REPORT NO. DOT-TSC-FRA-73-6

SURVEY OF FREIGHT CAR ROLLER BEARING REQUIREMENTS AND FAILURE MODES

John W. Lyons

JULY 1973
PRELIMINARY MEMORANDUM

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16. Abstract

AAR roller bearing requirements and interchange rules are presented and reviewed; also included and reviewed are rules covering adapters and grease for freight car bearings. Bearing fatigue theory, methods of fatique life calculations, and characteristic fatigue damage of sub-assembly components are reviewed. The results of a limited survey of bearing manufacturers, AAR records, and a railroad bearing repair shop to determine failure modes and identify critical sub-assembly components are presented, including a method for determining bearing assembly life from the lives of the components. The results of the study to date indicate that two major causes of bearing failures are worn adapters and loss of lubricant.

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PREFACE

The work discussed in this report was performed as part of an overall program at the Transportation Systems Center to provide necessary technology for greater reliability of railroad roller bearings, and to develop in-service inspection methods and criteria for this critical item. The program is sponsored by the Federal Railroad Administration, Office of Research, Development and Demonstrations. The program supports government activities designed to promote greater safety in railroad freight and passenger service.

This study was carried out by members of the Mechanical Engineering Division of the Transportation Systems Center. Major contributions were made by J. Lyons, R. Beatty and A. Lavery. The authors would like to acknowledge the information and assistance provided by the AAR regarding rules and regulations and by the Timken Bearing Company on the technical aspects of roller bearings.

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1. SUMMARY

AAR roller bearing and adapter requirements and interchange service rules are presented and reviewed; also included are bearing grease requirements and tests. Bearing fatigue theory, methods of fatigue life calculations, and fatigue damage of subassembly components are reviewed. The results of an initial survey to determine bearing failure modes and identify critical subassembly components are presented including a statistical method for determining bearing assembly life from the two-parameter Weibull life distributions of the components. Due to the scarcity of fatigue data on railroad roller bearings, this method could not be tested or applied to railroad roller bearings.

The survey results indicate that lubricant loss and worn adapters are major causes for bearing failures. Worn adapters can contact and thus loosen or dislodge seals from the bearing outer ring which permits the lubricant to be worked out of the bearing. In addition, worn adapters result in uneven loading of the bearing which enhances fatigue failure. Except in the case of hardened crown adapters, specification hardness requirements may be too low for optimum wear characteristics. However, the consequences of specifying higher hardnesses to promote longer adapter wear life have to be investigated since harder adapters may cause excessive wear in the side frame pedestal area.

2. INTRODUCTION

In 1954 the Association of American Railroads (AAR) permitted freight cars to be equipped with roller bearings to stem a large and increasing yearly number of car setouts due to hot boxes or journal failures occurring with the plain journal bearing equipped freight cars which typified the railroad industry at that time. Twelve years later in 1966, the AAR specified that all new and rebuilt cars of 100 ton capacity must be equipped with roller bearings. Since 1968, it has been mandatory that all new cars, and since 1970 that all rebuilt cars, be equipped with roller bearings. With the adoption of roller bearings, the number of freight car setouts per million car miles due to hot boxes has decreased from a high of 5.48 in 1957 to a low of 0.46 in 1971.

Figure 1 shows the increase in the percentage of cars equipped with roller bearings in the total railroad freight car fleet for the period 1 Jan. 1967 till 30 June 1972, at which time it is reported that 760,485 railroad and privately owned cars were equipped with roller bearings. On 1 Jan. 1973, it was estimated that more than 46 percent of the total fleet is equipped with roller bearings.

Figure 1 also shows the total fleet percentages of hot box setouts on roller bearing and plain bearing cars. Plain bearings account for about 91% of all hot box setouts as of the end of 1972, whereas roller bearings account for only 9%. It should be noted, however, that from January 1969 until about June 1972 hot boxes with roller bearings have increased essentially linearly from 2 to about 10%. The same trend in terms of the reduction of car miles (in million miles) per setout is shown in Figure 2 for the same period. This trend was noted in a recent FRA Office of Safety Report[1] in which the conclusions indicated (1) that roller

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[1] Journal Failure Report, October 1972, FRA office of Safety, Engineering and Accident Division

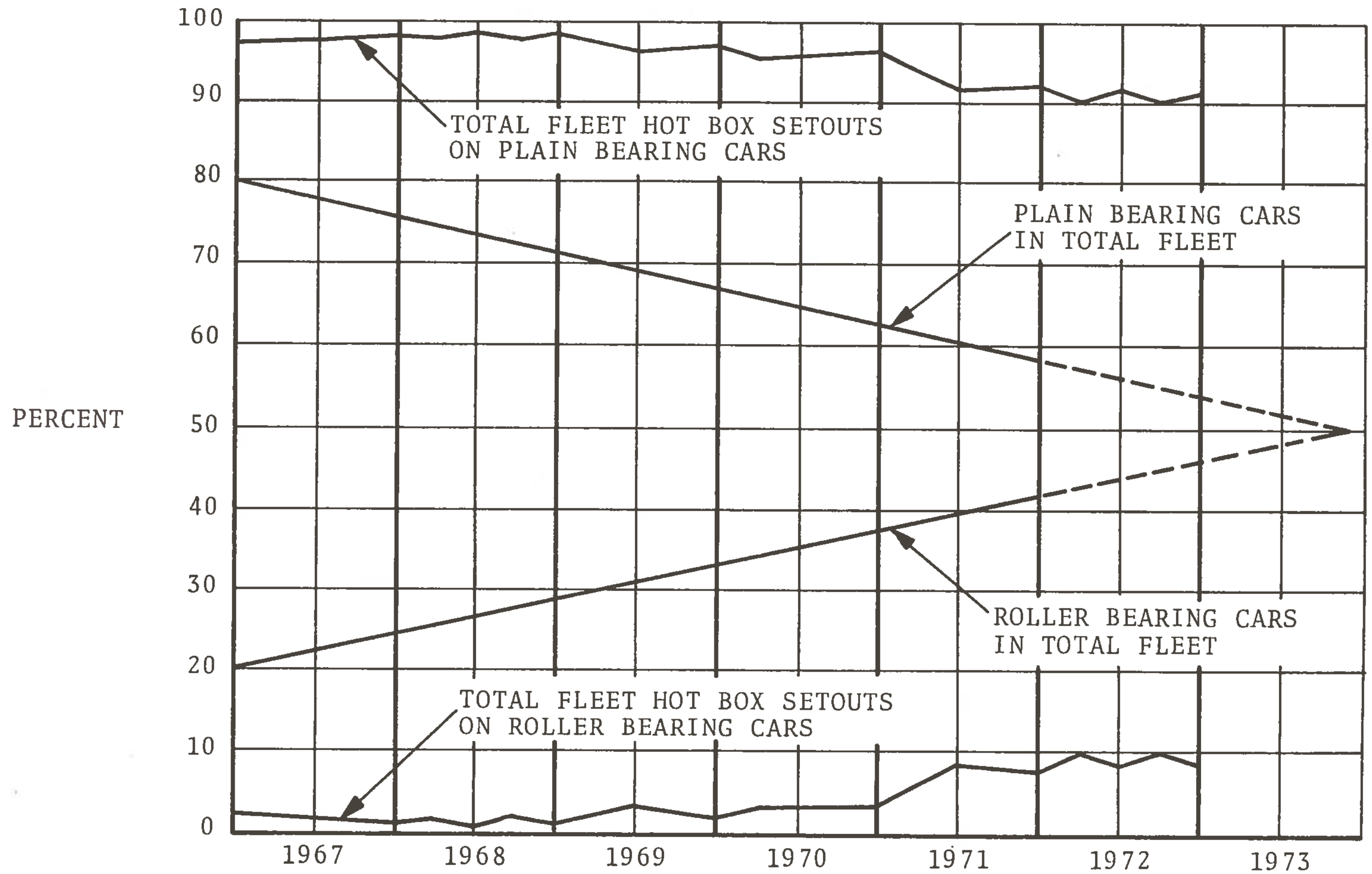

Figure 1 Percentages of Plain and Roller Bearing Cars and Hot Box Setouts in Total Freight Car Fleet

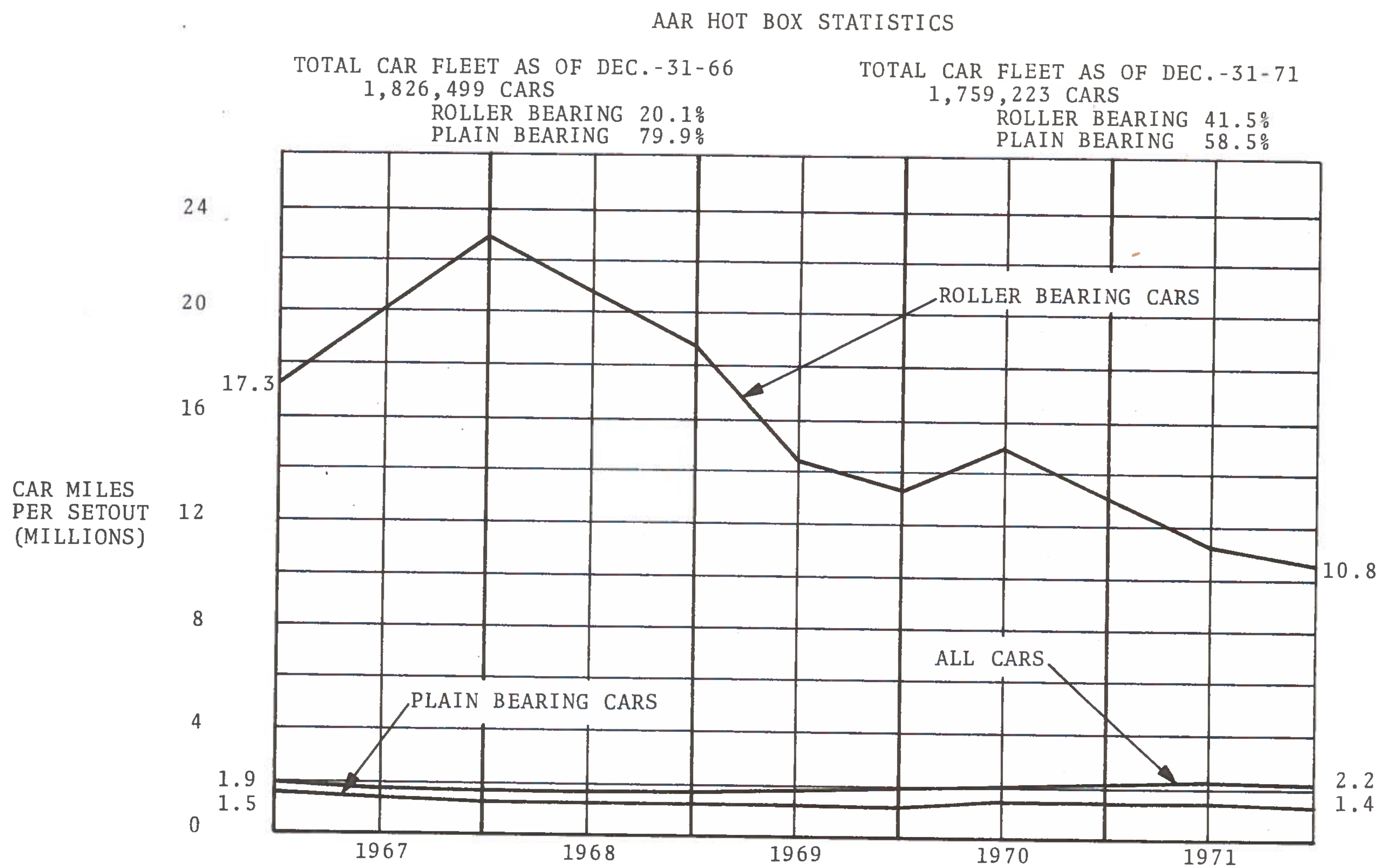

Figure 2 Car Miles (In Millions) Per Setout for Roller and Plain Bearing Cars for Period 1967 Thru 1971

bearings become less effective and more likely to heat with age, and (2) that the normal span of roller bearing life is about 10 years.

Considering that about 80% of all roller bearings in the freight car fleet are less than 10 years old and that the percentage of roller bearing hot boxes has increased about 2% in each of the past four years, a study of this critical truck component was initiated. The foremost goals in mind are improved in-service nondestructive inspection methods or improved reliability of critical subassembly components that fail and trigger failure of the whole bearing assembly.

In carrying out this study, it was felt that the best approach to the problem would be: 1) the collection of statistical data on bearing failure modes and critical subassembly bearing components; 2) the development or selection of a statistical method applicable to bearings permitting determination of the reliability levels of critical sub components; and 3) using the data and method above to identify the low reliability components. Once the weak links were identified, inspection procedures could be developed or component deficiencies eliminated which should lead to improved bearing performance. This interim report essentially covers the results obtained following this approach, along with other technical and specification information required for a proper understanding of railroad freight car roller bearings and the associated problems.

3. RULES COVERING ROLLER BEARINGS OPERATING IN INTERCHANGE SERVICE[1]

When cars equipped with roller bearing units are involved in derailment, wreck, fire, high water, or other similar incidents that could possibly damage roller bearings, the following applies:

> The wheel assemblies must be removed from the car or cars, assemblies inspected in accordance with the Wheel and Axle Manual, and the roller bearings dismantled and examined for defects in accordance with the Roller Bearing Manual. However, in event of minor yard derailments, where derailed wheels on empty cars have not moved more than one car length and where visual inspection reveals no evidence of distress to the bearing or adapter, no further examination of the bearing is required.

When cars equipped with roller bearing units are involved in a fire, the heat may be sufficient to temper the bearing rings and/or destroy the lubricant and/or seals. Special examination is required in such instances. Similar damage may also occur in car thawing operations, particularly where open flames are employed. Wheel assemblies are to be inspected in accordance with Wheel and Axle Manual and roller bearings dismantled and examined for defects in accordance with the Roller Bearing Manual with special attention being given to determine if excessive heat has affected any of the bearing parts.

Roller bearings used in interchange service are vulnerable to water. It affects the efficiency of the lubricant, and also results in water etching of the bearing parts. Therefore, when roller bearings are operated in a high water condition that results in water churning up around the seals, moisture will enter the interior of the bearing. When this occurs, wheels must be removed as soon as possible. Wheel assemblies must be inspected in accordance with Wheel and Axle Manual and roller bearings dismantled and examined for defects in accordance with the Roller Bearing Manual.

[1]AAR Roller Bearing Manual

Electrical current must never be passed through roller bearings as it may cause arcing within the bearing resulting in damage. All welding must be done with ground cable attached so that the circuit formed will not allow electrical current to flow through roller bearings.

A heating or cutting torch, when used around roller bearings, must never have heat directed on any portion of the roller bearing assembly. Care should be taken that cutting fragments are not directed toward the roller bearing assembly. A heating torch must never be used to heat lubricant fitting or plug for removal.

When bearing failure occurs due to overheating,* the complete roller bearing wheel assembly, with adapter at overheated bearing, must be properly identified giving car number and initial, date and location failed, lubrication date, detector millimeter reading, if available, and it should be shipped to the point designated by the respective railroad to determine the cause of failure. The adapter should be marked to indicate which end was inboard. The bearing manufacturer should be requested to send a service representative to make a joint inspection with a railroad representative to determine the cause of failure.

Plating, metal spray or any other method of building up bearing parts is prohibited. Abrasive cleaning materials (sandblasting, grit blasting, etc.) must never be used to clean outer rings, rollers, inner rings, thrust carrying end caps and backing rings and seals. However, external surfaces of outer rings and other parts subject to corrosion and paint accumulation may be cleaned by use of abrasive materials to expose possible damage. All parts treated in this manner must be cleaned before reuse.

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*Note: Normal bearing running temperatures can run up to 100°F above ambient. If the bearing temperature is in excess of 200°F above ambient the bearing is considered to have failed. This condition is called a "hot box" and requires that the car be "setout" to remove the bearing from service.

Bearing parts of different roller bearing manufactueres must never be mixed nor interchanged in the same assembly, except spacers, seals, locking plates, and lube fittings as designated by AAR.

Waste must not be used to clean roller bearings. Clean towels free from lint must be used.

4. AAR BEARING REQUIREMENTS

Specifications for Approved Journal Roller Bearings Applicable to Interchange Freight Cars are contained in the AAR Mechanical Division's Manual of Standards and Recommended Practices on pages D53 thru D58. Major specification requirements include the following:

1. Design - The major design requirements are that a roller bearing will be designed for a minimum life expectancy of 500,000 miles service with a load factor of 80% which is the full rail load in the radial direction for half the mileage. Life Expectancy is defined as that life at which no more than 10% of the bearing may have been replaced solely due to metal fatigue.

    a. Lubrication Seals - Lubrication seals shall be designed for a minimum life of 250,000 miles and shall be removable from bearing without damage to seal.

    b. Lubrication - Bearings and enclosures shall be designed for efficient periodic grease lubrication at a minimum interval of three year periods, and the amount of lubricant to be added at these intervals shall be the same for all corresponding sizes and types of bearings of all manufacturers. Manufacturers are permitted to pack their roller bearing units only with grease listed in Interchange Rule 26.

2. Manufacture

    a. Materials - The bearing manufacturer shall furnish information on the materials used for each part of the roller bearing submitted for approval. Standard designations such as AAR, ASTM, SAE etc., when available or an adequate description shall be shown for each material and the appropriate physical properties listed. Thereafter, no change in the specified physical properties of the material will be made without approval.

b. Dimensions - Journal roller bearings for each specific axle size shall have dimensions which will permit their use with journals, adapters, and side frames which are specified in the AAR Manual of Standards and Recommended Practices.

c. Marking - Components shall be permanently and legibly marked as follows:

1) Each AAR approved bearing assembly shall be marked with the manufacturers name or initials, the journal size, the letters AAR (and numbers) assigned by the AAR.

2) All parts used in the journal roller bearings except the rollers, non-separable roller assembly cages, and caps, locking plates, cap screws, snap rings, lubricant fittings, and vents shall be legibly and permanently stamped in the following sequence:

a) Manufacturers name or initials,

b) Manufacturers part number,

c) Month and year of manufacture.

3. Preliminary Acceptance, Inspections, and Tests:

a. Preliminary acceptance by the AAR Committee on Journal Roller Bearings, based on drawings and information submitted by the manufacturer, is obtained in accordance with a procedure outlined in Appendix A of the subject specifications and will not be discussed here.

b. Inspections and Tests - Two specimen journal roller bearings will be given detail inspection and tested at the AAR Research Center as outlined in Appendix B of this subject specification as follows.

1) Component Tolerances - These are checked for compliance with the requirements of the specifications.

2) Finish - Raceways and rollers are checked for a characteristic ground, polished or lapped finish. Surfaces shall be free from tool marks, chatter waves, grinding scratches, pits, rust, discoloration, and other surface imperfections.

3) Hardness

a) The rollers shall show a hardness value of 57 to 66 Rockwell C and the hardness of all rollers shall not vary by more than 5 points on the Rockwell C scale.

b) Bearing Rings - The hardness of all the bearing rings representing the test group of bearings shall not vary more than 5 points on the Rockwell C scale and no bearing ring shall measure less than 58 Rockwell C if made of alloy steel, or 50 Rockwell C if made of corrosion-resisting steel.

c) Thrust Surfaces - When not an integral part of bearing rings, hardness shall measure 40 Rockwell C or greater.

4. Corrosion Resistance Test - Bearings made of corrosion-resisting steel shall be subjected to a specified standard salt spray test after which bearing parts will be visually inspected for loss of surface finish and evidence of general pitting and corrosion.

5. Shock Test - Bearing rings, both corrosion resisting steel and alloy steel, will be dropped a vertical distance of four feet to strike an edge of a mild steel plate. Any splitting, cracking, chipping or significant deformation shall constitute a failure.

6. Cages - shall be inspected to ensure that they are constructed so as not to limit the proper function of the bearings to which they are fitted.

7. Standardization - Bearings shall be checked for conformity to AAR requirements for standardization.

8. Dynamic Test - Tests of a specimen journal roller bearing shall be made in the AAR Full Scale Test Machine under simulated service to determine the performance of the bearing and to determine loss of lubricant. After the test all components of the bearing shall be checked for wear or other defects. The AAR Roller Bearing test procedure is shown on the next page.

9. Seals - Where the seal in a seal case is a separable unit from the roller bearing housing, the unit shall be applied in such a manner that it cannot become loose or displaced in the housing or separated from the housing under conditions of normal service.

10. Seal Rings - Seal contact surface of seal rings shall be processed in a manner to provide life and operating characteristics consistent with seal performance requirements as specified under lubrication seals in the basic specification.

11. Grease System - The bearings shall have sufficient passages from grease filling hole to all parts of bearing to provide uniform distribution of grease throughout bearing.

12. <u>Vent</u> - A vent, or vents, or means of venting, of sufficient capacity shall be provided to purge to the atmosphere any excessive grease resulting from overfilling and to relieve excessive pressure due to a rise in temperature.

AAR PROCEDURE FOR JOURNAL ROLLER BEARING TESTS EMPLOYING LATERAL THRUST AND CONTROLLED AMBIENT TEMPERATURES

LOAD

20,000 lb. for 5 1/2 x 10 in. journal

26,250 lb. for 6 x 11 in. journal

30,000* lb. for 6 1/2 x 12 in. journal

THRUST

Six percent of load at 30 cycles per minute.

8:45 AM - Start under full load after overnight standing, increasing speed gradually to 80 mph in six minutes (9:00 am). Start lateral thrust at 8 mph. Record all temperatures at start; ambient plus bearing temperatures which are taken at each end of the outer ring on the surface of the ring O.D.

9:00 AM - Record all temperatures and maintain 80 mph for three hours, recording temperatures every 15 minutes.

12 NOON - Record temperatures, stop lateral thrust and reduce speed to zero at the end of 60 seconds. Allow to stand for two minutes, then accelerate gradually to 80 mph in two minutes, restarting lateral thrust at 8 mph.

12:05 PM - Maintain 80 mph and lateral thrust for 10 minutes, then repeat stop-and-start cycles every 15 minutes, ending the test at 4:30 pm, simulating 2050 freight car miles using 33 in. wheel diameter for the 5 1/2 x 10 and 6 x 11 in. bearings and 2235 miles using 36 in. wheel diameter for the 6 1/2 x 12 in. bearing for the four one-day runs. Record tempteratures every 15-minute cycle.

This permits one minute gradual deceleration, two minutes standing, two minutes acceleration and 10 minutes running, each 15-minute cycle, totaling 18 cycles.

Reverse direction of rotation at the start of each one-day run.

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* Machine Capacity

Each bearing test will run four days under the above schedule at the following ambient temperatures:

Monday - 90°F. after week-end standing.

Tuesday - 130°F.

Wednesday - 15°F.

Thursday - Minus 45° F.

Friday - Machine set-up time.

Upon completion of the above procedure, the bearing is dismounted and weighed, as is done before mounting, to determine the amount of grease loss, if any, through the seals. The bearing is then disassembled, a grease sample taken for analysis, and the bearing cleaned of grease.

Bearing races and rollers are inspected for wear and wear patterns, the roller cages for cracks or distortion and the seal elements for wear and deterioration, such as splitting, blistering, etc.

5. ROLLER BEARING ADAPTERS AND REQUIREMENTS

The rotating end cap roller bearing adapter is an intermediate part necessary to fit roller bearings into the pedestal opening of railroad car side frames and is designed to properly distribute the load on the rollers of the bearings. If excessive wear of the adapter occurs, off center loading of the bearing can result, the adapter bore can come in contact with the outer ring O.D. in the seal fit areas which will tend to loosen the seals or the thrust shoulders, or the thrust shoulders can come in contact with the inboard and outboard grease seals which will also tend to loosen the seals. Each of these conditions has an adverse effect on the operational life of the bearing and consequently, must be avoided. Adversely worn adapters can be detected visually in some cases with the bearing assembly in place in the truck and in all cases when the bearing is removed from the truck. Cracked, broken, warped, twisted, or otherwise distorted adapters must be scrapped and repair welding of cracked or broken adapters is prohibited.

ADAPTER SPECIFICATIONS

The AAR Mechanical Division Specification M 924-70, entitled Purchase and Acceptance of Journal Roller Bearing Adapters for Freight Cars, covers the requirements for adapters to be used with journal roller bearings including purchase and acceptance requirements. The major requirements cover materials, hardness, dimensional limits, workmanship and finish. Adapters intended for interchange service must be submitted for tests to ensure that hardness and dimension requirements are met.

Cast adapters may be made from pearlitic malleable iron, carbon steel, and ductile iron. Forged adapters are made from medium carbon (0.40 to 0.55C) steel. The hardness requirements for all adapters except those made from ductible iron are 179 Brinell (Rc8) minimum or may be heat treated to similar specifications as ductile iron adapters for greater wear. Ductile iron adapters with hardened crown and thrust shoulder surfaces require

350 minimum Brinell on the hardened surfaces to a minimum depth of 0.060 inch; for the remainder of the adapter the Brinell hardness requirement is 187 min to 241 max. The Brinell hardness range for ductile iron castings not having crown surfaces hardened is 217 to 255.

Considering that the purpose of the adapter is to properly distribute the load on the bearing, resistance to wear should be primary adapter requirement. This means that high hardness (strength) should be specified on the adapter wear surfaces. The hardened crown on the ductile iron adapter calls for a 350 Brinell (38RC) minimum hardness. This represents a 90,000 PSI higher strength on the wear surface than the strength of the wear surface an adapter with a 179 Brinell hardness. Increasing hardness however causes metals to become more brittle. With the proper alloy and heat treatment higher through hardness and toughness (resistance to brittle fracture) could be achieved. Unfortunately, the average cost would probably exceed the $8 average cost of the present adapters.

Another concern of the railroads is that a higher hardness adapter would probably result in greater wear of the side frame in the pedestal area. The AAR Specification for cast steel side frames calls for a 150 Brinell (RC<0) and 187 Brinell (10RC) minimum hardness for grade B and C steels respectively which are comparable to the 179 Brinell minimum specified for the majority of adapters.

Information on wear life and typical hardnesses of adapters and side frames was not gathered and is not generally known. Consequently, specific recommendations can not be made at this time. However, this area will be investigated in order to better define the problem in view of the importance of adapters to overall bearing performance.

6. JOURNAL ROLLER BEARING GREASE REQUIREMENTS

AAR Mechanical Division Specification M-917-64 covers grease for lubricating passenger service cars having Hyatt grease lubricated roller bearings with thrust blocks viz Grade A, and for lubricating all other types of grease lubricated roller bearings viz, Grade B. Basically, the grease shall be a smooth, well manufactured product of uniform quality, composed of high grade soaps, refined and filtered petroleum oils, suitable oxidation and rust inhibitors, and such other additives as are necessary for desired performance. The consistency of the grease shall be homogeneous, and free from corrosive matter, grit, rozen, waxes, talc, mica, graphite clay, clay compounds, or other fillers.

In addition to standard ASTM tests to determine properties of the mineral oil and other characteristics of the grease itself, interchange simulated service tests are required at the AAR Research Center on a roller bearing grease testing machine which contains rotating-end cap-type roller bearings. Briefly, the test consists of totally cleaning the bearings, inspecting and if necessary replacing defective or worn parts. The bearing assembly, excluding the end cap, the backing ring, and seal wear rings, is weighed, then filled with grease according to manufacturers' recommendations for weight of initial fill and distribution among the bearing elements. Finally the lubricated bearing must be reweighed and the weight of lubricant must be adjusted to correspond to the manufacturers' recommendations. The bearings are then mounted on a properly prepared roller bearing axle and the end cap, locking plate, and bolts applied per manufacturers' recommendations.

The bearings are run at a rate equivalent to 63 miles per hour for approximately 8 weeks. The accumulated mileage during this running period is about 49,392 miles. The grease that is purged from the bearing during this test is collected, weighed, and if in sufficient quantity subjected to a quarter-scale penetration measurement (ASTM Method D1403) and examined for homogeneity, oil separation, etc.

At the conclusion of the eight week test, each bearing is removed and carefully weighed (minus end cap, backing ring or seal wear rings), the grease in each bearing inspected, the bearing is cleaned with solvents and reweighed. The basis for determining the performance and acceptability of the bearing grease tested is the temperature of the bearing during test, data on the quantity of grease leakage, condition of leakage, condition of grease at final inspection, and the laboratory analysis of the grease removed from each bearing.

7. AAR APPROVED RAILROAD JOURNAL ROLLER BEARINGS

The AAR has conditionally approved three groups of rotating end cap bearings. The three groups are:

1) tapered roller bearings

2) cylindrical rollers limited lateral

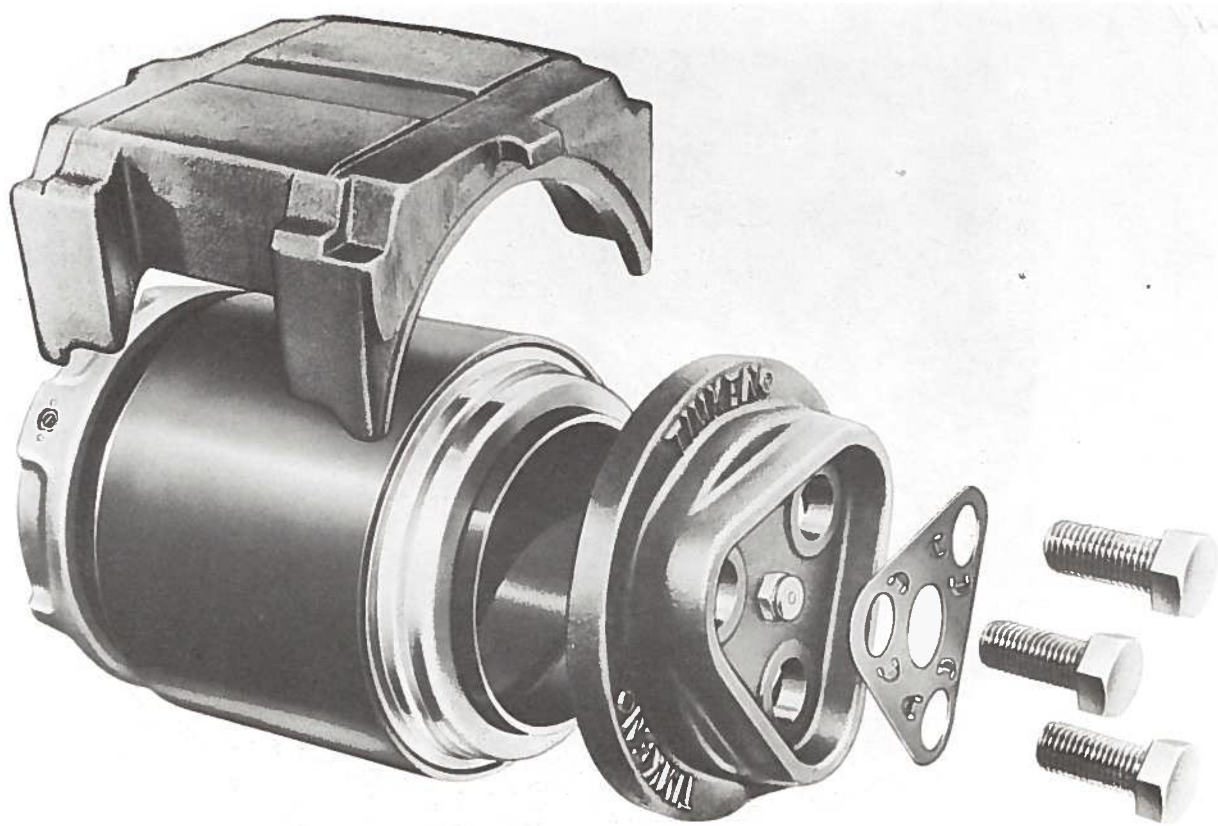
3) cylindrical roller extended lateral.

The tapered roller bearings account for about 92% of all the railroad roller bearings in use in the U. S. and Canada, and are manufactured by 11 companies, of which only five are U. S. firms.

7.1 ROLLER BEARING ASSEMBLY

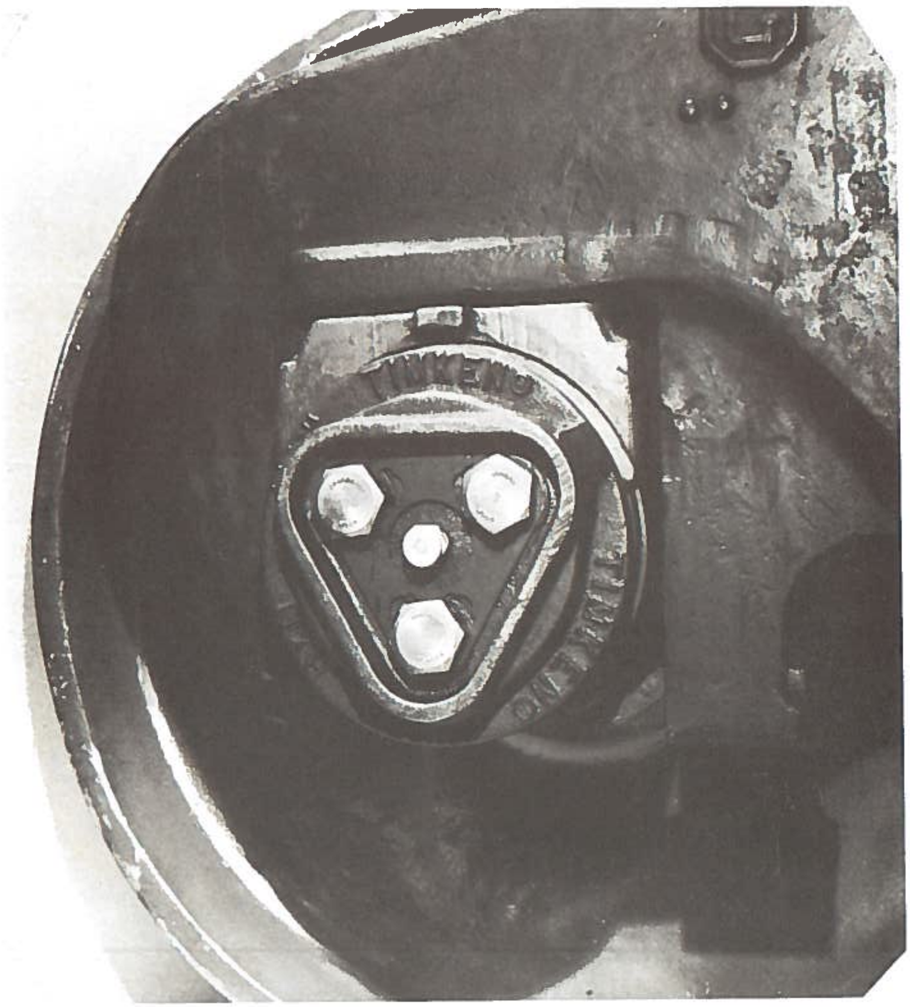
The typical roller bearing assembly (Fig. 3) consists of the bearing adapter (upper left), a self-contained preassembled, pre-adjusted, prelubricated and completely sealed roller bearing (lower left), the end cap (middle), and cap screws with locking plate (right). The roller bearing is installed on or removed from the axle with a bearing or wheel press with pressures ranging from 30 to 70 tons depending on the bearing size. The axle end cap and locking plate are secured to the axle with the cap screws and torqued tight from 110 to 460 foot pounds depending on bearing and cap screw size. Tabs on the locking plate are bent flat against the cap screw head to resist loosening of the cap screw.

In assembling the preassembled wheel set and bearing assembly to the truck, the truck frame is lowered into position on the bearing making sure that the adapters are properly seated on the bearing. Frame keys are bolted to the frame to prevent the bearing from becoming disengaged from the adapter. Figure 4 shows the bearing assembly in an AAR standard narrow pedestal frame truck. Figure 5 shows the bearing assembly in an "integral box" side frame modified for roller bearings.



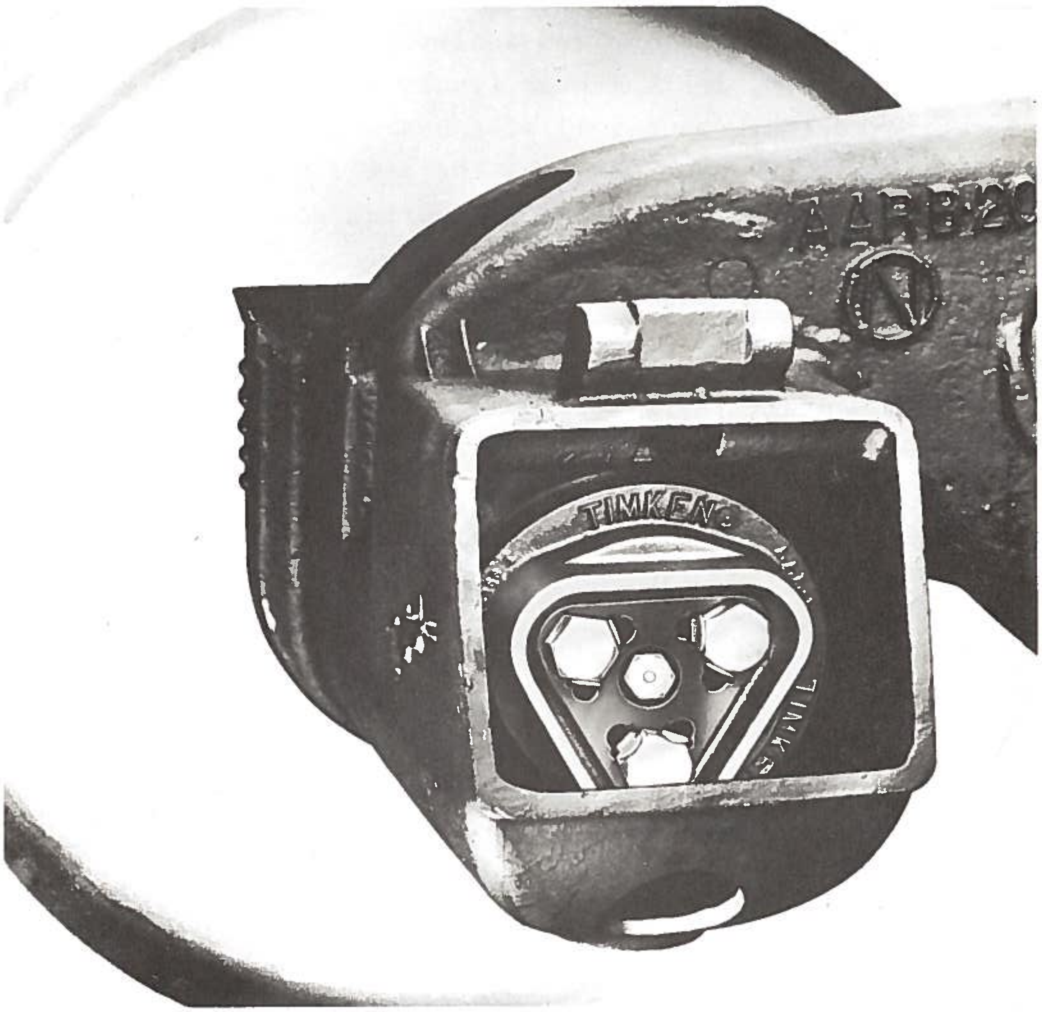
(Courtesy: Timken Co.)

Figure 3 Roller Bearing Assembly



(Courtesy: Timken Co.)

Figure 4 Roller Bearing Assembly in Narrow Pedestal Frame Truck



(Courtesy: Timken Co.)

Figure 5 Roller Bearing Assembly in Integral Box Side Frame Truck

7.2 BEARING SIZES AND LOADS

The vertical loads for the six different size bearings are shown in Table 1. Reliable or accurate information on lateral thrust loads, shock loads, and low frequency cyclic loads is not available. However, this information is being sought in a current contract[1] with the Illinois Institute of Technology Research Institute which is measuring and analyzing forces on critical components utilizing instrumented trucks and test cars. Lastly, it is estimated that the 6 x 11 size bearing constitutes about 45%, the 6-1/2 x 12 about 25%, and the 5-1/2 x 10 about 15% of the bearing failures for all roller bearing equipped freight cars.

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[1]Contract DOT-FR-20070, Analysis of Railroad Car Truck and Wheel Fatigue

TABLE 1. BEARING CLASSES, SIZES, AND MAXIMUM VERTICAL LOADS

Class	Size (axle journal diameter and length, inches)	Nominal Car Capacity, Tons, (current AAR designation)	Gross Rail Load, LBS[1] (on four axles)	Vertical Load on Each Bearing, Lbs, Approx.
B	4 1/2 x 8	30	103,000	12,000
C	5 x 9	44	142,000	16,750
D	5 1/2 x 10	55	177,000	21,000
E	6 x 11	77	220,000	26,250
F	6 1/2 x 12	100	263,000	31,500
G	7 x 12	125	315,000	38,000

[1] Gross Rail Load equals light weight of car plus loading

8. SUB ASSEMBLY COMPONENTS OF TAPERED ROLLER BEARINGS

A cross sectional drawing and a cross sectional photo of a roller bearing are shown in Figures 6a and b. with the different sub-assembly components labeled. Figure 7 shows the different components in the disassembled state. A description of bearing parts and various items is given below:[1]

ITEM	DESCRIPTION OF PARTS
Outer Ring	Cup or outer race
Inner Ring	Cone or inner race
Raceways	Surfaces of outer and inner ring on which rollers operate
Rollers	Cylindrical rollers or tapered rollers
Cage	Retainer; separator
Roller Assembly	(A) Rollers with cage only, when separable
	(B) Rollers with inner ring and cage, not separable
	(C) Rollers with outer ring, inner ring, spacer and cage, not separable
Spacer	Spacer, spacer ring; cone spacer
Seal	Seal proper, including inner case, if used
Seal Wear Ring	Ring on which seal rides or makes contact
End Cap	Cap at end of journal; axle end cap; locking cup; end cover
Backing Ring	Collar between bearing and journal fillet; axle collar; dust guard ring; enclosure collar
Adapter	The part which transfers load from truck to roller bearing
Cap Screws	End cap fasteners
Locking Plate	Cap screw locking device
Grease Fitting	Giant button head pressure fitting

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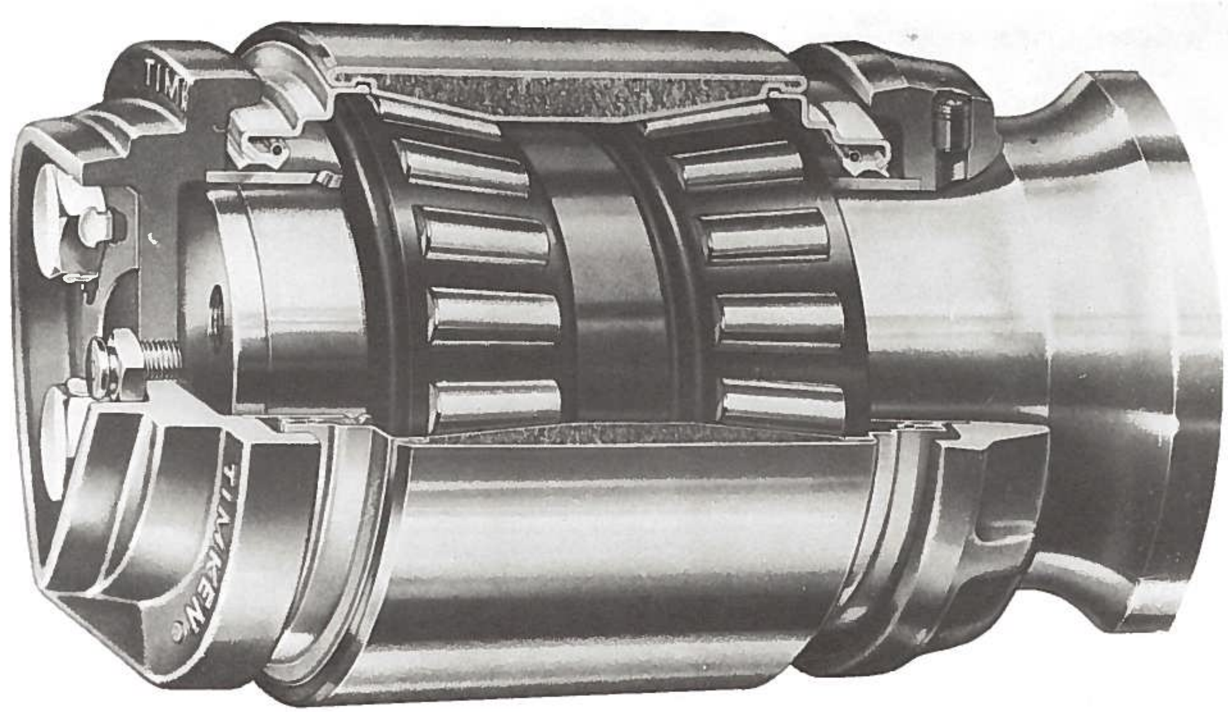
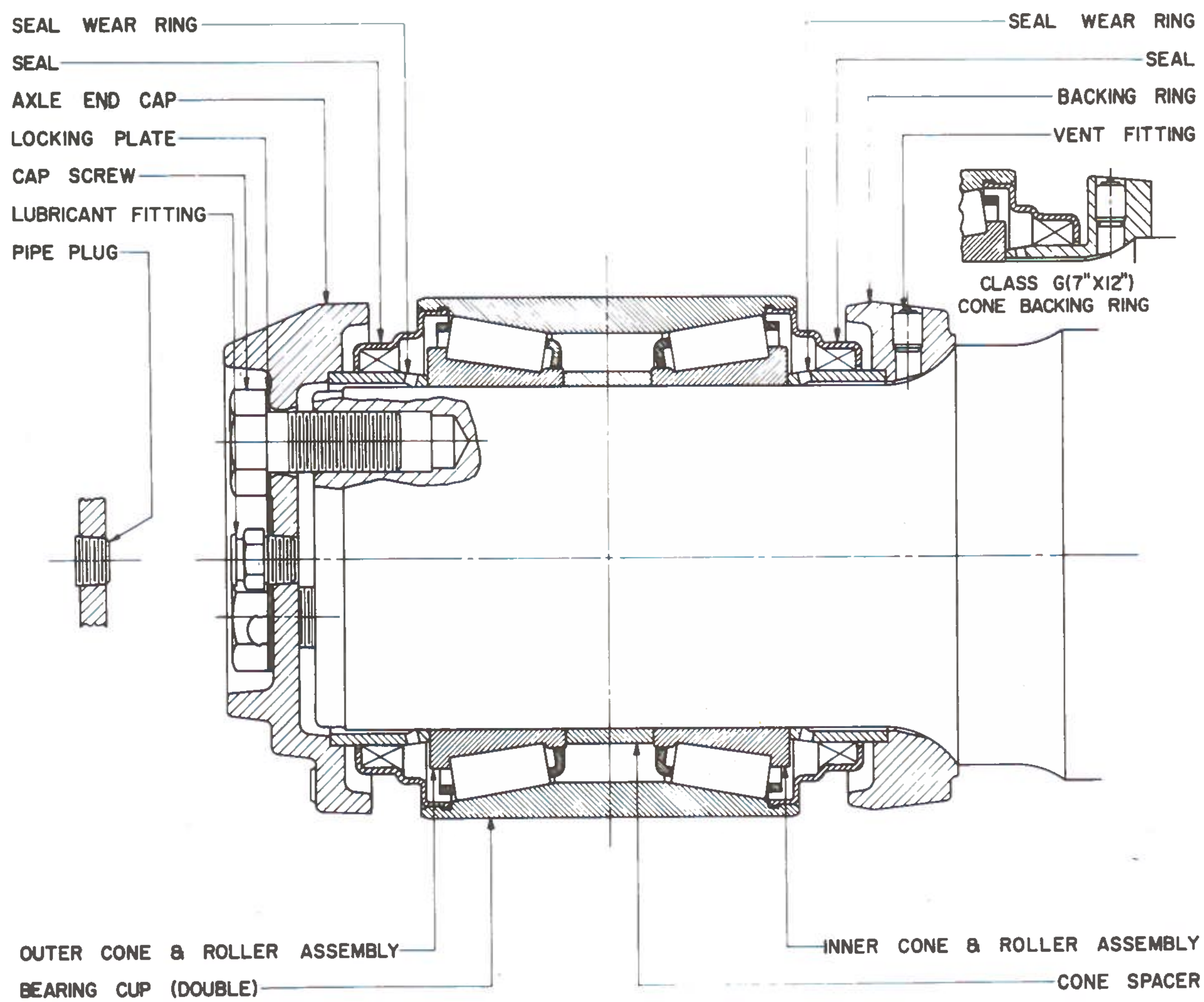
[1] Ref. AAR Roller Bearing Manual



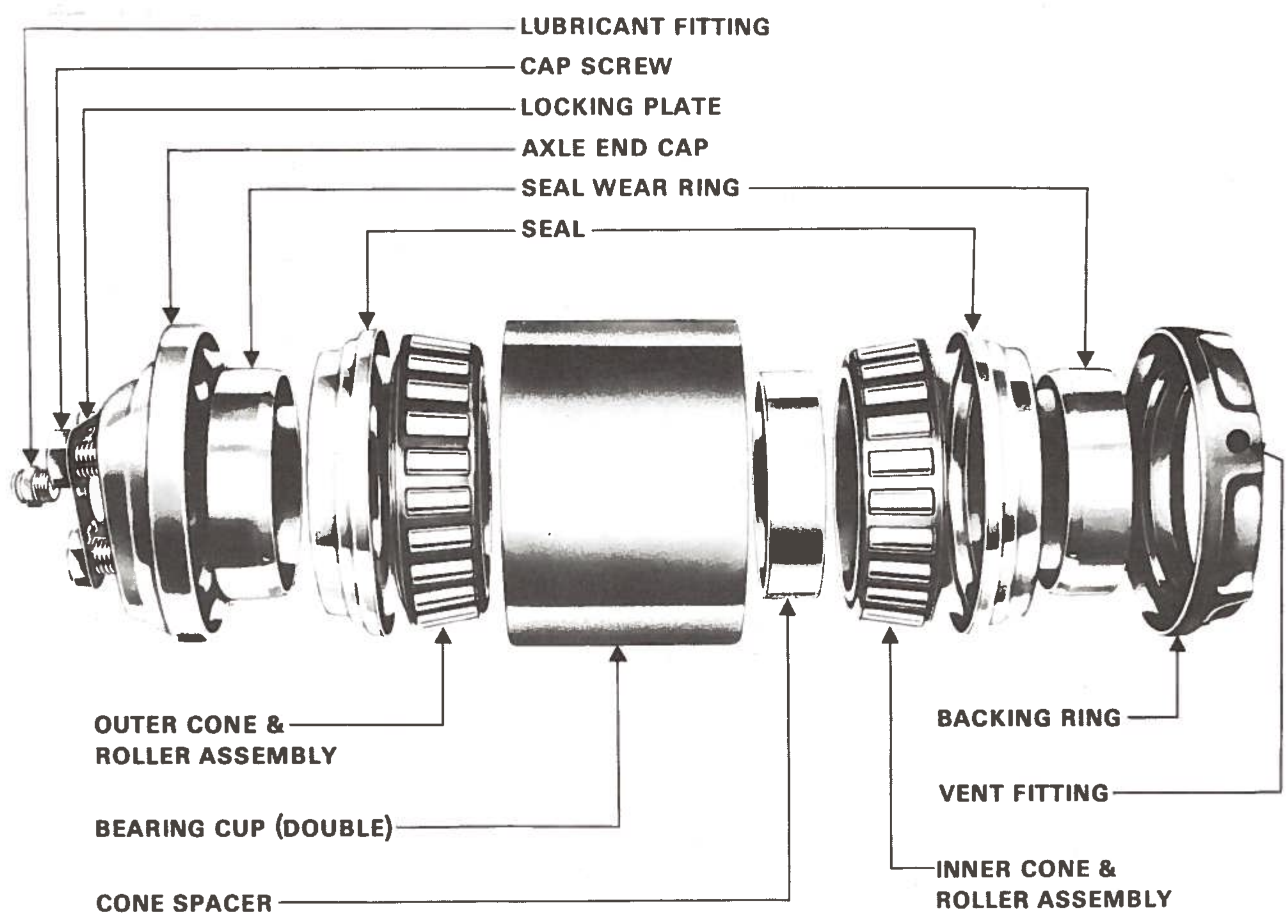
(Courtesy: Timken Co.)

Figure 6a. Cross Sectional Photo of Rotating End Cap Tapered Roller Bearing



(Courtesy: Timken Co.)

Figure 6b. Cross Sectional Drawing of Rotating End Cap Tapered Roller Bearing



(Courtesy: Timken Co.)

Figure 7 Disassembled Bearing Components

9. BEARING DAMAGE[1]

The type and discription of damage which characterizes defective subassembly bearing components is summarized below:

Damage	Description
Etching	Gray, or grayish black color, caused by water or acidity in the lubricant. Has little depth. Superficial water or acid etching, is acceptable after surfaces have been polished. Slight pitting after polishing is acceptable.
Stain Discoloration	Surface discoloration caused by moisture or acidity in the lubricant. Has no depth. Stains and discoloration having no depth, are not considered detrimental if they can be removed by polishing with a wire wheel or 320 grit abrasive cloth.
Corrosian Pitting & Rust	Black corrosion lines or pit marks. Can have some depth. Rust is build up on iron oxide, sometimes due to finger prints and is a form of advanced etching. Corrosion pitting or rusting which has advanced to severe pitting, and which cannot be removed by polishing the raceways and rollers with a wire wheel and polishing rouge, should be considered cause for rejecting parts so affected.
Heat Discoloration	Color visible, from faint straw to dark blue.
Fatigue Cracks	Minute cracks in load carrying surfaces, which are first indication of metal failure. They should be ground out to preclude spalling.

[1] AAR Roller Bearing Manual

Fatigue Spalling or Flaking	Originates as minute fatigue cracks and eventually pieces of metal drop out. This occurs in roller path of inner and outer rings and in roller surface. Incipient spalls may be repaired by grinding, as noted below, and returned to service. (If a spall results in a repaired spall larger than 3/8" on any side by 1/8" deep, the part must be rejected and not returned to service.)
Fracture	Crack extending across entire width of inner or outer ring or a roller is cause for scrapping.
Nicks	Surface damage due to rough handling or other abuse should be stored smooth if raised.
Smearing or Peeling	Surface roughness caused by transfer of metal from one surface to another as a result of galling. This can be caused by rollers sliding on roller path because of adverse lubricating conditions.
Brinelling	Surface indentations in raceways made by rollers under impact load. Brinelling is caused by the rollers being forced into the surface of either raceway while the bearing was subjected to heavy impact loading beyond its capacity.
Indentations	Visible recesses on surfaces of raceways or rollers, also called Fragment Indentations. Caused by foreign matter passing through bearing while loaded. Surface damage, usually caused by contaminants in the lubricant, is not considered as sufficient cause for rejecting bearing parts, unless the damage is such that roughness can be detected when the bearing is rotated by hand.

Electric Burns	Surface damage from passage of electric current causing localized craters, pits, fluting or corrugations. Pitting is the result of electric current passing through the bearing, such as when the ground cable is clamped to the rail or wheel for arc-welding repairs on cars, and is a reason for rejecting bearing parts affected. Fluting corrugations, resulting from electric current passing through the rotating bearing are sufficient cause for rejecting the parts of bearing affected.
Scoring	Parallel or concentric grooves which penetrate the original surface. They can be circumferential or axial.

10. BEARING FATIGUE[1]

GENERAL BACKGROUND

If a rolling bearing in service is properly lubricated, aligned, kept free of abrasives, moisture and corrosive reagents, and properly loaded, material fatigue is the limiting factor in bearing life. Structural materials in bearings subjected to an unlimited succession of repeated and/or reversed stresses do not exhibit an endurance limit and ultimately fail from metal fatigue. Fatigue is manifested as a flaking off of metallic particles from the surfaces of the rolling assembly components. The cause of fatigue is the large surface stress generated by the load acting on the rolling elements and the raceways. The large stress result from the small contact areas between the elements and even though the elemental loading may be moderate, the stresses are unusually large. It is not uncommon for some types of roller bearing to operate continuously with normal stresses exceeding 200,000 PSI compression and in some applications including endurance testing, the stresses may exceed 500,000 PSI compression on the rolling surfaces.

Since the effective area over which the load is supported increases rapidly with depth below the rolling surface, the high compressive surface stress does not permeate the entire rolling member. Therefore, contact fatigue of the rolling surface rather than bulk failure of the member is the significant factor in roller bearing design. Surface deformation due to the large contact stresses on the mating elements is generally of a low order of magnitude, for example, 0.001 inch or less in steel bearings.

The problem of bearing failure by fatigue has to be treated statistically since very little is known about the precise mechanism of rolling contact fatigue which causes the wide variations in bearing life. It is known, however, that the basic problem is one of material rather than design or extreme dimensional precision. Prior to 1950, each bearing manufacturer had his own method of establishing load and life ratings for his product. To overcome the

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[1] T.A. Harris, Roller Bearing Analysis, John Wiley & Sons, Inc. N.Y.

confusion which ensued from these practices, in 1950 the Anti-Friction Bearing Manufacturers Association (AFBMA) adopted a standardized bearing rating method based primarily on the results of a study by Lundberg and Palmgren who applied Weibulls statistical theory of material strength to the rolling contact problem. The AFBMA analytical methods for the determination of bearing loads and life are based on actual test data supplied by the manufacturers, and the validity of the rating methods were confirmed by the Bureau of Standards in an independent study using the same data.

ROLLER BEARING FATIGUE LIFE

In brief, it has been shown that the fatigue life of a tapered roller bearing can be very well approximated by the equation

$$L = \left(\frac{C}{P}\right)^{\frac{10}{3}}$$

where L is the number of million revolutions (or hours) that a specified percentage of bearings will fail to survive on account of fatigue causes. If the percentage is 10, then L = L 10 which is defined as the rating life; if the percentage is 50, then L = L 50, the median life. P equals the bearing load in pounds and C is the basic dynamic capacity which is defined as the constant bearing load (in pounds) that 90 percent of a group of similar bearings can endure for one million revolutions under given running conditions. Bearing manufacturers and the AFBMA standards provide information and methods for determining C as well as methods to evaluate the effect of conditions other than those arbitrarily adopted as standard. For example, doubling the bearing load reduces hours life to about one tenth and halving the load increase hours life about 10 times. Doubling the speed reduces life by one half, and reducing speed by one half doubles the life.

In summary, due to the statistical nature of bearing material fatigue the life of an individual bearing can not be determined but the fatigue life of a large batch of bearings can be rather accurately estimated. These estimates of reliable life are calculated using relationships derived by studying the statistical behavior of a large number of identical bearings essentially tested identically. As the paramaters improve or change, the various factors in the analytical relationships have to be redetermined or updated. For example, AISI 52100 steel was the bearing material used in the AFBMA standardization studies. This is an air melted steel recognized by the bearing manufacturers for many years as the best bearing material. In recent years, vacuum melted AISI 52100 has been introduced; it is a cleaner and more homogeneous material. It has been found that the fatigue life of bearings made with the latter material is 1.5 to 20 times longer than that obtained with the air melted 52100. Other factors, such as the hardness level of the bearing material, can also effect bearing life.

11. METHODS FOR DATA ANALYSIS

If information on the failure frequency of critical sub assembly components were available, the reliability levels of these components could be determined. Also, a method for predicting the fatigue life of a ball bearing assembly from the lives of the components was developed by Mayer[1]. An important application of this method is the prediction of the improvement in bearing assembly life which would be obtained by improvement of one or more components.

Bearing fatigue life experiments have shown that the cumulative percent failed versus life relationships for bearing components and the overall bearing assembly can be approximated by two-parameter Weibull distributions. The approach considers that the components of a ball bearing assembly, i.e., the ball complement and the inner and outer races, are arranged in series and the survival of the assembly requires survival of all the components. This approach is based on the theorem that the probability of the survival of the assembly is the product of the survival probabilities of the components. The survival function method, although not new, is applicable to bearing assemblies and may be applicable to other critical railroad truck components. However, since it was not possible to collect or develop any failure data on bearing components, the two parameter Weibull approach could not be tested. Data samples will have to be developed and the method tested before its applicability to railroad tapered roller bearings and other critical truck components can be determined. A detailed explanation of the two parameter Weibull life distributions of bearing components and their relationship with bearing assembly life is shown in Appendix A.

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[1]J. E. Mayer, Prediction of Ball Bearing Assembly Life from the Two Parameter Weibull Life Distributions of the Components, Annals of Reliability and Maintainability - 1970, Vol. 9 Assurance Technology Spinoffs, Ninth Reliability and Maintainability Conference, Ford Motor, Co., Detroit, Michigan.

12. ROLLER BEARING FAILURE MODE SURVEY

In order to get information on which subassembly components are failing and triggering the destruction of the overall bearing assembly, a survey of bearing failure statistics was carried out. FRA accident reports (T forms) were reviewed, personnel and records of a railroad roller bearing repair shop were interviewed and examined, two railroad roller bearing manufacturers were interviewed and AAR roller bearing statistics were reviewed. The main conclusion reached was that the records are not complete or detailed enough to single out defective components. The majority of bearing failures were attributed to loss of lubricant due to loose or leaky bearing seals and loose cap screws.

The following statistics were developed and provided by the Timken Bearing Company which investigated the reasons for car setouts due to their bearings on four railroads during 1971. Timken representatives made joint inspections of bearings with railroad representatives to determine the cause of the setouts.

Table 2 gives a breakdown of the types of cars that were setout. With the exception of the category "Other" all the car types are essentially high mileage cars that travel approximately 50,000 or more miles per year. The "Other" category is low mileage box cars which travel about 20,000 miles a year, the annual mileage of the average freight car. Based on this sample of 411 car setouts, 80.5% are high mileage cars indicating that the number of miles of bearing service is an important parameter in bearing reliability.

The reasons for the bearing problems which caused the setouts on the four railroads are shown in Table 3. The major problem area was loose seals which accounted for over 40% of the setouts. The "Other" category which constitutes the second major problem area covers a multiple of defects ranging from worn adapters and undersized axle to fatigue cracks and scored bearing surfaces.

TABLE 2. SETOUTS BY TYPE OF CAR

	Railroad A		Railroad B		Railroad C		Railroad D		TOTAL			
	No. Cars	% of Setouts	No. Cars	$ of Setouts	No. Cars	% of Setouts	No. Cars	% of Setouts	No. Cars	% of Setouts		
TTX	82	46.6	18	29.0	20	17.0	13	23.7	133	32.4	43.1	80.5
TTX Type	11	6.3	13	21.0	9	7.6	11	20.0	44	10.7		
Refrigerator	24	13.6	24	38.7	19	16.1	3	5.5	70	17.0	37.4	
Box	43	24.4	6	9.7	23	19.5	12	21.8	84	20.4		
Other	16	9.1	1	1.6	47	39.8	16	29.1	80	19.5		

(Courtesy: Timken Co.)

TABLE 3. CAUSE OF SETOUTS

	Railroad A		Railroad B		Railroad C		Railroad D		TOTAL	
	No. Bearings	% of Setouts	No. Bearings	% of Setouts	No. Bearings	% of Setouts	No. Bearings	% of Setouts	No. Bearings	% of Setouts
Loose Seals	97	52.1	50	80.8	11	8.9	14	25.4	172	40.4
Leaky Seals	26	14	3	4.8	0	0	2	3.6	31	7.3
Loose or Missing Cap Screws	26	14	1	1.6	10	8.1	10	18.2	47	11.0
Other	37	19.9	8	12.8	31	25.2	12	21.8	88	20.7
Found OK	0	0	0	0	71	57.7	17	30.9	88	20.7

(Courtesy: Timken Co.)

Over 20% of all the bearings were found to be perfectly satisfactory with no discernible reason for the setout. The high incident of satisfactory bearings with Railroad C suggests that they were having troubles with their hot box detectors or setout decision rational. Loose or missing cap screws were determined to be responsible for 11% of the setouts with leaky seals accounting for 7.3%. Surprisingly, the last two categories combined are less than any other single category. Based on all data reviewed, loose cap screws and leaky seals appeared to be cited with the same frequency as loose seals as the cause of bearing hot boxes

Bearing seals consist of a formed metal ring, which fits into the bearing cup (one seal at each end), joined to a rubber or plastic ring which touches and wears against the rotating seal wear rings (Figure 6). If the metal part of the seal becomes dislodged from the bearing cup, the rolling action essentially pumps the bearing free of lubricant. The seal can become dislodged if it is improperly installed, if the end cap touches against it due to loose or missing cap screws, or if excessive adapter wear permits the thrust shoulders to contact the bearing seals at either end of the cap.

Concurrent with shop inspections, Timken representatives also inspected freight cars in yards and rip tracks at various railroad locations in the U.S. and Canada. Table 4 shows the results of the inspection for 1971 and the first six months of 1972, along with the early results of Railroad E which started its own independent inspection of freight car bearings. Of a total of 27,351 cars inspected in the 18-month period, the two most numerous defects noted were worn adapters and loose seals which constituted 4.74 and 1.4 percent, respectively, of the total defects. Leaky seals and the "other" category comprised about 0.5% each of the total defects observed, and loose cap screws accounted for only 0.1% defects noted. Considering that worn adapters are about 3.4 times more plentiful than the next major defect, loose seals, and that worn adapters can cause off-center bearing loads and adversely effect the bearing outer ring and both grease seals, it is tentatively concluded that worn adapters are responsible for the majority of bearing failures.

TABLE 4. FREIGHT CAR INSPECTIONS

	Inspections Conducted by The Timken Company				Inspections Conducted by Railroad E	
	1971		1972			
	89' Flat Cars	Total	89' Flat Cars	Total	89' Flat Cars	Total
Cars Inspected	14,683	21,607	8,404	10,460	146	1,037
Defects Found:						
Cars w/Loose Seals	241 (1.64)	275 (1.27)	156 (1.86)	194 (1.85)	5 (3.42)	30 (2.89)
Cars w/Seal Leaking	14 (0.10)	122 (0.56)	22 (0.26)	24 (0.23)	- -	2 (0.19)
Cars w/Loose or Missing Cap Screws	18 (0.12)	21 (0.10)	6 (0.17)	10 (0.10)	1 (0.68)	8 (0.77)
Cars w/Worn Adapters	617 (4.20)	929 (4.30)	416 (4.95)	523 (5.00)	22 (15.07)	146 (14.08)
Other	55 (0.37)	92 (0.43)	73 (0.87)	91 (0.87)	3 (2.05)	48 (4.63)

(Courtesy: Timken Co.)

APPENDIX

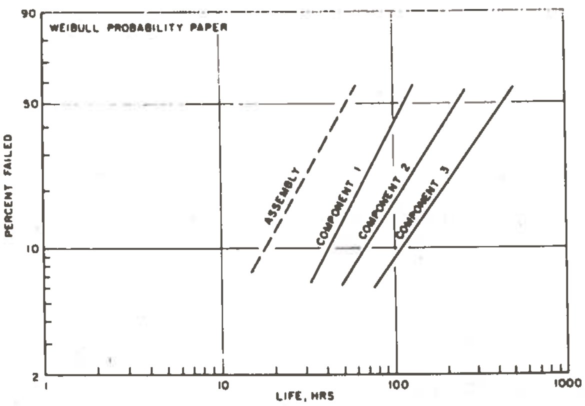
Summary of a report on The Prediction of Ball Bearing Assembly Life from the Two Parameter Weibull Life Distributions of the Components.

J.E. Mayer, Prediction of Ball Bearing Assembly Life from the Two Parameter Weibull Life Distributions of the Components Annals of Reliability and Maintainability - 1970, Vol 9 Assurance Technology Spinoffs, Ninth Reliability and Maintainability Conference, Ford Motor, Co., Detroit, Michigan

Meyer addressed the problem that if given the Weibull lines of the components (inner race, outer race, and balls) of a ball bearing assembly, can the Weibull line of the bearing assembly be determined if the components are arranged in series. This problem is illustrated in Figure A-1. Ball bearing components fail by a rolling contact fatigue mechanism and typical Weibull lines for the components are shown in Figure A-2. The Weibull slopes of the components can range from .6 to 2.4, mostly ranging from 1 to 2, and are not necessarily identical. To solve the basic problem, it must be shown that the performance of the ball bearing assembly can be computed as a function of the performance of its components. An important application of this solution would be the prediction of bearing assembly life improvement that would be obtained by improvements in one or more of the components.

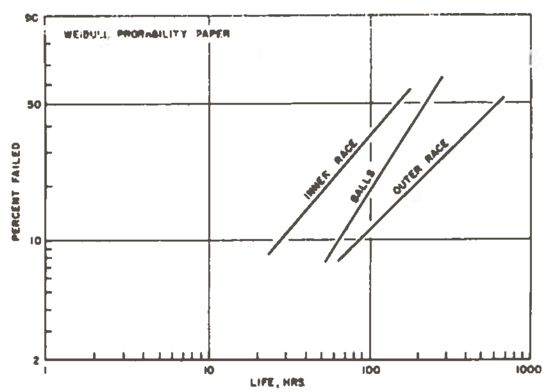
The survival of a ball bearing assembly requires survival of all the components. Utilizing the theorem that the probability of independent events is given by the product of their respective probabilities, the probability of survival (or reliability) of a ball bearing assembly, $R_a(x)$, was defined as the product of the reliability of the respective components. Therefore, if $R_1(x)$, $R_2(x)$ and $R_3(x)$ are the reliabilities of the inner race, balls, and outer race, respectively, and X is hours life

$$Ra(X) = R_1(X) \cdot R_2(X) \cdot R_3(X) \tag{1}$$



(Courtesy: Ford Motor Co.)

Figure A-1 Illustration of Two-parameter Weibull Cumulative Failure Distribution of Three Components and Cumulative Failure Distribution of the Assembly where the Components are Arranged in Series



(Courtesy: Ford Motor Co.)

Figure A-2 Typical Two-parameter Weibull Cumulative Failure Distributions for Components of a Ball Bearing Assembly and the failure distribution function (percent failed) for the assembly, Fa(X), is

$$Fa(X) = 1 - Ra(X) \tag{2}$$

It has been established that ball bearing fatigue life data for bearing assembly and component life (i.e., races and balls) is approximated by a two parameter Weibull distribution. This is shown in Figures A-3, A-4, A-5 and A-6 which are plots of the data in Table A-1 plotted on Weibull paper. In each plot, the solid line is the best least square line to fit the experimental points.

Therefore, it can be written that

$$F_1(X) = 1 - e - \left(\frac{X}{\theta_1}\right)^{b_1} \tag{3}$$

$$F_2(X) = 1 - e - \left(\frac{X}{\theta_2}\right)^{b_2} \tag{4}$$

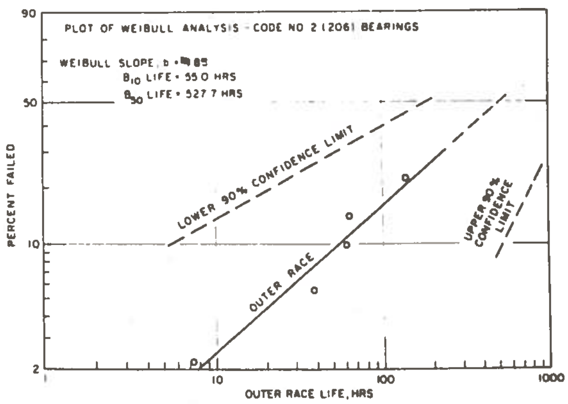
$$F_2(X) = 1 - e - \left(\frac{X}{\theta_3}\right)^{b_3} \tag{5}$$

where b is the slope of the Weibull curve and θ is a scale parameter defined by F(θ) = .632. By substitution

$$R_1(X) = e - \left(\frac{X}{\theta_1}\right)^{b_1} \tag{6}$$

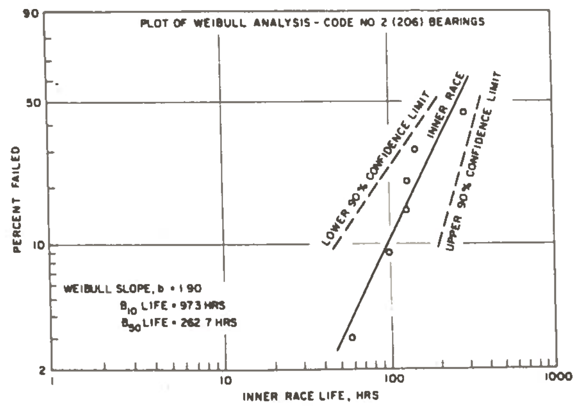
$$R_2(X) = e - \left(\frac{X}{\theta_2}\right)^{b_2} \tag{7}$$

$$R_3(X) = e - \left(\frac{X}{\theta_3}\right)^{b_3} . \tag{8}$$



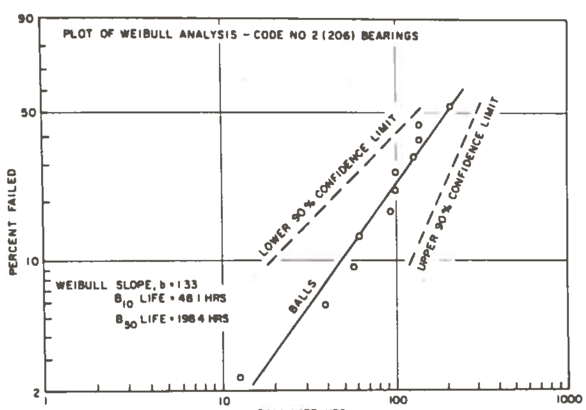
(Courtesy: Ford Motor Co.)

Figure A-3 Experimental Fatigue Life Data for the Outer Race



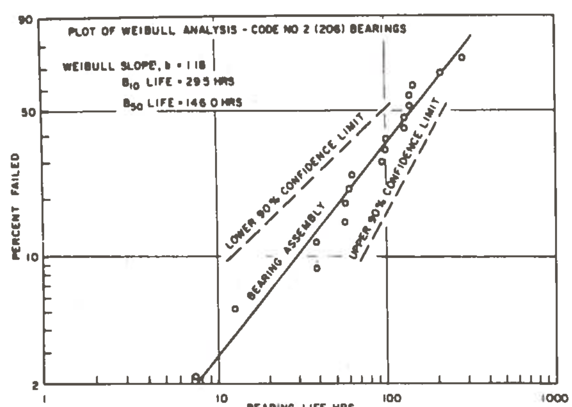
(Courtesy: Ford Motor Co.)

Figure A-4 Experimental Fatigue Life Data for the Inner Race



(Courtesy: Ford Motor Co.)

Figure A-5 Experimental Fatigue Life Data for the Balls



(Courtesy: Ford Motor Co.)

Figure A-6 Experimental Fatigue Life Data for the Ball Bearing Assembly

TABLE A-1. BEARING TEST RESULTS

Test No.	Bearing No.	Reason For Termination*	Failure Type**	Testing Time Hours
1	2-33	F	Ball	12.5
1	2-20	S	-	12.5
2	2-01	F	I.R.	128.5
2	2-09	S	-	128.5
3	2-17	F	I.R.	144.9
3	2-13	S	-	144.9
4	2-18	F	I.R.	285.7
4	2-22	S	-	285.7
5	2-26	F	I.R.	57.5
5	2-31	S	-	57.5
6	2-12	F	I.R., Ball	99.5
6	2-08	F	Ball	99.5
7	2-21	F	O.R., Ball	60.6
7	2-23	S	-	60.6
8	2-11	F	O.R.	38.7
8	2-37	F	Ball	38.7
9	2-32	F	Ball	137.7
9	2-30	F	O.R., Ball	137.7
10	2-34	F	I.R., Ball	127.5
10	2-03	S	-	127.5
11	2-10	F	Ball	94.1
11	2-29	S	-	94.1
12	2-25	F	Ball	208.7
12	2-40	S	-	208.7
13	2-06	S	-	1003.3
13	2-24	S	-	1003.3
14	2-15	F	O.R.	7.3
14	2-04	S	-	7.3
15	2-02	F	Ball	57.0
15	2-19	S	-	57.0
16	2-27	F	O.R.	63.8
16	2-05	S	-	63.8

* F = failure, S = suspension (Test procedure - two bearings were tested at the same time per test).

** I.R. = inner race spall, C.R. = outer race spall, Ball = ball spall

Substituting Equations (6), (7), and (8) into Equations (1) and (2)

$$R_a(x) = e^{-\left[\left(\frac{x}{\theta_1}\right)^{b_1} + \left(\frac{x}{\theta_2}\right)^{b_2} + \left(\frac{x}{\theta_3}\right)^{b_3}\right]} \tag{9}$$

$$F_a(x) = 1 - e^{-\left[\left(\frac{x}{\theta_1}\right)^{b_1} + \left(\frac{x}{\theta_2}\right)^{b_2} + \left(\frac{x}{\theta_3}\right)^{b_3}\right]} \tag{10}$$

In the bearing industry, it is common practice to express bearing lifetime as B_{10} which is defined as the life that 90% of a group of identical bearings will exceed before fatigue reaches the failure criterion. Therefore, expressing Fa(X) in terms of B_{10} and b by using the relationship

$$\theta_i = B_{10_i} \left(\ell n \frac{1}{1-.1}\right)^{-\frac{1}{b_i}} \tag{11}$$

where i = 1, 2, 3.

and $F(B_{10}) = .10$

the following equations are derived:

$$R_a(x) = \left(\frac{1}{1-.1}\right)^{-\left[\left(\frac{x}{B_{10_1}}\right)^{b_1} + \left(\frac{x}{B_{10_2}}\right)^{b_2} + \left(\frac{x}{B_{10_3}}\right)^{b_3}\right]} \tag{12}$$

and

$$F_a(x) = 1 - \left(\frac{1}{1-.1}\right)^{-\left[\left(\frac{x}{B_{10_1}}\right)^{b_1} + \left(\frac{x}{B_{10_2}}\right)^{b_2} + \left(\frac{x}{B_{10_3}}\right)^{b_3}\right]} \tag{13}$$

Using Equation (13), Fa(x) was computed from the component test data of Table 1 for X values ranging from 5 to 400 hours and compared with the experimental curve in Figure A-7. Between the 10% and 50% failed levels, the computed line is sufficiently straight that a Weibull approximation is reasonable.

In order to determine if (Fa(X)) comp could be employed to estimate the assembly life (with sufficient accuracy) the computed and experimental results were compared statistically and found to be sufficiently accurate. Therefore, (Fa(X)) comp, the computer value, can be approximated with a two parameter Weibull distribution.

Thus

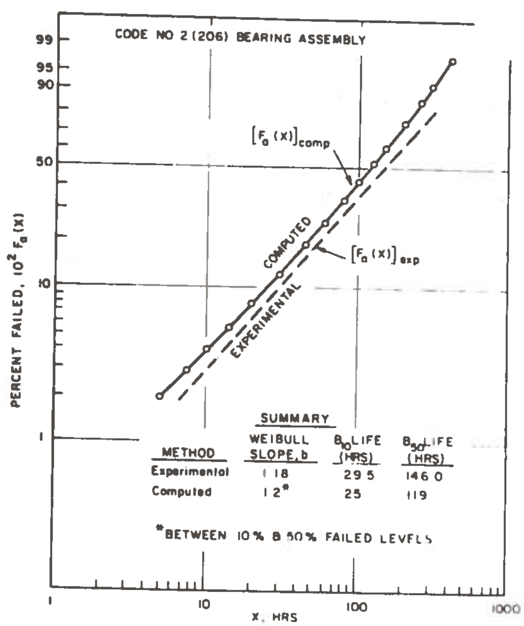
$$F_a(x)\text{ comp} = 1 - e^{-\left(\frac{x}{\theta_a}\right)^{b_a}} \tag{14}$$

and utilizing Equation (11)

$$F_a(x)\text{ comp} = 1 - \left(\frac{1\cdot}{1-.1}\right)^{-\left(\frac{x}{B10_a}\right)^{b_a}} \tag{15}$$

By equating Equations (15) and (13) and noting that for the identity to hold

$$\left[\frac{x}{B10_a}\right]^{b_a} = \left(\frac{x}{B10_1}\right)^{b_1} + \left(\frac{x}{B10_2}\right)^{b_2} + \left(\frac{x}{B10_3}\right)^{b_3} \tag{16}$$



(Courtesy: Ford Motor Co.)

Figure A-7 Comparison of Computed and Experimental Cumulative Failure Distributions for the Ball Bearing Assembly where the Computed Curve was Obtained by the Survival Function Method

By substituting B_{10_a} for X

$$1 = \left(\frac{B_{10_a}}{B_{10_1}}\right)^{b_1} + \left(\frac{B_{10_a}}{B_{10_2}}\right)^{b_2} + \left(\frac{B_{10_a}}{B_{10_3}}\right)^{b_3} \tag{17}$$

Using Equation (17), B_{10a} can be computed since b_i and B_{10i} are assumed to be known. Once B_{10A} is know, b_a can be found using Equation (24) rearranged as follows:

$$b_a = \frac{\ell n\left[\left(\frac{x}{B_{10_1}}\right)^{b_1} + \left(\frac{x}{B_{10_2}}\right)^{b_2} + \left(\frac{x}{B_{10_3}}\right)^{b_3}\right]}{\ell n\left(\frac{x}{B_{10_a}}\right)}$$

A comparison of experimental, computed, and Weibull approximated B_{10a} and b_a values is shown in Table A-2.

It is concluded that Equations (16) and (17) give sufficiently accurate values to allow their use in estimating the performance of the assembly from the performance of the parts.

TABLE A-2. COMPARISON OF EXPERIMENTAL, COMPUTED AND WEIBULL APPROXIMATED B_{10_a} AND b_a

Method	Code No. 2(206) Bearing Assembly	
	b_a	B_{10_a}, hours
Experimental	1.18	29.5
Computed	1.2	25
Weibull Approximated	1.20	24.7