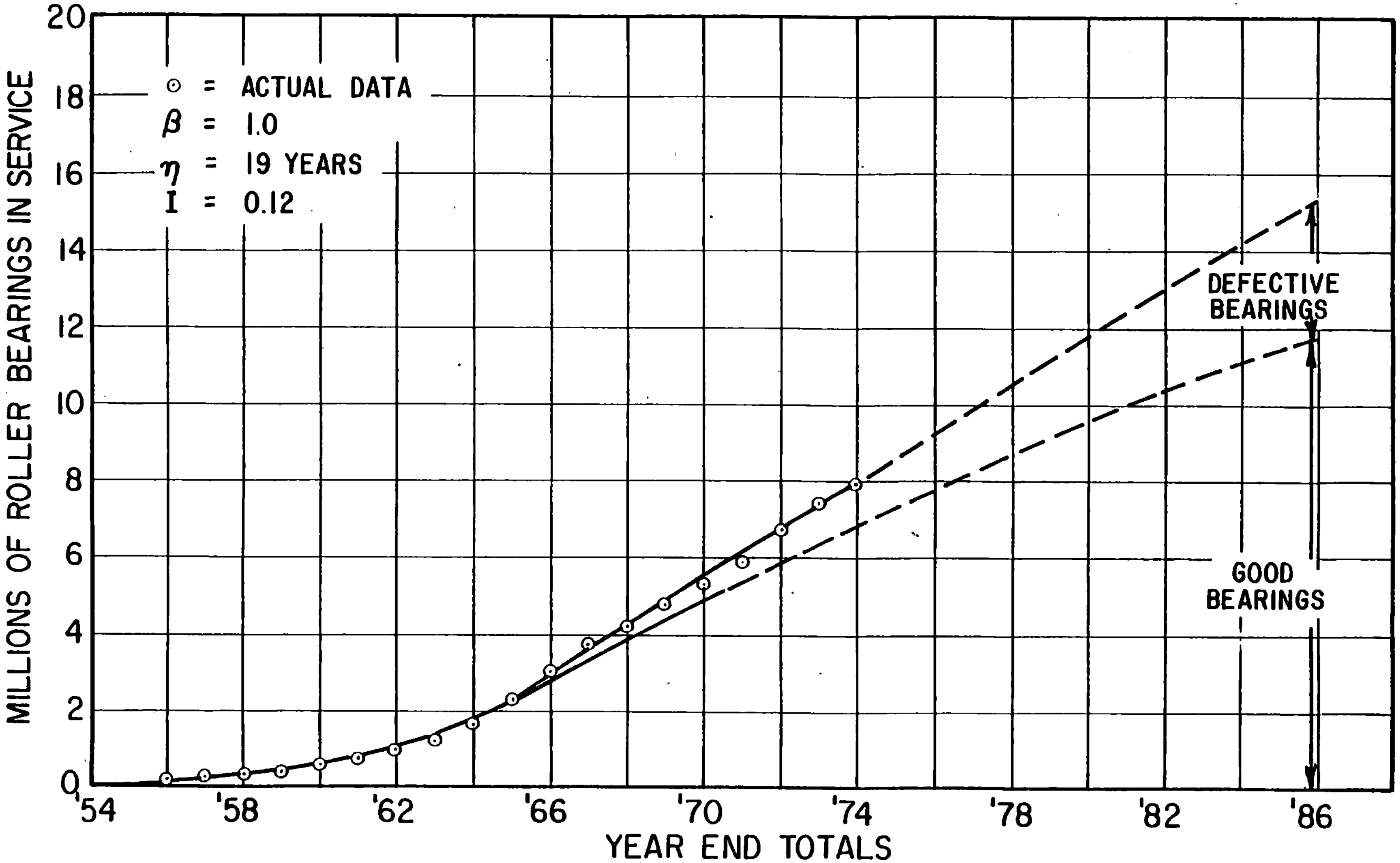


FIGURE 36. FREIGHT CAR ROLLER BEARING POPULATION IN THE UNITED STATES

constant, then the number of roller bearing failures occurring each year will rise in direct proportion to the bearing population (because the failure rate is constant).

If, however, the bearing age distribution should change in the future in such a way as to increase the average age, then we can expect an increasing failure rate, i.e., the number of bearings failing each year will increase at a faster rate than the population.

Each term in Equation (28) represents the number of bearings j years old. Figure 37 is a bar chart of Equation (28) showing the age distribution of bearings in the population as a function of time. The shaded section represents the estimated portion of the bearing population which contains condemnable defects.

At the present level of rework (I=.062), and linear population increase, the average age of the bearing population can be expected to increase to approximately 11 years by 1986.

As shown in Figure 32, the roller bearing population will eventually level off and the sales rate will equal the removal rate. It is shown in Appendix E that in the limit the average age of the population will approach the bearing characteristic defect life, which from Figure 16 is approximately 18 years.

4.7.4 Projected Failure Rate

The absolute number of bearings which will fail at year t is given by:

$$h^*(t)\ P^*(t)\ \Delta t_t = S(0)\Delta t_0\ R(t)\ h(t)\ \Delta t_t + S(1)\ \Delta t_1\ R(t-1)\ h(t-1) + \\ S(t)\ \Delta t_1\ R(0)\ h(0)\ \Delta t, \qquad (30)$$

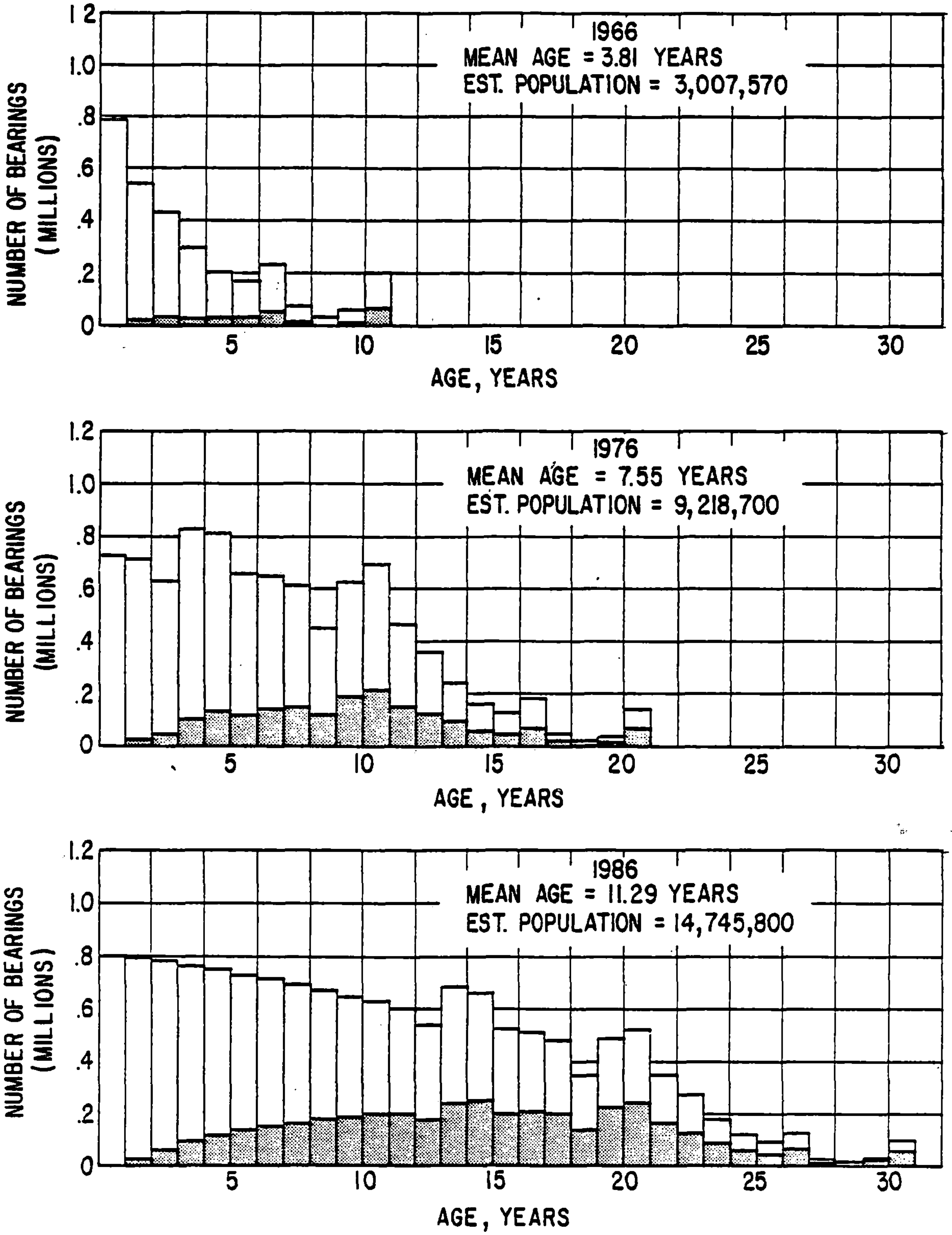


FIGURE 37. BEARING AGE DISTRIBUTION AS A FUNCTION OF TIME

or

$$h^*(t) = \sum_{i=0}^{t} \frac{S_i \Delta t_i \; R(t-i) \; h(t-i)}{P(t)} \; . \tag{31}$$

Figure 38 is a bar graph illustrating each of the terms in Equation (30) for the year 1986. The number of bearings failing and the failure rate in each age group are indicated. It is seen that although the failure rate increases continuously with bearing age the absolute number of bearings failing in each age group reaches a maximum at an age of about fifteen years.

Figure 39 shows a plot of Equation (30) as a function of time. The circles are the actual confirmed setouts and derailments as reported by the AAR. Thus, Equation (30) predicts that the absolute number of failures will rise 196% over the next ten years while the population size will grow by 70%. The increment beyond a linear increase is due to the aging of the population.

The rate of increase of the actual AAR experience through 1975 does not appear to be quite as high as the projection based on Equation (30). This projection is based on three measured quantities, the historical failure rate (Table 14), the historical defect rate (Figure 16) and the historical degree of rework (Figures 33 and 34). To make the projections more accurate, this historical data should be continuously updated and the projection rerun on a yearly basis. The fact remains that because the number of roller bearings in service is increasing and because the average age of these bearings is rising, the industry can expect an increase in the number of confirmed roller-bearing-caused setouts and derailments.

It is also instructive to ask what will happen to the failure rate as the population levels out in the future.

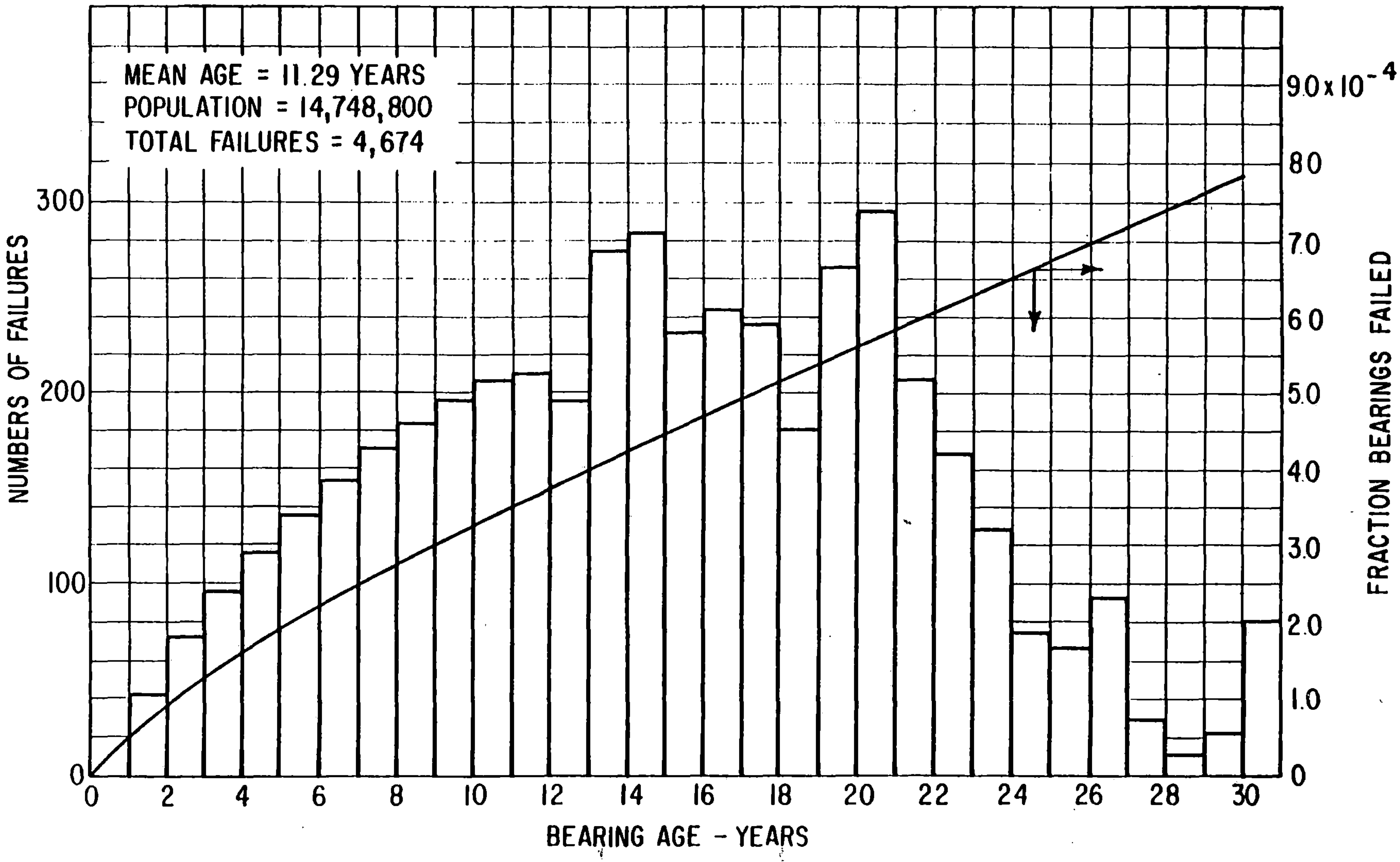


FIGURE 38. PROJECTED NUMBER OF BEARING FAILURES VERSUS BEARING AGE (1986)

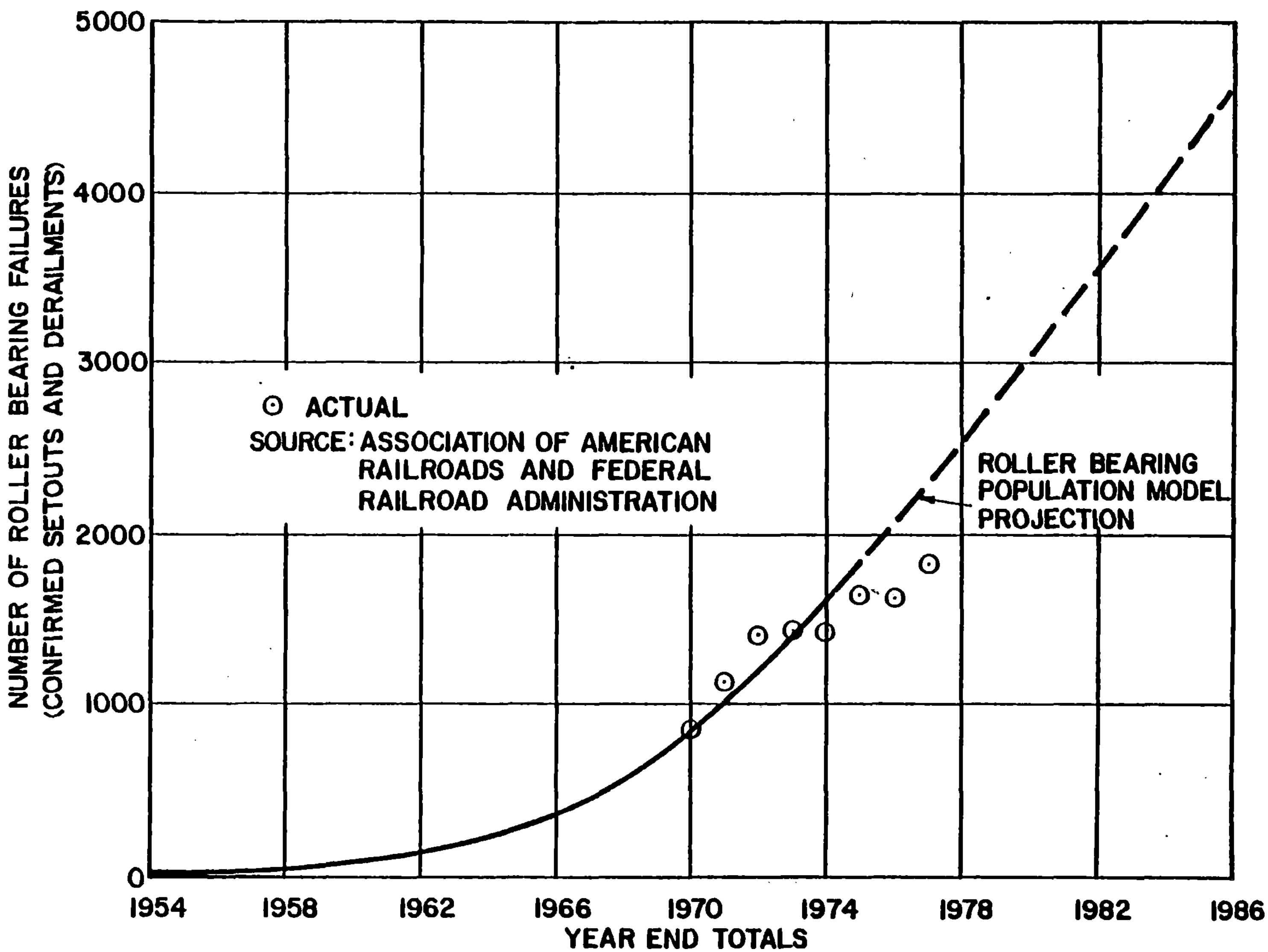


FIGURE 39. PROJECTED ROLLER BEARING FAILURES VERSUS TIME

If the failure distribution is a Weibull and S(t) = constant, it is shown in Appendix D that the failure rate can be expressed by:

$$h^*(t) = \frac{\int_0^t e^{-(\frac{t-\tau}{\eta})^\beta} \beta \, \eta^\beta (t-\tau)^{\beta-1} d\tau}{\int_0^t e^{-(\frac{t-\tau}{\eta})^\beta} d\tau} \tag{32}$$

or

$$h^*(t) = \frac{(1 - e^{-(t/\eta)^\beta})}{\int_0^t e^{-(\frac{t-\tau}{\eta})^\beta} d\tau} . \tag{33}$$

If $t \to \infty$, then $h^*(t)$ approaches a limit:

$$h^*(t) = \frac{\beta}{\eta\Gamma(\frac{1}{\beta})} = \frac{1}{\mu} . \tag{34}$$

The fact that the population failure rate approaches a constant as the population matures is an important conclusion. Further, the limiting failure rate is a function of only two parameters: the bearing characteristic life and the Weibull slope. The roller bearing industry is still in a growth phase and the failure rate is rising. As the total population stabilizes, the failure rate will approach the asymptote, $1/\mu$.

5. ACCELERATED TESTING

The previous discussion has indicated that either larger sample sizes or longer test times are required to increase the degree of protection afforded America's railroads by the present acceptance procedures. This means either an increase in the number of test machines or an increase in the test time. Another alternative is the use of accelerated testing.

Accelerated testing is achieved by subjecting the test units to conditions that are more severe than the normal ones. This results in shorter lives than would be observed under normal conditions. The results obtained at the more severe or accelerated conditions are then extrapolated to the normal conditions to obtain an estimate of the life distribution under normal conditions. Such testing provides a saving in time and expense compared with testing at normal conditions. Indeed, for railroad roller bearings, life at normal conditions is sufficiently long that testing at those conditions can be time consuming and costly.

Accelerated test conditions are typically produced by testing bearings and seals at higher levels of load, temperature, pressure, vibration, cycling rate, etc., or some combination of them than are encountered under normal conditions. The use of certain accelerating variables, as these are called, for a specific bearing or seal is usually established by engineering practice. For example, for accelerated testing of greases where temperature is an accelerating variable, the Arrhenius model is often used. Then life data obtained from units tested at different constant elevated temperatures are extrapolated to obtain an estimate of the life distribution at normal temperatures.

5.1 INVERSE POWER LAW MODEL

For bearings and seals tested at constant stress, the inverse power law model has frequently been used as a measure of life as a function of the stress. The assumptions of the model are:

(i) For any constant value, V, of the stress (which must be positive), the life distribution is Weibull where

(ii) The shape parameter, β, of the Weibull distribution is constant (i.e., independent of the stress) and

(iii) The characteristic life, η, at the 63rd percentile is an inverse power function of the accelerated variable, V; that is:

$$\eta(V) = (V_o/V)^n. \tag{35}$$

Here β, V_o, and n are positive parameters, characteristic of the component and the test method. Equation (35) is called the inverse power law.

In the case of roller bearings, Palmgren's equation is expressed as:

$$L_{10} = (\frac{C}{P})^{10/3} \times 10^6 \text{ revolutions}, \tag{8}$$

where:

$$L_{10} = \eta(V)\left[\ln\{-R(0.9)\}\right]^{1/\beta}, \quad n = 10/3 \tag{36}$$

and

$$V = P \qquad V_o = C$$

Under these assumptions, the fraction F(t;V) of units failing by time, t, under a constant stress, V, is:

$$F(t;V) = 1 - \left[\exp - \{t(V/V_o)^n\}^\beta\right], \quad t>0. \tag{37}$$

The 100F'th percentile, $t_F(V)$, of the life distribution for a stress value, V, may be written as:

$$\ln\{t_F(V)\} = n\ln(V_o/V) + (1/\beta)\,u(F), \tag{38}$$

where $u(F) = \ln\left[-\ln(1-F)\right]$ is the 100F'th percentile of the standard extreme value distribution.

For any two values V_1 and V_2 of the stress, the corresponding 100F percentiles $t_F(V_1)$ and $t_F(V_2)$ satisfy:

$$t_F(V_2) = (V_1/V_2)^n \, t_F(V_1). \tag{39}$$

This comes from Equation (37). Because the relationship (see Equation (39) holds for any percentage, 100F, it is sometimes written as:

$$t_2 = (V_1/V_2)^n t_1, \tag{40}$$

where the notation refers to any percentile. Thus, the relationship gives a test time t_2 at stress V_2 that is equivalent to a test time t_1 at another stress V_1. The test times are equivalent in the sense that the percentage failing is the same for both.

5.2 ACCELERATION FACTORS - FATIGUE

If we apply Equation (38) to the case of fatigue mode of roller bearings, we have:

$$\ell n \{t_F(P)\} = \frac{10}{3} \ell n \ (C/P) + (1/\beta)u(F) - (1/\beta)u(.1). \tag{41}$$

Considering a 6 x 11 railroad roller bearing, the following values apply:

C = 133,633 pounds (from Table 18)
β = 1.78 (from Figure 19)
n = 3.333 (from Equation 8).

Thus, on log-log paper, the relationships between the percentiles and the load are parallel straight lines as shown in Figure 40.

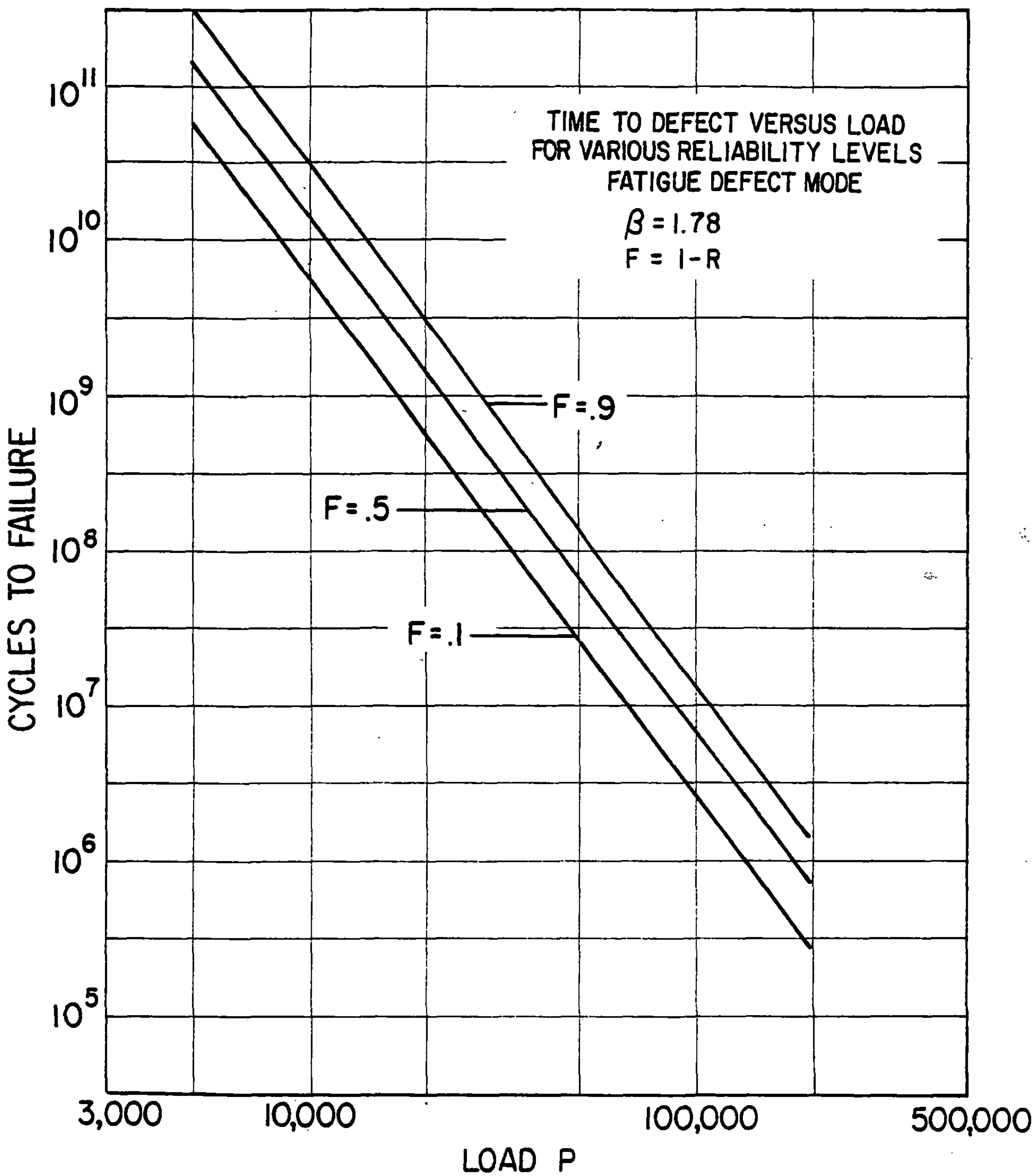


FIGURE 40. NUMBER OF CYCLES TO DEFECT VERSUS BEARING LOAD FOR VARIOUS RELIABILITY LEVELS

The life distributions for different stresses can also be depicted on Weibull probability paper as shown in Figure 41. On Weibull probability paper, the relationship (37) between the cumulative fraction of units failing and their age is a straight line. In Figure 41, the straight lines for the distributions at different stress levels are parallel. This comes from assumption (ii) that the distributions have the same shape parameter, β, which determines the slope of the distribution lines. The placement of the distribution lines for different stresses is determined by the characteristic lives (63.2% points on the straight lines) which are given by the relationship (35).

Now the present AAR procedure calls for loading the test bearings to the loads listed in Table 29. If we were to increase the test load to twice this load, the reduction in test time for a 6 x 11 bearing would be:

$$\frac{L_{10_T}}{L_{10_P}} = \left(\frac{P}{P_T}\right)^{3.33} = \left(\frac{26,250}{52,500}\right)^{3.333} = 0.099. \qquad (42)$$

Thus the saving in test time is roughly a factor of 10.

Referring to Equation (8) we could also increase the number of cycles, i.e., speed, to reduce test time. This approach, however, is not as effective since test time is reduced only linearly with speed.

These results are summarized in Table 30.

5.3 ACCELERATION FACTORS - BRINELLING

A railroad roller bearing supporting a freight car rolling on a track is subjected to a fluctuating load superimposed on a steady state load. The brinelling defect mode is influenced not only by the amplitude of the imposed load but also by the "apparent" frequency of its fluctuating component.

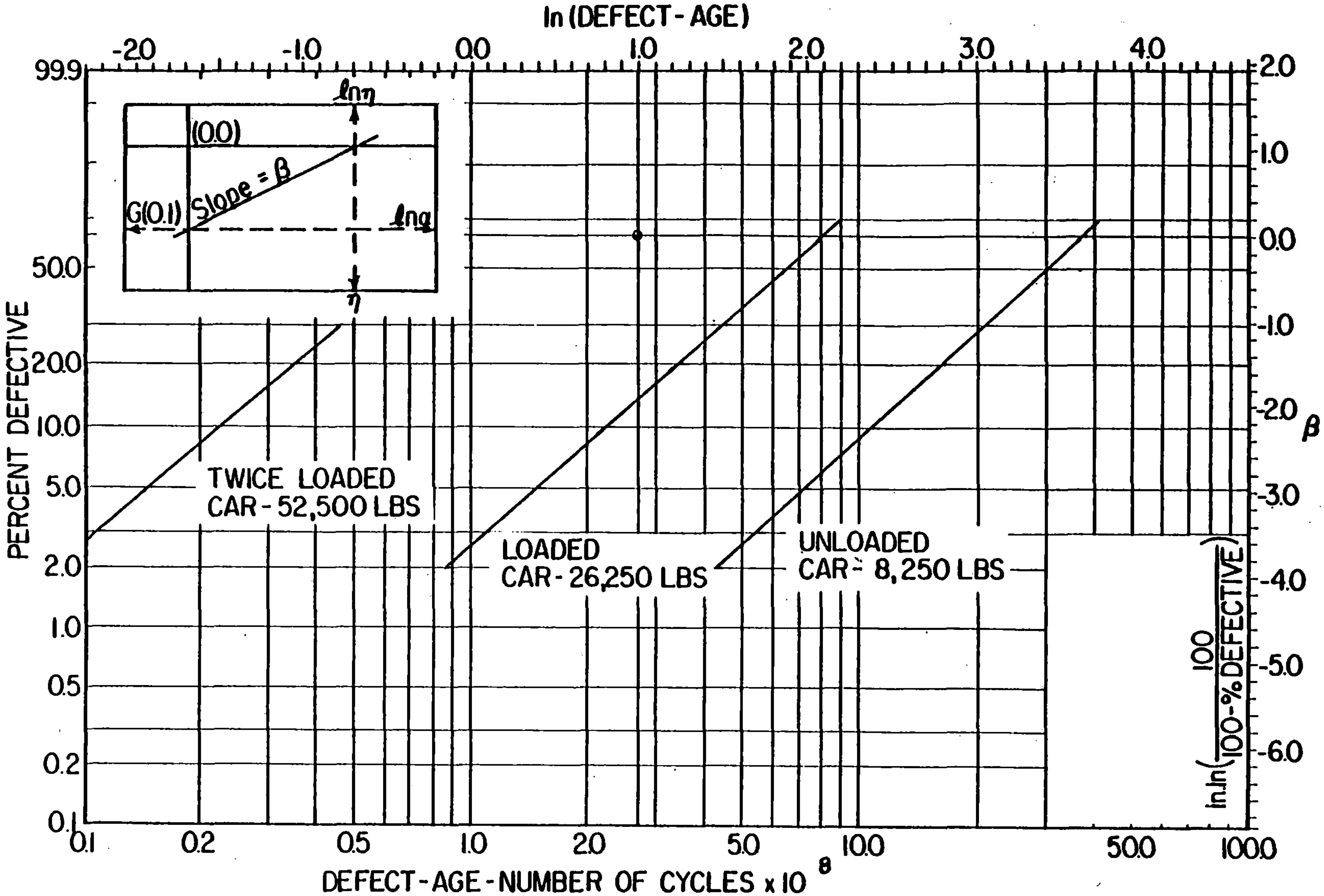


FIGURE 41. POWER LAW MODEL ON WEIBULL PROBABILITY PAPER

TABLE 29

CALCULATED FATIGUE LIFE
FOR ACCEPTANCE TEST CONDITIONS

The AFBMA formula for radial load rating in pounds for 1,000,000 Revs. is given in the following form:

$$C = f_c\{(i\ \ell_{eff}\ \cos\ \alpha_c)^{7/9}\ Z^{3/4}\ d^{29/27}\},$$

Where:

- f_c is a factor tabulated in the AFBMA standard for roller bearings at values of the parameter ($D\ \cos\ \alpha/d_m$)
- d is the mean roller diameter in inches
- ℓ_{eff} is the effective roller length in inches
- d_m is the mean pitch diameter of the roller complement in inches
- α_c is the contact angle (1/2 included cup angle) in degrees
- Z is number of rollers per row and i is the number of rows.

Numerical Values are tabulated for typical railroad tapered roller bearings.

Factor Name	5 1/2 x 10	6 x 11	6 1/2 x 12	7 x 12
d, Mean Roll Dia.	0.6968	0.7047"	0.84235	0.9094"
ℓ_{eff} Effective Roll Length	1.5373	1.5572	1.8653	2.0163
d_m Mean Roll Pitch Dia.	6.60978	7.0946"	7.99057	8.7964"
α_c 1/2 Included Cup Angle	10^o-0'	10^o-0'	10^o-0'	10^o-0'
Z No. Rolls per Row	23	24	23	23
i No. Rows per Bearing	2	2	2	2

AFBMA life ratings for AAR certification test loads are summarized below:

Bearing Size	Test Loads, lbs.		Life, L_{10} (Millions of Revs.)
	Radial	Axial	
5 1/2 x 10	20,000	1,200	299
6 x 11	26,250	1,575	141
6 1/2 x 12	30,000*	1,800	252
7 x 12	30,000*	1,800	402

* Test machine capacity.

TABLE 30

ACCELERATION FACTORS

FOR MAJOR DEFECT MODES

	Major Defect Modes		
Failure Model	Fatigue	Brinelling	Cone Bore
Acceleration Factors	Load, P Speed, N	Load, P Number of Cycles	Load, P Speed, N
Type of Law	Inverse Power Law	Inverse Power Law	Inverse Power Law
Form of Law	$L = \left[\frac{C}{P}\right]^{10/3}$		
Limit on Acceleration Factor	$P < 1/2\ C$	Bulk Fracture of the Material	Bulk Fracture of the Material

It would be highly desirable if the statistics of the load spectrum could be converted into statistics on the reliability of the bearing in the fluctuating load environment. Unfortunately, even in cases where a clear-cut failure model exists, the purely mathematical problems of translating the load spectrum statistics into useful reliability statistics are difficult.

If P(t) is the dynamic load imposed on the bearing, then at least three defect models can be envisioned:

(i) A brinell can occur the very first time P reaches a certain fixed limit, P_B.

(ii) A brinell can occur when the fraction of time for which $P>P_B$ is greater than some predetermined fraction, ε.

(iii) A brinell can occur due to an accumulation of damage. Each load excursion P causes a small but definite indentation which depends on the amplitude of the load. A condemnable defect occurs when the accumulation of these indentations reaches the brinelling limit.

The last defect model appears to most closely approximate the railroad bearing brinelling problem; we now consider this in greater detail.

In this model it is postulated that each load excursion or cycle of the random load P(t) produces an indentation which depends on the peak amplitude of the excursion. Each succeeding cycle inflicts additional damage and the brinell is said to be condemnable when the total damage reaches 100 percent. This model is described by Crandall and Mark (30) as it relates to fatigue.

For application to brinelling, we will assume that P(t) is a narrow-band process. This is not a poor assumption since Johnson's data (17-19) indicate that the frequency content of the railroad dynamic load environment is finite.

Following the model proposed by Crandall and Mark (30), one can ascribe an incremental damage to each cycle. Accumulating these damages leads to a total damage $\delta(t)$ for a time interval t. As t is increased, the damage $\delta(t)$ increases monotonically. At some time t_B the total damage reaches the size of a condemnable brinell and the bearing is condemned.

If we now consider an ensemble of load histories starting from an arbitrary origin in time as in Figure 42, the above procedure can be used to assign a value of failure time t_B to each sample of the ensemble. The failure times will vary randomly from sample to sample. To obtain a satisfactory statistical picture it would be desirable to know the probability density distribution $f(t_B)$ for the defect times. This distribution is, unfortunately, unknown. The distribution of brinelling damage is still unknown, but the central limit theorem can be invoked to show that the distribution tends toward the normal distribution as $t \to \infty$. Let ν_o^+ be the expected frequency of the narrow-band random load history P(t), i.e., the average number of zero crossings with positive slope per unit time. In time, t, then the expected number of "cycles" is $\nu_o^+ t$. The expected fraction of these cycles whose load amplitudes lie between P and P + dP is f(P)dP where f(P) is the probability density of the peaks. The expected number n(P) of such peaks is:

$$n(P) = \nu_o^+ t \quad f(P)dP. \tag{43}$$

The amount of brinelling damage due to a single load peak of amplitude P is from Equation (15):

$$\delta = \frac{kP^3}{d^2}. \tag{44}$$

If we multiply Equation (44) by the expected number of load cycles occurring between P and P + dP, we obtain the total damage due to all cycles having peaks between P and dP:

$$d\delta = \frac{kP^3}{d^2} \quad \nu_o^+ t \ f(P)dP. \tag{45}$$

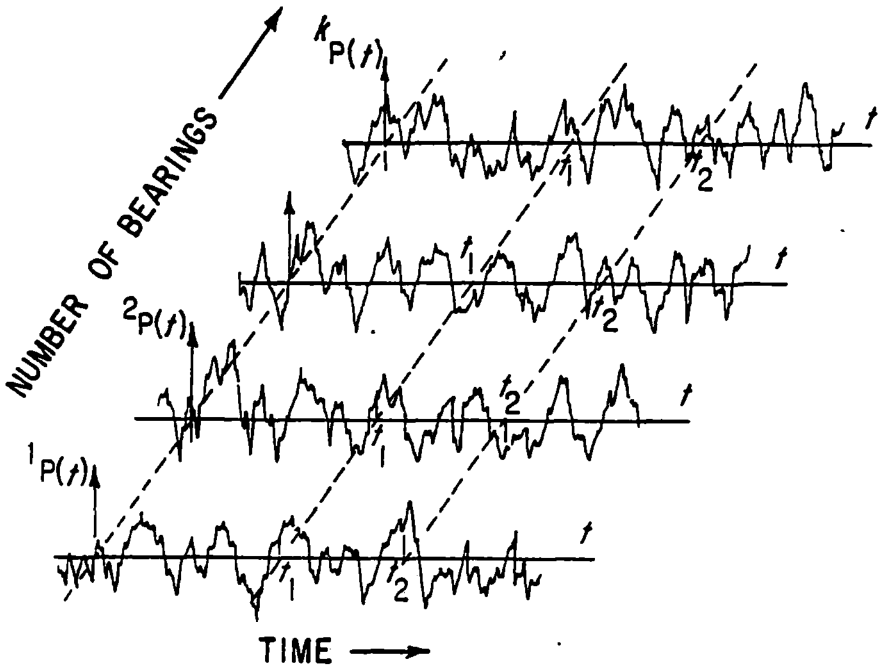


FIGURE 42. ENSEMBLE OF LOAD HISTORIES

The total expected brinelling damage is the sum of contributions like Equation (45) for all load excursions encountered. This sum is represented by the following:

$$\delta = \frac{\nu_o^+ tk}{d^2} \int_o^\infty P^3 f(P) dP \tag{46}$$

$$\delta = \frac{\nu_o^+ kt}{d^2} \quad 3/4 \sqrt{\quad} \tag{47}$$

Figure 43 shows qualitatively how the brinelling damage may accumulate with time. When the size of the brinell, δ, equals the condemnable size, δ_B, then the bearing is removed from service. We would expect the distribution of bearing ages about the mean to be distributed normally. Examination of data from Section 4 indicates that this conclusion is reasonably well verified by the experimental data.

This result indicates that the size of the accumulated brinell damage is linearly proportional to the expected number of cycles $\nu_o^+ t$; it also depends (nonlinearly) on the rms level σ_P of the load history, the geometry of the contact, and the constant k which appears in the brinelling law Equation (44) for the particular material involved. Thus, number of cycles and load appear to be the most likely acceleration factors.

5.4 ACCELERATION FACTORS - CONE BORE GROWTH

Of the possible phenomena responsible for cone bore growth discussed in Section 4.3.3.1, the plastic "ironing out" of the heavily loaded surface layers resulting in a system of residual stresses that expand the elastic "core" of the cone appears most likely.

The reasons for accumulation of plastic strain in rolling contact have been explored by a number of researchers. While some of the conventional plasticity theories are shown to account for some of the observed accumula-

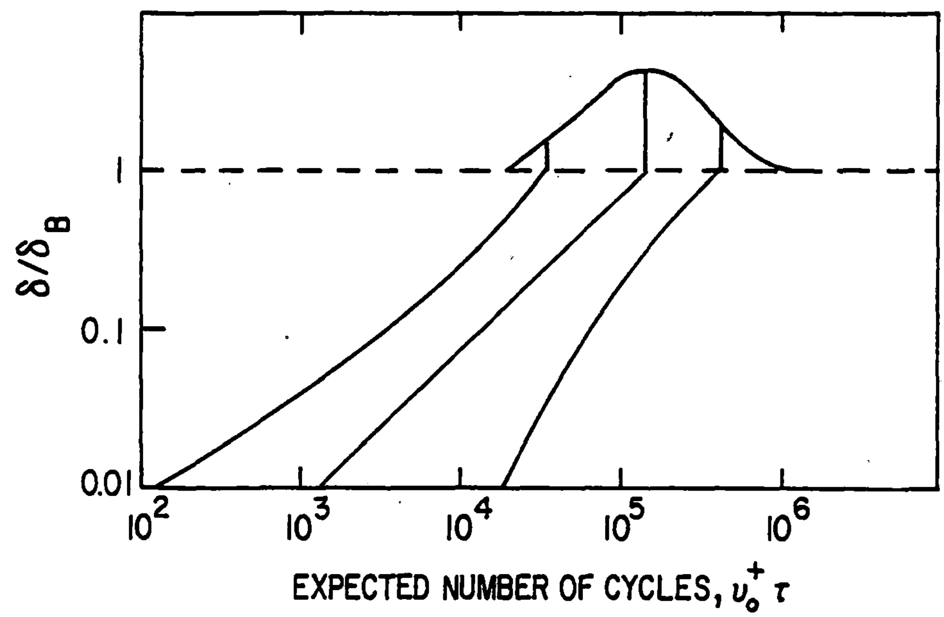


FIGURE 43. INCREASE IN BRINELLING DAMAGE WITH TIME

tion, it appears that the resistance to shear reversal in actual material is less than theoretically predicted. Nevertheless, even with the conventional perfectly plastic idealization, Johnson and Merwin (31) have demonstrated the cyclic operation of plastic "shakedown" or "incremental collapse" in rolling contact. A companion experimental paper by Hamilton (32) has shown good correlation.

Johnson and Merwin (31) and Hamilton (32) demonstrate the buildup of stress with repeated passage of the load. In the case where the maximum Hertzian compressive stress at the surface is about 3 times the conventional yield stress in simple tension (P_o = 5.5k), the high compressive residual hoop stresses penetrate to a depth approaching the width of contact and intensify with cycles. The width of the contact band between the roller and cone race is about0.010" for a 6 x 11 bearing at normal full static load.

The magnitude of plastic strain accumulation and depth of the plastically deformed layer required to produce bore growths of the magnitude observed have previously been evaluated in Section 4.3.3.1. From such an analysis it is seen that the accumulation of a maximum strain of only 0.2% in a layer 0.015" deep would account for a bore growth of 0.0005". It should also be noted that the residual hoop compressive stresses would be very high -- in excess of 150,000 psi. While these stresses are high, they must be viewed in terms of the high flow stresses required in this high hardness steel. Nevertheless, while reversed plastic strain can continue indefinitely, it appears clear that the continued accumulation of plastic hoop strain would diminish as the situation of reversed yielding due to residual stress magnitude is approached. Certainly brief periods of very high dynamic overload would accelerate this growth process but are not essential for some bore growth to occur. This analysis suggests that appropriate acceleration factors for cone bore growth are load and number of cycles.

6. DEMONSTRATION TESTING

The examination of the current railroad roller bearing acceptance procedure in Section 2 indicates that the dynamic test has a probability of accepting a poor quality bearing of more than 0.98. The current dynamic test can be made more stringent by increasing the number of test bearings and/or the number of test miles. It is further postulated in Section 5 that the total number of test machines and amount of test time to achieve a more favorable consumer risk could be minimized by resorting to accelerated testing. To demonstrate these ideas, accelerated life tests and tests with bearings having defective metallurgy were run.

In addition, a brinelling resistance test was devised and demonstrated. This latter test is proposed to offer the railroad industry more protection against a failure mode not currently covered under the present AAR Standard, D53-1971.

6.1 DESCRIPTION OF TEST APPARATUS

The test rig used to perform the accelerated life tests, tests with bearings having defective metallurgy and brinelling resistance tests (described in Sections 6.2, 6.3, and 6.5, respectively) is shown in Figure 44. Three bearing assemblies are mounted on the common shaft. The load is applied to the center bearing from above, through a standard railcar roller bearing adapter. The hydraulic cylinder to apply the load is mounted beneath the bearing on the underside of the bed of the machine with the plunger pointing down. Two large connecting rods, one on either side, carry the load to the top of the test bearing. A maximum of some 120,000 pounds of radial load can be applied. The shaft is belt-driven by a 30-hp, 440-volt motor. The support bearings are covered by sheet metal shrouds (not shown in Figure 44) into which outside air is forced to provide convective cooling.

Laboratory instrumentation utilized with the test rig included a twenty-four point temperature recorder, high frequency (50 kHz) accelerometers mounted on each bearing, and electrical contact resistance across the bearings. Automatic shutdown protection is incorporated and is triggered by high temperature, vibration, and motor current.

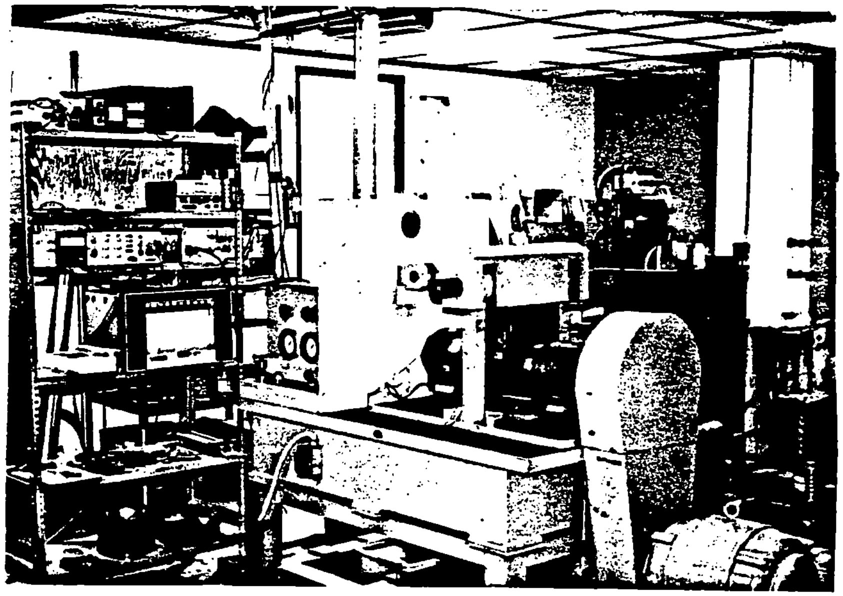


FIGURE 44. ROLLER BEARING TEST RIG FOR FAILURE PROGRESSION AND CERTIFICATION DEMONSTRATION TESTING

6.2 ACCELERATED TEST DEMONSTRATION

It has previously been shown that either larger sample sizes or longer test times would significantly reduce the railroads' risk in bearing certification. It was further suggested that accelerated testing at twice the rated load could reduce test times by a factor of ten, thereby reducing the cost of a more extensive certification procedure.

To illustrate this, the seven test bearings shown in Table 31 were run at twice the rated laod until the diagnostic instrumentation described in Section 6.1 indicated the presence of a defect. At this point the test was terminated.

As can be seen in Table 31, all of the bearings contained defects but not all of them were condemnable under AAR rules. In the subsequent analysis those bearings with condemnable defects were treated as failures and the remaining bearings as discontinued tests.

Using the same method described in Section 4.4.2.2, Table 32 presents the hazard table for the accelerated test failures. The table consists of 3 failure times for the bearings containing condemnable defects and 4 censoring times for the remaining bearing. The data have been ordered from smallest to largest without regard to whether they are censoring times or failure times. In the list of ordered times, the failures are each marked with an asterisk to distinguish them from the censoring times as discussed earlier.

The hazard value $h(x)$ for a failure time is the inverse of the number K units with a failure or censoring time greater than (or equal to) that failure time. The K value is given in parentheses next to the unit number. The cumulative hazard, $H(x)$, is the cumulative sum of all failure time preceding and including $h(x)$. Values for β and α are shown in Table 33. Failure time has been plotted against its corresponding cumulative distribution function in Figure 45.

TABLE 31

TEST RESULTS

(ACCELERATED BEARING TESTS)

Brg.	Test	Vendor	Speed RPM	Load Pounds	Ave Brg. Temp	Duration Hours	Equivalent* Miles at Full Load	Defect Description
102	3,5,7	A	672	52,000	250	306.8	197,599	#2 Cone Spalled (c) #2 Cup Fragment Indentation
101	2,4,6	A	672	52,000	267	299.4	192,833	Minor Fragment Indentation
204	C2A	A	672	52,000	230	207.2	133,450	Roller Spalled (c)
204	C4A	A	672	52,000	265	207.2	133,450	Roller Spalled (c)
207	C3A	A	672	52,000	222	200.0	128,813	Slightly Spalled Roller
212	C5A	B	672	52,000	250	192.9	124,240	Spalled Cone Cage Binds (c)
214	C4B	B	672	52,000	272	70.3	45,278	None
200	C1A	A	935	52,000	285	26.7	23,927	Pinpoint Roller Scale Pits

*Assumes 33 inch wheel

(c) — Condemnable

TABLE 32

HAZARD DATA FOR ACCELERATED LIFE DEMONSTRATION TEST

No.	Equivalent Miles at Full Load	Equivalent Years of Service	h Hazard	H Cumulative Hazard	Cumulative Distribution Function
1 (7)	23,927	2.04	-	-	-
2 (6)	45,278	3.86	-	-	-
3 (5)	124,240*	10.58	0.2	0.2	0.18
4 (4)	128,813	10.97	-	-	-
5 (3)	133,450*	11.37	0.333	0.5333	0.41
6 (2)	192,833	16.42	-	-	-
7 (1)	197,599*	16.83	1.0	1.5333	0.78

*Denotes Failure

TABLE 33

WEIBULL PARAMETERS
FOR ACCELERATED LIFE DEMONSTRATION TESTS

	Accelerated Test Data		Defect Data
	Equivalent Miles at Full Load	Equivalent Years of Service	Equivalent Years of Service
B_{10}	101,751	8.67	11.15
η	170,602	14.53	39.29
β	4.35	4.35	1.78

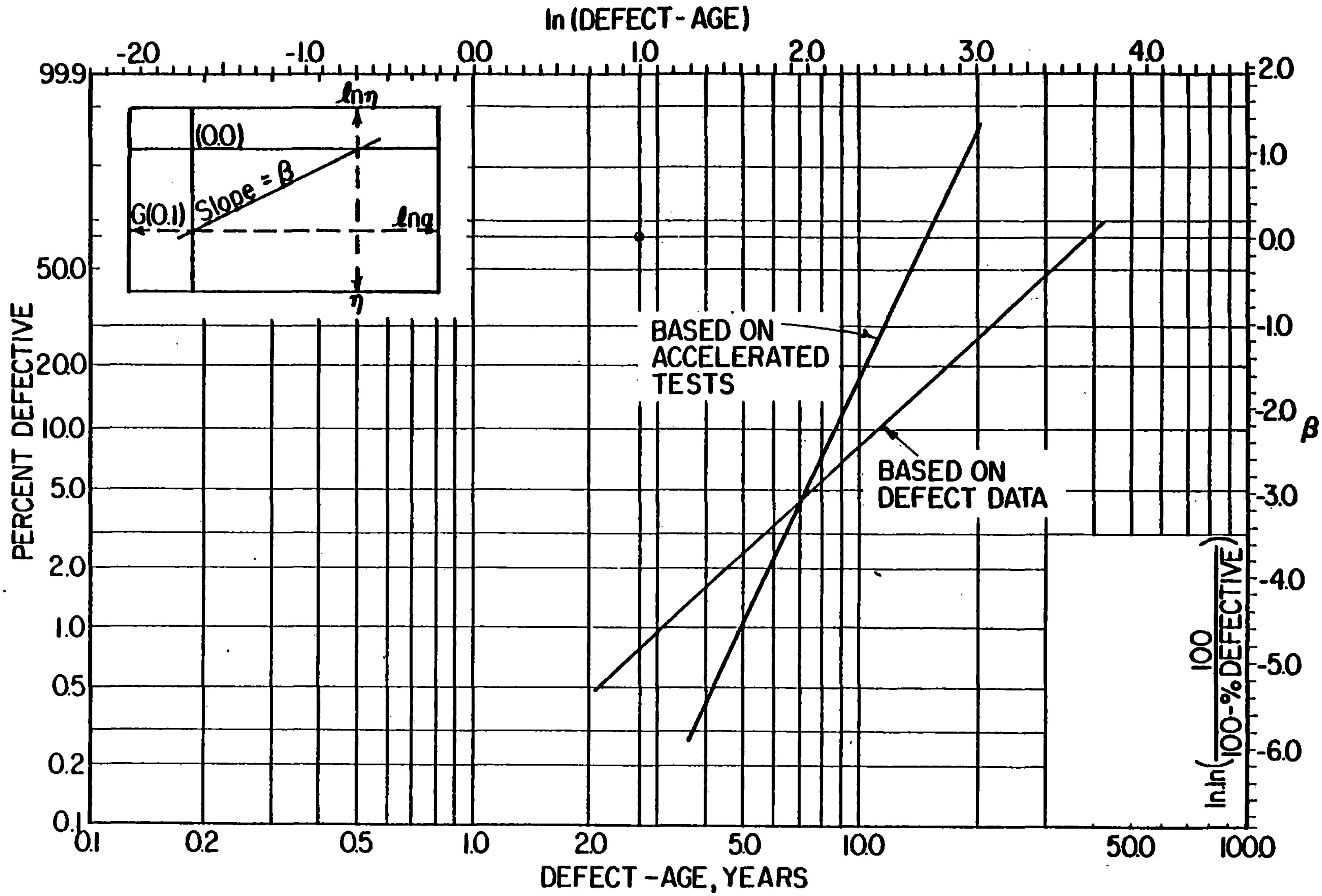


FIGURE 45. CUMULATIVE PERCENT DEFECTIVE - COMPARISON BETWEEN ACCELERATED TEST AND DEFECT DATA

6.2.1 Elastohydrodynamic Film Effects on Fatigue Life

As discussed earlier in Section 4.3.1, the magnitude of the lubricant film thickness in the bearing has an effect on the fatigue life of the bearing. The criterion that determines the effect of film thickness on fatigue failure appears to be closely related to the elastohydrodynamic film thickness/roughness ratio Λ defined as h/σ.

If one plots statistical life (L_{10}) of many homogeneous groups of rolling elements operated at varying Λ values, but at identical load, a curve of the type shown in Figure 46 results. The numerical values on the abscissa and ordinate vary, depending on the contact configuration, the chemistry of the lubricant, and the material of the contacts. In all cases, however, the curve has a "knee" as shown in Figure 46 around $\Lambda \simeq 1$. Above this point, at least up to $\Lambda = 4$, life increases gradually with speed. At some value below $\Lambda \simeq 1$ (depending on lubricant chemistry and probably on the contact material), there is an abrupt drop in spalling fatigue life.

Acceleration of fatigue testing by increasing load alone will force operation at a lower Λ. This will produce an erroneously low life. Similarly if speed is increased at constant load, the specific film will be increased, thus prolonging the L_{10} life. To provide equivalent conditions, the specific film Λ should be maintained constant between tests.

For a tapered roller bearing, the Dowson-Higginson formula,

$$\Lambda = \frac{h}{\sigma} = \frac{1.6^{\;0.6}\ (\bar{\eta}U)^{0.7} E'^{\,0.03}\ R^{0.43}}{\sigma Q^{0.13}}\ , \qquad (48)$$

where:

Λ = specific film thickness = $\frac{h}{\sigma}$

h = lubricant film thickness (in.)

$\bar{\eta}$ = viscosity $\left(\frac{\text{lb-sec}}{\text{in}^2}\right)$

U = rolling speed (in./sec.)

E' = $1/2 \left[\frac{1-\nu_1^{\,2}}{E_1}\right] + \left[\frac{1-\nu_2^{\,2}}{E_2}\right]$ (in^2/lb)

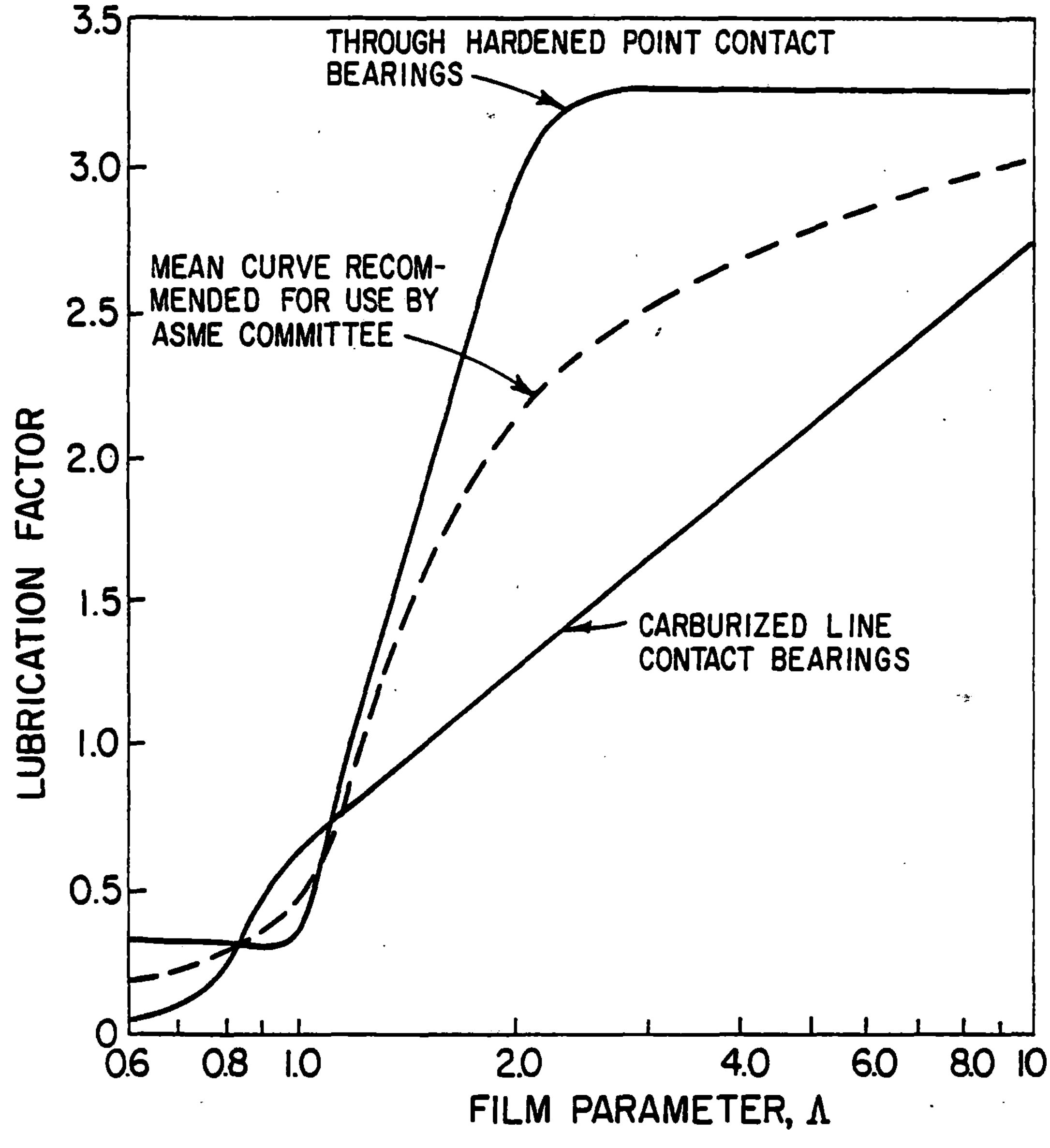


FIGURE 46. LUBRICATION FACTOR F AS A FUNCTION OF FILM PARAMETER Λ (AFTER REFERENCE 12)

R = roller radius (in.)

σ = surface roughness (in.), and

Q = roller load (lb.)

provides a simple scaling factor.

For identical bearings operating on the same lubricant with the same surface roughness, the scaling law reduces to:

$$\frac{\Lambda_1}{\Lambda_2} = \frac{\left[\frac{(\eta N)^{0.7}}{P^{0.13}}\right]_1}{\left[\frac{(\eta N)^{0.7}}{P^{0.13}}\right]_2} \; . \qquad (49)$$

Table 34 presents a comparison between the average test conditions and the average service conditions. The value of the specific film thickness is considerably lower than that experienced in service. However, both values of Λ are below 0.6. The effect on life below $\Lambda = 0.6$ appears to be independent of Λ (see Figure 45) and we would therefore not expect to see a lubrication effect on life between test and service.

Although the accelerated tests came reasonably close in predicting the bearing L_{10} fatigue life, the difference between the values of β deduced from the defect data and the accelerated test data was excessively large. There may be several reasons for this.

- The sample size for the accelerated test data was relatively small (3 failures) and the difference may be accounted for by the statistical scatter.
- The test load may have been above the range where the model describing the effect of load on life distribution (Equation (35)) was no longer valid.
- A more likely explanation may be the result of more than one active failure mode, each of which may be described by a separate model. Moreover, the life distribution and the load dependence may

TABLE 34

COMPARISON BETWEEN

TEST AND APPLICATION DATA

	Service	Test
Average Train Speed (mph)	27.7	65
Average Rotational Velocity (rpm)	285	672
Average Load (lbs)	26,250	52,500
Average Temperature (^oF)	125^oF	250^oF
Lubricant Viscosity (cs)	52	7.2
Λ	0.4	0.17

$$\frac{\Lambda_1}{\Lambda_2} = \frac{\left[\dfrac{(672 \times 7.2)^{0.7}}{52{,}500^{0.13}}\right]_1}{\left[\dfrac{(285 \times 52)^{0.7}}{26{,}250^{0.13}}\right]_2} = .42$$

not be adequately described by any one model. The condition of the test bearings suggests that this may have been the case since the roller tracks were glazed. Some lubricant degradation may have resulted from the relatively high test temperatures (Table 31) and contributed to a shorter than expected L_{10} life and larger than expected Weibull slope.

6.2.2 Ferrographic Analysis of Accelerated Test Grease

Following each of the certification demonstration tests, samples of grease were collected from five different locations from each bearing. These locations were:

1. Behind grease seal -- A side
2. Cage surface -- A side
3. Spacer area
4. Cage surface -- B side
5. Behind grease seal -- B side.

Five of these samples were subjected to ferrographic analysis by the Naval Air Engineering Center (NAVAIRENGCEN) to evaluate the feasibility of relating the quantity and nature of wear debris to the condition of a bearing.

The pertinent conditions associated with each of the five selected samples are summarized in the following table:

Sample No.	Bearing No.	Sample Location	From Test No.	Description of Damage
38	203	3	C2A	Moderately Spalled Cone
58	209	3	C4A	Spalled Roller
63	212	3	C5A	Heavily Spalled Cone (see Reference 5)
64	212	4	C5A	
65	212	5	C5A	

Table 35 summarizes the results. The following comments summarize the observations made during the analysis.

Sample 38. Sample contained large quantities of black oxide and carbon. Temper coloration was noted on various ferrous metallic particles, indicating high operating temperatures. The amount of particles indicate that a component was in an abnormal wear mode.

Sample 58. Sample contained large quantities of black oxide and carbon. Nonferrous metallic particles were also detected in slightly greater number than in other samples observed. Slight temper coloration was also observed on ferrous metallic particles.

Sample 63. Sample contained large quantities of carbon and ferrous metallic spheres, which can be attributed to rolling contact fatigue. Slight coloration was observed on the ferrous metallic particles. Ferrous metallic laminar particles were of sufficient quantity to indicate roller contact fatigue.

Sample 64. Sample as received was extremely deteriorated, indicating high temperatures in the sample area. Large amounts of black oxides, carbon and polymer were observed, with temper coloration noted on the larger ferrous metallic particles. The amount of large ferrous particles compared to small ferrous particles also indicates a severe were situation was occurring.

Sample 65. Sample contained an extremely large quantity of particles, both ferrous and nonferrous, approximately 4 times the amount observed in the other samples. Large quantities of carbon, friction polymer and oxide spheres were present, with large ferrous metallic particles similar to those observed in Sample 64 also present. Temper coloration was very evident on ferrous metallic particles.

TABLE 35

SUMMARY OF FERROGRAPHIC ANALYSIS

WEAR MODE CHARACTERISTIC	Sample Number: / Sample Location:	38 / 1	58 / 2	63 / 3	4	63 / 5	64 / 6	65 / 7
Beiby Particles		M	M	M		M	M	H
Fatigue Chunks (typical gear surface fatigue)		F	F	F		F	F	M
Laminar Particles (gears or rolling bearings)		M	M	M		M	M	H
Cutting Wear Particles (high unit pressure)		F	F	F		F	F	F
Spheres (fatigue cracks in rolling bearings)		F	F	H		H	F	M
Corrosive Wear Particles		F	F	M		M	N	N
Oxide Particles (includes rust)		M	F	M		M	F	F
Dark Metallic-oxide Particles (typical hard steels)		M	H	M		M	M	H
Severe Wear Particles		F	M	M		M	M	M
Nonferrous Metallic		F	M	F		F	M	M
Nonmetallic, Crystalline		M	M	M		M	H	H
Nonmetallic, Amorphous (i.e. friction polymer)		M	M	M		M	M	M
DENSITOMETER DATA (TYPE 7056) % AREA COVERED								
Reading at 54 mm		11	13	8		8	9	58
Reading at 50 mm		3	3	3		3	1	48
Reading at 10 mm		2	4	1		1	.9	14

NONE N   FEW F   MODERATE M   HEAVY H

Based upon the work performed on the railroad roller bearing grease samples as well as work performed under the NAVAIRENGCEN Wear Particle Analysis Program, the following preliminary conclusions and recommendations have been made:

1. It is feasible to identify a severe state of wear in railroad roller bearings using grease analysis techniques.

2. Prime indicators of railroad roller bearings wear are:
   Elemental Analysis
   Particle Size Distribution
   Particle Morphology.

3. Particle morphology appears to be a good indicator of railroad roller bearing wear. Particles peculiar to roller contact fatigue were found in sufficient quantities to facilitate an acceptable analysis.

4. Total particle count exhibited wide variations in reflecting wear state relative to the actual bearing condition. This may be attributed to sensitivity to sample technique and grease conditions. More samples will need to be analyzed to establish a realistic sensitivity trend.

5. Trend analysis (a series of samples from one railroad roller bearing from beginning of test to failure) as opposed to individual sample analysis (one sample from the failed bearing) as utilized in this effort, would be the best approach in the development of a grease analysis correlation effort.

6. A blue temper coloration appearing on ferrous metallic wear particles in several of the samples, coupled with the presence of black and red oxides, indicates that these samples have been subjected to extreme temperatures. Observations of some grease breakdown confirms this indication.

7. The seal cavity locations (locations 1 and 5) appear to be the best location for collecting grease samples.

6.3 TESTS OF METALLURGICALLY DEFECTIVE BEARINGS

To further demonstrate the discriminating power of a laboratory acceptance test, bearings of "poor" metallurgical quality shown in Table 36 were subjected to extended life testing under full load conditions of 26,000 pounds. It was intended that this group of bearings would possess all the externally measurable qualities of an acceptable bearing (size, tolerance, surface finish, and hardness) but would have inferior metallurgical properties that would likely lead to premature fatigue failure.

This group of metallurgically defective bearings consisted of two lots. Both lots incorporated new rollers, new cages, and cups manufactured prior to 1966 made from unmodified AISI (which were known to be inferior from the standpoints of brinelling resistance and wear). The cups were reground to "new" surface finish and taper tolerances.

The first lot of bearings had cones produced from AISI 1050 steel, not bearing quality. After machining, these new races were heated to 1600°F and oil-quenched. After grinding, the hardness of the end faces of the cones was Rc 50-55. The surface hardness of the rolltrack was Rc 50 and the hardness dropped to Rc 40 at 0.010 inches below the rolltrack surface. Since these cones did not possess the AAR required hardness of Rc 58, a second lot was produced. However, four of this lot of bearings were subjected to testing while awaiting the manufacture of the second lot. These bearings were designated 201, 202, 203, and 205.

The second lot of bearings had cones produced from AISI 1040 steel, not bearing quality. After machining, these races were carburized to a depth of 0.080 inches, reheated to 1600°F and oil quenched. After grinding, the hardness on the end faces of the cones was Rc 61-62. The surface hardness of the rolltrack was Rc 61, and was Rc 50 to a depth of 0.025 inches. Due to the poor hardening characteristics of the steel, the case was not fully developed

TABLE 36

TESTS OF METALLURGICALLY DEFICIENT BEARINGS

Brg.	Test	Cone Material AISI	Speed RPM	Load Pounds	Ave. Brg. Temp.	Duration Hours	Equivalent* Miles at Full Load	Defect Description
201	C1A	1050	935	26,250	285	26.7	2,451	Spalled Cones (c)
202	C1A	1050	935	26,250	235	26.7	2,451	Spalled & Seamed Cones (c)
203	C2A	1050	672	26,250	195	130.2	8,589	Spalled Cones (c), Rollers with Scale Pits
205	C2A	1050	672	26,250	215	130.2	8,589	Spalled Cones & Rollers (c)
206	C3A	1040	672	26,250	185	200.0	13,194	One Roller with Scale Pits
208	C3A	1040	672	26,250	185	200.0	13,194	Slightly Spalled Rollers, 1 Cup Slightly Spalled
209	C4A C4B	1040	672	26,250	200	147.3	9,717	Slightly Spalled Rollers, Cups Slightly Spalled
210	C4A C4B	1040	672	26,250	210	147.3	9,717	Scale Pits on 4 Rollers, 1 Cone Spalled
211	C5A	1040	672	26,250	180	192.9	12,726	2 Rollers with 20-30 Scale Pits
213	C5A	1040	672	26,250	200	192.9	12,726	Scale Pits on 1 Roll, 1 Cone Spalled

*Assumes 33 inch wheel
(c) Condemnable defect

at the back rib. Consequently, surface hardness of the rolltrack immediately adjacent to the rib was Rc 54. Six of these bearings (designated 206, 208, 209, 210, 211 and 213) were subjected to extended life testing.

6.3.1 AISI 1050 Tests

The bearings made from AISI 1050 would have all been rejected during the AAR laboratory inspection since their cone hardness was below the required 58 Rockwell C. Despite the easily detected cone softness the bearings, all ran more than the 2050 miles required by the AAR dynamic test before a condemnable defect was detected.

6.3.2 AISI 1040 Tests

This batch of bearings did exhibit externally acceptable properties and would most likely have passed the laboratory inspection. Moreover, all passed the dynamic test by a wide margin. In fact, all were removed from test before generating condemnable defects.

These tests tend to reinforce the conclusion that the present dynamic test does not offer sufficient discriminating power to differentiate between good bearings and bearings with known metallurgical defects.

6.4 BRINELLING TEST DEMONSTRATION

The current laboratory inspection does not include a brinelling test as such. It does call for a shock test, which requires dropping the bearing ring a vertical distance of 4 feet to strike on edge on a mild steel plate having dimensions of 1/2 x 15 x 15 inches. Any splitting, cracking, chipping, or significant deformation constitutes a failure. This is a form of a brinelling test; however, the specification does not define "significant deformation."

In order to define the brinelling resistance distribution of the railroad roller bearing, the test bearings shown in Table 37 were subjected to brinelling loads. Two groups of bearings were tested, one group 10 years old and another new group. Each group contained three bearings from each of the three major domestic manufacturers, for a total of nine bearings.

The test bearings with all but four rollers removed in each were loaded to 123,200 pounds in the test rig shown in Figure 44. For each bearing cup and cone, 16 indentations were made. This entire load is taken by one roller in each row so that the maximum roller load is:

$$Q_{max} = \frac{123,200}{2} \times \cos\alpha_c = 60,644 \text{ pounds.} \quad (50)$$

The equivalent radial load required to achieve this maximum individual roller load in a full complement bearing can be calculated from the approximate Equation (11):

$$P = \frac{iZ\,Q_{max}}{5\cos\alpha_c} = \frac{2 \times 24 \times 60,664}{5 \times 0.9848}; \quad (51)$$

thus

$$P = 591,360 \text{ lbs}$$

As seen from Figure 20 and Table 23, at this load one would expect a permanent indentation of approximately 3 mils.

The mean of the measured indentation depths measured is shown in Figure 47 and, as can be seen, the measured depth is considerably less than the estimates based on the AFBMA criterion. This means that the AFBMA method of calculating the static load capacity of tapered roller bearings is conservative.

Like the results of Wickstrand (16), the scatter found in the measured data was extremely large. This is illustrated in Table 38 where the mean value, the standard deviation, and the variance are shown for each set of test bearings.

TABLE 37

BRINELLING TEST BEARINGS

BRG. NO.	MANUFACTURER	AGE Cup	AGE Cone A	AGE Cone B
300	A	1965	1966	1966
301	A	1966	1966	1965
304	A	1966	1966	1966
303	B	1965	1965	1964
306	B	1964	1968	1963
312	B	1972	1972	1972
307	C	1965	1965	1965
311	C	1965	1965	1965
313	C	1965	1965	1965
302	A	1976	1976	1976
309	A	1976	1976	1976
310	A	1976	1976	1976
305	B	1975	1975	1975
308	B	1975	1975	1975
317	B	1975	1975	1975
314	C	1975	1976	1976
315	C	1975	1976	1976
316	C	1975	1976	1976

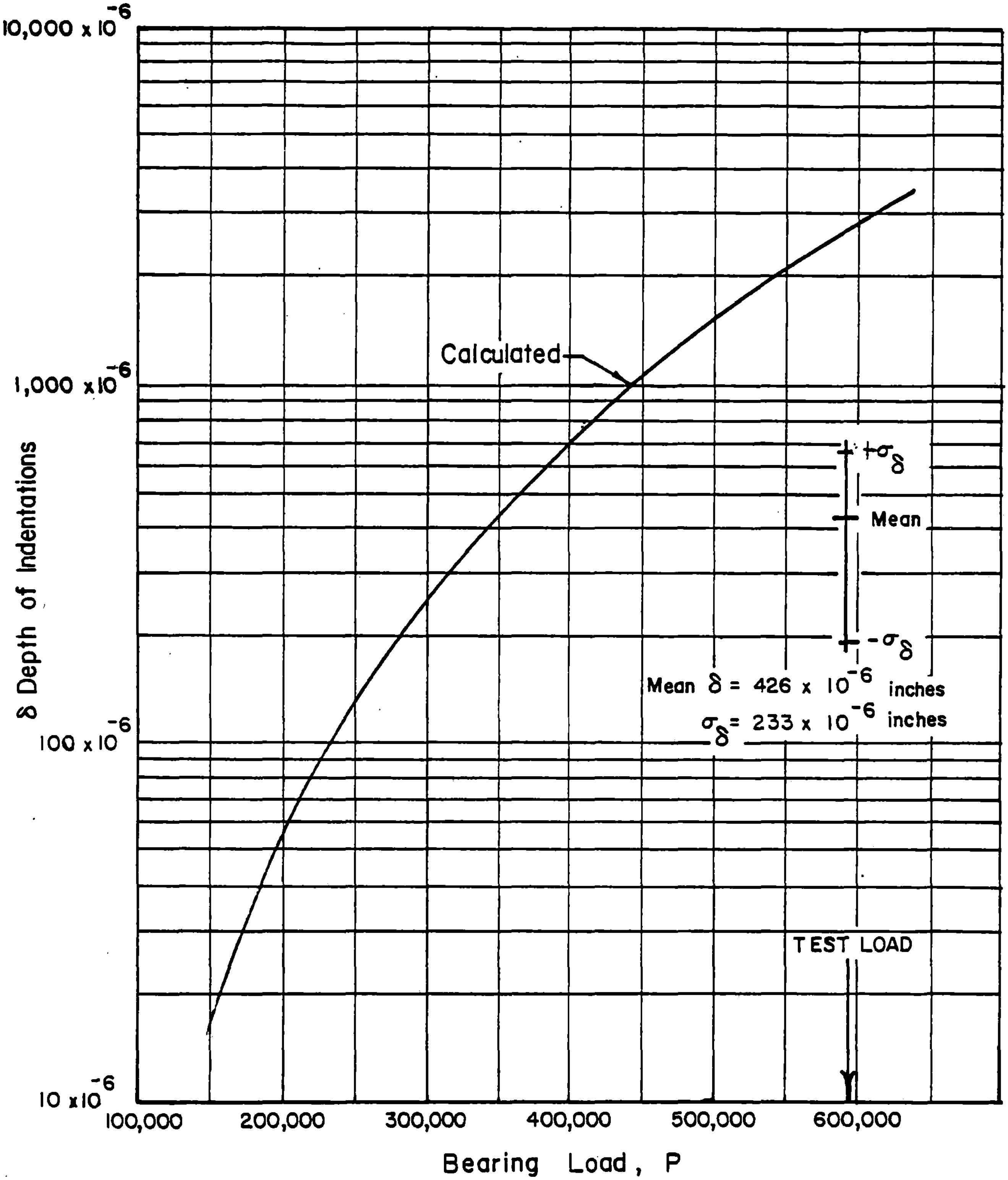


FIGURE 47. COMPARISON BETWEEN MEASURED BRINELLING DEPTH AND AFBMA CRITERION - 6 x 11 BEARING

TABLE 38

MEAN INDENTATION OF SAMPLE

BEARINGS — MICROINCHES

CONES									
AGE	VENDOR A			VENDOR B			VENDOR C		
	Mean μin	Std Dev μin	Variance μin^2	Mean μin	Std Dev μin	Variance μin^2	Mean μin	Std Dev μin	Variance μin^2
OLD	648	327	107,465	429	60	3,651	460	87	7,594
NEW	561	67	4,514	145	127	16,379	210	39	1,486

CUPS									
AGE	VENDOR A			VENDOR B			VENDOR C		
	Mean μin	Std Dev μin	Variance μin^2	Mean μin	Std Dev μin	Variance μin^2	Mean μin	Std Dev μin	Variance μin^2
OLD	551	186	34,759	456	19	365	435	270	72,870
NEW	331	334	111,915	74	133	17,578	188	43	1,873

The wide scatter found in the brinelling resistance of supposedly identical bearings will have a significant influence on our recommended certification procedures discussed later in Section 7.1.

Relationship Between Brinelling and Cone Bore Growth

The brinelling process is a measure of the plastic strength of the bearing material, and is analogous to an indentation hardness test. Consider the cylinder testing on a flat surface shown in Figure 48a. If the surfaces are pressed together with a load Q, they will at first deform elastically according to Hertz' classical equations. At this stage, the area of contact $A = \ell b$ will be proportional to $Q^{2/3}$, while the mean pressure over the area of contact will be $p_m = Q^{1/3}$. The way in which A and p_m vary with Q is shown in Figure 48b.

As the load Q is increased, the mean pressure P_m increases until it reaches a value such that at a critical point within the softer material the elastic limit is exceeded. This occurs at the region where the shear stresses are at a maximum. The Hertizian analysis shows that this region is situated at a point z about 0.5b below the center of the contact area. The elastic limit is just exceeded at this point when:

$$p_m = 1.1 \sigma_y, \tag{52}$$

where σ_y is the elastic limit of the softer metal as found in pure tension (or frictionless compression) experiments. At this stage the metal around z (Figure 49a) is plastic and yields irreversibly. The material outside this region has not yet reached the conditions for plasticity and its deformation is still essentially elastic. Consequently, when the load is removed only a very slight amount of residual deformation remains.

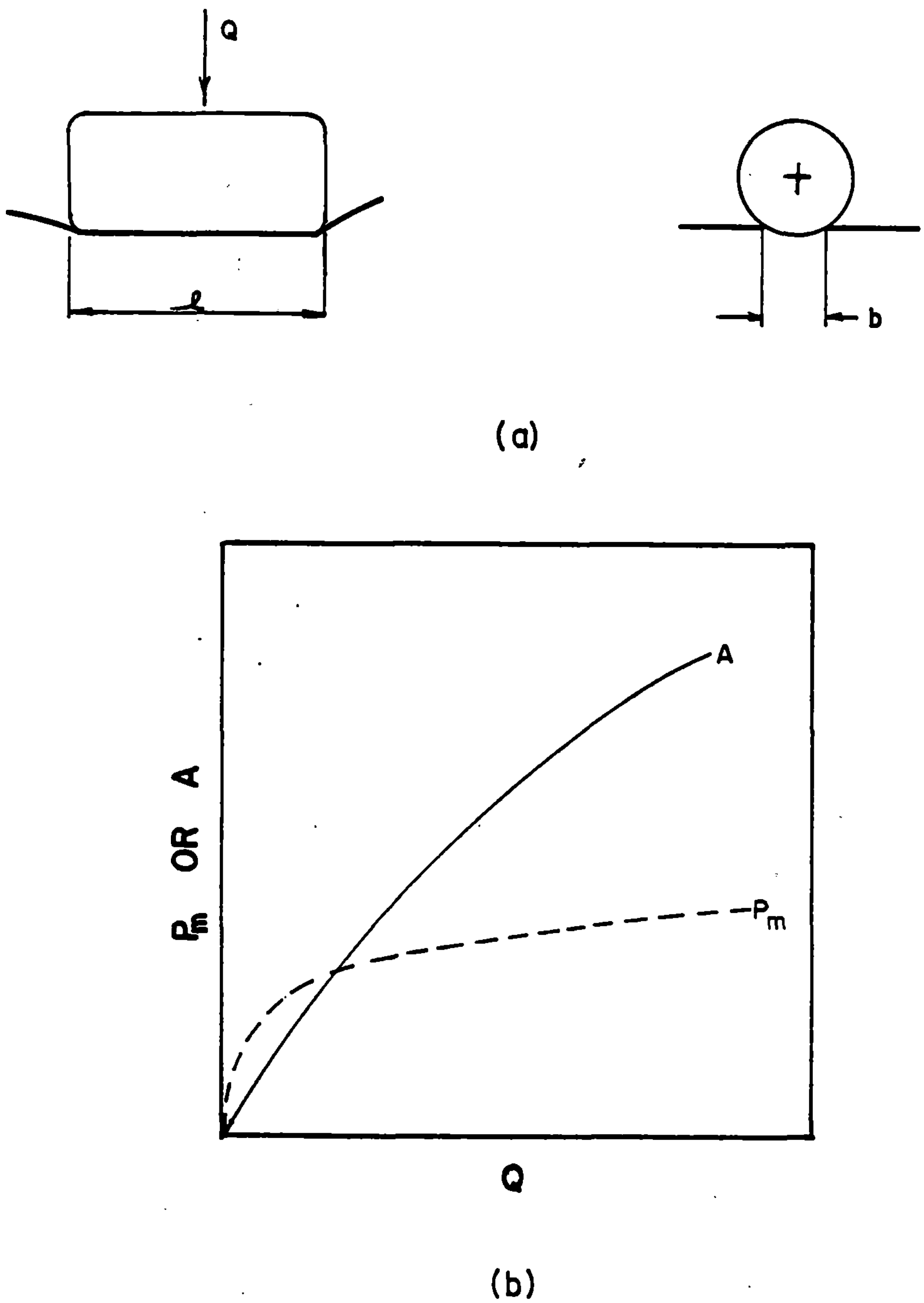


FIGURE 48. RELATION BETWEEN CONTACT AREA AND FLOW PRESSURE

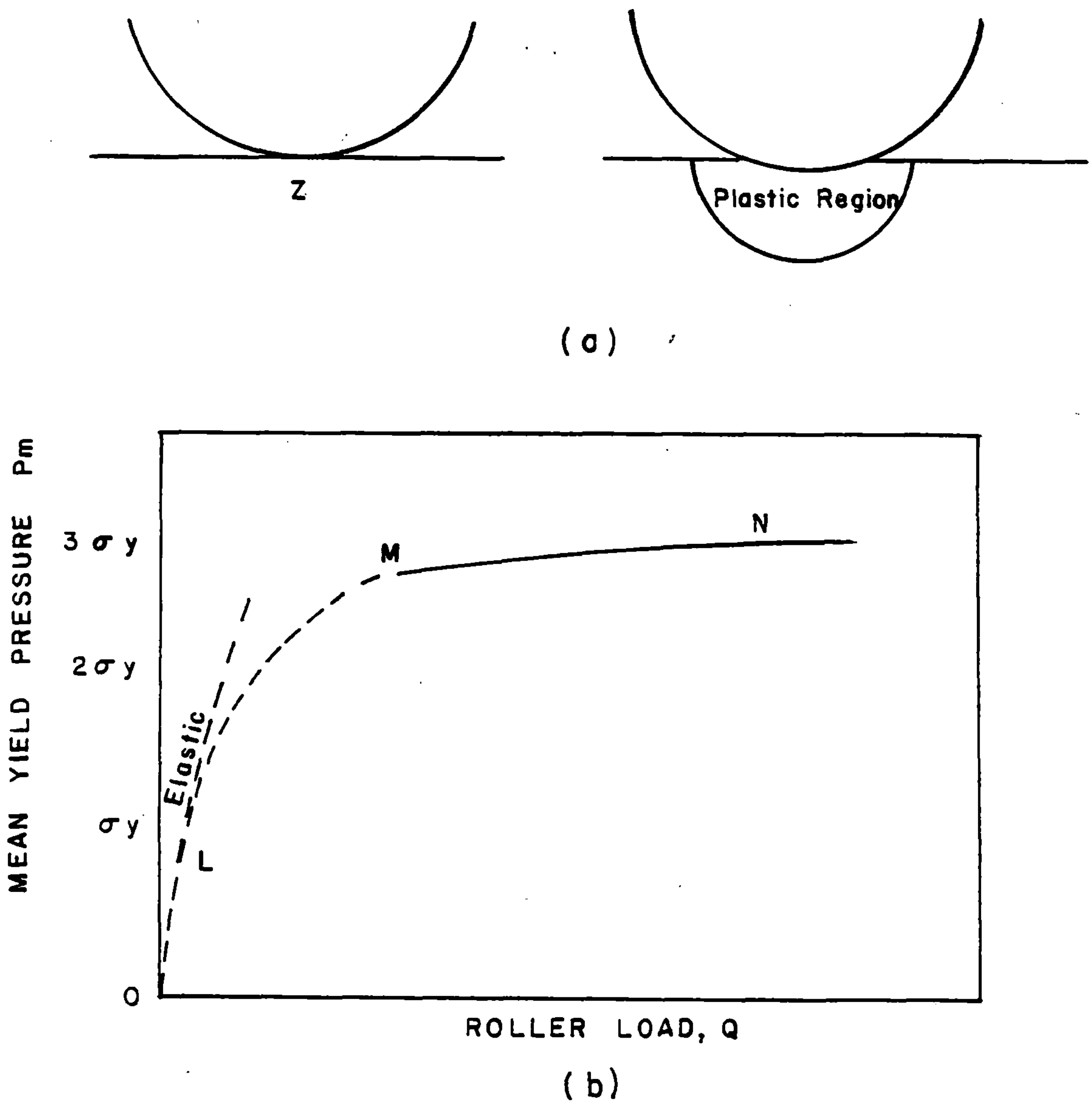


FIGURE 49. SCHEMATIC REPRESENTATION OF BRINELLING CHARACTERISTICS

If the load is now increased further, the area of contact A and the mean pressure p_m rise. The region of plasticity around z grows rapidly, and a stage is soon reached at which the whole of the material around

$$p_m = c\sigma_y \tag{53}$$

where c has a value of approximately 3.

If the load is still further increased it is found that although the size of the deformed area increases, Equation (51) is still valid, provided that the deformed area is not too large compared with the size of the specimens and that the elastic limit σ_y does not increase as a result of the plastic deformation produced - i.e., provided there is no work hardening. In practice, of course, it is impossible to find a metal that does not work harden.

Referring to Figure 49b, we may now describe graphically the variation of p_m with load for materials which do not work harden. The portion OL represents the increase of p_m with Q over the purely elastic range where the deformation is completely reversible. At the point L, where p_m reaches a value of about $1.1\sigma_y$, the onset of plastic deformation commences. There is a gradual increase in p_m and a value is reached at about $p_m = 2.8\sigma_y$ where "full" plasticity occurs. The mean pressure is now more or less independent of the force Q and follows the curve MN.

If the deformation beyond L still followed the elastic equations, the load at which p_m becomes equal to $2.8\sigma_y$ would be $(2.8/1.1)^3$, i.e., about 16 times the load at L. That is to say that full plasticity would be reached at a load about 16 times that at which the onset of plasticity occurs.

The preceding discussion is confined to materials that do not work harden. If the metal is capable of work hardening, the formation of this indentation itself will produce an increase of the elastic limit σ_y. Theoretical considerations and practical measurements show that the elastic limit around the indentation will not be constant but will vary from point to point. Nevertheless, we may assume an average representative value of the elastic limit which is related to the mean pressure p_m by a relation of the same type as Equation (51). Experimental investigation shows that, in fact, the elastic limit σ_{ye} at the edge of the indentation may be used for this purpose. If p_m is compared with σ_{ye} we find that over a wide range of indentation sizes p_m equals $c\sigma_{ye}$, where c has a value lying between 2.7 and 3. Further, if b is the width of the indentation and d the diameter of the cylinder, the depth of the indentation is completely defined by the dimensionless ratio b/d, and it is found that the strain is approximately proportional to the depth of indentation or:

$$\varepsilon = c\,\frac{\delta}{d}. \tag{54}$$

Combining this with Equation (22):

$$\Delta D = \frac{c\,(D_2^2 - D_1^2)\,D\delta}{(D_2^2 - 1)d}, \tag{55}$$

i.e., the cone bore growth should vary approximately linearly with the depth of indentation.

Referring to Figure 50, one can see that this is indeed the case. This result suggests that the brinelling resistance can also be used as a measure of cone bore growth resistance.

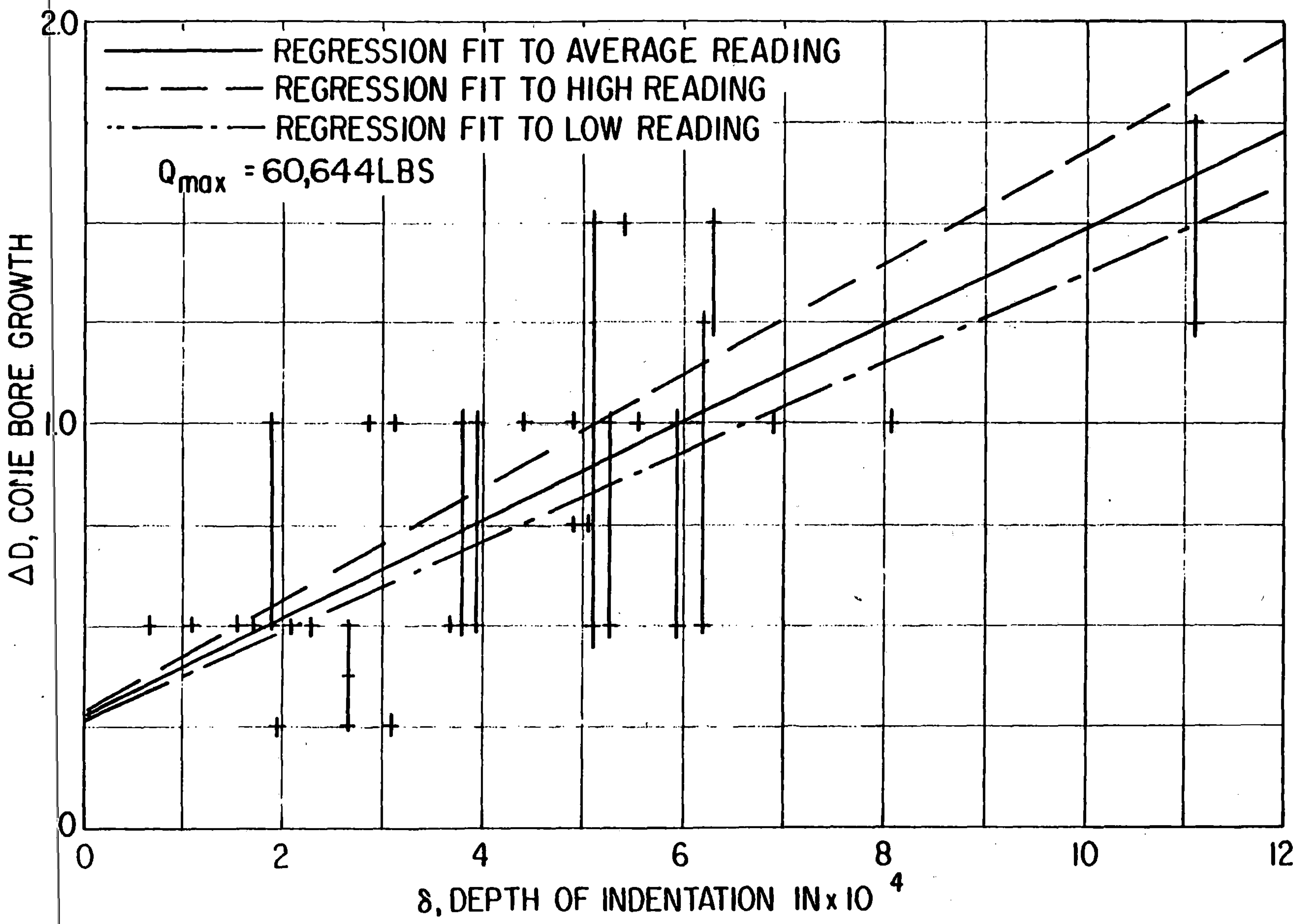


FIGURE 50. CORRELATION BETWEEN CONE BORE GROWTH AND BRINELLING RESISTANCE

7. RECOMMENDED CERTIFICATION PROCEDURE

The preceding sections have described the concept of quantifying consumers' and producers' risks based upon attributes tests. It has been shown both analytically and experimentally that the present certification procedure is biased in favor of producer and that significant improvements in consumer protection could be achieved by making the procedure more stringent.

There are, in general, two areas in which the procedure could be modified to provide a greater degree of protection to the consumer. The first involves adding a test to measure brinelling resistance and the second involves adding a larger number of test bearings and running them for a greater number of miles.

7.1 BRINELLING TEST

In addition to the shock test currently used by the AAR, it is suggested that at least three of the candidate bearings be subjected to a brinelling test similar to that described in Section 6.4. Sixteen impressions should be made in each bearing under an equivalent bearing load of 600,000 lbs. No one loading would be allowed to generate a condemnable brinell per AAR specification (Reference 4).

The cone bore resulting from these repeated loadings should also be measured and no growth allowed to exceed 0.0015 inches.

The addition of this test would provide some degree of protection against the two most prevalent failure modes which are not addressed by the present specification.

7.2 GENERAL CERTIFICATION PLAN

A signicant improvement in consumer protection can be achieved by making the general certification plan more stringent. A two-stage test plan similar in concept to the current AAR certification procedure is suggested. These two stages consist of a laboratory test stage and a field test stage following successful completion of the laboratory test. These stages are shown schematically in Figure 51.

7.2.1 Effect of Stage Variables

The generalized certification procedure was first studied by artificially limiting the procedure to the first stage (normally, the laboratory dynamic test stage). To do this, the approval number n_{a1} and rejection number n_{r1} were selected so that $n_{r1} - n_{a1} = 1$; i.e., either approval or rejection is decided in the first stage. In addition, n_{a1} was selected to be 0 (no failures allowed). This value of n_{a1} represents the most consumer-protective procedure and was used so that the characteristics of the procedure in protecting the consumer could be observed.

The results of the analysis are given in Figure 52. The figure shows the probability of accepting a bearing whose L_{10} defect life is either 100,000 miles (assumed to represent a "poor quality" bearing) or 500,000 miles (assumed to represent a "good quality" bearing). The horizontal axis is the number of miles to which the bearings are tested (L_{t1}). These miles can be actual miles (representative loading) or the equivalent miles produced by accelerated testing. The several curves on the plot are for various numbers of test bearings (n_1) -- each bearing being successfully tested to the number of miles given on the horizontal axis. The plot indicates that for a given number of bearings under test, the probability of accepting a good or poor quality bearing decreases as the number of test miles increases. In addition, for a fixed number of test miles, the probability of accepting a good or poor quality bearing decreases as the number of bearings is increased.

For the certification procedure to be useful, the procedure should be used at the point of maximum vertical separation between the good and poor quality bearing curves. Also, this separation should be as large as possible. It

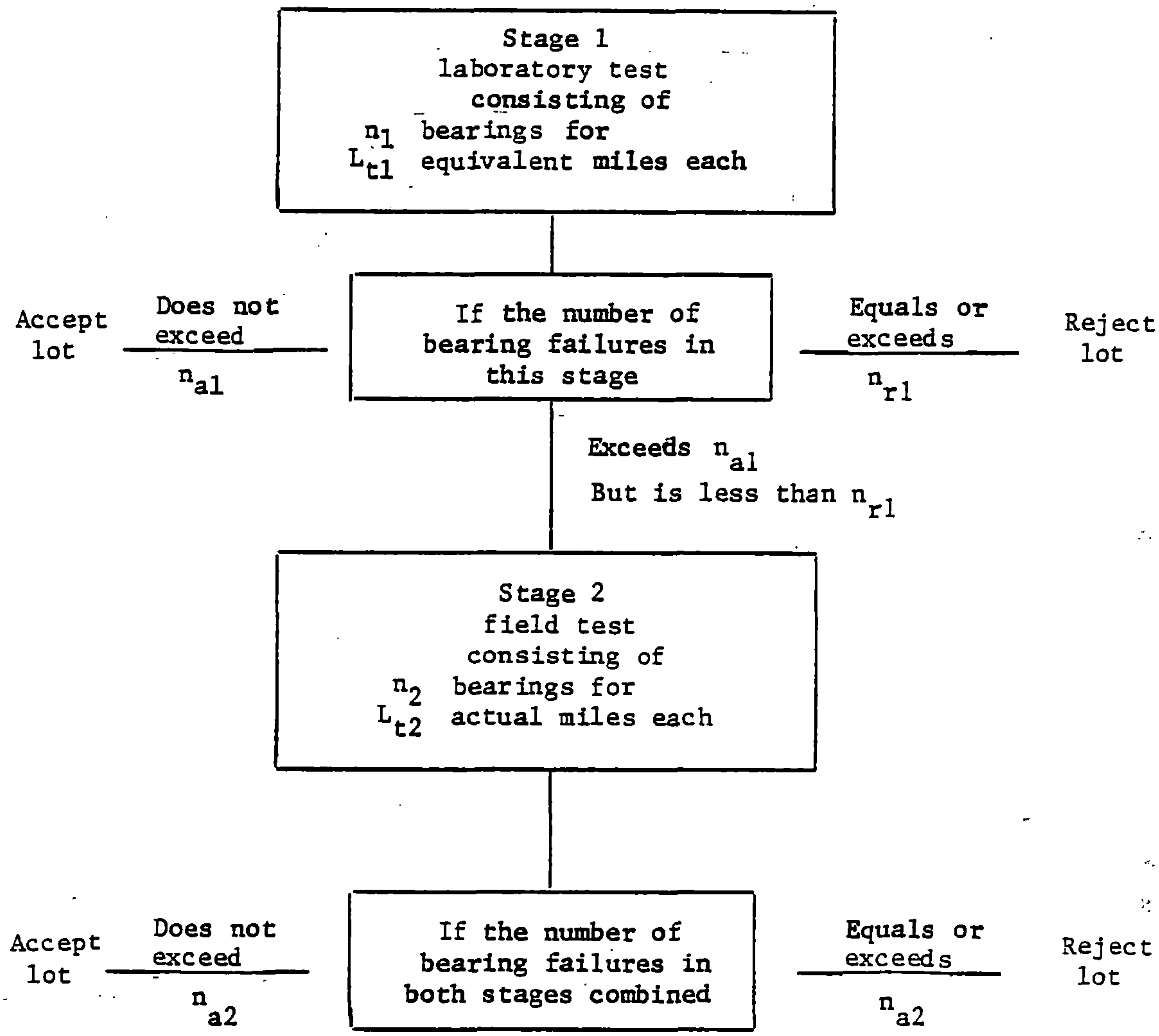


FIGURE 51. GENERAL CERTIFICATION PROCEDURE

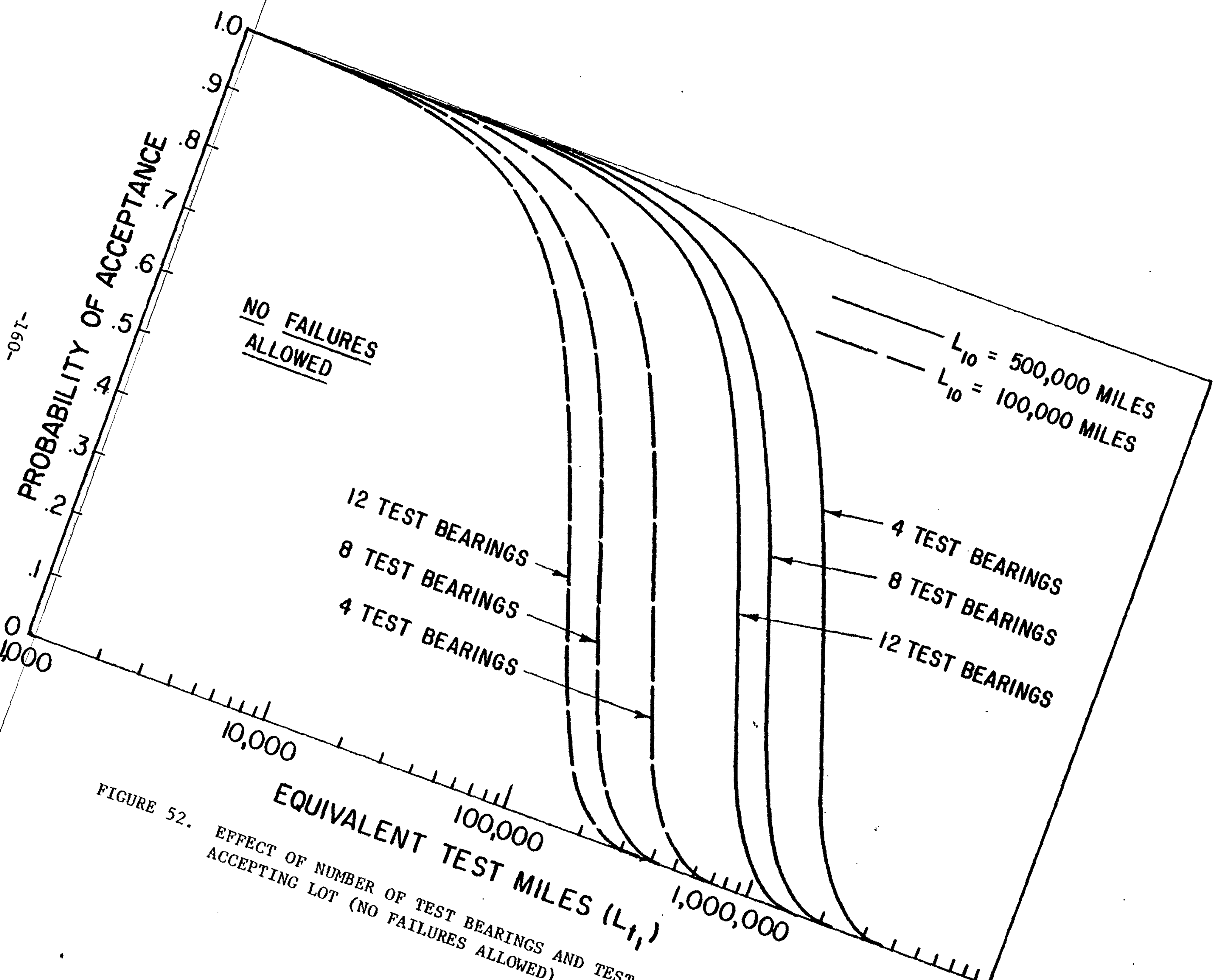


FIGURE 52. EFFECT OF NUMBER OF TEST BEARINGS AND TEST MILES ON PROBABILITY OF ACCEPTING LOT (NO FAILURES ALLOWED)

is interesting to note that the maximum discrimination is the same for <u>any</u> number of bearings (at the <u>appropriate</u> test mileage) and for <u>any</u> mileage (at the <u>appropriate</u> number of bearings). Also, the probability of accepting a good (or poor) quality bearing is the same at all points of maximum discrimination.

The effect of n_{a1} on the acceptance procedure was studied by changing n_{a1} from 0 to 1. The procedure is now less consumer-protective in that either zero or one defective bearings still result in acceptance. More than one defective bearing causes rejection.

The results of the analysis with $n_{a1} = 1$ are given in Figure 53. The figure shows that, qualitatively, the characteristics of the procedure with $n_{a1} = 1$ are the same as those with $n_{a1} = 0$. However, all curves have been moved to the right (when compared to those of Figure 52). This indicates that in order to obtain the maximum discrimination between the good and poor quality bearing, more miles must be run — number of bearings is constant or more bearing must be tested — number of miles run is constant as compared to the case when $n_{a1} = 0$.

It should be noted that the maximum good/poor quality bearing discrimination is larger for $n_{a1} = 1$ than for $n_{a1} = 0$. This maximum discrimination is about 0.76 and is, as before, independent of the number of bearings (at the appropriate number of miles).

Figure 53 was used as the basis for studying the effect of adding an additional stage to the already present stage 1. To do this, n_{a1} was specified so that bearing approval was not possible in stage 1. Consequently, if either 0 or 1 defective bearings were found in stage 1, stage 2 was undertaken. If two defective bearings were found in stage 1, the bearing lot was rejected.

In stage 2, the most consumer-protective policy consistent with stage 1 was used; i.e., $n_{a2} = 1$.

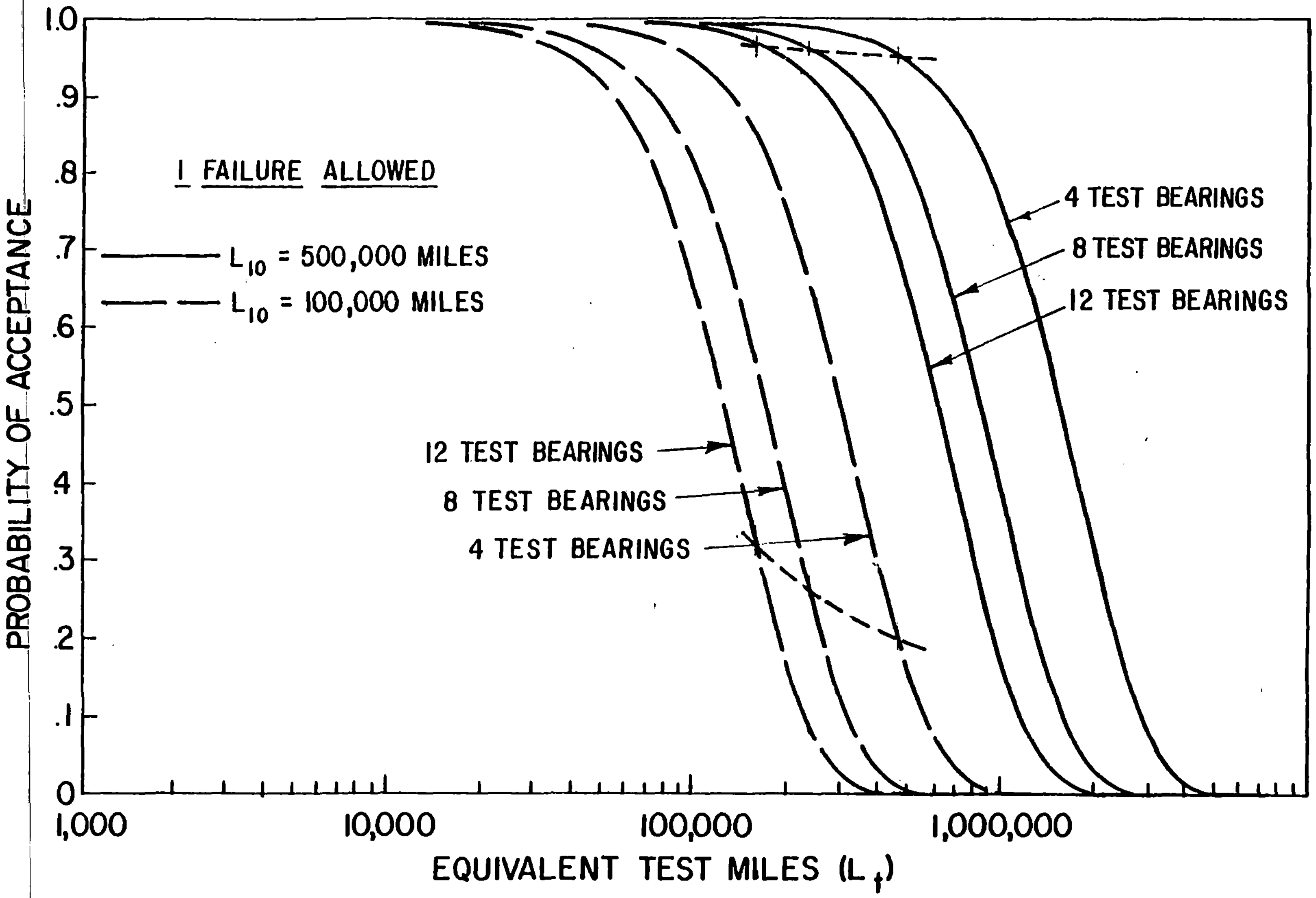


FIGURE 53. EFFECT OF TEST BEARINGS AND TEST MILES ON PROBABILITY OF ACCEPTING LOT (ONE FAILURE ALLOWED)

Figure 54 shows the result of adding stage 2 to an already present stage 1. The already present stage 1 is that for $L_{t1} = 473{,}500$ test miles and $n_1 = 4$ bearings -- see Figure 53. In Figure 54 the probability of accepting a good or poor quality bearing by the entire two-stage certification procedure is plotted versus the number of stage 2 test miles. (These stage 2 test miles are actual usage miles.) Results for $n_2 = 8$, 40 and 64 are given.

Figure 54 indicates that adding the second stage into the already present first stage does not affect the probability of accepting either a good or poor quality bearing unless n_2 and/or L_{t2} are large. It also shows that the discrimination between good and poor quality bearings can be increased from that in stage 1 and that for a given n_2 (l_{t2}), aN L_{t2} (n_2) can be selected to provide maximum discrimination. As before, this maximum discrimination is the same for any n_2 (at the appropriate L_{t2}) and for any L_{t2} (at the appropriate n_2).

Changing n_{a2} from 1 to 2 causes the curves in Figure 54 to change to those in Figure 55. Since $n_{a2} = 2$ is less consumer protective than is $n_{a2} = 1$, it is not surprising to find that the curves in Figure 55 are to the right of those in Figure 54. In addition, the maximum discrimination has increased from that of Figure 54 and for a given number of bearings is at a point to the right of that in Figure 54.

Two additional plots (Figures 56 and 57) were constructed for the present broadened analysis of the certification procedure. These plots give results corresponding to those in Figures 54 and 55; however, a different "already present" stage 1 was used to which stage 2 was added. This different stage 1 is that shown in Figure 53 ($L_{t1} = 157{,}833$ miles and $n_1 = 12$ bearings). These stage 1 values of L_{t1} and n were not selected arbitrarily. They were chosen so that the product $L_{t1} \cdot n_1$ used for Figures 54 and 55 was the same as that for Figures 56 and 57. In stage 1, the product $L_t \cdot n$ represents a measure of the demand on the test machine. For $L_t \cdot n = 1{,}894{,}000$ the curve of equal test machine demand is shown in Figure 53 as a dashed line.

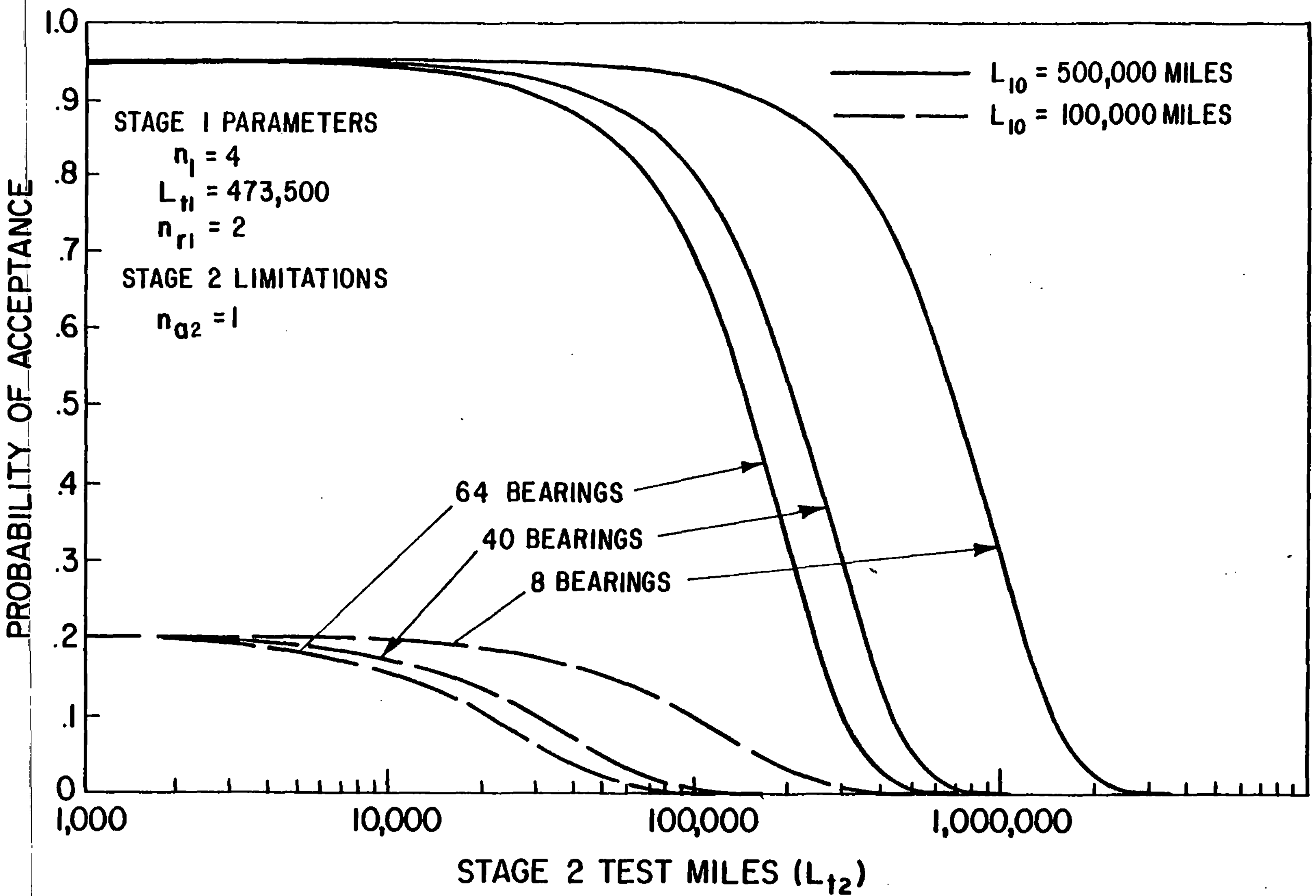


FIGURE 54. EFFECT OF STAGE 2 TEST MILES AND NUMBER OF TEST BEARINGS ON PROBABILITY OF ACCEPTING LOT

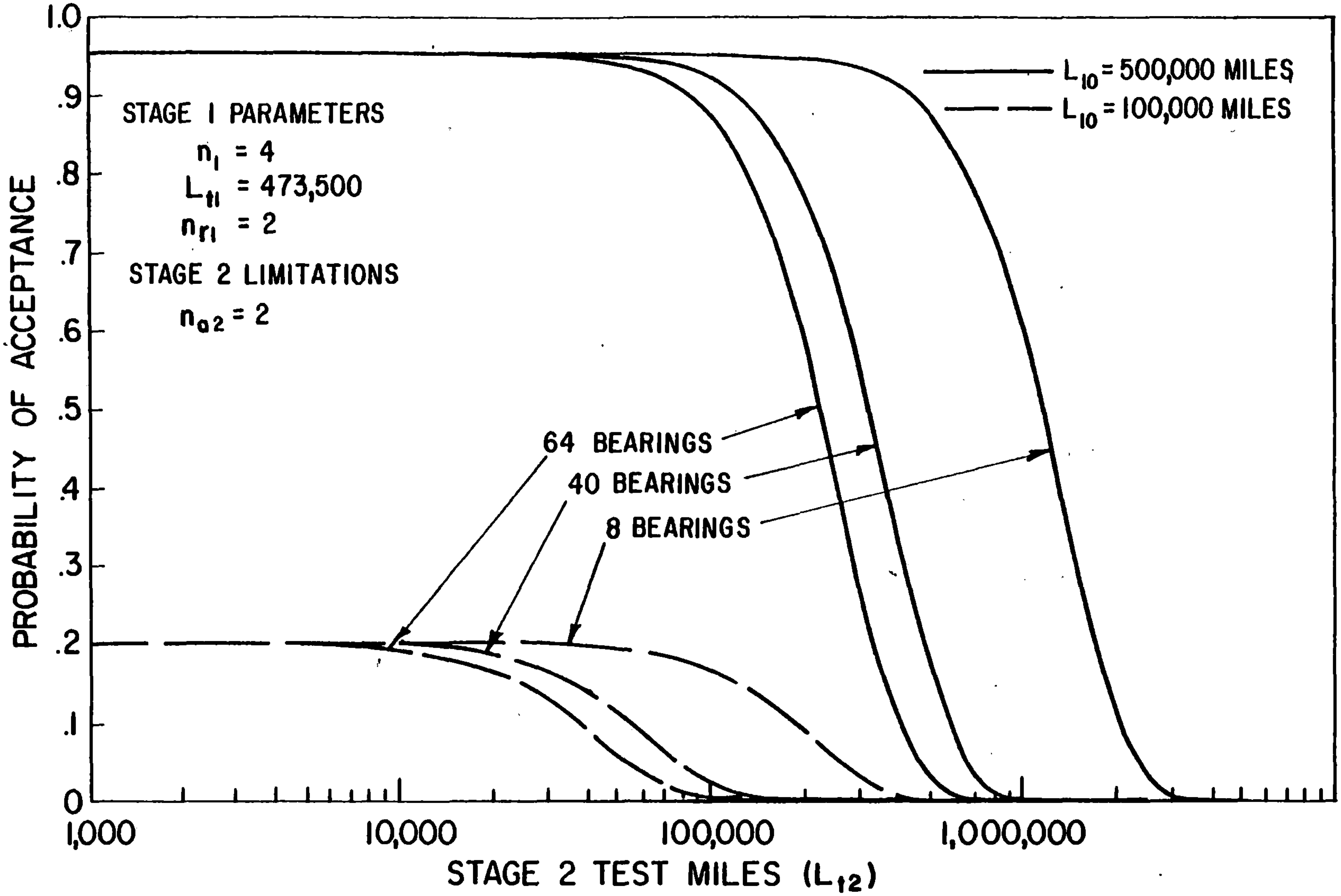


FIGURE 55. EFFECT OF STAGE 2 TEST MILES AND NUMBER OF TEST BEARINGS ON PROBABILITY OF ACCEPTING LOT (CONTINUED)

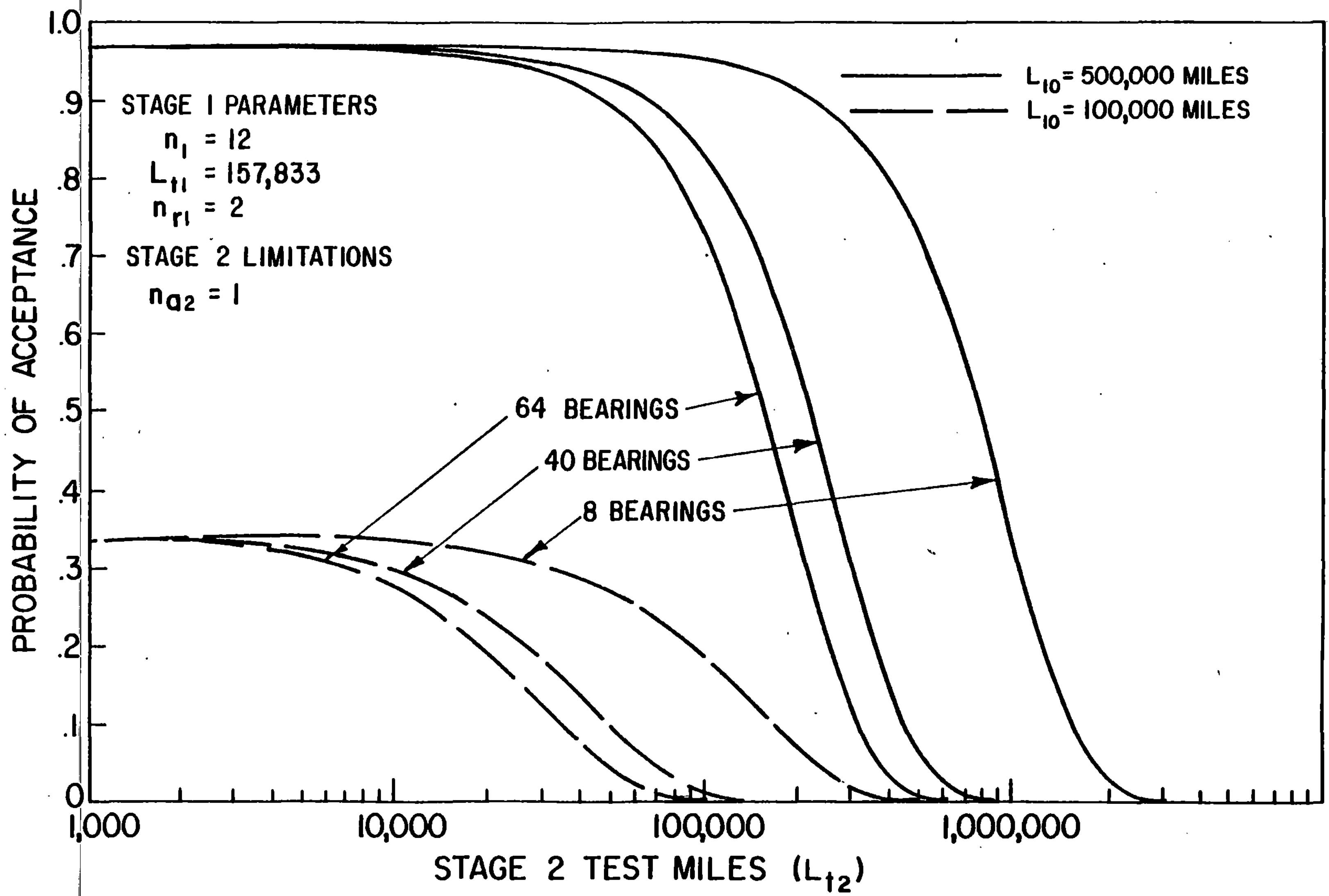


FIGURE 56. EFFECT OF STAGE 2 TEST MILES AND NUMBER OF TEST BEARINGS ON PROBABILITY OF ACCEPTING LOT (CONTINUED)

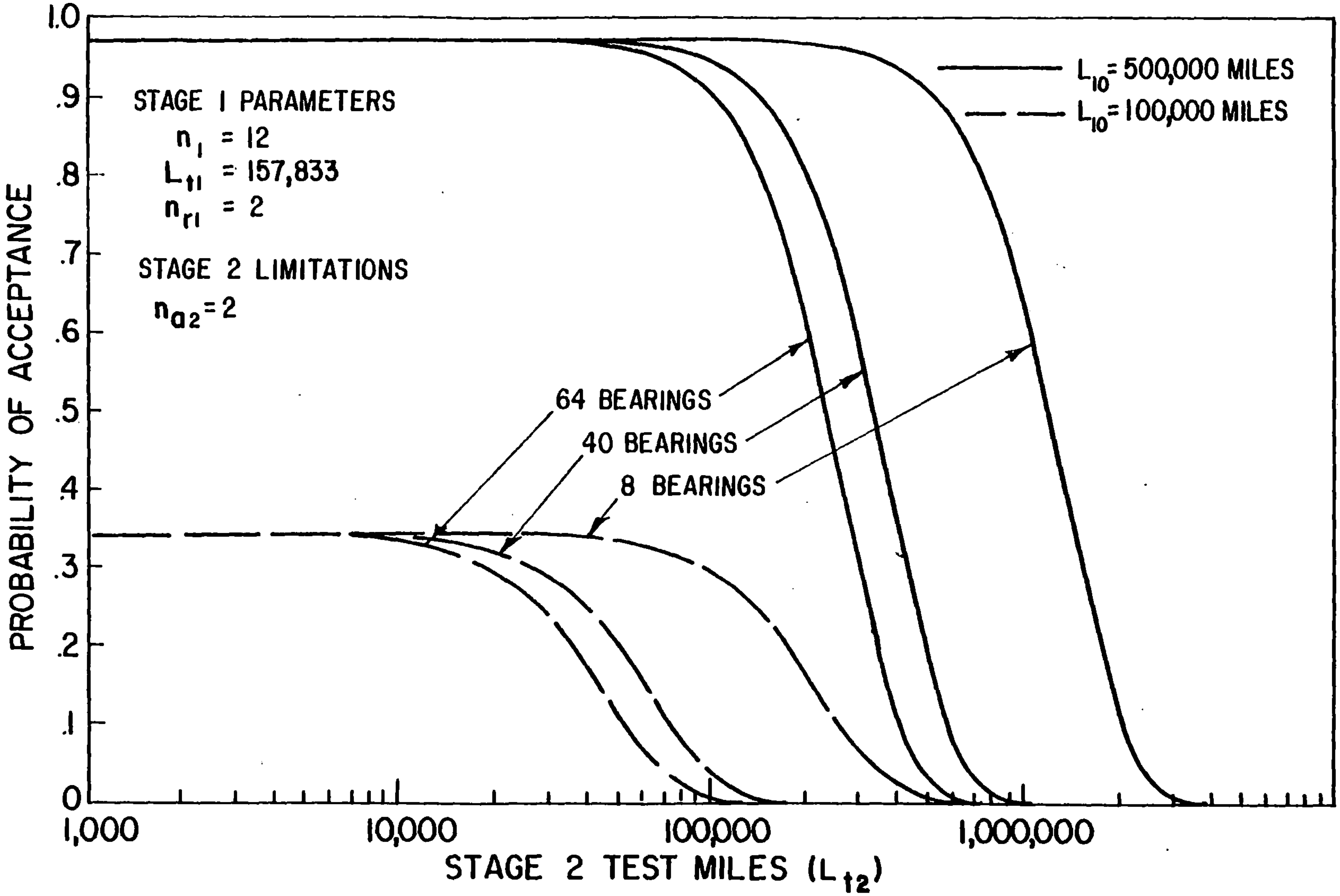


FIGURE 57. EFFECT OF STAGE 2 TEST MILES AND NUMBER OF TEST BEARINGS ON PROBABILITY OF ACCEPTING LOT (CONTINUED)

Figures 56 and 57 show that, qualitatively, the characteristics of the certification procedure are the same as those for the previous stage 1. For low values of L_{t2} and n_2, no change in the probability of accepting a good or poor quality bearing is obtained. Also, as before, the maximum discrimination obtained is greater with the less consumer-protective procedure of Figure 57 as compared with that of Figure 56. Also, this maximum discrimination requires more stage 2 miles (for a given number of stage 2 bearings) or more stage 2 bearings (for a given number of stage 2 miles).

It is of interest to compare Figure 54 with Figure 56 and to compare Figure 55 with Figure 57. Although a different stage 1 was used for Figures 54 and 55 than for Figures 56 and 57, the comparison sugests that for the two stage 1 conditions used, the maximum discrimination obtainable in the second stage is essentially independent of the conditions of the first stage. It should be noted that the probability of accepting a good (or poor) quality bearing is not the same in Figure 54 as it is in Figure 56 at the point of maximum discrimination. This is true also for Figures 55 and 57.

7.2.2 Example of Acceptance Plan Alternatives

The preceding curves (Figures 52 through 57) showed the effect on the probability of acceptance by varying:

1. The number of stages in the acceptance procedure,
2. The number of bearings subjected to acceptance testing,
3. The number of equivalent miles imposed upon the bearings tested, and
4. The number of failures permitted during the test process.

Needless to say, there are a number of ways in which the aforementioned variables can be manipulated to arrive at a acceptance procedure. Furthermore, the range of each of the variables could be broadened.

In an effort to bring the results of the risk analyses into better focus, an example of one possible acceptance procedure is presented in this section. This procedure, which is based upon the same approach as was used to produce the curves previously presented, is shown in Figure 58.

The following ground rules or initial assumptions were used in producing the procedure shown in Figure 58.

1. Four bearings would be subjected to laboratory (stage 1) testing.
2. Five car sets (40 bearings) would be subjected to field (stage 2) testing.
3. Two failures would be permitted during the procedure.
4. Field testing would be permitted only after the laboratory testing had demonstrated a consumer risk equal to or less than 0.2.
5. Testing would be terminated and the bearing would be termed "acceptable" at the point where the consumer's risk equalled 0.1.
6. At the termination of testing, the manufacturer's risk would not exceed 0.05.

As can be seen from Figure 58, there are least five ways a given lot of bearings could gain acceptance under the aforementioned assumed ground rules. To illustrate the method by which such a procedure can be generated, consider the lower route to acceptance shown in Figure 58, i.e., one failure in each stage.

Referring to Figure 53 (stage 1 or laboratory test only and one failure allowed), it is seen that in the case of four test bearings, a probability of acceptance of 0.2 is reached at 473,500 miles. Thus, at this point, the lot can be released for the second stage, or field testing. Referring now to Figure 55, it is seen that in the case of 40 test bearings (5 car sets), 58,000 field miles with one additional failure allowed (for a total of two failures) is required to reduce the proba-

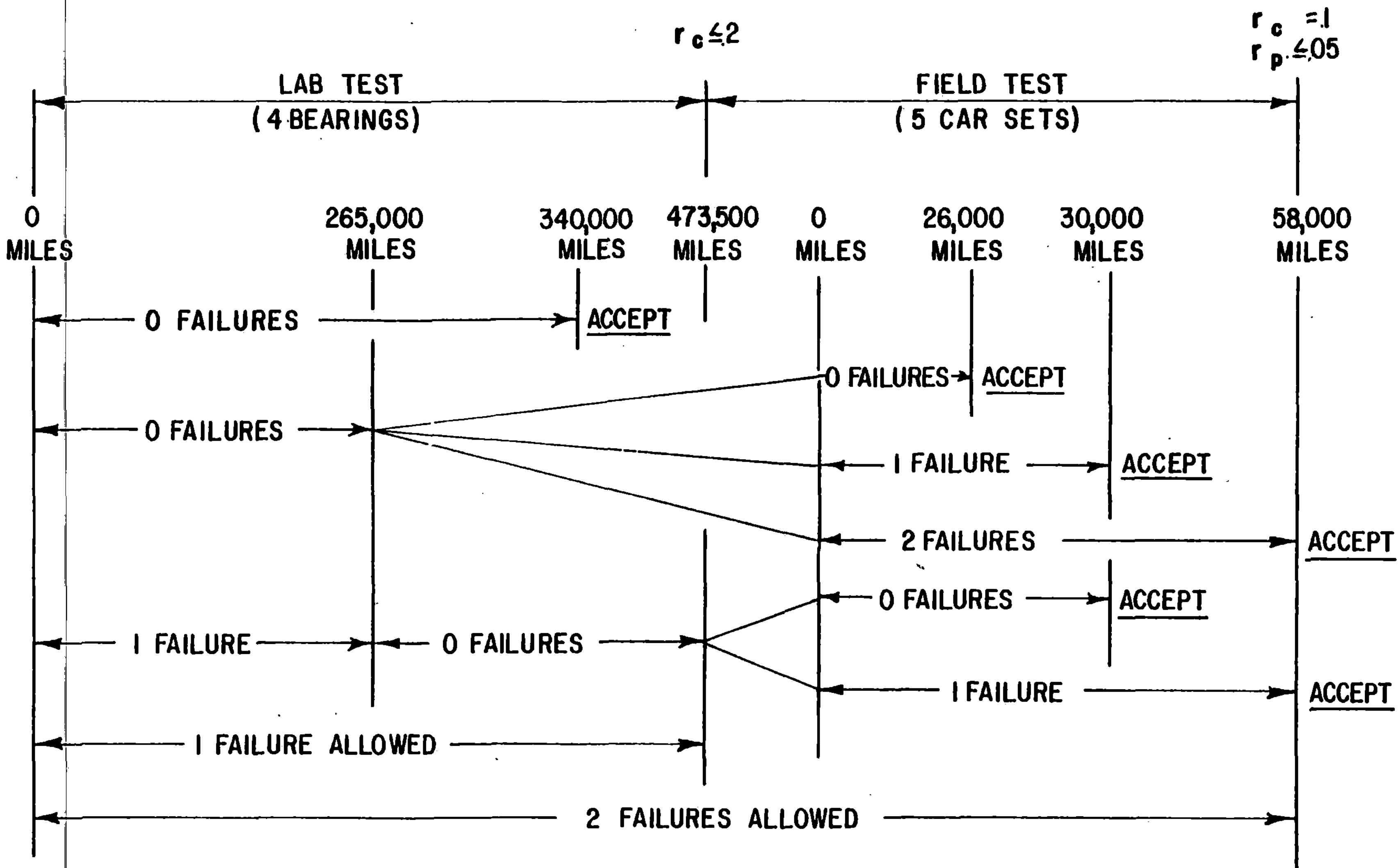


FIGURE 58. EXAMPLE OF RAILCAR ROLLER BEARING CERTIFICATION PROCEDURE DESIGNED TO PRODUCE A MAXIMUM CONSUMER'S RISK OF 0.1

bility of acceptance of a 100,000 mile L_{10} life bearing (consumer's risk) to 0.10. At this point, the probability of accepting a 500,000 mile L_{10} bearing is approximately 0.95. Thus, the producer's risk is 0.05 (1.00 - 0.95). Similar routes to acceptance can be generated using Figures 52 through 57 depending upon the risk limits and the number of test bearings one desires to impose.

Although there is an infinite number of certification schemes that could be generated, depending upon the range of variables and initial ground rules, the one presented in Figure 58 gives a reasonably good framework within which to work for example purposes

Figure 58 shows at least the trend of the results of all risk factor analyses conducted. The important ones are:

1. To get the consumer's risk down to the 0.05 to 0.2 region, it is necessary to accumulate a large number of miles (hundreds of thousands) on the bearings subjected to testing.
2. If possible, it is better to accumulate a large number of miles on a small number of bearings than to accumulate a small number of miles on a large number of bearings since increasing the number of test bearings does not reduce the required number of test miles proportionally. This is because it is necessary to run well into the life of the bearing before any reasonable distinction in bearing quality can be discerned. (Infant mortality effects are not included here since the Weibull model used to characterize the bearing defect behavior exhibits a monotonically increasing hazard.)

Clearly, the large number of miles could be accumulated either in the laboratory or in the field. However, if the main test area is in the field, the probability of field failures could be high if a good prefield screening test was not performed. Thus, it appears that the laboratory test should be the one where an acceptable consumer's risk is basically established and would allow the field test to further enhance (lower) the consumer's risk without unduly raising the producer's risk. The scheme shown in Figure 58, in principle, accomplishes these objectives.

8. REFERENCES

1. Association of American Railroads, "Year Book of Railroad Facts," Washington, D.C., 1974.

2. Federal Railroad Administration, "Journal Failure Report," October, 1972.

3. Locks, M.O., RELIABILITY, MAINTAINABILITY, AND AVAILABILITY ASSESSMENT, Spartan Books, Rochelle Park, N.J., 1973.

4. Association of American Railroads, Operation and Maintenance Department, Mechanical Division, "Roller Bearing Manual," 3rd Edition, May 1, 1974.

5. Waldron, W.D., Frarey, J.F., and Smith, A.I., "Improvement of Railroad Roller Bearing Test Procedures and Development of Roller Bearing Diagnostic Techniques -- Volume 2, Diagnostics" Prepared for U.S. Department of Transportation, Transportation Systems Center, Cambridge, MA, by Shaker Research Corporation, FRA/ORD-81/28,II, March 1981.

6. Harris, T.A., ROLLING BEARING ANALYSIS, John Wiley and Sons, Inc., New York, 1966.

7. Moyar, G.J. and Morrow, J., "Surface Failure of Bearings and Other Rolling Elements," University of Illinois Engineering Experiment Station Bulletin 486, Urbana, November 1964, Bulletin 468.

8. Moyer, C.A., and McKelvey, R.E., "A Rating Formula for Tapered Roller Bearings, " SAE Transactions, Vol. 71, 1963.

9. "Load Ratings and Fatigue Life for Roller Bearings," AFBMA Standard 11, Approved June 1972.

10. Jones, A.B., "The Life of High-Speed Ball Bearings," ASME Transactions, Vol. 74, No. 5, July 1952, pp. 695-703.

11. McGrew, J.M., et al, "Elastohydrodynamic Lubrication -- Preliminary Design Manual," United States Air Force Report AFAPL-TR-70-27, November 1970.

12. Journal of ASME "Life Adjustment Factors for Ball and Roller Bearings," September 1971.

13. Lundberg, G., Palmgren, A., and Bratt, E., "Static Capacity of Ball and Roller Bearings," Kullagertidningen, No 3, 1943.

14. Palmgren, A., Lundgerg, G., Anderson, B., and Bratt, E., "The Dependence of Static Capacity on Hardness," Kullagerdidningen, No. 3., 1945.

15. Palmgren, A., BALL AND ROLLER BEARING ENGINEERING, Burbank and Co. Philadelphia, PA, 3rd Edition, 1959, p. 53.

16. Wickstrand, N.M., "Depth of Permanent Indentations in Flat Plates Due to Loaded Cylindrical Rollers," Journal of Lubrication Technology, Trans. ASME, 92F 163-178 (1970).

17. Johnson, M.R., "Analysis of Railroad Car, Truck and Wheel Fatigue -- Part 1: Service Load Data and Procedures for the Development of Fatigue Performance Criteria," U.S. Department of Transportation, 1975, Report No. FRA-OR&D-75-68.

18. Johnson, M.R., "Effects of Longitudinal Impact Forces on Freight Car Truck Bolsters," Report No. FRA-ORD&D-75-10, Prepared for Federal Railroad Administration, September 1974.

19. Johnson, M.R., "Structural Adequacy of Freight Car Truck Castings and Wheels," Prepared for Department of Transportation under Contract No. DOT/TSC 727, October, 1976.

20. Muir, H., Averback, B.C., and Cohen, M., "The Elastic Limit and Yield Behavior of Hardened Steels," Trans. ASM, Vol. 47, 1955, pp. 380-407.

21. Johnson, K.L., "A Shakedown Limit in Rolling Contact," University Engineering Laboratory, Cambridge, England, 1961.

22. Landgraf, R.W., Morrow, J.D. and Endo, T., "Determination of the Cyclic Stress-Strain Curve," Journal of Materials, Vol. 4, No. 1, March, 1969, pp. 176-188.

23. Mackenzie, C.T. and Benham, P.P., The Engineer, Decenber 28, 1962 (quoted by Coffin, L.F., Trans. ASM, Vol. 60, 1967, pp. 160-175).

24. Moyar, G.J., "A Mechanics Analysis of Rolling Element Failures," PH.D. Dissertation, University of Illinois, Urbana 1960.

25. Timoshenko, S., STRENGTH OF MATERIALS: PT. II, ADVANCE THEORY AND PROBLEMS, 3rd Edition, Van Nostrand (Princeton, N.J.), 1956.

26. Moyar, G.J., "Performance Comparison of Radial Lip Seals for Railyard Journal Bearing Service," Brenco, Inc., February 1975.

27. Nelson, W., "Hazard Plotting for Incomplete Failure Data," Journal of Quality Technology, Vol. 1, No. 1, January 1969.

28. Horve, L.A., "Statistical Interpretation of Shaft Seal Performance," Society of Automotive Engineers, SAE Paper 741044, October, 1974.

29. Wilson, D.S., "Reliability Model for Miniature Blowers, Final Report" Prepared under RADC Contract F30603-74-C-0131, Rome Air Development Center, Rome, NY, February, 1975.

30. Crandall, S.H. and Marks, W.P., RANDOM VIBRATION MECHANICAL SYSTEMS, Academic Press, New York, N.Y. 1963.

31. Merwin, J.F., and Johnson, K.L., "An Analysis of Plastic Deformation in Rolling Contact," Proceedings of the Institution of Mechanical Engineers (London), 1963, Vol. 177, No. 25.

32. Hamilton, G.M., "Plastic Flow of Rollers Loaded Above the Yield Point," Proceedings of the Institution of Mechanical Engineers, (London) 1963, Vol. 177, Number 25.

33. Bowden, F.P., and Tabor, D., THE FRICTION AND LUBRICATION OF SOLIDS, PART I, Oxford University Press, London, 1964.

34. Weibull, W., "A Statistical Theory of the Strength of Materials," Proceedings of the Royal Swedish Institute for Engineering Research, No. 151, Stockholm (1939).

35. Leiblein, J. and Zelen, M., "Statistical Investigation of the Fatigue Life of Deep Groove Ball Bearings," Research Paper 2719, Journal of Research, U.S. National Bureau of Standards, Vol. 57, 1956.

36. Johnson, L.G., "The Statistical Treatment of Fatigue Experiments," General Motors Research Laboratories Report 202, 1959. Elsevier Publishing Co., New York, NY, 1964.

37. Corten, H.T., "Statistical Fracture Strength Concepts," Short Course on Fracture Mechanics, Univeristy of Illinois, Urbana, July 1965.

38. Johnson, W.A. and Mehl, R.F., "Reaction Kinetics in Processes of Nucleation and Growth," Trans. AIME, Vol. 135, 1939, p. 416.

39. Austin, J.B. and Rickets, R.L., "Kinetics of Decomposition of Austenite at Constant Temperature," Trans. AIME, Vol. 135, 1939, p. 396.

40. Avrami, M., "Kinetics of Phase Change, Part I - General Theory," Journal of Chem Physics, Vol. I, p. 1103 (1939).

Idem, "Kinetics of Phase Change, Part II - Transformation - Time Relations for Random Distribution of Nuclei," Journal of Chem Physics, Vol. 8, p. 212 (1940).

41. Mann, N.R., Schafer, R.D., and Singpurwalla, N.D., METHODS FOR STATISTICAL ANALYSIS OF RELIABILITY AND LIFE DATA, John Wiley and Sons, New York, NY, 1974.

42. Kao, J.H.K., "A Summary of Some New Techniques on Failure Analysis," Proceedings of 6th National Symposium on Reliability and Quality Control in Electronics, 1960.

APPENDIX A

BEARING DEFECT DISTRIBUTIONS

Plots of the results in Tables 15 and 16 are given in Figures A-1 through A-22. For these plots, the defects were regarded as not producing catastrophic bearing failure.* In addition, it was noted that the overwhelming majority of the bearings had not previously been reworked. For both reasons, the number of defective bearings of a given age was taken to be the accumulated sum of defects which had occurred since the bearings were new. In other words, for the plots and for the determination of the Weibull parameters the cumulative distribution, F, was taken to be that represented by the proportion of defective bearings.

For each plot a regression analysis was run to fit the Weibull distribution to the data. The results of the regression are shown in Figures A-2 through A-24; the line obtained for each case is drawn on the appropriate plot. For some cases, the data for the first two years was not included in the regression. These cases were those for which the confidence interval for those years was large.

The results contained in the plots and in the regression figures are discussed in detail in Section 4.2.2.

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*Bearings with broken rollers were regarded as potential producers of catastrophic failures.

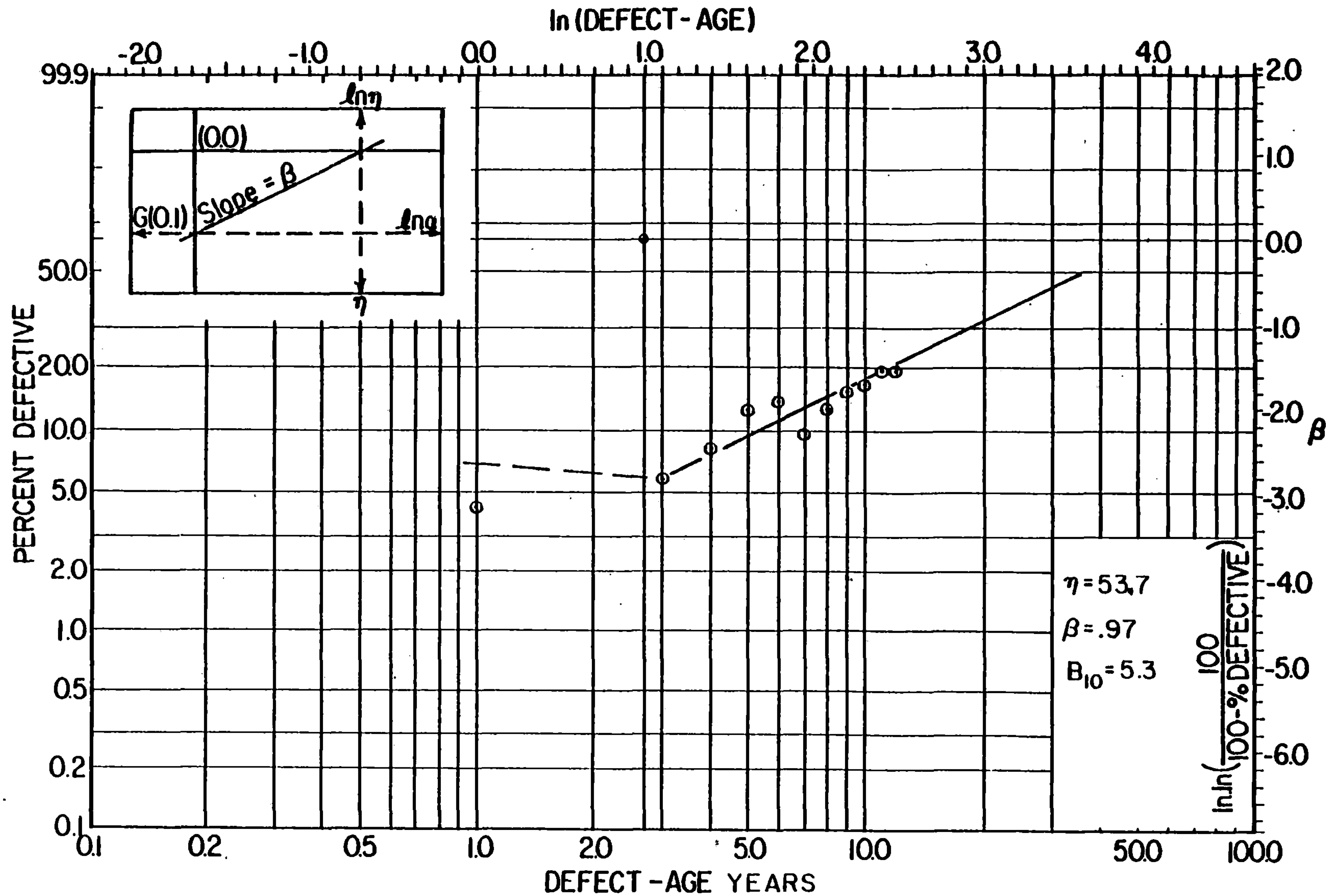


FIGURE A-1. PERCENT DEFECTIVE VERSUS DEFECT AGE - ALL CONE DEFFECTS

12	19.48	.2166645852165
11	19.42	.2159197062248
10	16.62	.1817617135354
9	15.26	.1655824407897
8	12.8	.1369658550731
7	9.71	.1021434736687
6	13.4	.1438703704197
5	12.76	.1365072446546
4	8.29	8.65387614E-02
3	5.84	6.01747230E-02

ALPHA = 53.72182075124
BETA = .9746282383292
B10 = 5.338073395593

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.299285952806	1	.299285952806
RESIDUAL	5.30994822E-02	8	6.63743528E-03
TØTAL	.35238543506	9	

F= 45.09060203252

CØEFF. ØF DETERMINATIØN= .8493141969816
CØEFF. ØF CØRRELATIØN= .92158244177
STANDARD ERRØR ØF ESTIMATE= 8.14704564E-02

FIGURE A-2. WEIBULL REGRESSION RESULTS - ALL CONE DEFECTS

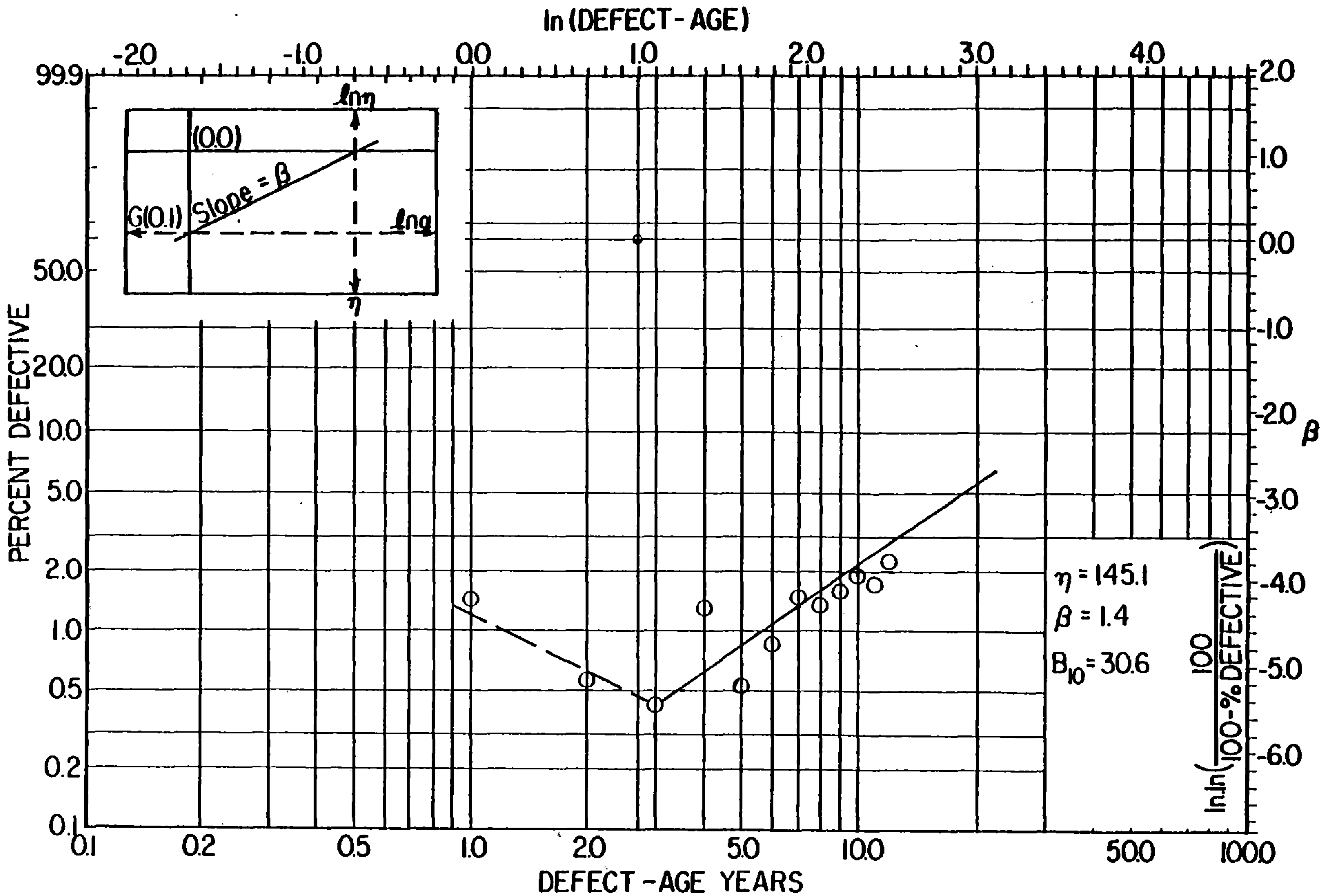


FIGURE A-3. PERCENT DEFECTIVE VERSUS DEFECT AGE - SPALLED CONES

12	2.34	2.36781273E-02
11	1.78	1.79603253E-02
10	1.93	1.94886765E-02
9	1.65	1.66376411E-02
8	1.39	1.39975096E-02
7	1.51	1.52151658E-02
6	.86	8.63719339E-03
5	.53	5.31409482E-03
4	1.27	1.27813343E-02
3	.43	4.30927158E-03

ALPHA = 145.1070933867
BETA = 1.4450795126Q2
B10 = 30.5757348811I

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.2590036992539	1	.2590036992539
RESIDUAL	9.33817358E-02	8	1.16727169E-02
TØTAL	.35238543506	9	

F= 22.18881000814

CØEFF. ØF DETERMINATIØN= .7350011478477
CØEFF. ØF CØRRELATIØN= .85732207941
STANDARD ERRØR ØF ESTIMATE= .10804034883

FIGURE A-4. WEIBULL REGRESSION RESULTS - SPALLED CONES

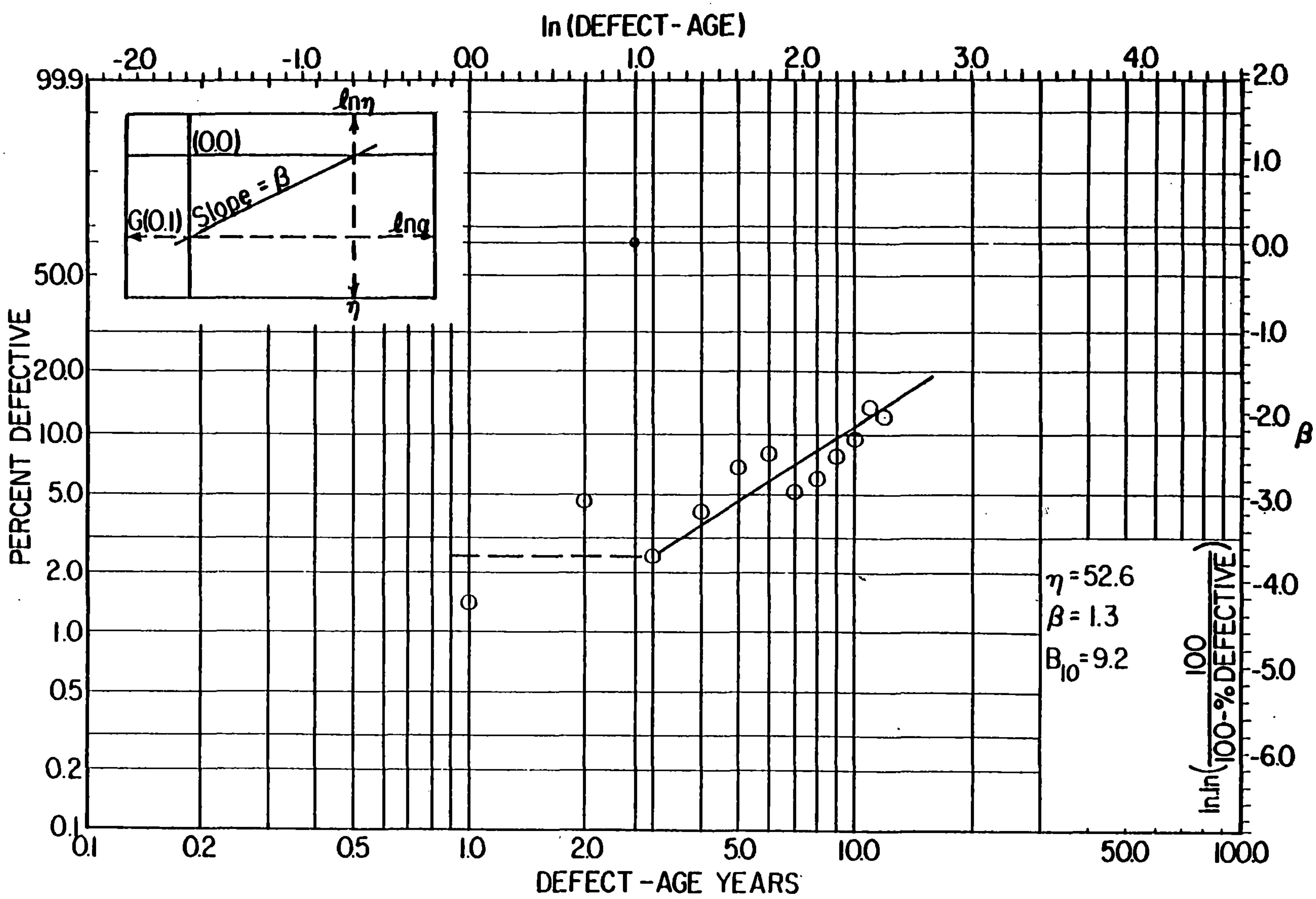


FIGURE A-5. PERCENT DEFECTIVE VERSUS DEFECT AGE - OVERSIZED CONES

12	12.4	.1323891880458
11	13.61	.1462982576173
10	9.75	.1025865887751
9	8.04	8.38164860E-02
8	6.29	6.49652788E-02
7	5.24	5.38228066E-02
6	8.27	8.63207064E-02
5	7.06	7.32160623E-02
4	4.25	4.34295579E-02
3	2.5	2.53178079E-02

ALPHA = 52.57797205403
BETA = 1.290435435452
B10 = 9.192714817355

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.2829076003862	1	.2829076003862
RESIDUAL	6.94778346E-02	8	8.68472933E-03
TØTAL	.3523854350б	9	

F= 32.57529273495

CØEFF. ØF DETERMINATIØN= .8028356800219
CØEFF. ØF CØRRELATIØN= .895601098209
STANDARD ERRØR ØF ESTIMATE= 9.31918952E-02

FIGURE A-6. WEIBULL REGRESSION RESULTS - OVERSIZED CONES

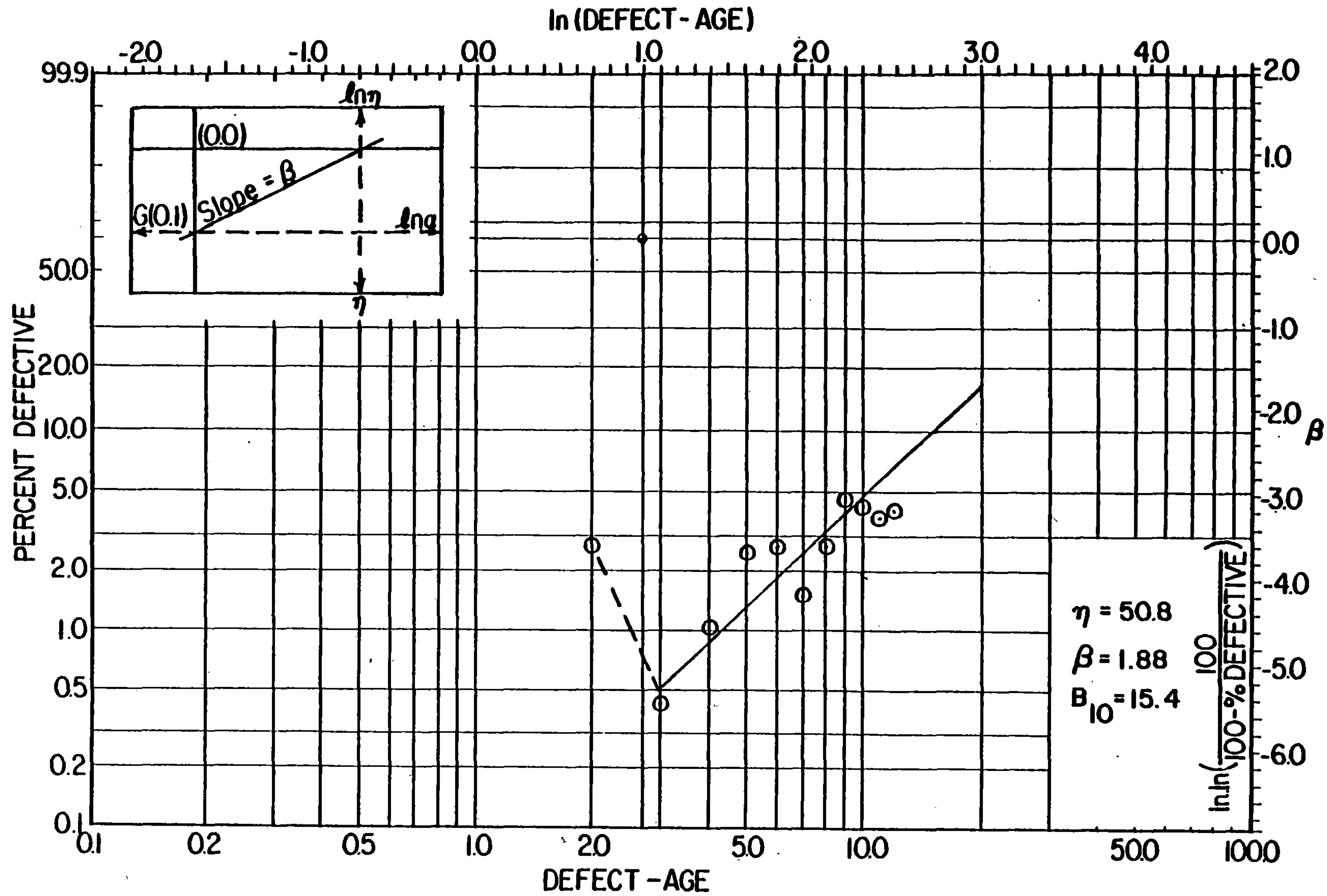


FIGURE A-7. PERCENT DEFECTIVE VERSUS DEFECT AGE - BRINELLED CONES

Age	% Brinelled Cones	
3	.43	4.30927158E-03
4	1.06	1.06565801E-02
5	2.53	2.56255476E-02
6	2.76	2.79880365E-02
7	1.51	1.52151658E-02
8	2.7	2.73711967E-02
9	4.64	4.75109826E-02
10	4.35	4.44744901E-02
11	3.76	3.83251143E-02
12	4.1	4.18642040E-02

ALPHA = 50.85284372729
BETA = 1.880794744364
B10 = 15.37027344605

REGRESSION TABLE

SOURCE	SUM OF SQ.	DEG.FREEDOM	MEAN SQ.
REGRESSION	.2786965794012	1	.2786965794012
RESIDUAL	7.36888556E-02	8	3.48370724E-02
TOTAL	.35238543506	9	

F= 30.25657835607

COEFF. OF DETERMINATION= .7908856373526
COEFF. OF CORRELATION= .88931751211
STANDARD ERROR OF ESTIMATE= 9.59745120E-02

FIGURE A-8. WEIBULL REGRESSION RESULTS - BRINELLED CONES

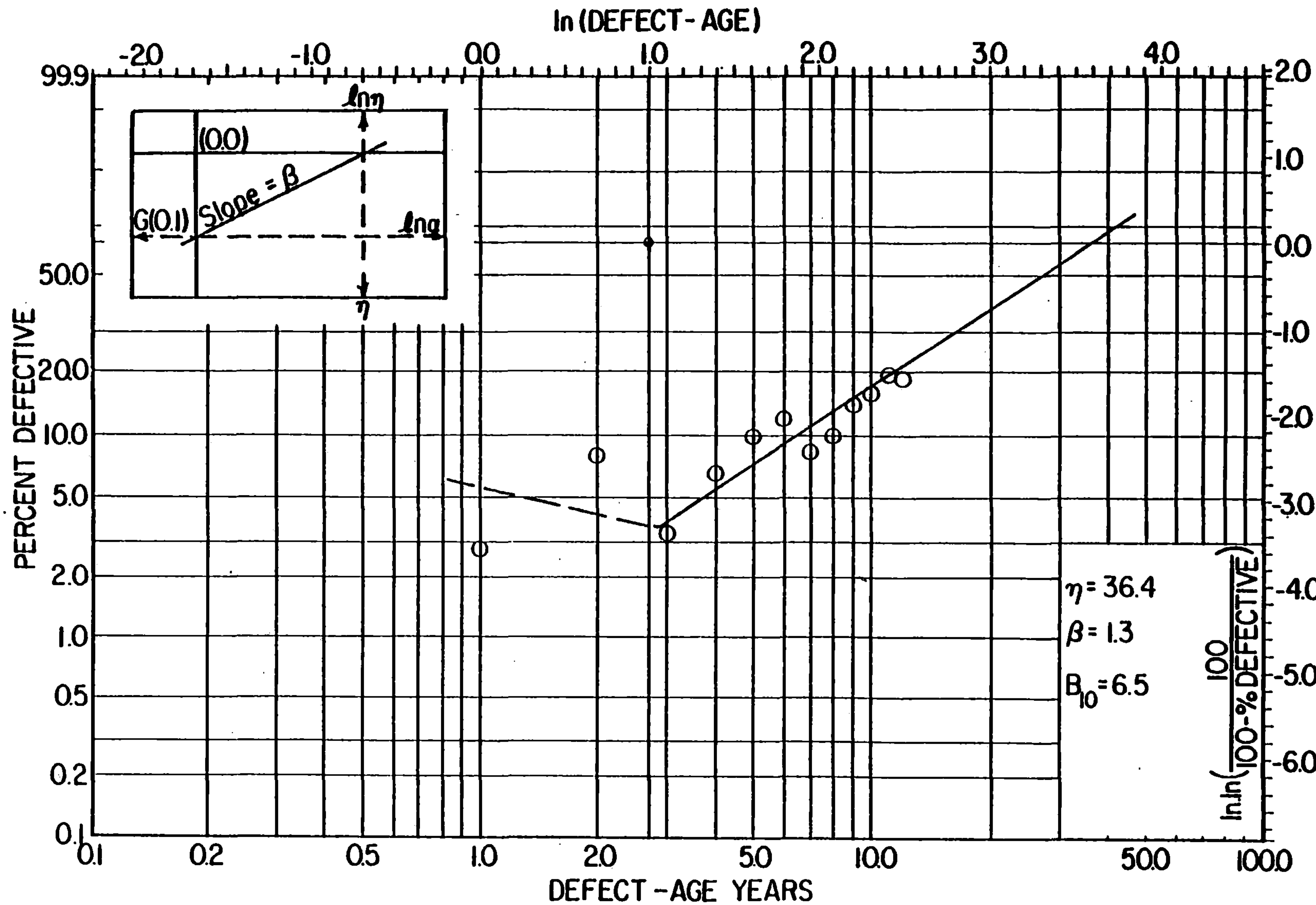


FIGURE A-9. PERCENT DEFECTIVE VERSUS DEFECT AGE - OVERSIZED, SPALLED, AND BRINELLED CONES

12	18.31	.2022385906343
11	18.89	.2093639279059
10	16.03	.1747105937926
9	14.29	.1542006810773
8	10.38	.1095916766301
7	8.26	8.62116968E-02
6	11.89	.1265841521095
5	9.96	.1049161699496
4	6.58	6.80647309E-02
3	3.36	3.41774518E-02

ALPHA = 36.42479760406
BETA = 1.310026364664
B10 = 6.536768806696

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.3060820752985	1	.3060820752985
RESIDUAL	4.63033597E-02	8	5.78791997E-03
TØTAL	.35238543506	9	

F= 52.8829142205

CØEFF. ØF DETERMINATIØN= .8686002452013
CØEFF. ØF CØRRELATIØN= .93198725592
STANDARD ERRØR ØF ESTIMATE= 7.60783804E-02

FIGURE A-10. WEIBULL REGRESSION RESULTS - OVERSIZED, SPALLED AND BRINELLED CONES

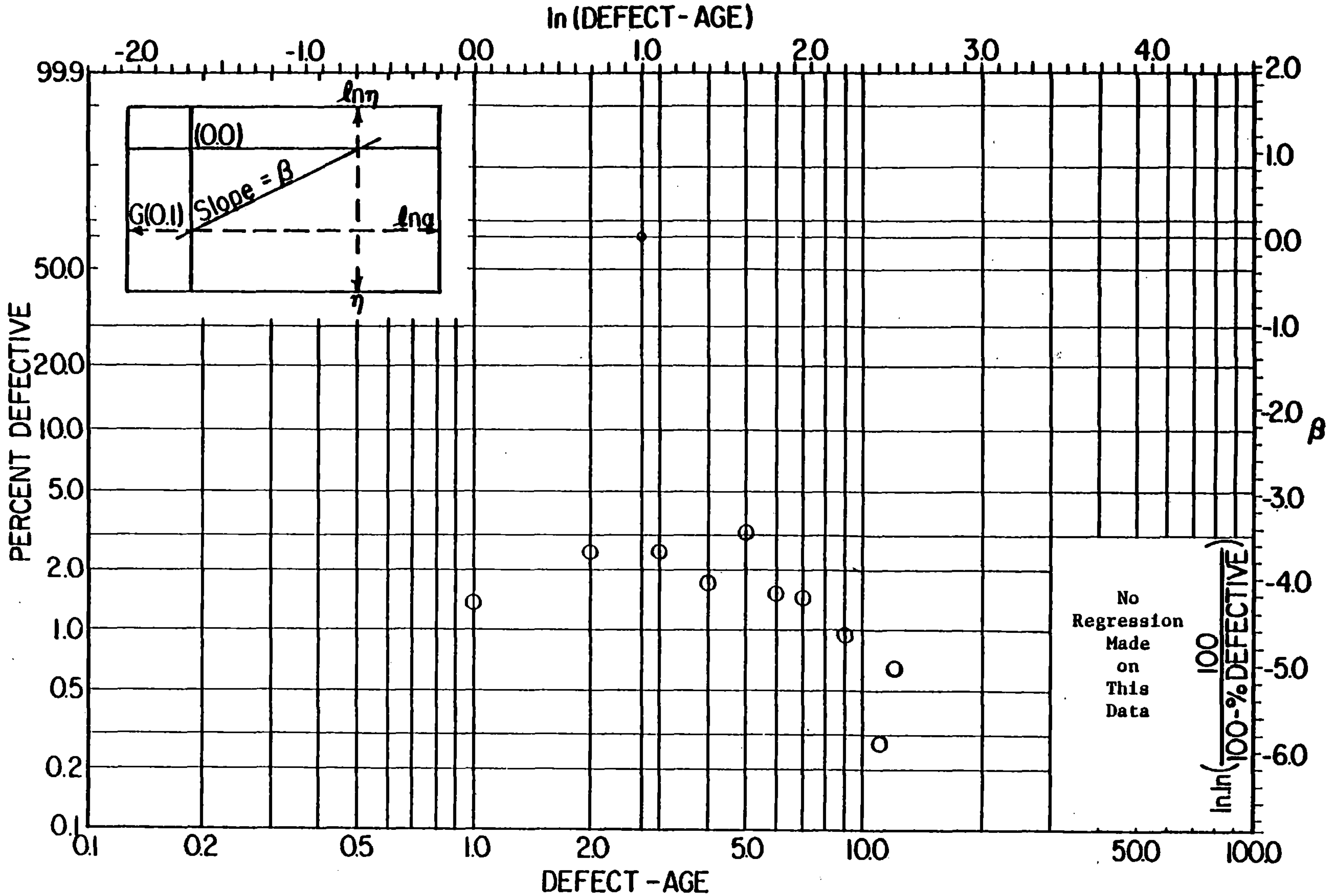


FIGURE A-11. PERCENT DEFECTIVE VERSUS DEFECT AGE - OTHER CONE DEFECTS

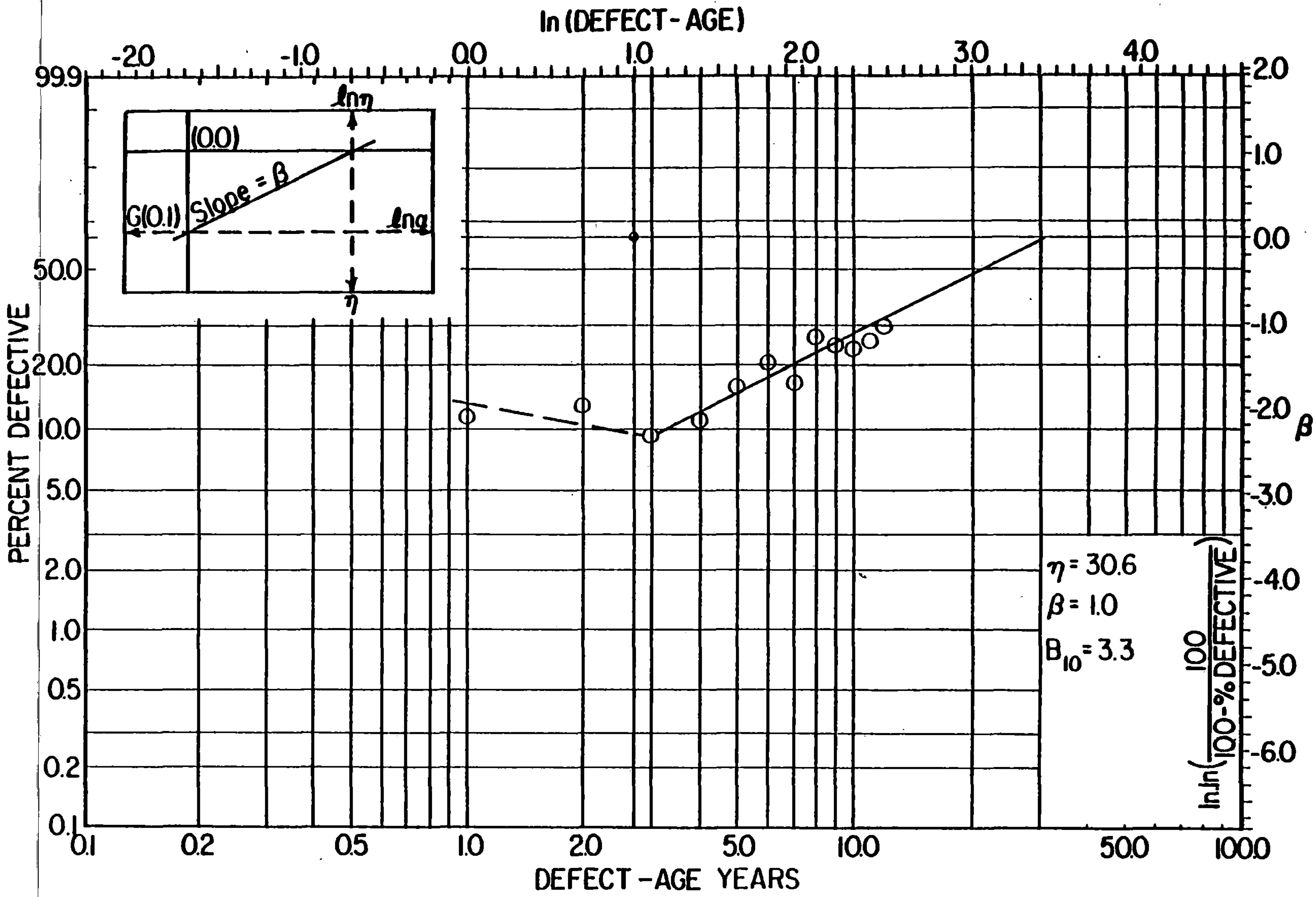


FIGURE A-12. PERCENT DEFECTIVE VERSUS DEFECT AGE - ALL CUP DEFECTS

12	30.62	.3655715444463
11	26.65	.309927681399
10	24.66	.283158983724
9	25.51	.2945052977999
8	27.03	.3151217882104
7	16.85	.1845239803491
6	21.65	.243984217186
5	16.71	.1828416920408
4	11.71	.1245433350746
3	9.34	9.80539404E-02

ALPHA = 30.56997831825
BETA = 1.007761578648
B10 = 3.277163045865

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.3165372535139	1	.3165372535139
RESIDUAL	3.58481815E-02	8	4.48102269E-03
TØTAL	.35238543506	9	

F= 70.63951137534

CØEFF. ØF DETERMINATIØN= .8982699681104
CØEFF. ØF CØRRELATIØN= .94777105258
STANDARD ERRØR ØF ESTIMATE= 6.69404413E-02

FIGURE A-13. WEIBULL REGRESSION RESULTS - ALL CUP DEFECTS

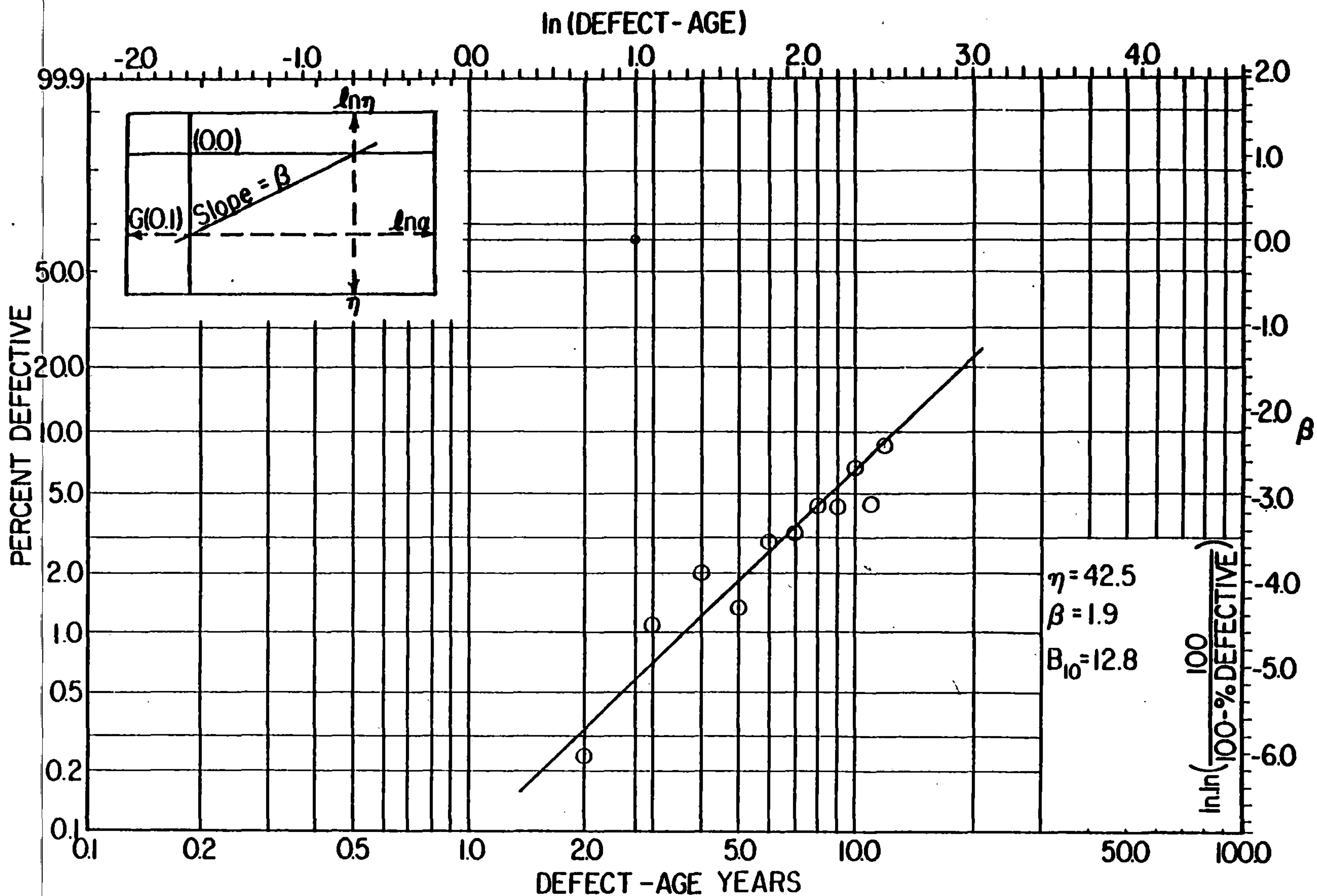


FIGURE A-14. PERCENT DEFECTIVE VERSUS DEFECT AGE - SPALLED CUPS

12	8.625	9.01982679E-02
11	4.548	4.65466826E-02
10	6.847	7.09268833E-02
9	4.366	4.46417806E-02
8	4.452	4.55414469E-02
7	3.201	3.25335223E-02
6	2.934	2.97790264E-02
5	1.349	1.35818167E-02
4	2.034	2.05497062E-02
3	1.112	1.11822894E-02
2	.238	2.38283670E-03

ALPHA = 42.4540916489 8
BETA = 1.881515254329
B10 = 12.83763111596

REGRESSION TABLE

SOURCE	SUM OF SQ.	DEG.FREEDOM	MEAN SQ.
REGRESSION	.5576068839866	1	.5576068839866
RESIDUAL	5.68298358E-02	9	6.31442620E-03
TOTAL	.6144367198175	10	

F= 88.30681776088

COEFF. OF DETERMINATION= .9075090501663
COEFF. OF CORRELATION= .95263269426
STANDARD ERROR OF ESTIMATE= 7.94633639E-02

FIGURE A-15. WEIBULL REGRESSION RESULTS - SPALLED CUPS

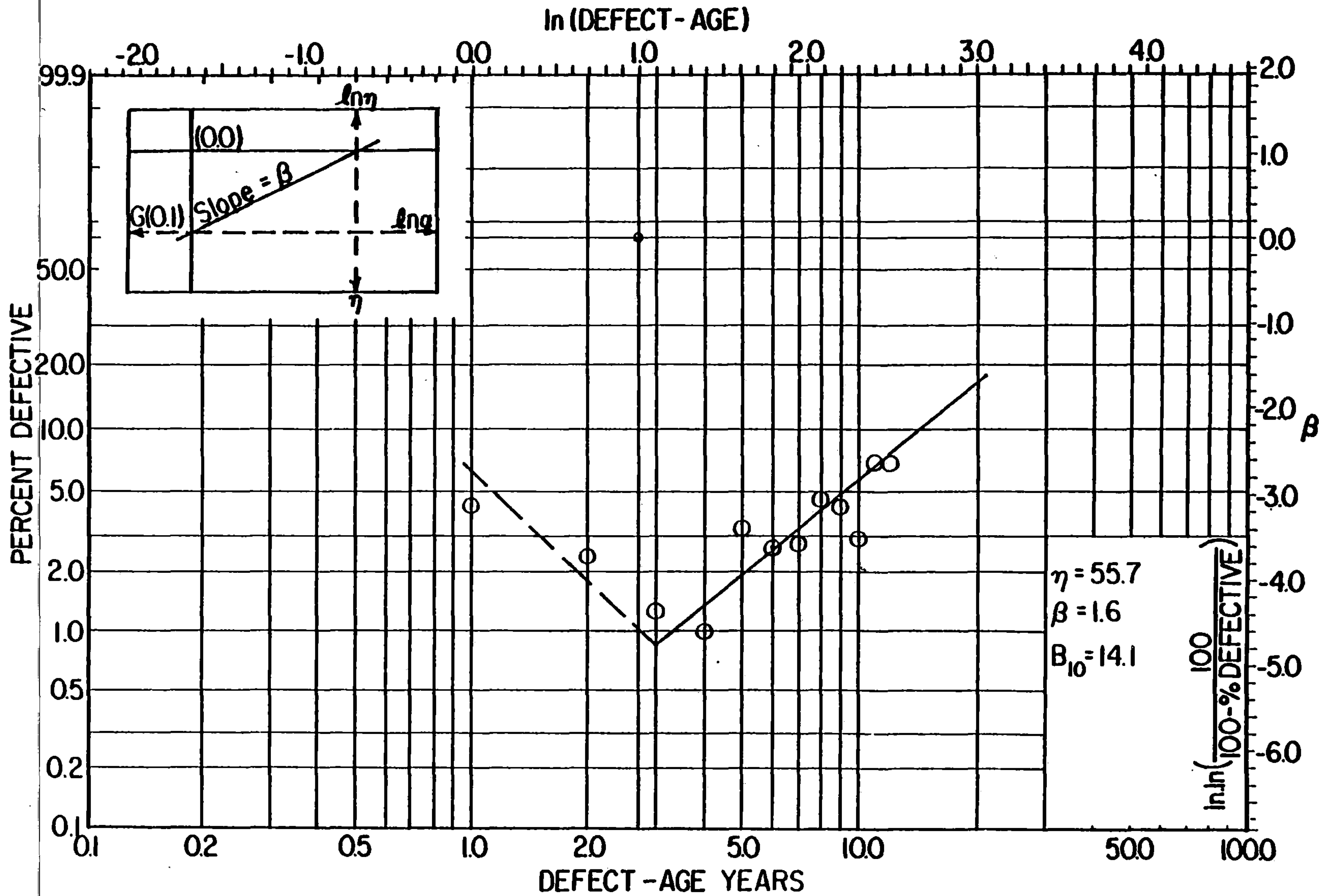


FIGURE A-16. PERCENT DEFECTIVE VERSUS DEFECT AGE - OVERSIZED CUPS

12	7.12	7.38618485E-02
11	7.05	7.31084718E-02
10	2.997	3.04282801E-02
9	4.32	4.41608957E-02
8	4.76	4.87701643E-02
7	2.81	2.85023604E-02
6	2.666	2.70218230E-02
5	3.43	3.49020520E-02
4	1	1.00503358E-02
3	1.3	1.30852395E-02

ALPHA = 55.65224332013
BETA = 1.642080485461
B10 = 14.13540620869

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.2696818486919	1	.2696818486919
RESIDUAL	8.27035863E-02	8	1.03379482E-02
TØTAL	.352385435O6	9	

F= 26.08659290702

CØEFF. ØF DETERMINATIØN= .7653036188797
CØEFF. ØF CØRRELATIØN= .87481633437
STANDARD ERRØR ØF ESTIMATE= .1016757016

FIGURE A-17. WEIBULL REGRESSION RESULTS - OVERSIZED CUPS

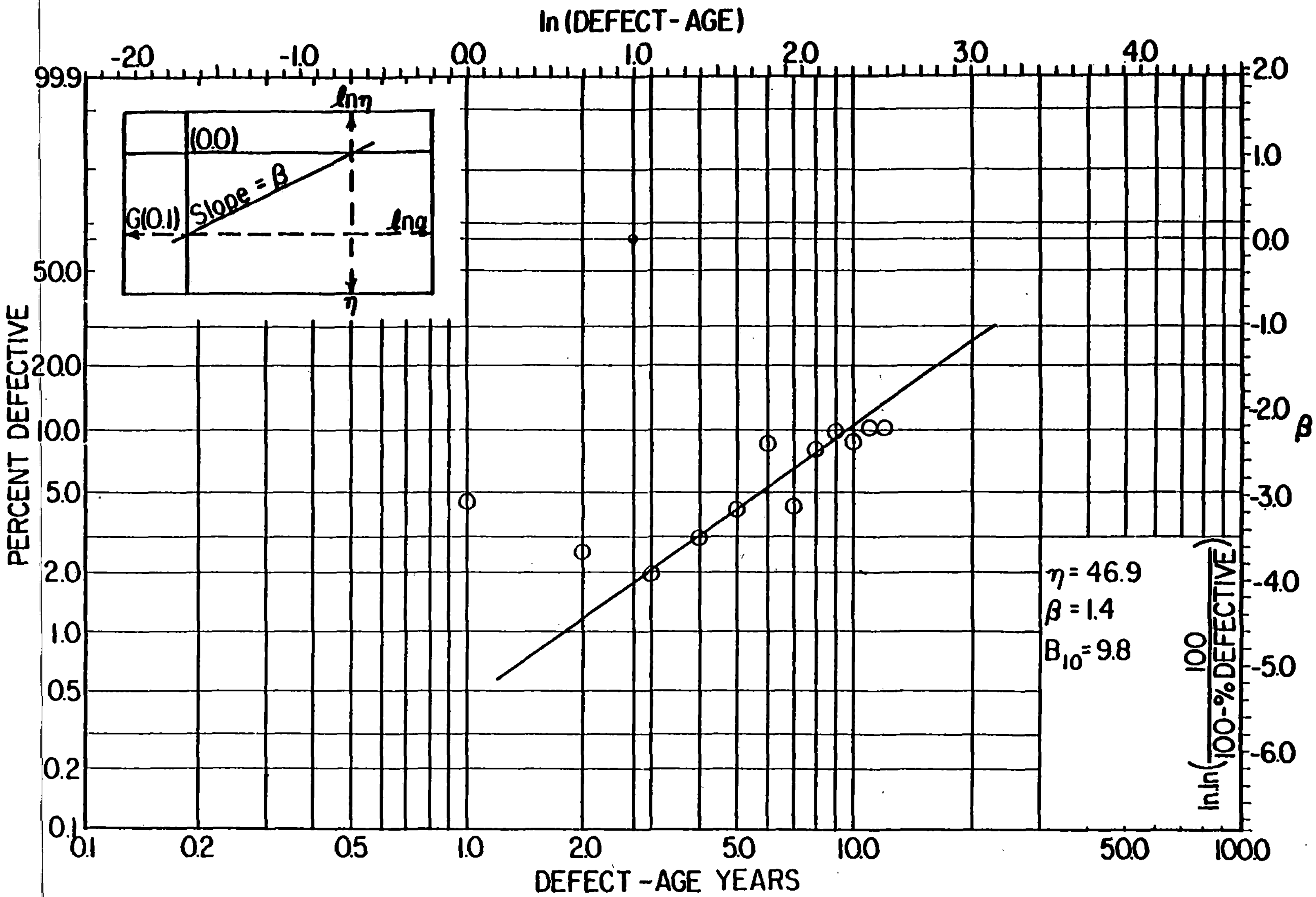


FIGURE A-18. PERCENT DEFECTIVE VERSUS DEFECT AGE - BRINELLED CUPS

12	10.52	.1111550493662
11	10.39	.109703265092
10	8.84	9.25539816E-02
9	9.82	.1033625129951
8	8.01	8.34903104E-02
7	4.37	4.46836076E-02
6	8.64	9.03624400E-02
5	4.27	4.36384570E-02
4	3.05	3.09748042E-02
3	2	2.02027073E-02

ALPHA = 46.95418179422
BETA = 1.435577266509
B10 = 9.792328074613

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.3003748373424	1	.3003748373424
RESIDUAL	5.20105977E-02	8	6.50132471E-03
TØTAL	.352385435056	9	

F= 46.20209734537

CØEFF. ØF DETERMINATIØN= .8524042354113
CØEFF. ØF CØRRELATIØN= .92325740474
STANDARD ERRØR ØF ESTIMATE= 8.06307925E-02

FIGURE A-19. WEIBULL REGRESSION RESULTS - BRINELLED CUPS

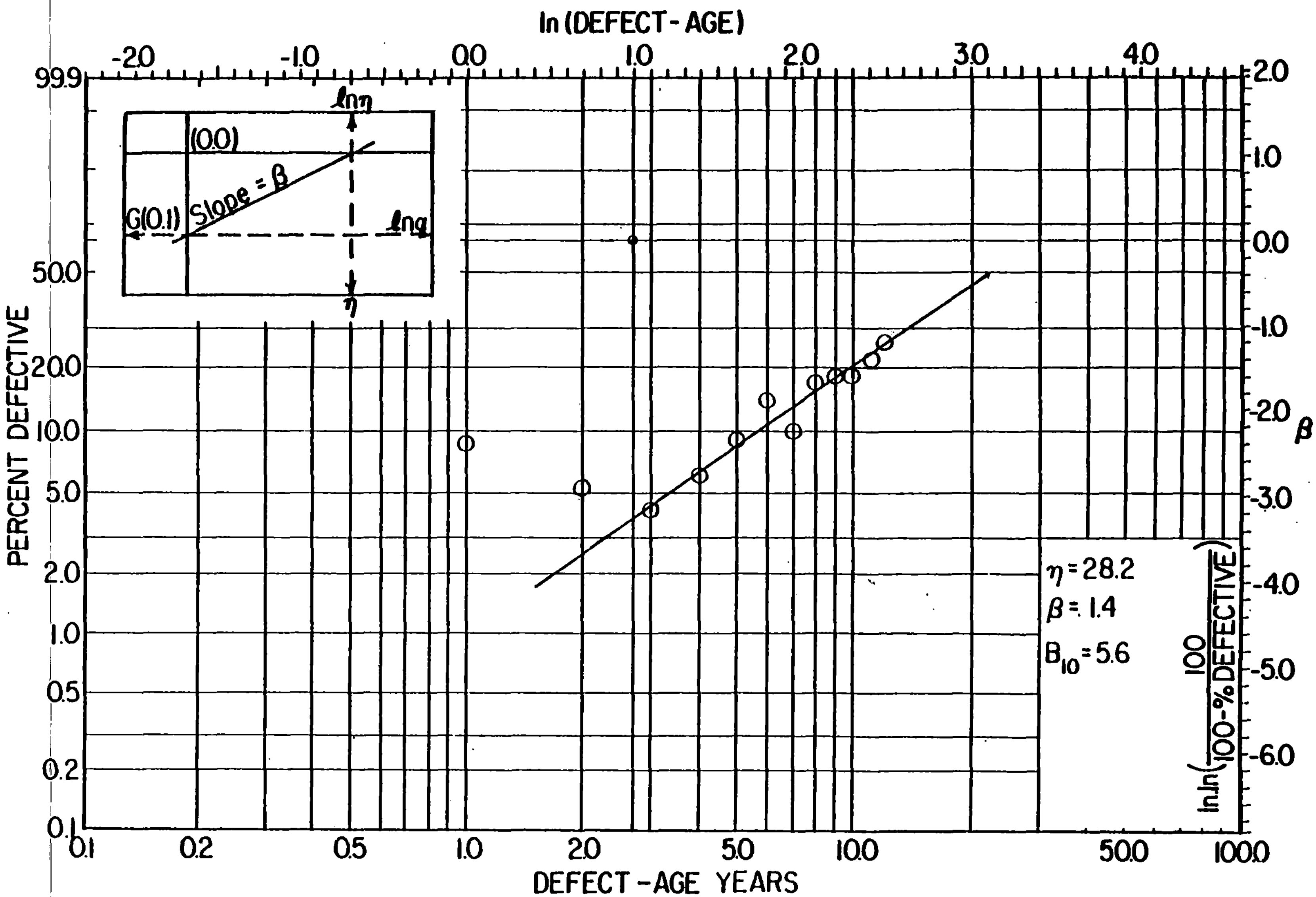


FIGURE A-20. PERCENT DEFECTIVE VERSUS DEFECT AGE - OVERSIZED, SPALLED AND BRINELLED CUPS

12	26.27	.304760413986 5
11	21.99	.2483331623877
10	18.687	.2068642806182
9	18.51	.2046898726559
8	17.22	.1889836996675
7	10.38	.1095916766301
6	14.236	.1535708479622
5	9.05	9.48602810E-02
4	6.08	6.27268299E-02
3	4.41	4.51019739E-02

ALPHA = 28.17257849094
BETA = 1.399417916822
B10 = 5.642187161743

REGRESSIØN TABLE

SØURCE	SUM ØF SQ.	DEG.FREEDØM	MEAN SQ.
REGRESSIØN	.3339640395481	1	.3339640395481
RESIDUAL	1.84213955E-02	8	2.30267443E-03
TØTAL	.35238543506	9	

F= 145.0331118866

CØEFF. ØF DETERMINATIØN= .947723731803
CØEFF. ØF CØRRELATIØN= .97351103322
STANDARD ERRØR ØF ESTIMATE= 4.79861900E-02

FIGURE A-21. WEIBULL REGRESSION RESULTS - OVERSIZED, SPALLED AND BRINELLED CUPS

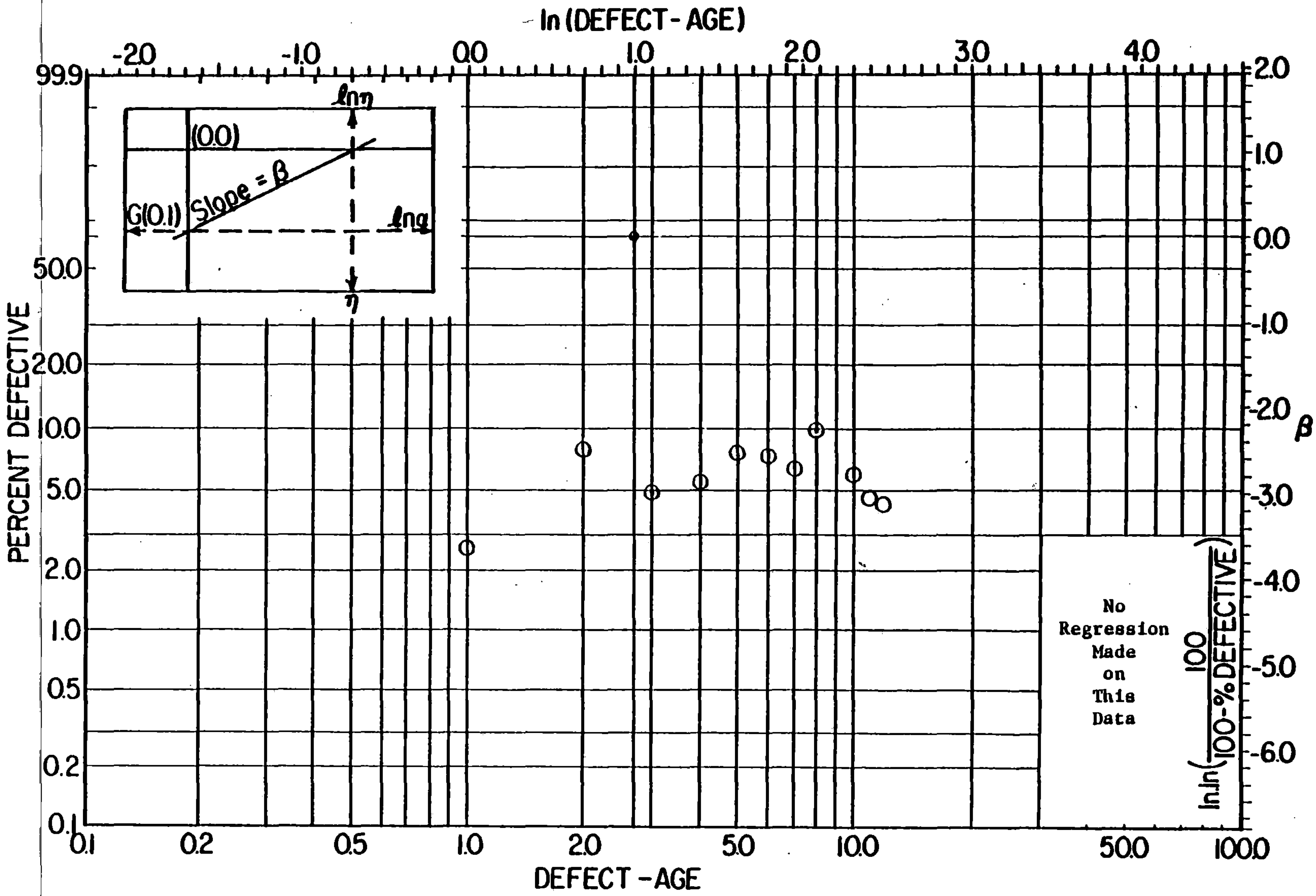


FIGURE A-22. PERCENT DEFECTIVE VERSUS DEFECT AGE - OTHER CUP DEFECTS

APPENDIX B

GLOSSARY OF SEAL DEFECTS

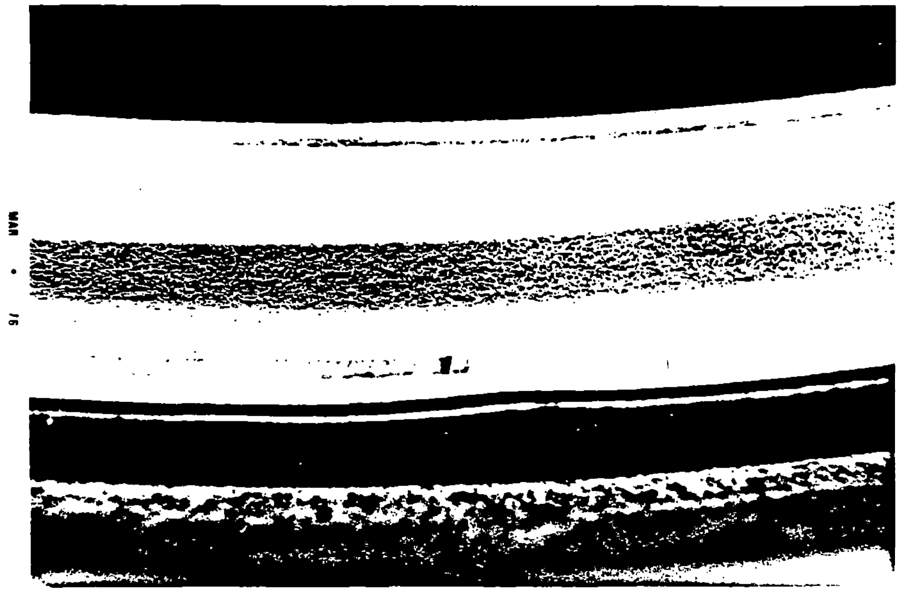


FIGURE B-1. TYPICAL WEAR PATTERN

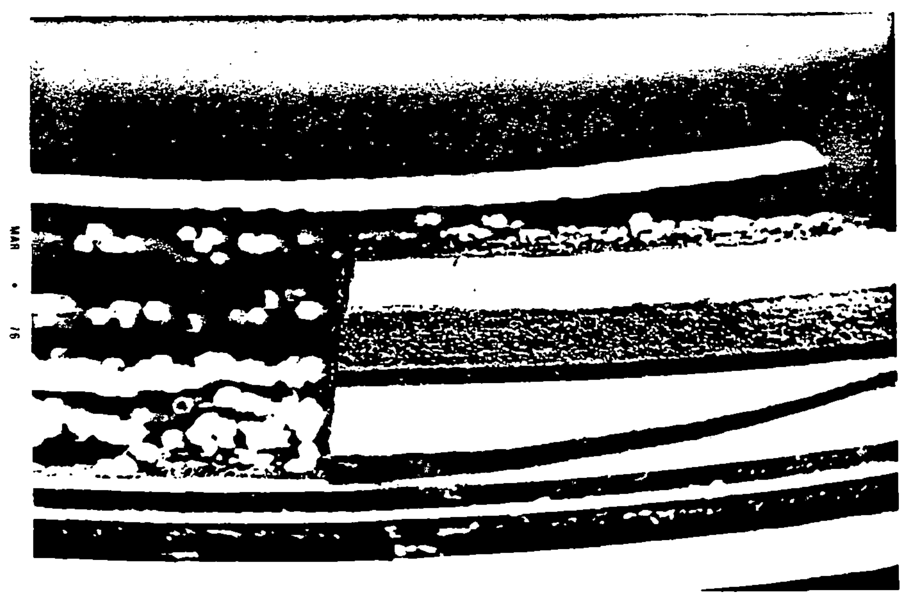


FIGURE B-2. BROKEN

FIGURE B-3. BLISTERED SEAL

FIGURE B-4. CRACKED SEAL

APPENDIX C

SEAL DEFECT DISTRIBUTIONS

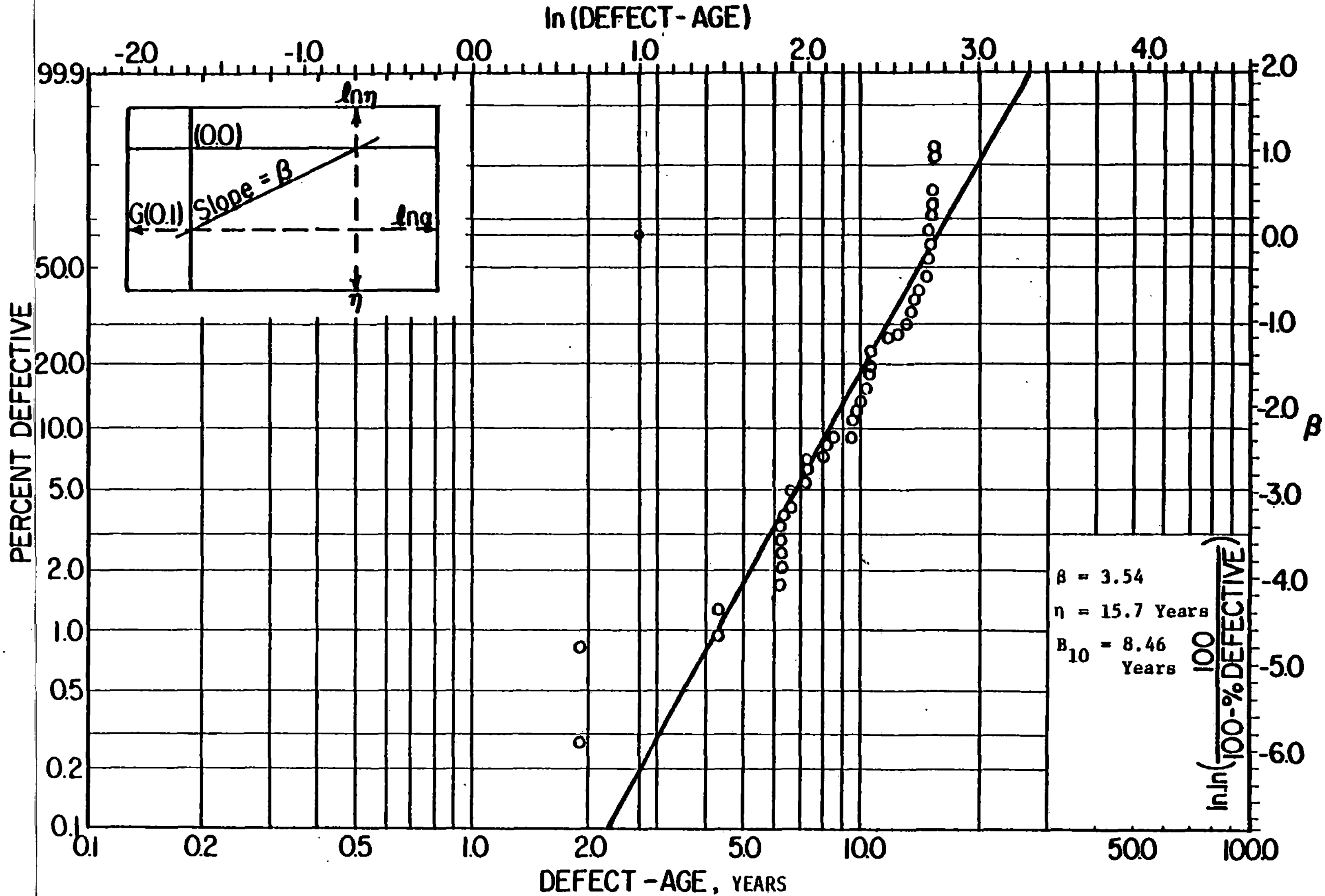


FIGURE C-1. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE, WEAR OF ALL SEALS

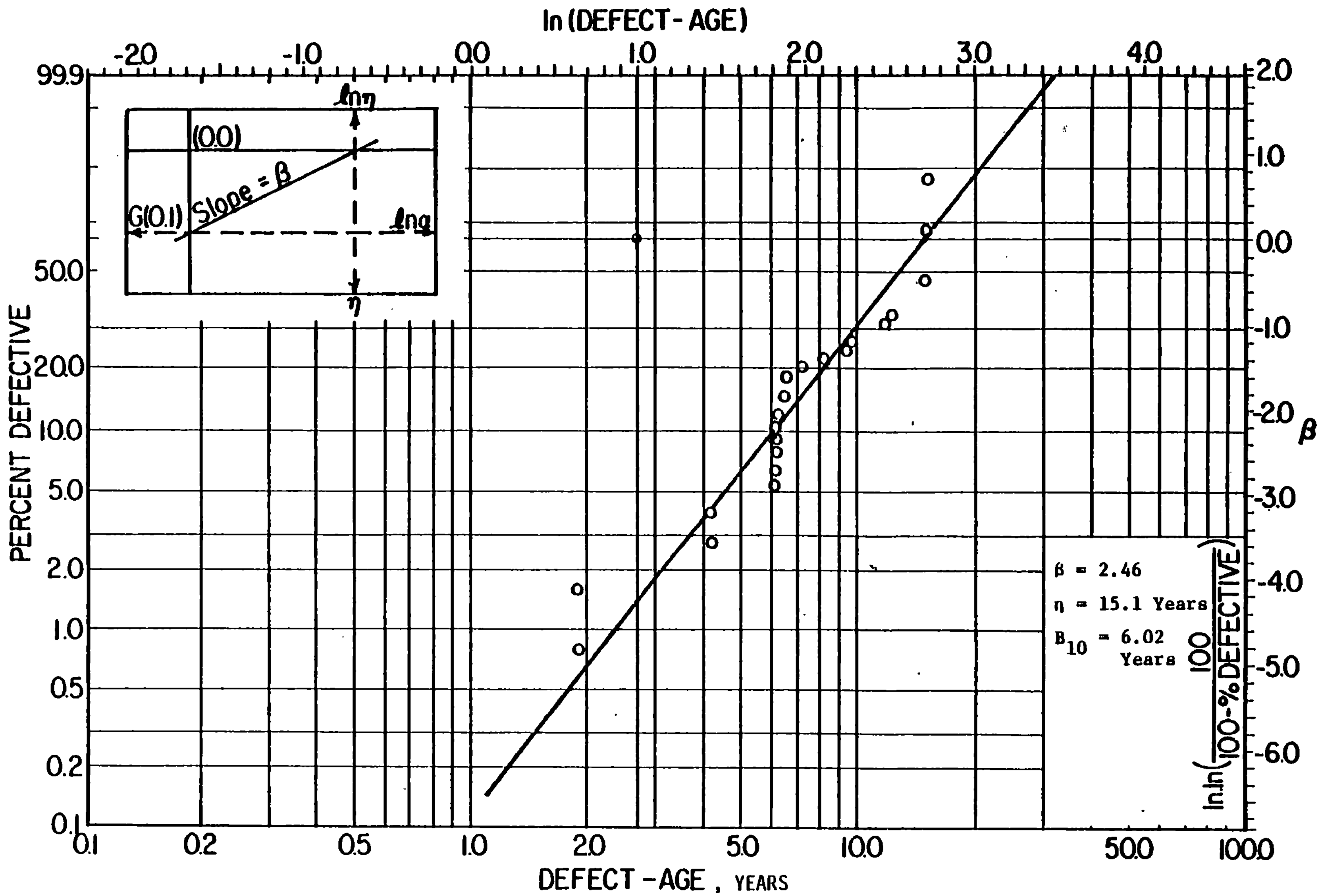


FIGURE C-2. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE, WEAR OF 6 x 11 BEARING SEAL

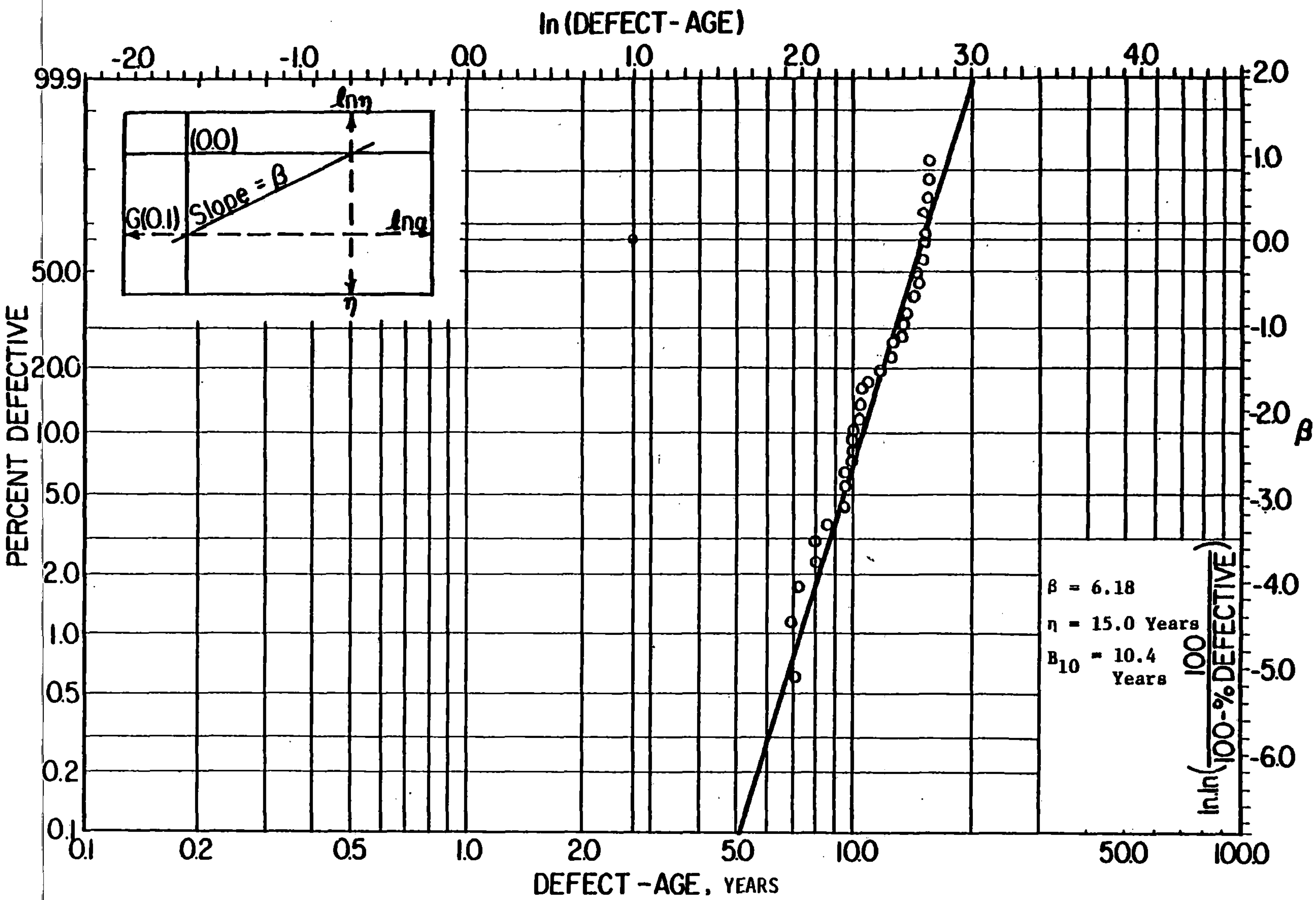


FIGURE C-3. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE; OTHER (BLISTERED, CRACKED, ETC.)

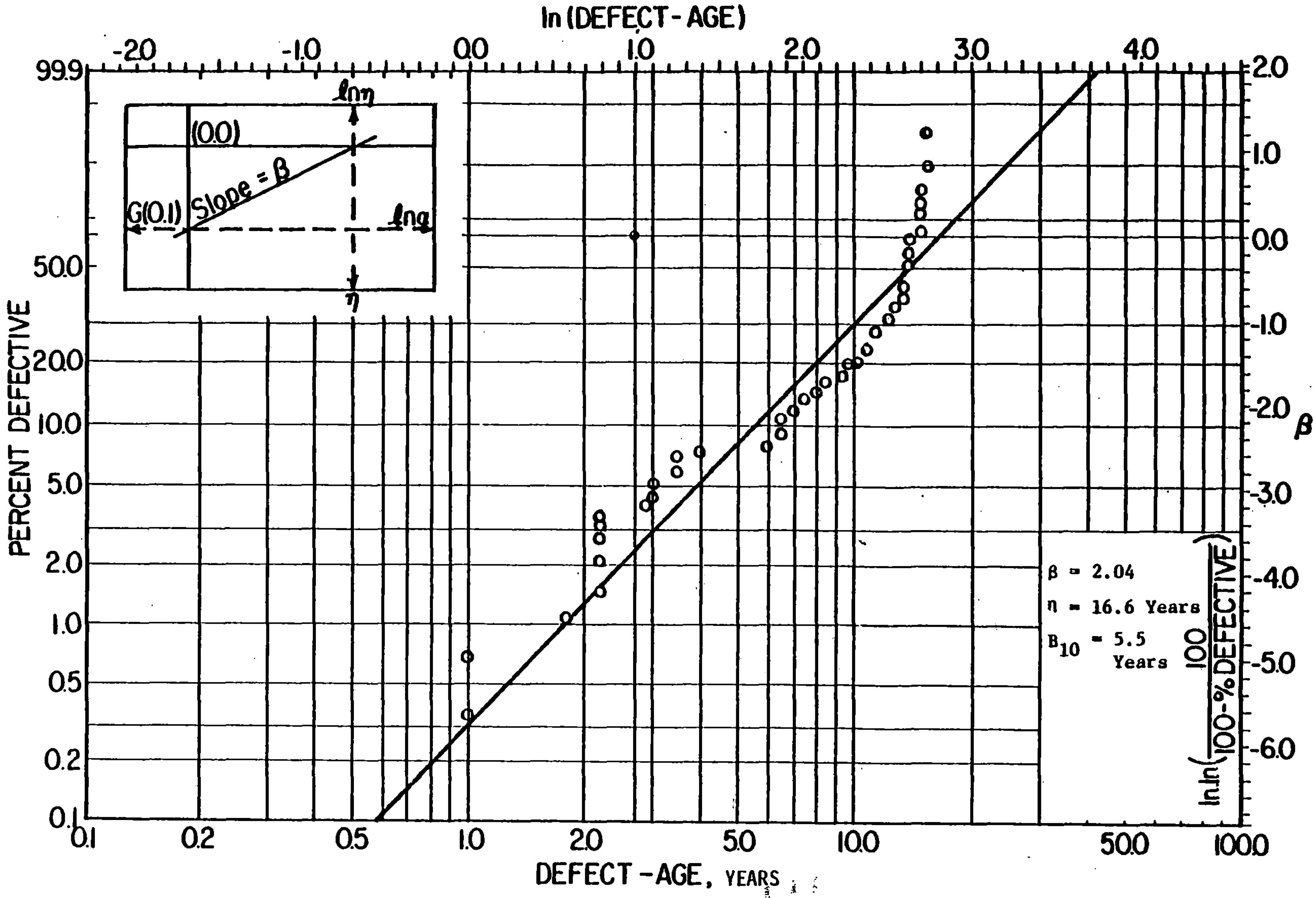


FIGURE C-4. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE, OVERSIZED OR UNDERSIZED DIAMETER

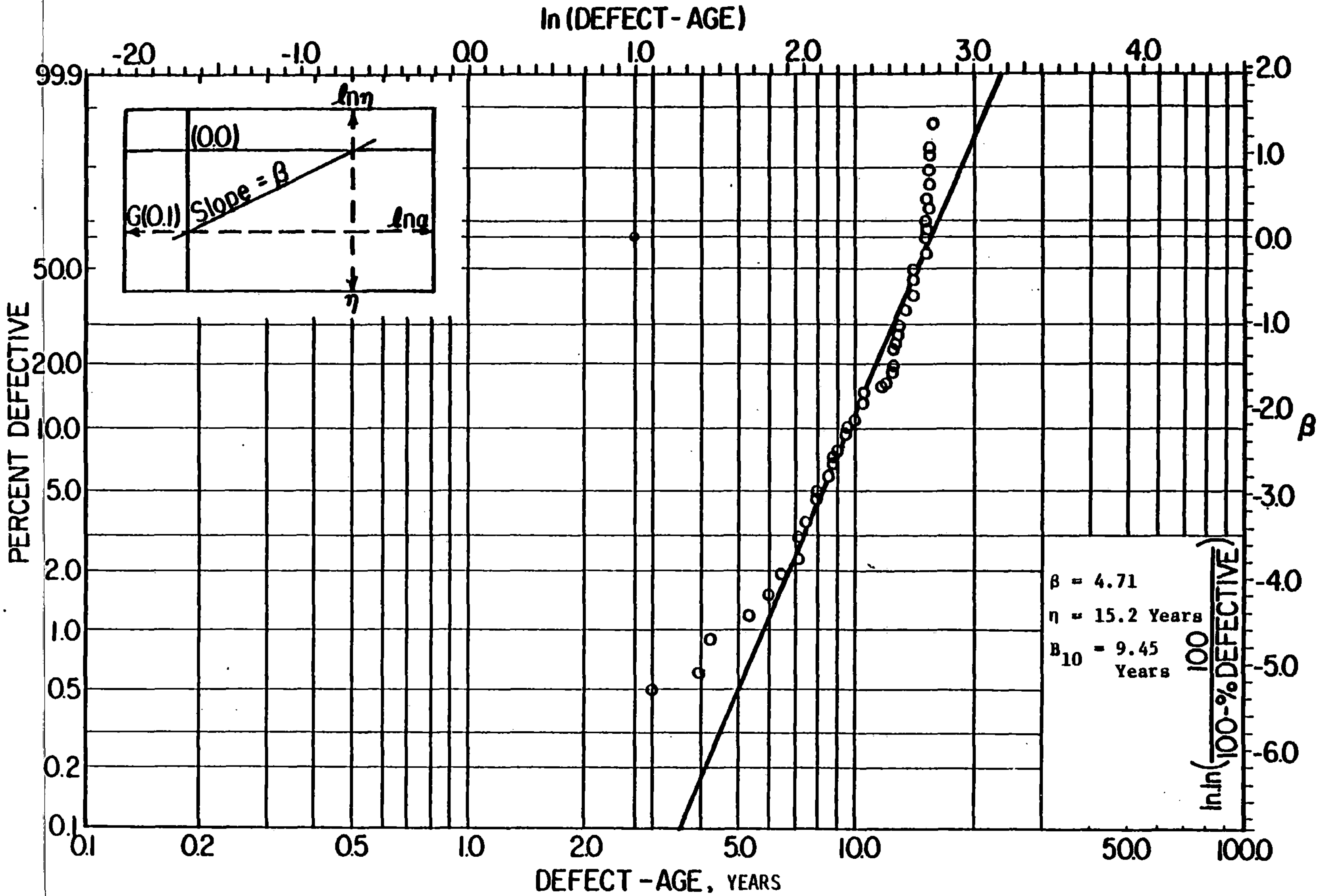


FIGURE C-5. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE, FIT

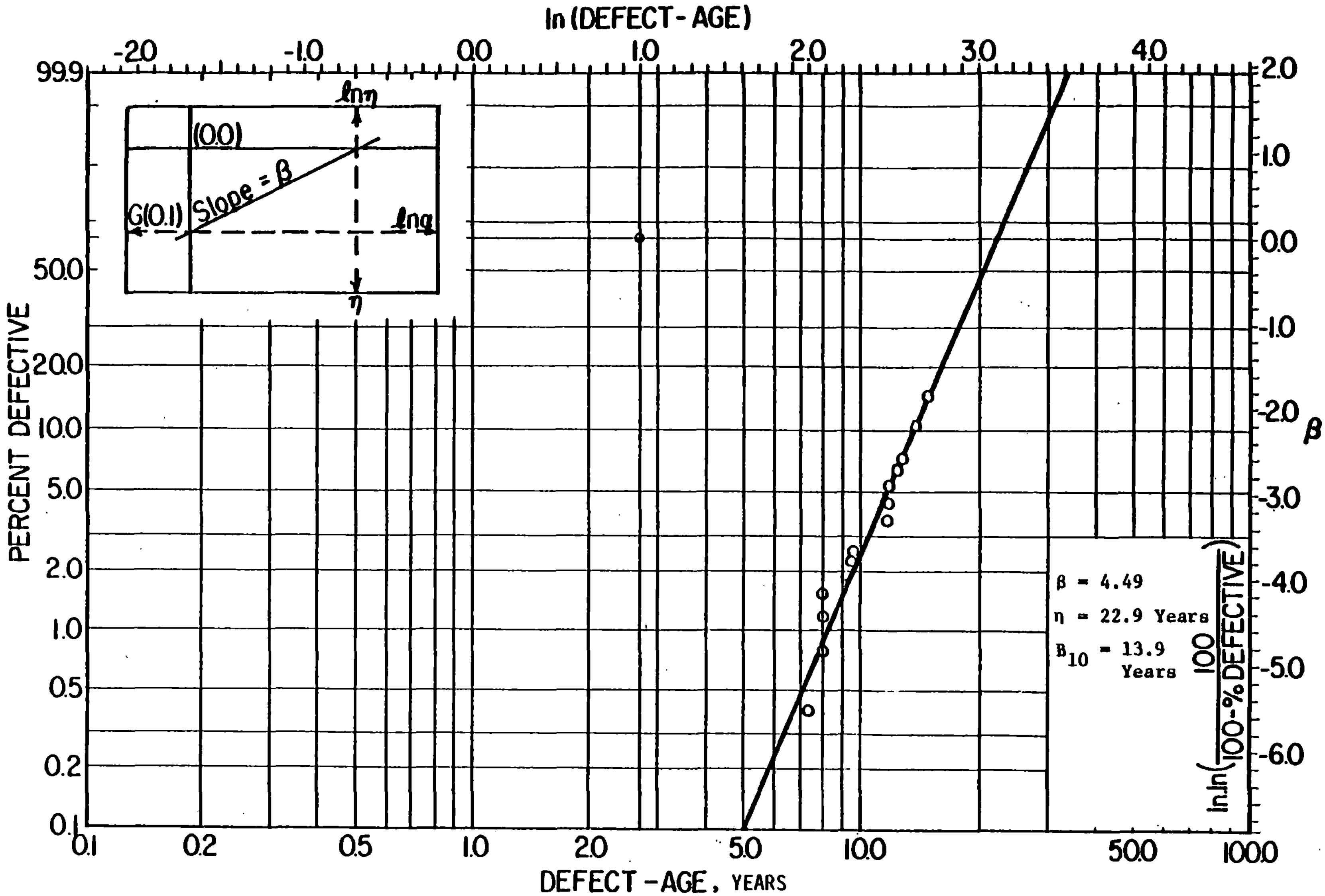


FIGURE C-6. CUMULATIVE PERCENT DEFECTIVE VERSUS AGE; OTHER (BLISTERED, CRACKED, ETC.)

APPENDIX D

WEIBULL DEFECT ANALYSIS

INTRODUCTION

The Weibull distribution was named after Wallodi Weibull, a Swedish engineer, due to his publication of applications of the distribution in strength of materials and the rupture of solids. His work became better known after 1951 when it was published in the Journal of Applied Mechanics (34). In the middle 1950's, Leiblein and Zelen (35) of the National Bureau of Standards used the Weibull distribution to describe the fatigue life of ball bearings. Later, Johnson (36) at the General Motors Research Laboratories successfully described wear and fatigue of rolling contacts with Weibull statistics.

Although Weibull statistics have been applied most frequently and successfully to fatigue data, Weibull's analysis has been applied as well to the distribution of mechanical strength and toughness as illustrated by Corton (37); thus, the applicability of Weibull analysis to brinelling, which is a strength - sensitive (i.e., hardness-sensitive) occurrence, is perhaps not too surprising. Furthermore, load spectra which are most commonly plotted on normal probability paper as percent exceedances or cumulative probability versus load can just as readily be plotted on Weibull paper. Bore growth appears to be a time-dependent process and is believed to be related to either metallurgical transformations and/or to the accumulation of cyclic (micro) plastic deformation in the bore. Many metallurgical transformations have been found to obev rate equations of the type proposed by Johnson and Mehl (38). Austin and Rickets (39) and Avrami (40). All of these rate equations are identical in basic form to the Weibull equation; thus, again the validity of Weibull statistics for bore growth may not be too surprising.

In recent years, there has been a large amount of intensive study given to the Weibull distribution resulting in a variety of mathematical derivations of the Weibull distribution which have, in turn, led to improved methods of parameter estimation. These developments have been summarized by Mann, Schafer, and Singpurwalla (41).

MATHEMATICS OF THE WEIBULL DISTRIBUTION

The Weibull equation has three parameters which make it a very general distribution capable of representing a wide variety of data. It has become particularly popular for representing defect data because of the many different shapes this distribution may assume. The simplest form of the cumulative distribution function, following Kao (42), is:

$$F(t) = 1 - \exp^{-\left(\frac{t-\gamma}{\alpha}\right)^{\beta}}, \tag{D-1}$$

where

t is time

α is the scale parameter

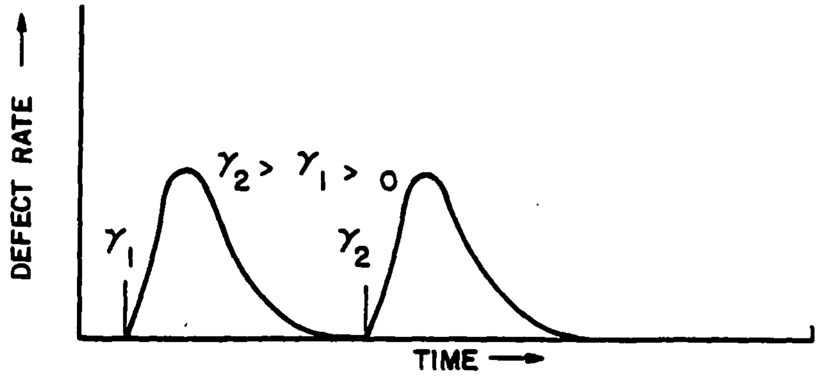
β is the shape parameter

γ is the location parameter.

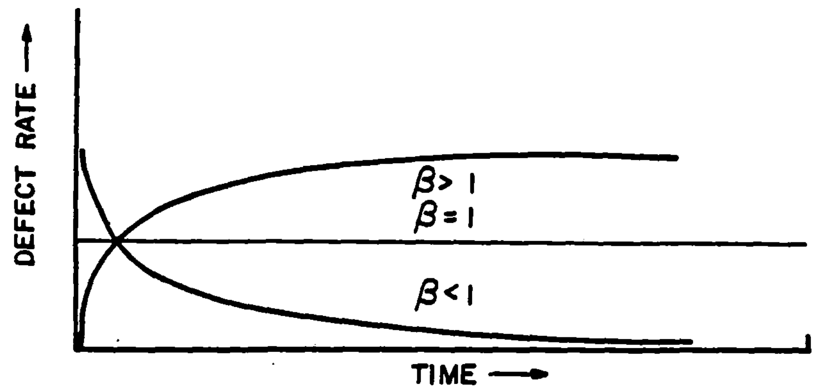
The location parameter, γ, has the effect of moving the distribution along the time base as shown in Figure D-1. When defects may be expected to begin as soon as an item is placed in use, then $\gamma = 0$. On the other hand, many items have, or at least are expected to have, some period of defect-free operation and/or use where γ is then some positive time after zero. This defect-free period is often called the guarantee or warranty period for such products as have been designed and manufactured with the intent of providing some satisfactory lifetime under a broad spectrum of use conditions.

The shape parameter, β, determines how the shape of the Weibull failure function varies with time. When $\beta>1$, we know that the defect rate is increasing with time; for $\beta=1$, the failure rate is uniform or constant over time, a condition also known as the exponential failure distribution; and for $\beta<1$, the defect rate is decreasing with time. These relationships are shown in Figure D-1(b).

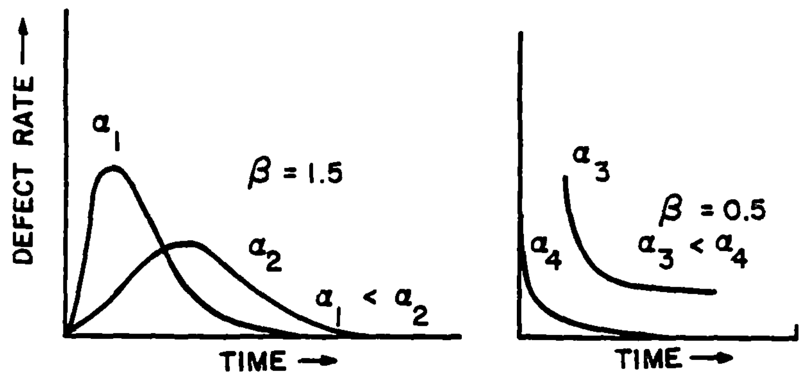
The scale parameter, α, also affects the shape of the Weibull distribution. For $\beta>1$, increasing values of α cause the defect rate curve to flatten out, while for $\beta<1$ increases in α also flatten out the defect rate curve for a fixed β as well as giving a relative defect rate which is always lower as shown in Figure D-1(c).



(a) EFFECT OF THE LOCATION PARAMETER



(b) EFFECTS OF THE SHAPE PARAMETER



(c) EFFECTS OF THE SCALE PARAMETER

FIGURE D-1. EFFECTS OF THE WEIBULL PARAMETERS

DESCRIPTION OF WEIBULL PROBABILITY PAPER

Weibull probability paper is derived from a rearrangement of Equation (D-1). By transposing, inverting, and letting $\alpha = \eta^{\beta}$, we obtain:

$$\frac{1}{1-F(t)} = \exp\left(\frac{t-\gamma}{\eta}\right)^{\beta}. \tag{D-2}$$

Now, taking logarithms twice on each side, we have:

$$\ln \ln \frac{1}{1-F(t)} = \beta \ln(t-\gamma) - \beta \ln \eta, \tag{D-3}$$

which has the linear form: $y = mx + b$.

The values of $y = \ln \ln[1/(1-F(t)]$ yield the right-hand vertical scale on Weibull paper, Figure D-2. The actual values are multiplied by 100 to convert to percentages. The values of $\ln(t - \gamma)$ give the top horizontal scale. However, to facilitate plotting of data, direct entry scales are provided. The left-hand vertical axis is scaled in cumulative percent, $F(t)$. The bottom horizontal axis allows direct plotting of the observed times to failure.

Other features of Weibull paper are the principal ordinate, running vertically from 0.0 on the top scale to 1 on the bottom scale; and the principal abscissa, running from 0.0 on the right-hand scale to 63.2% on the left scale.

There is also a circled cross mark at the intersection of the principal ordinate and abscissa. To estimate the value of β, an auxiliary line is drawn parallel to the Weibull line passing through the circled cross mark, and down to the left until it intersects the principal ordinate. The point of intersection is then projected to the right-hand scale and the value of the intercept there is the estimate, $\hat{\beta}$, for that Weibull line.

The value of the B_{10} and characteristic lives can be read directly from the bottom scale.

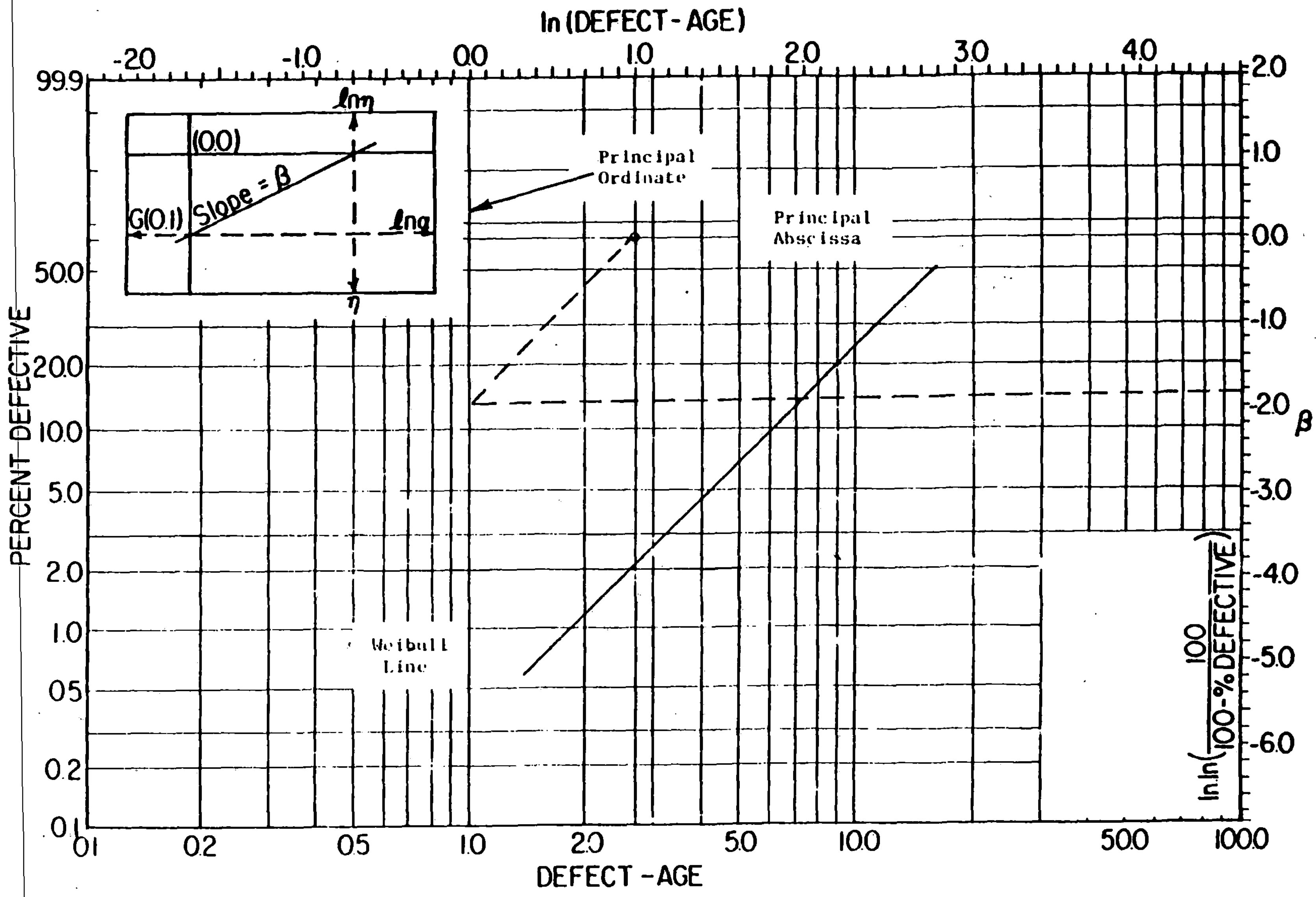


FIGURE D-2. LAYOUT OF WEIBULL PROBABILITY PAPER

THE METHOD OF LEAST SQUARES ON TRANSFORMED DATA

There are several ways (41) to estimate the shape and scale parameters of a Weibull distribution from a set of life test data. One of the simplest approaches is to use the method of least squares on the transformed data. It has already been shown in Equation D-3 that the Weibull distribution can be linearized by twice taking the logarithm of Equation D-2. In Equation D-3 the dependent variable, y, is given by:

$$y = \ln\ln \frac{1}{1-F(t)},$$

and the independent variable x by:

$$x = \ln (t-\gamma).$$

Equation D-3 is most appropriate if we are interested in predicting the fraction defective for a given bearing age.

In the present study, however, we are more interested in predicting life, L_{10} and η, for a given fraction defective, 10% and 63.2%, respectively. To do this, we rearrange Equation D-3 to give:

$$\ln(t-\gamma) = \frac{1}{\beta} \ln\ln \left[\frac{1}{1 - F(t)}\right] + \ln\eta, \tag{D-4}$$

which also has the form:

$$y = mx + b,$$

but with:

$$\begin{aligned} y &= \ln(t-\gamma) \\ m &= 1/\beta \\ x &= \ln\ln \left[\frac{1}{1 - F(t)}\right] \\ b &= \ln\eta. \end{aligned}$$

Using the standard least squares method, we have regressed $\ln(t-\gamma)$ on $\ln\ln \frac{1}{1 - F(t)}$ to determine the values of B_{10}, η, and β presented in this report.

It should be recognized that the regression of $\ln\ln \frac{1}{1 - F(t)}$ on $\ln(t-\gamma)$ will in general lead to different values of B_{10}, η, and β than those obtained by the regression of $\ln(t-\gamma)$ on $\ln\ln \frac{1}{1 - F(t)}$.

Tables D-1 and D-2 are comparisons of the cone and cup Weibull parameters calculated by the two different regressions. In most cases, the estimated values of B_{10} are within 10 percent of one another. The difference between the two estimates of the characteristic lives is much larger than the difference between estimates for the B_{10} lives. This is a result, not so much of the regression, as of the fact that we are extrapolating outside the range of the data, i.e. from 12 years to 30-436 years.

In general, the difference between the two estimates of the Weibull parameters is a measure of the scatter in the raw data.

TABLE D-1

COMPARISON OF CONE WEIBULL PARAMETERS OBTAINED BY TWO DIFFERENT REGRESSIONS

	Regression of $\ell n(t-\gamma)$ on $\ell n \ell n \frac{1}{1-F(t)}$			Regression of $\ell n \ell n \frac{1}{1-F(t)}$ on $\ell n(t-\gamma)$		
	B_{10} Years	η Years	β	B_{10} Years	η Years	β
All Defects	5.4	53.7	0.97	5.1	77.3	0.83
Spalling	30.6	145.1	1.45	52.4	435.5	1.06
Oversize	9.2	52.6	1.29	9.9	86.6	1.04
Brinelling	15.4	50.6	1.88	13.6	45.1	1.91

TABLE D-2

COMPARISON OF CUP WEIBULL PARAMETERS OBTAINED BY TWO DIFFERENT REGRESSIONS

	Regression of $\ell n(t-\gamma)$ on $\ell n \ell n \frac{1}{1-F(t)}$			Regression of $\ell n \ell n \frac{1}{1-F(t)}$ on $\ell n(t-\gamma)$		
	B_{10} Years	η Years	β	B_{10} Years	η Years	β
All Defects	3.3	30.6	1.01	3.0	36.2	0.91
Spalling	12.8	42.5	1.88	13.8	51.8	1.71
Oversize	14.1	55.6	1.64	17.6	105.6	1.26
Brinelling	9.8	47.0	1.44	10.4	65.4	1.22

APPENDIX E

BEARING POPULATION CHARACTERISTICS

In this appendix, we will derive some general observations about a population of components which have been introduced into the market place at some time $(t = 0)$. The size of the population is increased by sales, $S(t)$, each year and decreased by failures, $G(t)$, each year. The population will have two phases of interest -- a growth phase where the sales will exceed the failure rate, and a steady-state phase where the sales will equal the failure rate.

In the growth phase, the size of the population of any time, t, can be described by the following:

t	Sales	Population Size
0	$\dot{S}(0)\ \Delta t_0$	$(\dot{S}(0)\ \Delta t_0)R(0)$
1	$\dot{S}(1)\ \Delta t_1$	$(\dot{S}(0)\ \Delta t_0)R(1) + \dot{S}(1)\ \Delta t_1\ R(0)$
2	$\dot{S}(2)\ \Delta t_2$	$(\dot{S}(0)\ \Delta t_0)R(2) + \dot{S}(1)\ \Delta t_1\ R(1) + \dot{S}(2)\ \Delta t_2\ R(0)$
3		
.		
.		
.		
t	$\dot{S}(t)\ \Delta t_t$	$(S(0)\ \Delta t_0)R(t) + S(1)\ \Delta t_1\ R(t-1) + \ldots + S(3)\ \Delta t_1 R(t-2) + \dot{S}(t)\ \Delta t_t\ R(0)$

The size of the population at the th year is simply

$$P_t^* = \sum_{i=0}^{t} \dot{S}_i\ R(t-1)\ \Delta t_i$$

As Δt_i becomes small, the population size can be expressed as:

$$P_t^* = \int_0^t \dot{S}(t)\ R(t-\tau)\ d\tau.$$

For the special case where $\dot{S}(t)$ = constant and the failure distribution is exponential, integration gives:

$$P_t^* = \dot{S}\ \delta(1 - e^{-t/\delta}).$$

This equation merely states that the population will reach an asymptote given by $\dot{S}$ as t becomes large. For the Weibull distribution, the asymptote is $\dot{S}\mu$; where μ is the mean life.

The failure rate in any year can be expressed by

$$G^*(t) = \dot{S}(0)R(t)\ \frac{R(t) - R(t+1)}{R(t)} + \dot{S}(1)\ R(t-1)\ \frac{R(t-1) - R(t)}{R(t-1)}$$

$$+ \dot{S}(t)\ R(0)\ \frac{R(0) - R(1)}{R(0)}.$$

For $\dot{S}$ = constant, this simply becomes:

$$G^*(t) = \dot{S}\{R(0) - R(t+1)\},$$

and for the Weibull distribution:

$$G^*(t) = \dot{S}\{1 - e^{-(t/\eta)^{\beta}}\}.$$

For $\beta < 1.0$, the failure rate rises more quickly in the early years and approaches the replacement rate gradually. However, for $\beta > 1$, the failure rate will remain low for quite some time and will suddenly rise quite rapidly to the replacement rate. The latter behavior will be typical of roller bearings.

The average age of the population is given by:

$$\bar{t}^* = \frac{\int_0^t \tau\ \dot{S}(\tau)\ R(t-\tau)\ d\tau}{\int_0^t \dot{S}(\tau)\ R(t-\tau)\ d\tau}.$$

For $\dot{S}$ = constant and an exponential failure rate:

$$\bar{t}^* = \frac{\delta\{1 - e^{-t/\delta}(t/\delta + 1)}{1 - e^{-t/\delta}}.$$

As t becomes large, the average age of the population approaches the characteristic life of the population.

Once the population age becomes greater than the characteristic life, the population size will level out. The population at any year n will be:

$$P(t) = \dot{S}(0)\ \Delta t_o\ R(t) + \dot{S}(1)\ \Delta t_1\ R(t-1)\ \ldots + \dot{S}(t)\ R(0).$$

Dividing by P(t):

$$1 = \frac{\dot{S}(0)\ \Delta t_0\ R(t)}{P(t)} + \frac{\dot{S}(1)\ \Delta t_1\ R(t-1)}{P(t)}\ \ldots + \frac{\dot{S}(t)\ R(0)}{P(t)}$$

it is seen that $\frac{\dot{S}_i\ R(t-i)}{P(t)}$, $i = 0, 1, \ldots t$

is simply the probability density function of the population at age t. It can also be expressed as:

$$f(\tau) = \frac{\dot{S}(\tau)\ R(t-\tau)}{P(t)}.$$

For a constant sales rate and an exponential distribution:

$$f^*(\tau) = \frac{1\ e^{-\{(t-\tau)/\delta\}}}{\delta\ (1 - e^{-t/\delta})}$$

The cumulative distribution function

$$F^*(\tau) = \int_0^{\tau} \frac{\dot{S}(\tau)\ R(t-\tau)\ d\tau}{P(t)}$$

and in the special case for $\dot{S}$ = constant and an exponential distribution

$$F^*(\tau) = \frac{(1 - e^{-\tau/\delta})}{(1 - e^{-t/\delta})}$$

We are also interested in the instantaneous failure rate of the population, which is the proportion of the total population which will fail in small increment time Δt at time t. The absolute number which will fail at year t is given by:

$$h^*(t)\ P(t)\ \Delta t_t = \dot{S}(0)\ \Delta t_0\ R(t)\ h(t)\ \Delta t_t + \dot{S}(1)\ \Delta t_1\ R(t-1)\ h(t-1) + \ldots$$

$$+ \dot{S}(t)\ \Delta t_1\ R(0)\ h(0)\ \Delta t_t$$

$$h^*(t) = \sum_{i=0}^{t} \frac{\dot{S}_1\ \Delta t_i\ R(t-i)\ h(t-i)}{P(t)},$$

and for t small:

$$h^*(t) = \int_0^t \frac{S(\tau)\ R(t-\tau)\ h(t-\tau)\ d\tau}{P(t)},$$

which for a constant sales rate and an exponential distribution is simply:

$$h^*(t) = \frac{1}{\delta}.$$

This conclusion means that for an exponential failure rate the instantaneous failure rate of the total population is a constant and is the same as the instantaneous failure of each segment of the population introduced at year 0, 1, 2, ... t and still present in the total population at year t.

In the case where the failure distribution is a Weibull and S = constant:

$$h^*(t) = \frac{\int_0^t e^{-\left(\frac{t-\tau}{\eta}\right)^\beta} \beta/\eta^\beta (t-\tau)^{\beta-1} d\tau}{\int_0^t e^{-\left(\frac{t-\tau}{\eta}\right)^\beta} d\tau},$$

or

$$h^*(t) = \frac{(1 - e^{-(t/\eta)^\beta})}{\int_0^t e^{-(\frac{t-\tau}{\eta})^\beta} d\tau}.$$

If $t \to \infty$, then $h^*(t)$ approaches a limit:

$$h^*(t) = \frac{\beta}{\eta\Gamma(\frac{1}{\beta})} = \frac{1}{\mu}.$$

The fact that the population hazard rate approaches a constant as the population matures is an important conclusion. Further, the limiting hazard rate is a function of only two parameters: the component characteristic life and the Weibull slope. The roller bearing industry is still in a growth phase and the hazard rate is rising. As the total population stabilizes, the hazard rate will approach the asymptote, $1/\mu$.

If the roller bearing population should start to decline sometime in the future, the hazard rate will again rise. However, this possibility is not an immediate concern.

APPENDIX F

REPORT OF NEW TECHNOLOGY

Essentially all known basic diagnostic techniques applicable to the roller bearing failure problem were reviewed, and several were identified as potentially feasible. Although conceptual adaptations of existing techniques were identified, no inventions appear to have resulted from this work.

However, this study did result in an improvement in knowledge about the experimental feasibility vibration based diagnostic approaches. Included in this work was an actual demonstration of a relatively simple diagnostic system in an actual railroad wheel shop. With minimal future development, such a system could be widely deployed to test bearings for gross defects without removing them from the axle. The experimental work is covered in Section 3.

Section 4 described an improved mathematical model developed to perform cost-benefit analyses of innovative railcar roller bearing diagnostic approaches and procedural (operational) modifications.