TASK FORCE REPORT

ON

THE M-2 AXLE/BEARING FAILURE INVESTIGATION

Prepared for

the

Federal Railroad Administration

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Prepared by

U.S. Department of Transportation
Transportation Systems Center
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FOREWORD

This report on the investigation of the axle/bearing failures on M-2 commuter rail cars was requested by the Administrator of the Federal Railroad Administration (FRA) to support his role in regulating the safety of the U.S. rail transport system. This sixty-day study defines the possible failure mechanisms of the axle/bearing system, assesses the effectiveness of interim and longer-term countermeasures imposed, and examines the possibility that there exists a risk of such an incident occurring in other rail car fleets with similar design configurations.

The engineering analyses and site visits needed to support this study were conducted primarily by technical experts from the Transportation Systems Center, the FRA, and in certain specific disciplines, technical experts from the university, government and industry sectors. The numerous site visits to commuter and urban rail properties were of critical value to our efforts and we received outstanding cooperation everywhere. The assistance of the FRA Office of Safety and the Urban Mass Transportation Administration, respectively, in arranging these visits is greatly appreciated.

The insight and diversity of views of the large number of people contacted during this study were extremely helpful in addressing the many facets of this problem. Based on the data and information available, however, the Task Force takes sole responsibility for its reported conclusions and believes that they are valid and supportable.

We would be remiss if we did not mention the wide range of interest in the results of this study expressed by the rail industry at large and the government community. Axle/bearing problems, while certainly of heightened concern when they occur on passenger trains, are common throughout the industry and the general information on axle/bearing design, maintenance and inspection practices presented in this report will be valuable for other related applications.

We are particularly grateful to the FRA Administrator, Robert W. Blanchette, his Special Assistant, Robert R. Collins, and Joseph W. Walsh, Associate Administrator for Safety, for their support during the conduct of the study. We also acknowledge the valuable assistance of the staffs of the FRA Office of Safety and the Office of Research and Development.

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EXECUTIVE SUMMARY

On April 4, 1982, and November 3, 1982, M-2 commuter trains operated by Conrail between New Haven, Connecticut, and New York City, lost wheels during revenue operation. Fortunately, no deaths, injuries, derailments or serious property losses resulted from either incident although the potential for serious consequences was apparent.

When the remedial actions taken by the Federal Railroad Administration (FRA) and Conrail following the April incident failed to prevent a recurrence of the wheel loss problem (termed a wheel "burnoff"), the FRA Administrator took immediate action. An Agreement between and among the FRA, Conrail, the Connecticut Department of Transportation and the Metropolitan Transportation Authority was signed on November 11, 1982. This Agreement contained three major provisions:

1. The maximum operating speed of the M-2 fleet was reduced from 70 mph to 55 mph.

2. Daily initial-run pellet inspections and en route inspections for axle/bearing overheat at 30-mile intervals were mandated.

3. The M-2 fleet would undergo a retrofit program, replacing hollow axles with solid axles by October 15, 1983.

The FRA Administrator also established a Task Force to further investigate the safety considerations of the M-2 axle/bearing problem. The Task Force was given sixty days to address three major objectives:

1. To define the failure mechanism of the M-2 axle/bearing system.

2. To assess the effectiveness of the countermeasures embodied in the Agreement of November 11, 1982.

3. To determine the risks of related failures to other passenger fleets with similar axle/bearing configurations (approximately 2400 cars).

This report is the product of the Task Force effort.

The Task Force activities included site visits to commuter rail and urban rail properties, meetings with equipment suppliers, data collection and analysis and engineering analyses. It must be noted that in several instances the Task Force was presented with contradictory information which was difficult to reconcile in the time available. Further, the engineering analyses have not been subjected to verification tests. Based on the data and information available, however, the conclusions contained herein are believed to be valid and supportable.

In general, two paths to ultimate failure were identified. The first path begins with an improper axle/bearing/wheel assembly and, relatively speaking, is the more rapid route to failure. The second path is traceable to the M-2 inboard bearing/hollow axle design configuration. An examination of the facts concerning actual M-2 and related passenger car wheel burnoffs indicates that both paths have been followed in past incidents.

Analyses of the data collected indicate that the potential for a burnoff is also influenced by vehicle weight, speed, and inspection and maintenance practices, but precise relationships among these variables were not established given the limited time available to the Task Force.

Specific conclusions derived from the Task Force efforts are grouped around the three Task Force objectives.

Failure Process Conclusions

- There are two causes that can initiate the failure process.
  - Improper bearing assembly procedures.
  - Characteristics of the M-2 axle/bearing design configuration.
- Past practices that allowed M-2 wheels to be changed without rebuilding bearings could result in excessive mileage on journals and bearings.
- Recent improvements in maintenance and assembly procedures on the M-2 fleet have improved the safety margin.
- The current M-2 hollow axle design cannot assure that the bearing will be effectively restrained against fretting.

Countermeasure Conclusions

- The 55 mph speed limit on the M-2 fleet is considered appropriate as an interim safety measure.
- Thermal runaway that begins just after the last en route inspection before terminal layover may result in a burnoff before the first en route inspection on the next run.
- The 30-mile interval for intermediate inspection for excess heat provides a safe margin for detection of incipient failures but requires that the inspection procedure used can reliably detect bearings that have reached a temperature of 250 degrees Fahrenheit.
- The melt stick procedure is unreliable for detection of incipient bearing failures.
- Utilization of a positive indication of bearing temperature would allow the possibility of relaxing the inspection interval.

- All automated wayside and onboard detectors reviewed require additional development, test and evaluation.

- Retrofit of the M-2 fleet with solid axles is a sound decision.

Fleet Survey Conclusions

- Inboard bearing configurations with hollow axles should not cause concern in general.

- The similarity of the characteristics of the New Jersey Arrow cars with the M-2 cars is cause for concern.

- The mileage interval between bearing rebuild is a critical parameter.

- Appropriate intermediate inspections between rebuilds are still essential to protect any fleet against the consequences of an improperly assembled bearing.

Specific recommendations are grouped as near term and long term activities.

Short Term Recommendations

- Continue 55 mph speed restrictions and daily initial-run axle pellet inspection for the M-2 fleet.

- Expand pellet inspection to cover all layover trains.

- Use a device such as a hand-held pyrometer to measure temperature in the axle bore which, when combined with more frequent pellet checks, can provide a more accurate and reliable inspection technique while also providing a more flexible schedule of en route inspections.

- Continue the solid axle retrofit of the M-2 vehicle.

- Notify operators of importance of mileage interval for bearing rebuild.

- New Jersey Transit should continue observance of the 150,000 mile bearing rebuild interval on Arrow Cars and insure proper bearing assembly and quality control procedures.

Longer Range Recommendations

- Develop an FRA Safety Inspector training program on bearings and axles.

- Urge operators (through AAR and APTA) to adopt more uniform bearing assembly, maintenance and inspection procedures.

- Urge industry to develop automated wayside or onboard detection devices for overheated inboard bearings.

- Contact European and Japanese operators to benefit from their experience with train bearings in high speed applications.

- Hold a briefing for industry on the results of this investigation.

1.0 BACKGROUND

On two occasions in 1982, M-2 commuter cars operated by the Consolidated Rail Corporation (Conrail) between New Haven and New York City lost a wheel during revenue operation. The first incident occurred April 4, 1982, at Southport, Connecticut and involved a commuter train traveling at about 70 mph and carrying over 200 passengers. The 33-inch, 750-pound wheel left the right of way, rolled approximately 1500 feet and hit the side of a church. The train did not derail and in fact continued about 2 miles to the next station before the train crew became aware of the situation.

The second incident occurred November 3, 1982, near New Rochelle, New York and involved an M-2 commuter train carrying 700-800 passengers including about 100 passengers in the car which lost the wheel. In this case the train had slowed from its 70 mph operating speed to about 35 mph and was passing through a crossover when the wheel separated. The wheel then traveled about 400 feet and came to rest between the rails of an adjacent track. The train did not derail, but it was stopped and the passengers were transferred to buses.

The preliminary investigations of these two incidents indicated that there is reason for concern regarding a recurrence. Wheels were also lost from M-2 vehicles April 28, 1981, August 2, 1977, and July 22, 1977. Records also show a history of axle/bearing problems indicating conditions that could have eventually caused a wheel to separate from an axle.

The M-2 cars constitute a 244 vehicle fleet that is between 7 and 10 years old. General Electric (GE) was the manufacturer of the M-2 cars. Bearings have been supplied by both Brenco and Timken.

Approximately 2400 other rail passenger vehicles have axle/bearing configurations that are similar to the M-2 design, i.e., hollow axles with inboard bearings. Although reported incidents of axle/bearing problems of this severity on other passenger operations are rare, the possibility of the failure mechanisms that have affected the M-2 vehicle eventually affecting other fleets is also cause for concern.

The April, 1982 incident prompted the Federal Railroad Administration (FRA) Office of Safety to initiate an investigative effort with technical support from the Transportation Systems Center (TSC). Conrail and GE had already begun an investigation of the problem. The resulting analyses were reviewed by TSC and additional analyses and testing were carried out with the cooperation of Conrail and with contractual support from Ensco, Inc. The results were summarized in a TSC memo to the FRA Deputy Associate Administrator for Safety, dated October 1, 1982. The analyses and test results indicated that the M-2 hollow axle/bearing system was subject to wear from heavy loads and repeated dynamic bending deflections. It was hypothesized that this eventually could lead to overheating, bearing seizure and wheel loss.

After the April, 1982 incident, Conrail expanded an ongoing inspection scheme using temperature sensitive pellets which were inserted in the axle bore beneath the bearing seat (journal) to identify bearings at the initial stage of overheat. A retrofit campaign to replace all hollow axles in the fleet with solid axles was subsequently initiated.

It appeared that the situation was under control with the implementation of these actions, but the November incident served to intensify the concern of all parties and to prompt further remedial actions. Although these incidents did not cause substantial damage and there were no injuries, the potential for a serious accident was apparent and the FRA Administrator took immediate action. An agreement by and among the FRA, Conrail, the Connecticut Department of Transportation (Conn-DOT) and the Metropolitan Transportation Authority (MTA) was signed on November 11, 1982.

The agreement included continuation of the daily pellet inspection before the initial trip and introduction of en route inspection points at 30-mile intervals (New Rochelle, New York and Norwalk, Connecticut) to check for excess heat in the axle. This inspection involved stopping the train and applying a 200 degree F melt stick to the end of each axle. A speed restriction of 55 miles per hour was also imposed. Finally, it was agreed that the fleet would be retrofitted with solid axles by October 15, 1983.

In addition, the FRA Administrator established a task force to further investigate the safety considerations of the M-2 axle/bearing problem. The task force effort was initiated November 23, 1982, and this report is the product of that sixty-day effort.

2.0 OBJECTIVES OF STUDY

The potential results of additional wheel loss incidents involving M-2 or other similar commuter cars provide the impetus for developing a better understanding of the failure mechanism and assessing current countermeasures, thereby providing information to reduce the risk of future wheel loss incidents. This general goal was addressed by defining three interrelated objectives: (1) to define the failure mechanism of the M-2 axle/bearing system; (2) to review the countermeasure programs being employed on the M-2 fleet; and (3) to determine the risks of wheel loss to passenger rail fleets with similar axle/bearing configurations.

2.1 The Failure Mechanism

The wheel loss incidents that have occurred have been attributed to a process termed "burnoff". The burnoff process involves the heating of the axle in the area of a seized bearing to such a high temperature that the axle softens and fails, and the wheel and axle stub separate from the vehicle. There is ample evidence of this high temperature condition in the recorded wheel loss incidents and in other cases where a complete failure did not occur. The first key objective of this study is, therefore, to determine why in some rare instances abnormally high temperatures are generated at the axle/bearing interface of the M-2 commuter rail cars. This objective then involves defining and tracing back the sequence of events to a basic cause or causes of the burnoff phenomenon.

2.2 Review of Countermeasure Programs

Under terms of the agreement dated November 11, 1982, speed restrictions and daily initial-run and intermediate 30-mile inspection procedures have been applied to the M-2 car fleet as discussed previously. The second objective of the study is to review these countermeasures, taking into account what is discovered regarding the failure mechanism, and to reach a judgment regarding their likely effectiveness. Thus it must be determined whether, given the fleet age, condition, maintenance, and operating conditions, a burnoff can develop between the en route inspection points at speeds not exceeding 55 mph. In addition, the axle retrofit program will be evaluated for its effectiveness as a permanent solution.

2.3 Risks to Other Rail Passenger Cars

There are about 2400 other rail passenger vehicles which have axle/bearing configurations similar to the M-2 vehicles. The circumstance(s) accounting for M-2 burnoff incidents may possibly develop in some of these other car fleets. The third objective of the study is to determine whether there is a possibility that the M-2 failure mechanism will occur elsewhere. Although design similarities are of importance, maintenance procedures and physical evidence of non-critical deterioration to the parts involved are also indicators of possible problems.

3.0 APPROACH

The first step was to identify and define the major elements of the problem. These were determined to be:

1. The engineering design of the axle/bearing system and its application to the fleets in question;

2. Axle/bearing inspection intervals and procedures;

3. Axle/bearing maintenance and assembly procedures;

4. Operational characteristics of the fleet.

Two study teams were established to collect and analyze the two classes of information needed to meet the task force objectives. A Failure Assessment Team was created to develop engineering answers to questions about the cause(s) of burnoff incidents on M-2 vehicles and thus concentrated on item (1), above. An Operational Fleet Assessment Team focused its efforts on assessing the axle/bearing history and the inspection, maintenance and assembly procedures for other passenger vehicle fleets (Items (2), (3) and (4)). It is important to point out that there was deliberate interaction between the two teams, and in several instances individuals served on both teams. The study teams were comprised of individuals from TSC and FRA, as well as experts from universities and private industry. Consultations were held with key members of the manufacturing industry, in particular, the bearing manufacturers. The team contributors and their resumes are included in the Appendix. The Urban Mass Transportation Administration assisted in arranging site visits to urban properties.

The Failure Assessment Team (Team 1) developed failure scenarios composed of initial or underlying causes, intermediate-stage conditions or indicators of physical deterioration, and the ultimate failure process. While all three parts of the scenarios and the causal process that link them are important for an adequate understanding of the problem, specifying the characteristics of the intermediate stage is critical if short term inspection and operational countermeasures are to be properly assessed.

The information collected by the Operational Fleet Assessment Team (Team 2) has two primary uses, namely, the assessment of wheel-loss risks to the other vehicle fleets, and the verification or rejection of failure scenarios considered likely explanations of the M-2 incidents. This second use of the Team 2 data was a key input to Team 1 analyses and deliberations since, for the failure theories to be accepted, they had to be consistent with the observed fleet behavior.

To clarify nomenclature used in subsequent sections, a description of rail car axle/bearing/wheel configurations is provided in Section 4. The findings and results of the Team 1 effort which focuses on the failure mechanism of the M-2 axle/bearing are discussed in Section 5. Countermeasures in effect for the M-2 vehicle, and inspection requirements and techniques in general, are discussed in Section 6. The findings and results of the Team 2 effort covering the fleet survey are discussed in Section 7. The conclusions and recommendations are listed in Section 8.

4.0 WHEELSET CONFIGURATION AND ASSEMBLY

The configuration of a wheelset (axle, bearings, wheels, etc.) and the procedures employed to assemble it have important influences on the failure mechanisms which may arise in service. This section focuses attention on those aspects of configuration and assembly which relate to the kind of failure progression observed in the M-2 fleet.

Section 4.1 discusses the all-purpose tapered roller bearing, which is widely employed in the national freight fleet as well as on the M-2 and other passenger fleets. Section 4.2 discusses the application of the roller bearing to an outboard-bearing wheelset, the configuration for which industry standards have been established. Section 4.3 discusses the inboard-bearing wheelset, which is employed by the M-2 and other passenger fleets. The sequence of presentation is intended to focus attention on the differences between inboard- and outboard-bearing wheelsets as well as the elements of outboard-bearing standards which have been applied to inboard-bearing assemblies.

4.1 The Tapered Roller Bearing

Two basic design objectives govern the configuration of a tapered roller bearing. First, the bearing must operate as close as possible to the ideal condition of pure rolling motion in order to minimize sliding-friction losses. Second, the bearing must be able to support lateral thrust forces.

Figure 4-1 shows how a cylindrical roller moving on a cylindrical race achieves the first objective. In pure rolling motion the roller's forward speed V is the product of its angular speed A and radius R, and the roller completes one trip around the race in the time $T=\pi D/AR$. These conditions are satisfied everywhere along the length of the roller; thus it can operate without sliding for any choice of design dimensions for the roller radius R and race diameter D. The roller is free to slide laterally, however, so that the cylindrical bearing is unable to meet the second objective of supporting lateral thrust forces.*

A bearing with rollers running on a conical race can support most of the thrust by contact rather than sliding forces. However, the roller must be tapered to operate in pure rolling motion. As shown in Figure 4-2(a), the ratio of race diameter to roller radius, D/R, must be the same at both ends of the roller to make the "once-around" rolling trip time the same everywhere along the length of the roller. As shown in Figure 4-2(b), this kinematic requirement is satisfied if and only if the inner and outer races are the surfaces of two cones (indicated by lines OB and OC) which have a common apex on the axle centerline.

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*Strictly speaking, a cylindrical bearing will support some thrust until the sliding-friction resistance is overcome, but other provisions are required to limit lateral play and support larger thrusts. For example, cylindrical bearings are used in some diesel locomotive trucks, where lateral play is limited by separate friction bearings attached to the truck frame and positioned to engage the end of the axle.

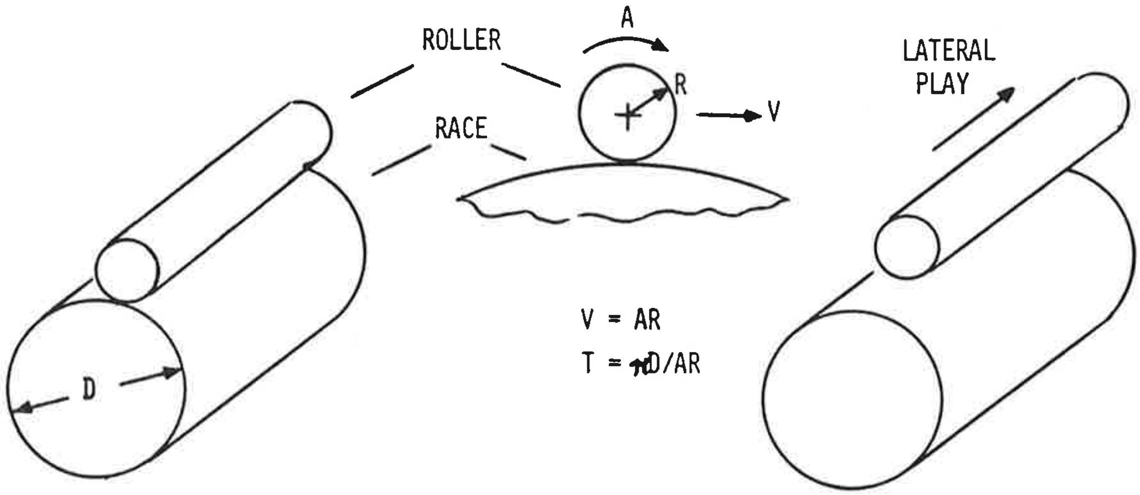
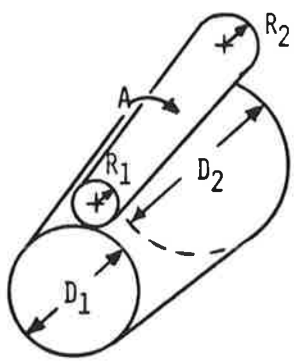


FIGURE 4-1 CYLINDRICAL ROLLER ON A CYLINDRICAL RACE



$V_1 = AR_1$

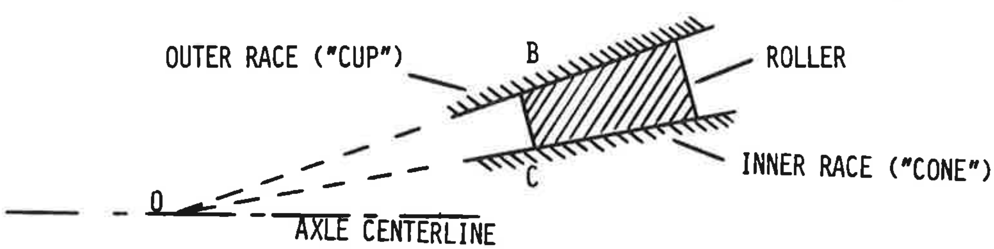
$T_1 = \pi D_1/AR_1$

$V_2 = AR_2$

$T_2 = \pi D_2/AR_2$

$T_1 = T_2$ REQUIRES $D_1/R_1 = D_2/R_2$

(A) KINEMATIC REQUIREMENT FOR PURE ROLLING MOTION



(B) GEOMETRY OF ROLLING COMPONENTS

FIGURE 4-2. TAPERED BEARING KINEMATICS AND DESIGN

Bearing hardware cannot attain the ideal conditions discussed above, but proper assembly results in a close approximation. The critical elements of the assembly will be reviewed with reference to Figure 4-3, a schematic section through the short-F bearing on the M-2 and similar to the standard-F bearing used on 100-ton freight cars. The illustration has been simplified by omitting the components that do not directly participate in rolling-contact motion or load transfer through the bearing.

The bearing stack is formed with five components: two cones, two seal wear rings, and a spacer ring. The manufacturer supplies the cones preassembled with their rollers and cages (not shown) which keep the rollers in place and separated from each other. When the stack is assembled in the cup, the bearing must have a small lateral play (typically between 0.020 and 0.026 inch). The play is measured on a fixture designed for the purpose, and can be adjusted to meet the specification by substituting a slightly shorter or longer spacer ring. The measurement is performed at the bearing rework bench and is called "bench lateral". When a bearing has passed the bench lateral test, the seals and grease charge are added. The bearing is then ready for application since other inspection requirements and dimensional checks such as cone bore diameter were made before assembly.

The bearing is designed to have an "interference fit" (also called "press fit") between its cones and the axle. This means that the axle diameter exceeds the cone bore diameter by a small amount (typically 0.0020 to 0.0045 inch for an F-bearing) before application, so that the cones are expanded when the bearing is pressed onto the axle. The seal wear rings (but not the spacer ring) have a similar interference fit. The expansion has two effects:

- A pressure is established between the bearing stack and the axle.
- The lateral play decreases.

The axle is coated with castor oil or mineral oil to prevent surface damage during the pressing operation, and this lubricant film remains trapped between the mating surfaces. The film has a modest friction coefficient which, together with the fit pressure, provides some resistance to relative motion between the axle and the bearing stack. However, the resistance is small in comparison with some of the service loads that will be available to cause such motion, and clamping forces must be placed on the ends of the stack to prevent the motion. The methods by which clamping forces are supplied are discussed in Sections 4.2 and 4.3.

The lateral play is measured again after application and must meet a "running lateral" specification (typically zero to 0.020 inch; the cup must be able to be turned freely by hand if the running lateral is zero). Within these limits the bearing will have a stack height within specified dimensions, and the race surfaces of the cup and cones will closely approximate the ideal kinematic conditions shown in Figure 4-2 (b) with the roller running up against the cone ribs.

Even a perfectly assembled bearing is not ideal, however, because the loads transferred from the cup to the cones have small thrust components which squeeze the rollers outward against the cone ribs, where sliding friction

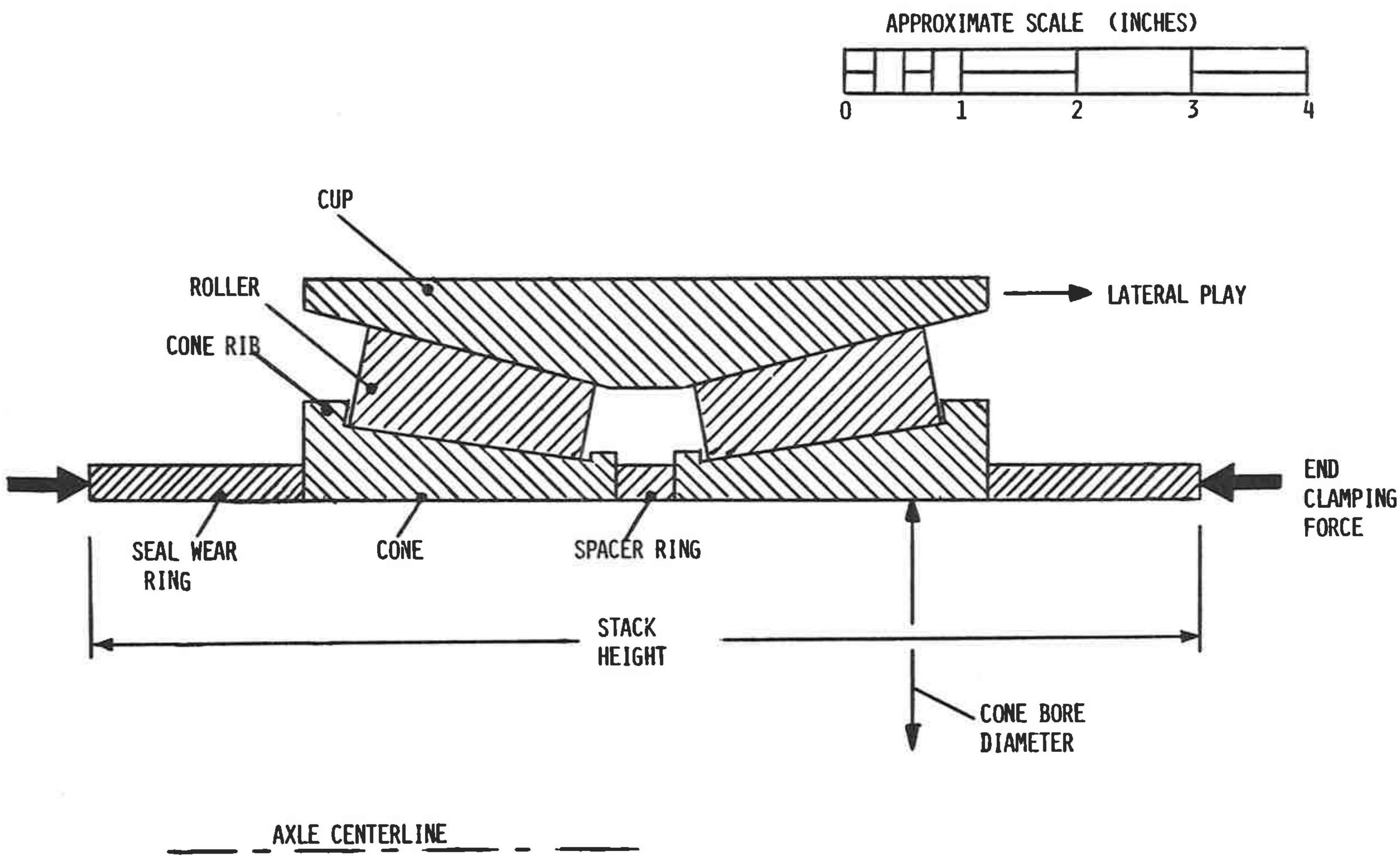


FIGURE 4-3. SHORT-F BEARING APPLIED TO M-2 CAR

generates heat. The grease lubricates the sliding surfaces, however, keeping the friction coefficient and heat generation low. Typical bearings tend to run warm (about 100 F above ambient temperature) and thereby undergo a slight additional expansion which decreases the running lateral play.

Loss of clamping and/or certain kinds of service damage can increase the running lateral in a tapered roller bearing. A field condemning limit of 0.030 inch is used to screen for bearings which should be removed from service. A bearing that has any running lateral will engage in creep rather than pure rolling motion. Creep is a combination of rolling and sliding and generates frictional heat. The creep and heat generation are low for bearings with modest running lateral. However, bearings which exceed the field condemning limit have large creep motions and heating rates that are believed to induce damaging temperatures.

4.2 Outboard-Bearing Wheelset

A simplified schematic section through an F-size wheelset is shown in Figure 4-4 to illustrate the discussion of assembly procedure for the outboard-bearing configuration. The figure also shows the static loads and corresponding bending moment diagram created by a fully loaded 100-ton freight car. Note that the greatest bending moment is attained at the wheel seats, and is three to four times the moment at the bearing seats. The static loads curve the axle, and the amount of curvature is proportional to the bending moment.

The assembly procedure must establish proper wheel gauge as well as proper bearing application. The sequence of operations is as follows:

- o Press wheels onto axle.
- o Install backing rings and press bearings onto axle.
- o Install end caps, locking plates, and cap screws.

The wheel seats are lubricated with linseed oil or a mixture of linseed oil and white lead to prevent surface damage during the press operation. Large interference fits (typically 0.005 to 0.007 inch) are used between the wheel seat and hub bore to retain the wheel in place under service loads. The force required to press a wheel onto the axle cannot be controlled directly, but will range from 70 to 120 tons in a proper installation. Gauge is established by monitoring the distance between the backs of the wheel rims ("back-to-back" measurement) as the second wheel is pressed.

Backing rings are installed between the wheel hubs and bearing seats to cover the axle shoulders and to provide a surface to support the bearing stack. The bearings are now pressed on and checked for running lateral, as described in Section 4.1. From 4 to 12 tons of force is required to press a bearing. When the bearing reaches the backing ring, the force is increased by an additional 20 to 35 tons ("spiking") to make sure that there is no end clearance between the edges of the backing ring and inboard seal wear ring.

Finally, the end-cap hardware is installed. Note that the edge of the end cap engages the edge of the outboard seal wear ring (see Figure 4-4).

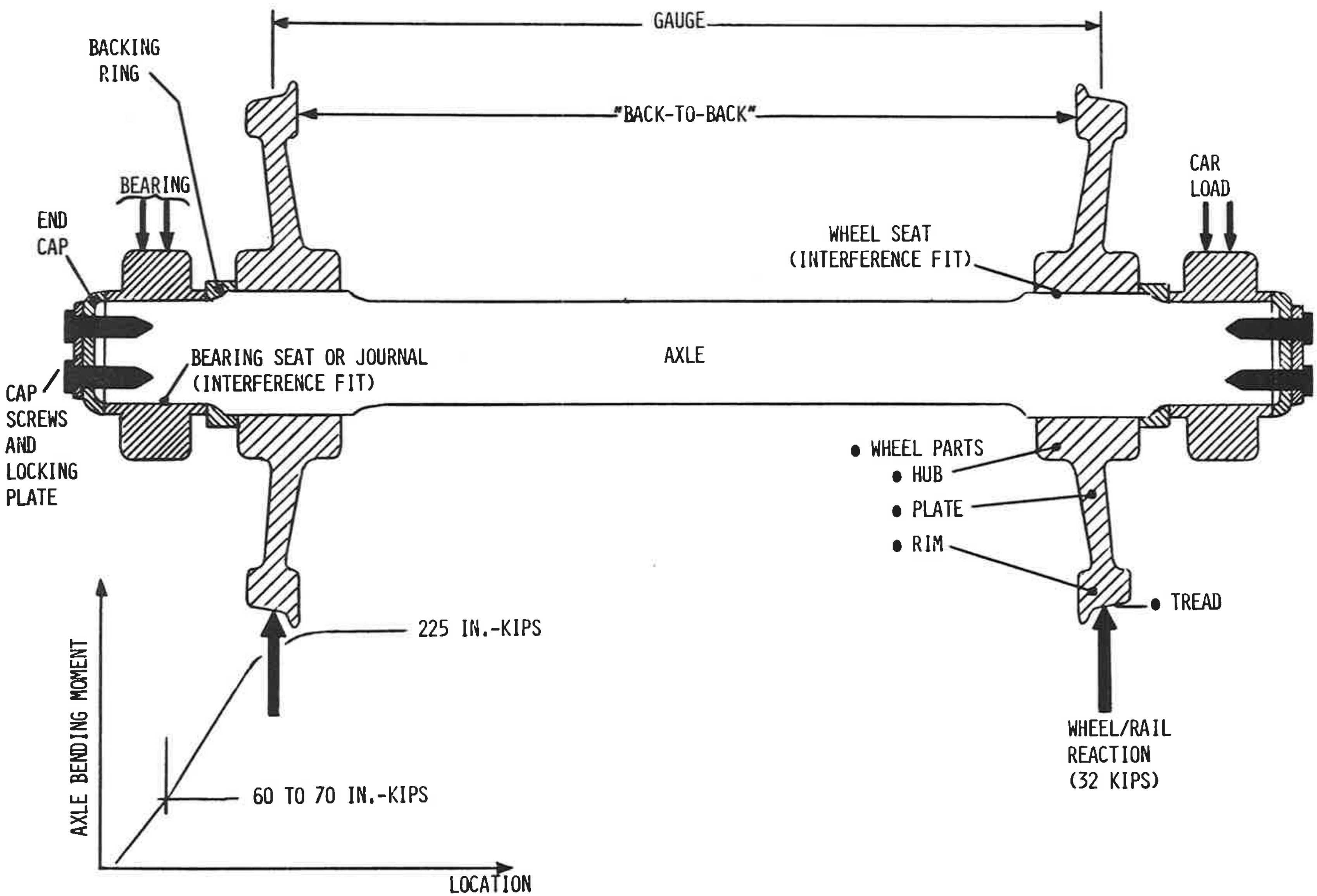


FIGURE 4-4. CONFIGURATION AND LOADING OF OUTBOARD-BEARING WHEELSET

Thus, the end cap establishes an end clamping force on the bearing stack (see Figure 4-4) as the cap screws are tightened. A torque wrench must be used to tighten the cap screws, and each must be set up with 360 to 390 lb.ft. of torque for an F-bearing. If the minimum torque specification is met, there will be about 22 tons of end clamping force on the bearing stack.

4.3 Inboard-Bearing Wheelset

Figure 4-5 illustrates the configuration of a typical inboard-bearing wheelset and shows the static loading corresponding to an M-2 "A"-car with full seated passenger load. (The forces and maximum bending moment shown here were measured on car 8562 during an operational loads test conducted in September 1982.) Note that the bearing seat is in the zone of greatest bending moment for the inboard-bearing axle. Also note that even though the M-2 car loading is about 50 percent less than the 100-ton freight car loading, the bending moment (and hence axle curvature) at the M-2 bearing seat is about twice as much as at the freight bearing seat (see also Figure 4-4).

The assembly procedure for the inboard-bearing wheelset must achieve the same objectives as required for the outboard-bearing wheelset. The approach is different, however, as is the sequence of operations:

- Press drive gear and ground ring onto axle.
- Press bearings onto axle.
- Press wheels onto axle.
- Check for clearance at ends of bearing stacks and measure wheel gauge.

Parts of the drive gear must first be pressed onto the axle and the gearbox installed if the axle is powered. These operations do not affect the bearings, however, and need not be discussed for the present purpose.

The second operation is to press the bearings and check them for running lateral, as described in Section 4.1. The bearing stack is butted against a raised part of the axle ("locating ring") and rests over relief grooves* under part of each seal wear ring. The pressing and spiking forces are similar to those in the outboard-bearing application; the 35-ton upper limit on spiking is particularly important to prevent permanent deformation of the locating ring.

The wheels are now pressed on with the same lubrication practice and the same ability to control forces as for the outboard-bearing application. Here arises a significant difference, however, in the approach which must be used to clamp the bearing stack. After the wheel has reached its 70 to 120-ton pressing force limit, the operation is continued until the back of the wheel hub engages the outboard seal wear ring, and the wheel is then spiked with an additional 20 to 35 tons.

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*The relief grooves serve the same function as the shoulders on an outboard-bearing axle, namely to prevent the pressing operations from leaving residual tension in the axle.

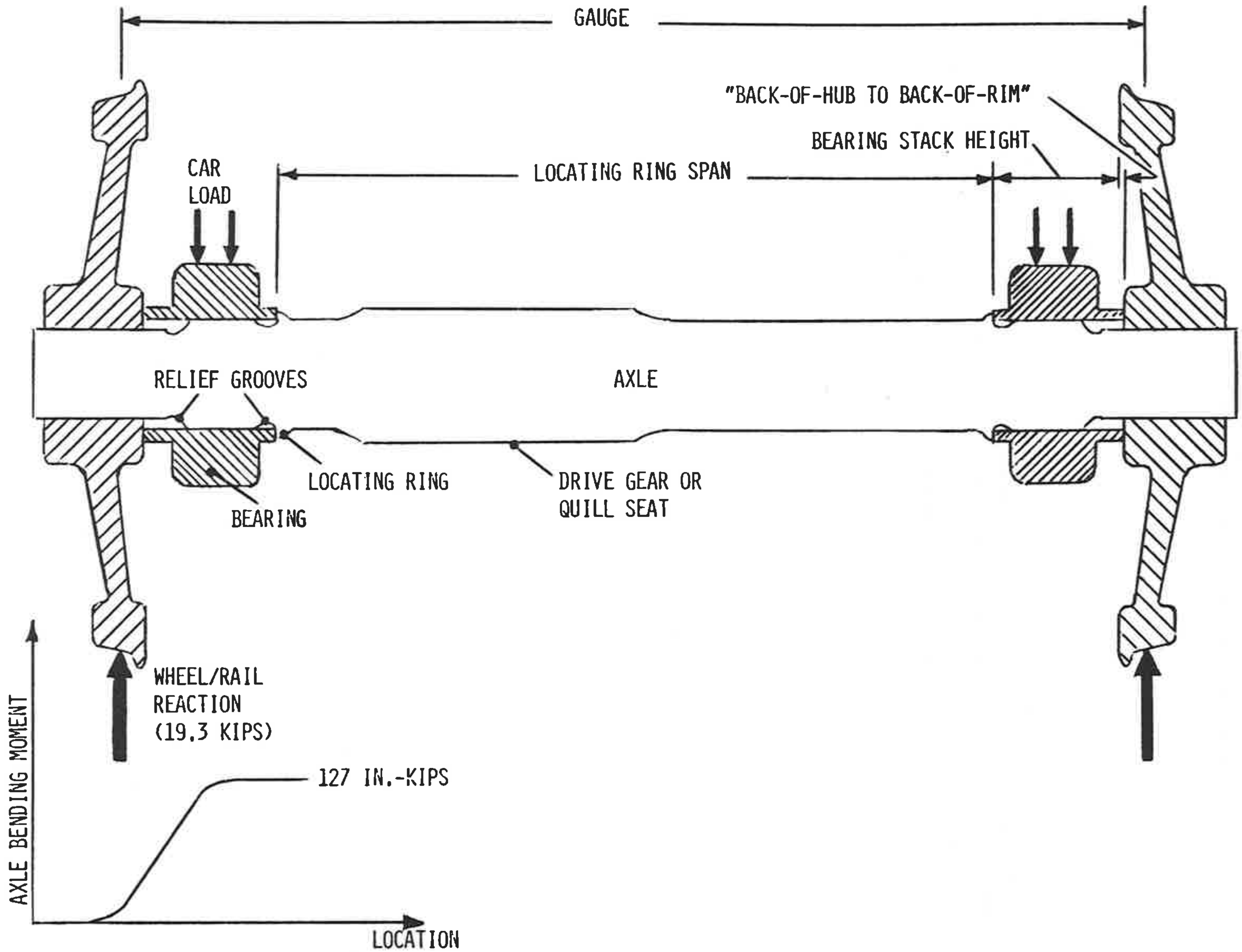


FIGURE 4-5. CONFIGURATION AND LOADING OF INBOARD-BEARING WHEELSET

In contrast to the force-controlled clamping supplied by the end-cap hardware on an outboard bearing, the wheel-spiking operation does not provide positive control because the axle and bearing stack relax when the press is relieved. The resultant clamping force after this "springback" effect depends upon the combination of wheel pressing and spiking forces achieved, and upon the relative flexibilities of the axle and bearing stack. In extreme cases it is possible to finish with a small end clearance resulting in no clamping force. Hence, the final step in the assembly procedure is to check for clearance by attempting to insert a feeler gauge between the seal wear rings and the locating ring on one side, and wheel hub on the other. Clearances exceeding 0.002 inch can be detected and rejected by this procedure, but smaller clearances will enter service undetected. The detection limit is fixed by the practical consideration that a feeler gauge less than 0.002 inch thick is too flexible to use in such an inspection.

Wheel gauge cannot be controlled directly on an inboard-bearing axle because the clamping procedures take precedence. Instead, wheel gauge is measured after completing the assembly, and is controlled indirectly by dimensional tolerances on the following components with machining typically done by the sources indicated (see Figure 4-5):

- Bearing stack height (commercial or dedicated bearing rework shop).
- Locating ring span (axle supplier).
- Back-of-hub to back-of-rim dimension on the wheel (railroad wheel shop or supplier).

The foregoing circumstances can create a risk of improper assembly. If tight dimensional control is not enforced at all sources, following the proper assembly procedure may lead to an out-of-gauge wheelset. The risk arises because the assembly shop personnel, being familiar with established practice for outboard-bearing axles, may be tempted to back a wheel off after spiking or to omit the spiking operation entirely in order to achieve a "back-to-back" dimension (see Figure 4-4) within specification.

5.0 FAILURE CAUSES AND PROGRESSION

The behavior of the axle/wheel/roller-bearing system in the railroad environment will be discussed in five phases to identify probable original causes and estimate detection capabilities. The five phases are:

- o Initial conditions of assembly;
- o Modification of the assembly conditions by thermal expansion associated with operational warmup;
- o Mechanical degradation phase;
- o Incipient failure phase;
- o Final failure.

Figure 5-1 is a schematic diagram of the failure cause and progression paths which will be discussed in detail in Sections 5.1 through 5.5. The discussion will make comparative reference to systems other than the M-2 hollow axle where necessary to highlight differences expected in other fleets.

5.1 Initial Conditions of Assembly

Initial conditions include: (1) improper assemblies and (2) limitations imposed by a particular design configuration on the effectiveness of an assembly that was accomplished according to prescribed practices.

5.1.1 Improper Assembly

One of the important factors in an improper assembly is excessive clearance at the end of the bearing stack. Excessive clearance results for inboard bearings when the wheels are improperly pressed. Improper pressing can occur when a wheel with a back-of-hub to back-of-rim dimension below specification is pressed to gauge, or is spiked and then backed off to achieve gauge. Excessive clearance results for freight bearings when a bearing (outboard) is applied with cap screws installed below minimum torque specification and/or with cap-screw locking plate missing. Improper assembly is more likely for inboard bearings than for freight bearings because tolerance control is required at more than one production station in the inboard bearing assembly process and because of the influence of standard practice for outboard-bearing assemblies.

Interference fit below specification is another important factor in an improper assembly. The specified range of fits is achieved in theory if separate but complete checks are carried out on journal and cone bore diameter. It is possible that spot checks may have been considered adequate in the past, and thus that some assemblies might have started service with less than the minimum specified interference fit.

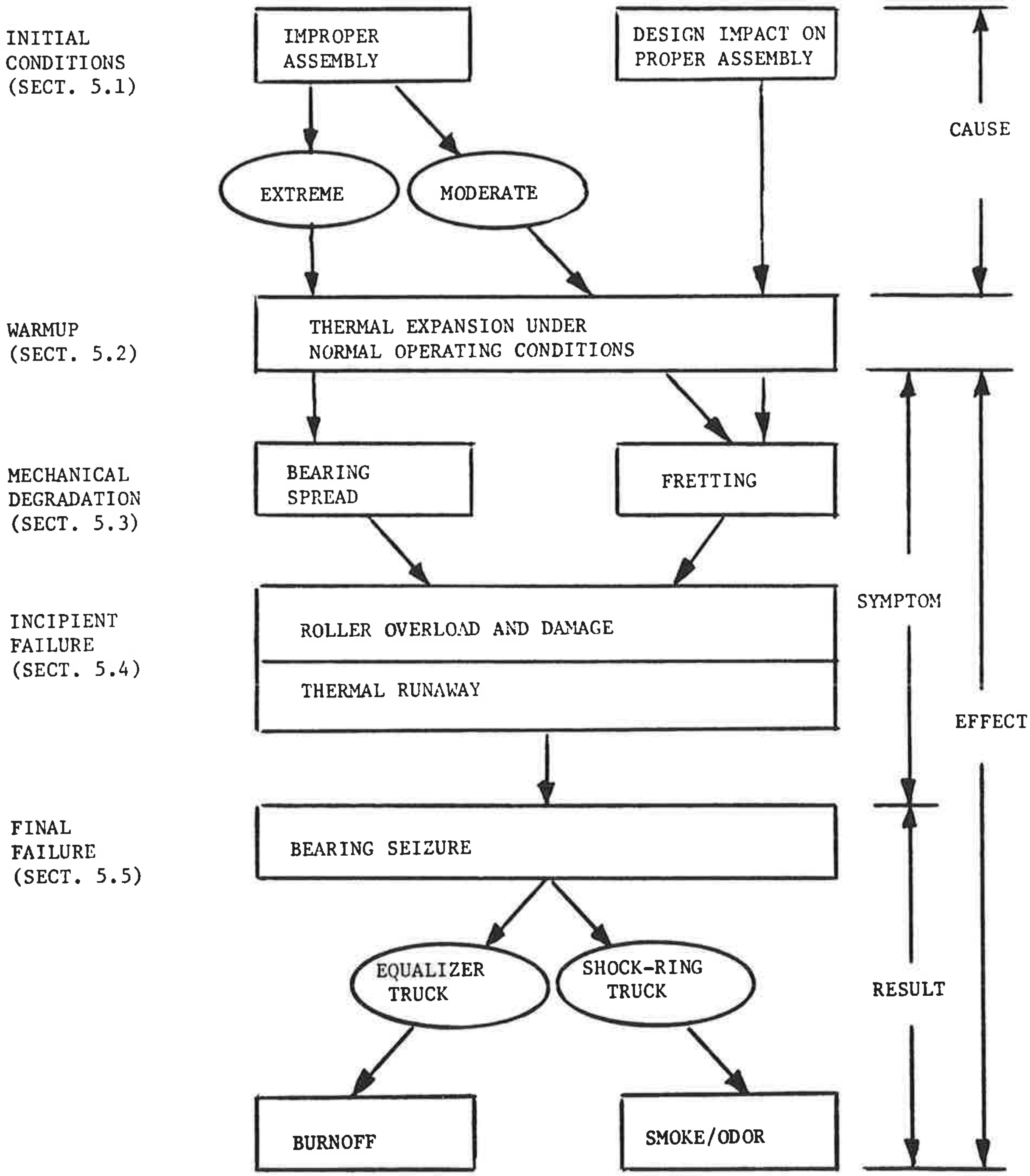


FIGURE 5-1. SUMMARY OF FAILURE CAUSES AND PROGRESSION

5.1.2 Impact of Design Configuration

The inboard-bearing system (even with a solid axle) is less tolerant to normal assembly practices and revenue environment than the outboard-bearing system. The additional effects of a hollow versus solid axle will be discussed separately. There are two specific disadvantages associated with the inboard location of the bearing:

- The configuration depends on a relaxing system to establish bearing end-clamp force. The relaxation or "springback" occurs when the press is relieved after the wheel is spiked. Calculations for a nominal (70-ton) wheel-press force show that an M-2 solid axle retains about half the clamping force established by cap screws in the corresponding freight application. Decreased clamping force means decreased resistance to relative motions between the bearing cone and the axle.

- The configuration places the bearing at a location where the axle is curved by the normal static loads imposed on the bearings and the corresponding wheel/rail reaction loads. The rate of fretting wear (see Section 5.3.2) is proportional to axle curvature in an unclamped bearing. The axle curvature at the inboard M-2 bearing is twice the curvature at an outboard 100-ton freight bearing (see Section 4). Curvatures of the magnitude experienced by railroad axles have no effect on the fatigue of bearing rollers or raceways and thus are not considered in the application design formulas established by the Anti-Friction Bearing Manufacturers Association.

Designing an inboard application with a hollow rather than a solid axle further decreases the tolerance of the system with respect to normal assembly procedures and revenue service. In this case, the decrease results from the extra flexibility of the hollow axle. The two principal flexibility effects correspond to the two inboard-application effects discussed above:

- Increased compression flexibility results in a further decrease of end-clamping force. Calculations for a nominal M-2 assembly show that a clamping force of 4.2 tons (net) is expected for the hollow axle, as compared with 10.8 tons for the solid axle and 22 tons for the equivalent freight bearing application.

- Increased bending flexibility means increased axle curvature for given static load. For the M-2, the difference between the solid and hollow axles is about 15 percent increase in curvature.

The inboard-bearing/hollow-axle system is especially critical when normal variations of shop practice in pressing and spiking wheels are considered. Wheel-pressing force depends on both the wheel/axle interference fit and the friction coefficient established by the mixture used to lubricate the fit. The pressing force cannot be controlled, and based on past experience with outboard-bearing axles, will vary between 70 and 120 tons in assemblies considered acceptable. Shop personnel can control the spiking force within a tolerance, estimated to range between 20 and 35 tons.

Calculations show that the "springback" effects under certain combinations of these conditions can leave the assembly with end clearance in excess of 0.0005 inch, i.e., enough to leave the bearing stack unclamped even after it has warmed up and expanded in normal operation (see Figure 5-2). The clearance situation is significant because the amount is likely to be less than the detectability limit of the prescribed 0.002-inch feeler gauge check and because any clearance means no end clamping force.

Another important impact of the inboard-bearing configuration involves wayside detectability of an overheated bearing. For equal load and speed, inboard bearings provide less time margin than outboard bearings for detection before critical temperatures are reached because the inboard bearing is cooled less efficiently. Also, overheat temperatures are more difficult to detect because of limited access to the bearing location. The hollow axle is more advantageous than the solid axle in this respect because the bore can provide access to a good detection point.

5.2 Thermal Expansion

The initial interference fit between bearing and axle is established at shop temperatures, i.e., at about 65 F. The bearing and journal operating temperatures are much higher, however, even under normal-operating conditions. Energy loss and heat transfer calculations show that a normally operating bearing will run at about 100 F above ambient temperature, i.e., at a temperature of 175 F on a typical warm day (75 F). Additional calculations show that, for the M-2 hollow axle, the journal will run at about 5 F cooler than the bearing, i.e., at 170 F for the typical warm day.

Normal operating conditions will thus impose thermal expansion effects upon the initial assembly, with temperature increments of 105 F for the journal and 110 F for the bearing. These increases change the interference fit because the bearing material expands more with heating than the axle material. Calculations show that the following effects result from operating a 65 F assembly on a warm day:

- If there was an initial clamping force in the assembly, the force is increased by 5.8 to 6.3 tons, the higher figure corresponding to a solid axle. The increased clamping force reduces the press-fit (radial) pressure, but the net effect benefits the assembly, i.e., increases its resistance to degradation.

- With or without initial clamping force, the bearing cone bore expands more than the journal. The net effect is equivalent to a loss of 0.0008 inch of fit on the diameter, i.e., about 40 percent of the representative 0.002-inch light initial fit. This effect is detrimental, i.e., it decreases the ability of the assembly to resist degradation.

In a force-clamped assembly, the net result of the two effects is an increase in resistance because the benefit from the added clamping force outweighs the detriment from partial loss of fit. Assemblies which have lost the clamping force by "springback", however, will experience a net decrease in resistance if they are in the "critical region" defined in Figure 5-2.

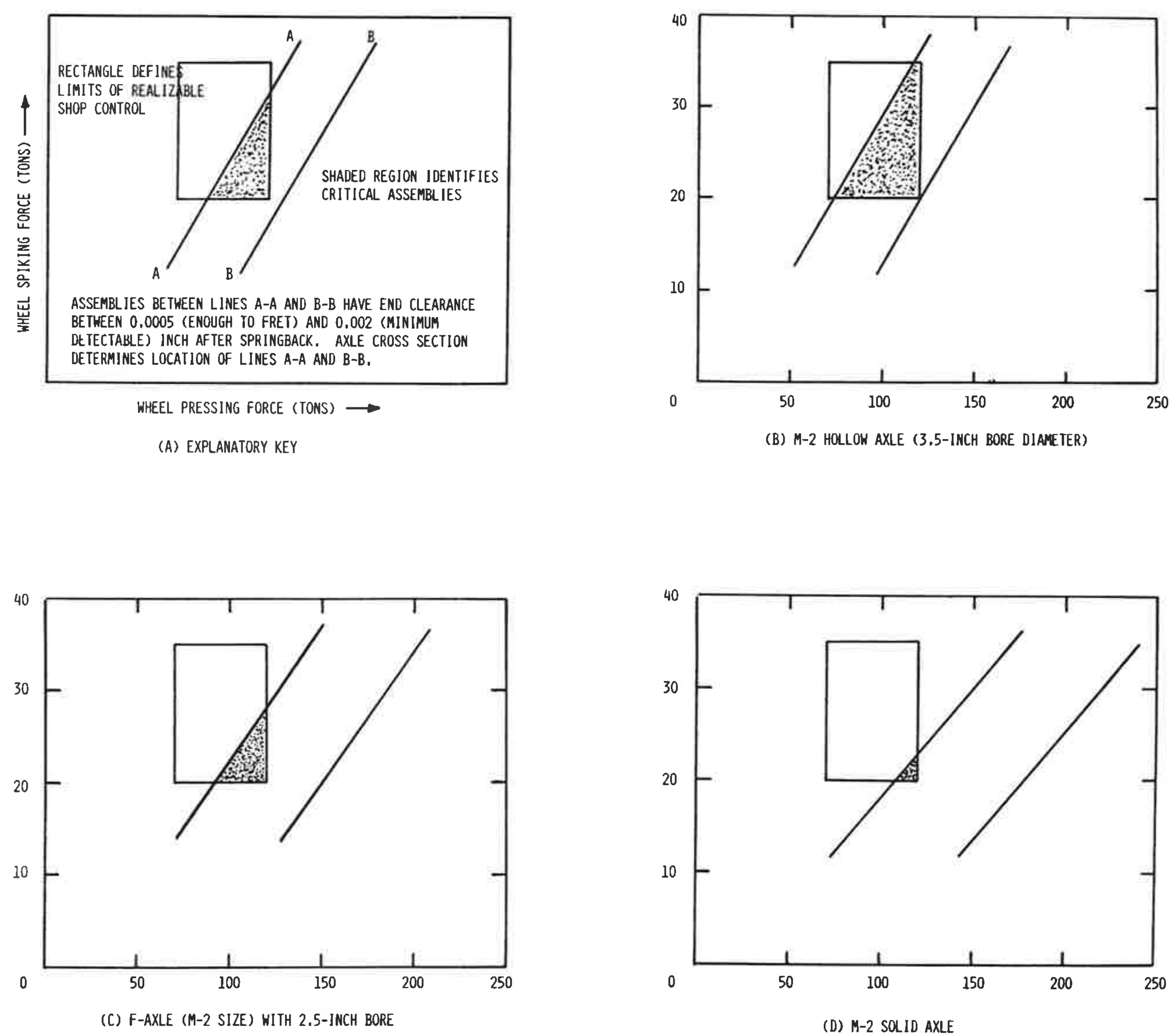


FIGURE 5-2. VULNERABILITIES OF INBOARD-BEARING AXLES TO CRITICAL CLEARANCE

5.3 Mechanical Degradation Phase

The mechanical degradation phase involves processes which consume months or years (i.e., tens to hundreds of thousands of miles) and which aggravate the initial assembly conditions to a point at which the moving components of the bearing become overloaded.

Two possible paths of mechanical degradation have been identified, one faster than the other in relative terms. Which path is followed depends upon the initial assembly conditions (as improved or degraded by the thermal expansion discussed in Section 5.2) and the specific system configuration. These paths are examined in Sections 5.3.1 and 5.3.2; both are found to be significant.

5.3.1 Bearing Spread

An improper assembly (excessive clearance) will permit the bearing to spread, even if the initial interference fit is within the specified tolerance. In extreme cases of improper assembly this is the faster path to ultimate failure. Two sources of dynamic loading are available to spread an unclamped bearing with excessive clearance:

- When carbody vertical loads are transferred through the bearing cones, lateral thrusts also develop. The lateral thrust provides the spreading force. For a 0.002-inch initial fit degraded by warm-day operating temperatures, static load is sufficient to spread a freight bearing or an M-2 hollow axle bearing. The M-2 solid axle requires a dynamic load factor of 1.11 to spread the bearing, but this is still a frequently occurring event. Initial fits exceeding 0.002 inch can still be spread by vertical dynamic loads, but the required load factors are larger and the spreading action will thus be slower.

- Lateral thrust forces due to curve negotiation and track appliances, can also spread bearings. The required force level is 4,000 to 4,600 lb. (the higher figure corresponding to solid axles) for the 0.002-inch initial fit on a warm day. This amount of thrust is expected routinely, even assuming that both bearings on the axle share the available thrust equally. Again, higher thrusts are required for initial fits exceeding 0.002 inch, and the spreading action will be slower.

Based on the foregoing results, it is expected that bearings in excessive-clearance assemblies will spread rapidly out to the limits of available clearance. The rolling components are then in an overloaded state.

Spread bearings can be detected by checking running lateral or bearing stack height. The critical factor in setting an inspection interval for this check is the mileage which a spread bearing can accumulate before reaching the thermal runaway part of the incipient failure phase.

5.3.2 Fretting

Absent excessive end clearance, axle/bearing assemblies are potentially subject to fretting degradation. Fretting means the repeated and frequent occurrence of minute relative motions between components, in this case between the bearing cones and the journal. Motions in the range of 0.0002 to 0.002 inch (5 to 50 micrometers) are sufficient to cause fretting. The fretting process gradually wears material away from the contact surface. In the case of press fits such as the axle/bearing system, the wearing reduces the interference fit, and the wear rate increases as the fit is reduced. This process results in the slower path to ultimate failure.

The following mechanisms were investigated as potential sources of fretting, with the results indicated:

- Axle flexure may cause relative motion with respect to the cones of an inboard bearing. (Outboard bearings are not as severely affected because they are located on the part of the axle with less curvature.) To cause relative motion, the available static or dynamic axle bending moment must first overcome any clamp force in the bearing stack and then overcome the resistance at the cone-bore/axle interface.

    - In bearings held by clamp forces, even the extreme dynamic events (e.g., dynamic load factors of 1.6 to 2.0 which occur at rates of one event or less per mile as determined in operational tests on M-2 cars in simulated revenue runs) do not provide sufficient bending moment to overcome the clamp force in a 0.002-inch fit assembly on a warm day.

    - Where end clearance exists, however, the static bending moment on an M-2 axle (full-seated car load) is sufficient to overcome the interface resistance and to produce relative motions in the range of 12 to 14 micrometers, i.e., well within the fretting regime. The occurrence rate for these motions is once per axle revolution, i.e., about 600 events per mile.

- Axle twisting may cause relative motion at an inboard bearing on a powered axle. (Outboard bearings are not affected because they are located on an untwisted part of the axle.) To cause relative motion, the available torques from lateral thrust and drive motor must first overcome the end clamping and interface resistances. Calculations show that, even under the most favorable assumptions for motion, the M-2 system will not develop more than 4.4 micrometers of relative motion. This figure corresponds to the unclamped hollow axle. The axle twist mechanism must therefore be viewed only as a secondary contributor at most, particularly when one notes that the occurrence rate for drive motor torque to cause the motion, as determined by operational test, is of the order of one event or less per mile.

- "Ovalizing" was investigated as a potential mechanism. The term refers to the fact that a hollow axle tends to deform to an elliptical or oval shape under the bearing load. Approximate calculations show that the M-2 hollow axle may develop a gap between the journal and cone bore for about 67 degrees of arc around the underside of the axle, considering full-seated static load. (The existence of such a gap under these conditions would imply that proposed ultrasonic inspection to detect wear gaps would be subject to a high rate of false alarms.) Further calculation was made to estimate the amount of relative motion associated with the ovalizing condition. The result was from one to ten percent of the minimum amount required for fretting, i.e., ovalizing is not viewed as a significant mechanism in the case of the M-2. Calculations made by MTA consultants, apparently of a more extensive nature, suggest that ovalizing produces a fretting motion of the same order of magnitude as produced by axle flexure. The exact significance of ovalizing thus remains to be determined, and may affect estimates for the rate of the fretting process. However, the ovalizing mechanism does not affect any of the major conclusions regarding inspection of the existing M-2 fleet or the M-2 retrofit program.

- Finally, cup walking was investigated for possible correlation with a fretting source. Roller bearing cups are observed to "walk" slowly around the axle during service. This is expected behavior; its absence would be cause for concern. Axle/bearing fretting would be related to cup walking if the walking phenomenon were caused by high torques. Such torques would have to be much larger than the normal-operation friction torque in a roller bearing in order to make the cup walk, and would thus also be able to move the cone on the axle. The absence of any other mechanism for walking the cup without large torques would constitute compelling indirect evidence that the rollers are subject to momentary operation at high friction and high torque, which would then cause fretting motion between the cone and the axle. An alternate creep mechanism has been found, however, for which rough calculations indicate that the cup can walk without effect on the cone/axle interface. Cup walking is presently considered to be uncorrelated with cone/axle fretting, but the true level of correlation can only be determined by test.

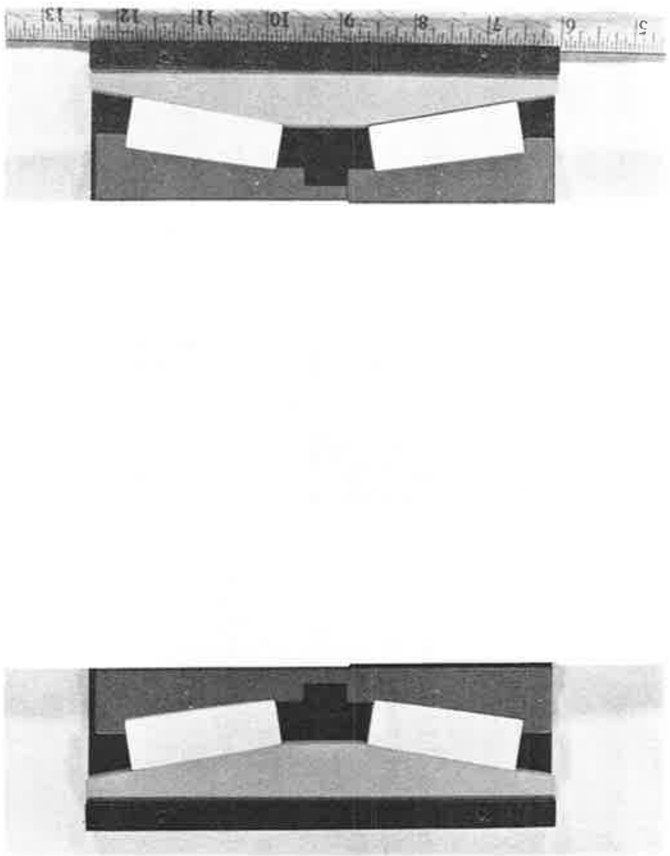
The foregoing assessment suggests the following specific sets of initial assembly conditions which can cause serious fretting:

- Improperly assembled solid or hollow axles with modest end clearance, i.e., more than 0.0005 inch but less than the excessive amount which would "short-circuit" the mechanical degradation phase by permitting the more rapid bearing spread mechanism to dominate.

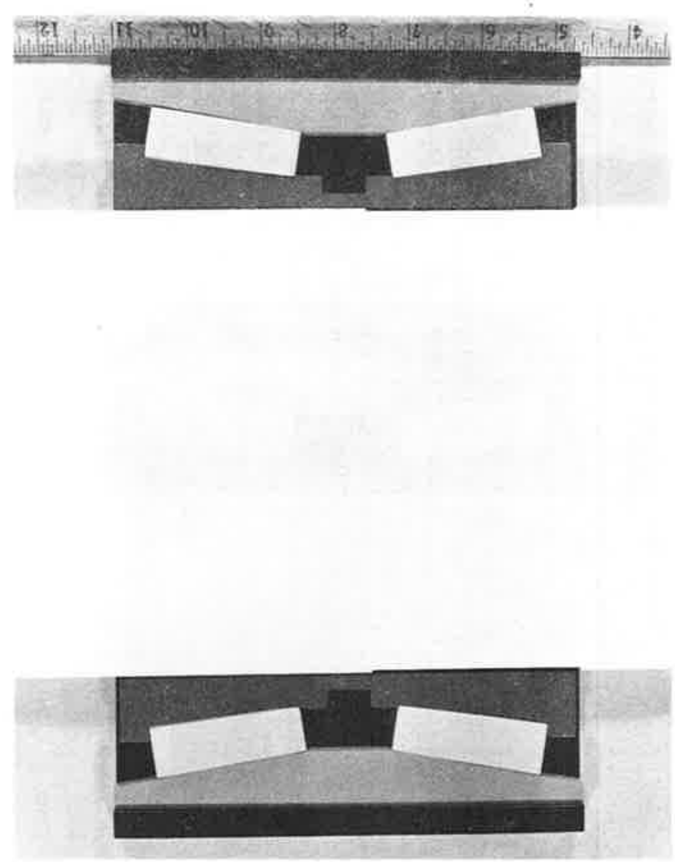
- Properly assembled hollow axles with critical combinations of wheel pressing and spiking forces within accepted shop limits (see Figure 5-2).

Both of the situations described above are believed to have occurred in the M-2 fleet hollow axles.

Axle/bearing systems subject to fretting will experience a wearing away of material from the cone bore and from the part of the journal under the cone. An independent calculation by an expert consulting for the Metropolitan Transit Authority indicates that about 200,000 miles of service is required



(C) BEARING WITH PROPER STACK HEIGHT BUT RIGHT CONE RUNNING ON AXLE GROOVE; CUP SHIFTED TO LEFT AND LINE CONTACT ESTABLISHED ON LEFT ROLLER; NOTE LOSS OF CONTACT WITH RIGHT ROLLER



(D) SAME AS (C) BUT CUP SHIFTED TO LINE CONTACT WITH RIGHT ROLLER; NOTE CONCENTRATION OF CONTACT ON LEFT ROLLER

FIGURE 5-3. KINEMATIC MODEL OF NORMAL AND ABNORMAL CONDITIONS IN M-2 BEARING (CONTINUED)

- o Both of the foregoing effects will contribute to increases in the steady-state and transient torques which the bearing develops. These increased torques will accelerate the axle grooving process and may be sufficient to start grooving the axle under a spread bearing. One probable instance of the latter situation has been observed in the M-2 fleet (axle removed from car 8669).

Discussions with representatives of the roller bearing manufacturers developed the information that the duration of the roller overload and damage sub-phase cannot be determined by engineering calculations. The principal reason for this is that the rolling components are operating in a configuration well removed from the normal operating conditions which have been tested. It is known, however, that the damage rate will be higher for bearings operating at static loads which are higher in proportion to their rated loads. Thus, for example, the duration of this sub-phase would be expected to be much shorter for a standard-F freight bearing operating under a loaded 100-ton car than for the short-F bearing operating under the M-2. Conversely, it should be much longer for the E-bearing under a Bay Area Rapid Transit car because the load on that bearing is much lighter, relative to the rated load, than is the M-2 load on its bearing.

5.4.2 Thermal Runaway

At some point in the damage sub-phase, heat production in the bearing may concentrate in one cone and increase from a normal operating rate of about 2,900 Btu/hr to about 4,000 Btu/hr. At this point the steady-state bearing temperature which would be required to reject the heat to the surroundings exceeds the grease destruction temperature (about 300 to 325 deg F), and the bearing cannot remain in steady-state operation. This is the beginning of thermal runaway. Heat transfer calculations show that:

- o The initial part of the runaway is gradual; about one hour is required to take an M-2 bearing from 250 F to 300 F on a hot (100 F) day, assuming that the car is operating with full-seated load at 55 miles per hour.

- o The grease vaporizes and/or carbonizes at about 300 F; this process does not consume any significant amount of time and is neglected.

- o The bearing is now operating "dry", and the heat generation rate increases to at least 10,000 Btu/hr. For a full-seated car at 55 mph, it is calculated that 49 minutes are required to take the bearing from 300 F to 600 F. However, the actual heat generation rate for a dry bearing should be verified by testing to determine the actual time available.

At 600 F the rollers have entered the regime where they can soften and deform under load; this is defined as the end of thermal runaway and the end of the incipient failure phase.

5.4.3 Additional Field Experience

On January 21, 1983, confirming evidence was obtained when car 8525 was temporarily removed from service at New York Grand Central Terminal because of a suspected pellet melt in one axle. This axle was removed from the car and disassembled for inspection at the Harmon Shop. Figure 5-4 is a photograph of the suspect journal, showing a groove, approximately 0.12 inch on the axle diameter, under the inboard cone position. This cone had lost all grease except for parafin residue, and the rollers were heat-discolored corresponding to a temperature of approximately 450 F. In addition, the small ends of the rollers in the outboard cone were heat-discolored and spalled (Figure 5-5), and spalling appeared at corresponding locations on the cup. These observations suggest that the rolling components had been overloaded in the manner represented by the model illustrations of Figure 5-3 (C/D), and that the equipment had reached the starting point of thermal runaway.

5.5 Final Failure

Once the rollers have reached 600 F and started to soften, the bearing quickly seizes and the final failure phase has started. What happens in this phase depends principally on the type of truck in which the bearing is operating. The following three situations appear to represent all of the possibilities:

- In a freight truck, the lowest-friction interface can be between the bearing and its journal. The heat generation rate increases and the axle rapidly reaches 1400 - 1600 F. A burnoff occurs with the carbody and bolster dropping to interfere with the running gear. FRA accident statistics analyzed by the Association of American Railroads show that burnoffs on these outboard bearing axles often result in a derailment.

- The M-2 and other passenger fleets have a pedestal (equalizer-beam) type inboard-bearing truck. Again, the lowest-friction interface after seizure can be between the bearing and the axle, and the failure will become a burnoff. The final failure phase is estimated to be short (of the order of a minute, or about one mile at 55 mph), and little difference is expected between passenger and freight duration. However, none of the six burnoffs which have occurred on M-2 cars and similar Arrow cars has resulted in a derailment. This difference is believed to result from the circumstances that the wheel rather than the bearing is detached from an inboard-bearing burnoff, and that the remnants of the truck can somehow utilize the drive-gear load path to continue supporting the carbody load in a tricycle fashion. There is no justification, however, for relying on such behavior to avert a derailment, particularly if unpowered trucks must be considered.

- The M-1 and other passenger fleets employ a truck in which the primary suspension is replaced by a rubber shock-isolation ring wrapped around the bearing housing. In this case, the lowest-friction interface after seizure appears to be between the bearing and the shock ring. The destroyed bearing rotates with the axle and thus protects the journal from thermal extremes and burnoff, while friction burns the

FIGURE 5-4. GROOVED JOURNAL FROM CAR 8525 AFTER PELLET MELT

FIGURE 5-5. OUTBOARD CONE WITH SPALLED ROLLERS FROM CAR 8525

shock ring. The shock-ring type design inadvertently has its own "detection" system, i.e., the smoke and odor from the burning rubber is easily detected. However, no data is available to establish a safe burn time or distance.

6.0 EVALUATION OF COUNTERMEASURES

The M-2 wheel loss incidents have led to the adoption of several countermeasures aimed at reducing the risk of additional failures and suggestions for other risk-reducing ideas. The purpose of this section is to evaluate the adequacy of countermeasures already adopted and others that have been offered as possible candidates. Particular attention will be given to those countermeasures specified in the November 11, 1982 agreement (see Section 1.0).

6.1 Factors Related to Inspection Requirements

The utilization of appropriate inspection techniques provides the last defense against catastrophic failure. In the case of the M-2 vehicles, the interim steps taken, while decreasing the risk of failure significantly, are costly in terms of both delays and financial burdens to the operator. The analyses of Section 5 provide justification for the burden imposed by these short-term safety measures and provide insight into the inspection requirements for a long-range solution.

6.1.1 Shop Inspection

Inspections for excess running lateral or stripping the axle and inspecting the journal surface require shop facilities and are impractical unless long inspection intervals can be established. This safe interval would correspond to the mechanical degradation phase plus the roller overload and damage sub-phase. The mechanical degradation phase is the dominant part of the duration for failures controlled by fretting, and the safe interval in this case is often compatible with typical wheel-change schedules. If the failure is controlled by an excessive-clearance improper assembly, however, the mechanical degradation phase may be short, and the roller overload/damage sub-phase may be the dominant part. Testing is required to quantify this sub-phase in order to determine whether the safe interval overlaps wheel-change intervals, or whether automated wayside inspection, onboard detection, or other measures should be required to protect fleets against the possibility of improper assemblies.

To avoid the necessity of dropping the axle, other alternatives have been considered. For example, an ultrasonic technique for detecting loss of fit at the axle/bearing interface is under investigation. The inability of conventional piezoelectric transducers to establish a satisfactory contact with the inner surface of the hollow axle appears to be overcome by using an electromagnetic acoustic transducer and detector. Preliminary tests demonstrated the ability to detect loss of fit but also revealed missed detection and/or false detections depending on the orientation of the transducer. A second transducer configuration is presently being constructed to try to improve the situation. If the ultrasonic technique can be developed sufficiently to provide reliable detection of loss of fit without false alarms, it can be applied more frequently and at less cost than axle stripping. However, it is doubtful that the ultrasonic technique can reliably detect spread bearings.

Another possible approach for frequent inspection is the measurement of available bearing stack space. A simple go/no-go gauge could be considered. This technique could provide a less costly control to protect a fleet against spread bearings.

6.1.2 Heat Generation and Propagation

The heat generated by a properly functioning bearing can be readily transferred to the atmosphere by the axle/bearing system with an adequate margin of safety. However, when the axle/bearings are in the failure process the system's limited capacity to transfer heat results in increased bearing temperatures. This accelerates the failure process.

Some form of wayside inspection is essential to detect systems which have progressed into the later part of the incipient failure phase. The analysis in Section 5.4 indicates that the temperature of the bearing begins to rise above normal (about 175 F) in the incipient failure phase and that at about 250 F the thermal runaway is approaching the point at which the grease will burn, leaving the bearing in critical condition. A heat detection system that triggers a warning when the bearing temperature reaches 250 F would prevent a wheel loss since it is estimated the failing bearing must travel at least 55 miles at 55 mph between the events of 250 F temperature and the grease-burn point. It is desirable that any heat detection system be automated to minimize errors and reduce delays for inspection. Such an automated heat detection system could be either onboard the vehicle or located at track wayside.

Viewing inside the axle bore under the bearing is a good place to inspect; the temperature at this location will stay within 5 to 10 F of the bearing temperature. Partial loss of mechanical contact between the cone bore and axle was considered in this assessment and was found not to materially affect the heat conductivity between the cone and the axle. The reason for lack of effect is that the mechanical contact under the load zone is shared around the rotating axle with sufficient frequency to maintain the transfer of heat.

Results of the heat transfer analysis depend on several variables. The heat generation rate of the bearing and ambient temperature have a dominant effect on the calculated temperature at the candidate measurement locations on the axle/bearing system. Figure 6-1 shows the calculated temperatures at key locations for normal bearing operation and a failing condition for ambient temperatures of 30 F and 70 F.

The results of the calculations were compared with bearing manufacturer and operator knowledge about normal operating temperatures, and with the results obtained from a limited number of freight bearing destruction tests conducted at the Transportation Test Center (TTC). The comparisons showed that the calculations were generally consistent with both sets of data. The TTC report also appears to show that car weight and speed are critical parameters in the onset of thermal runaway. It seems of particular value to pursue further tests of this nature.

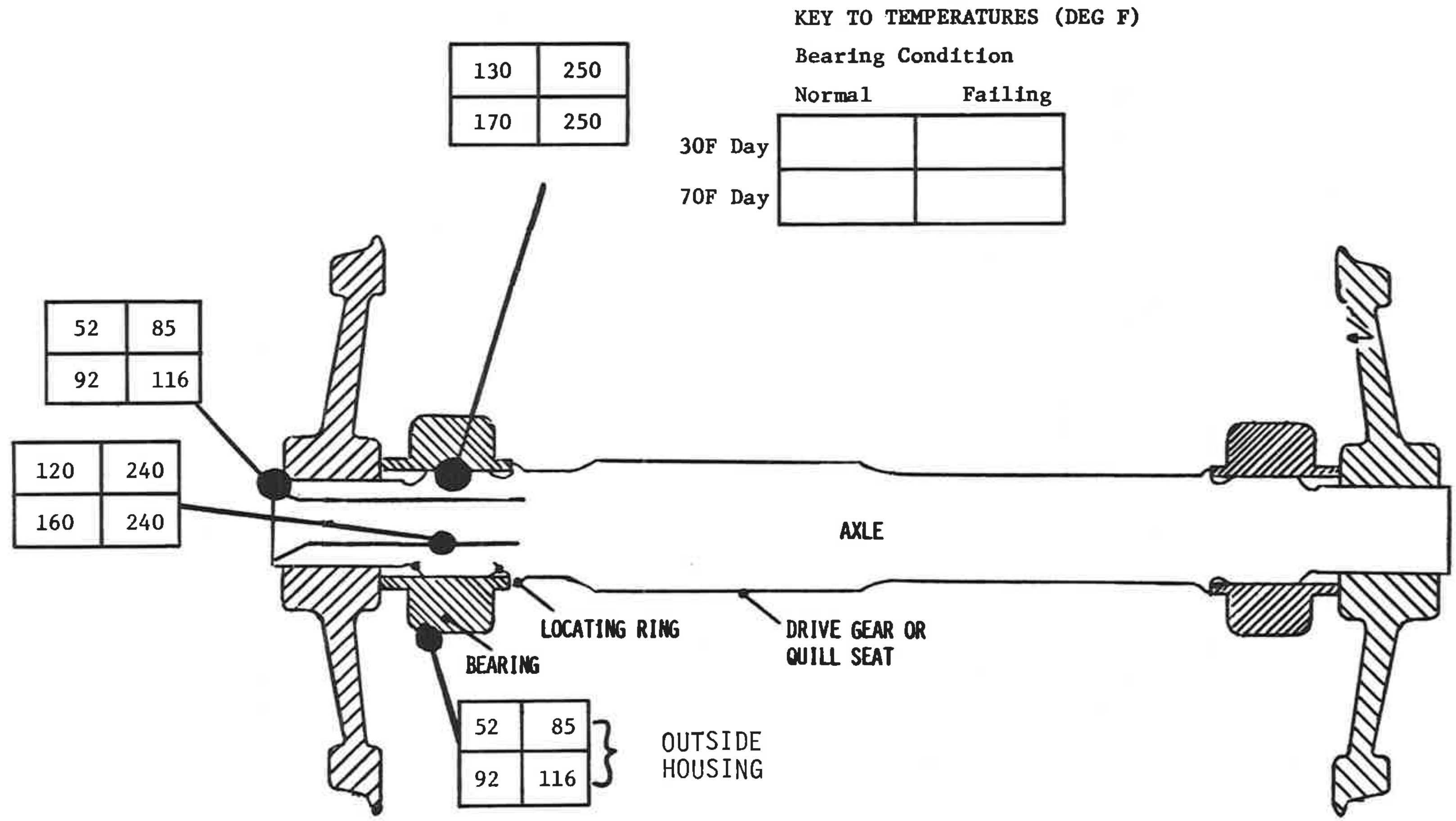


FIGURE 6-1. CALCULATED TEMPERATURES AT CANDIDATE MEASUREMENT LOCATIONS

6.2 Implications for Current Wayside Inspection

If the last wayside inspection occurs just before the 250 F point in a thermal runaway condition and if no inspection is conducted at the next terminal, then the inspection interval is effectively doubled. In the case of Norwalk to New Haven, the interval would be 68 miles. Even if the failure is in a "layover" car which allows the bearing and axle to cool to ambient temperature, then calculations show that the thermal runaway may restart and reach burnoff before the first wayside inspection on the next run. Therefore, inspection of pellets in all trains at New Haven is an essential supplement to the en route program. Appropriate temperature measurements made immediately after train arrival would accomplish the same purpose.

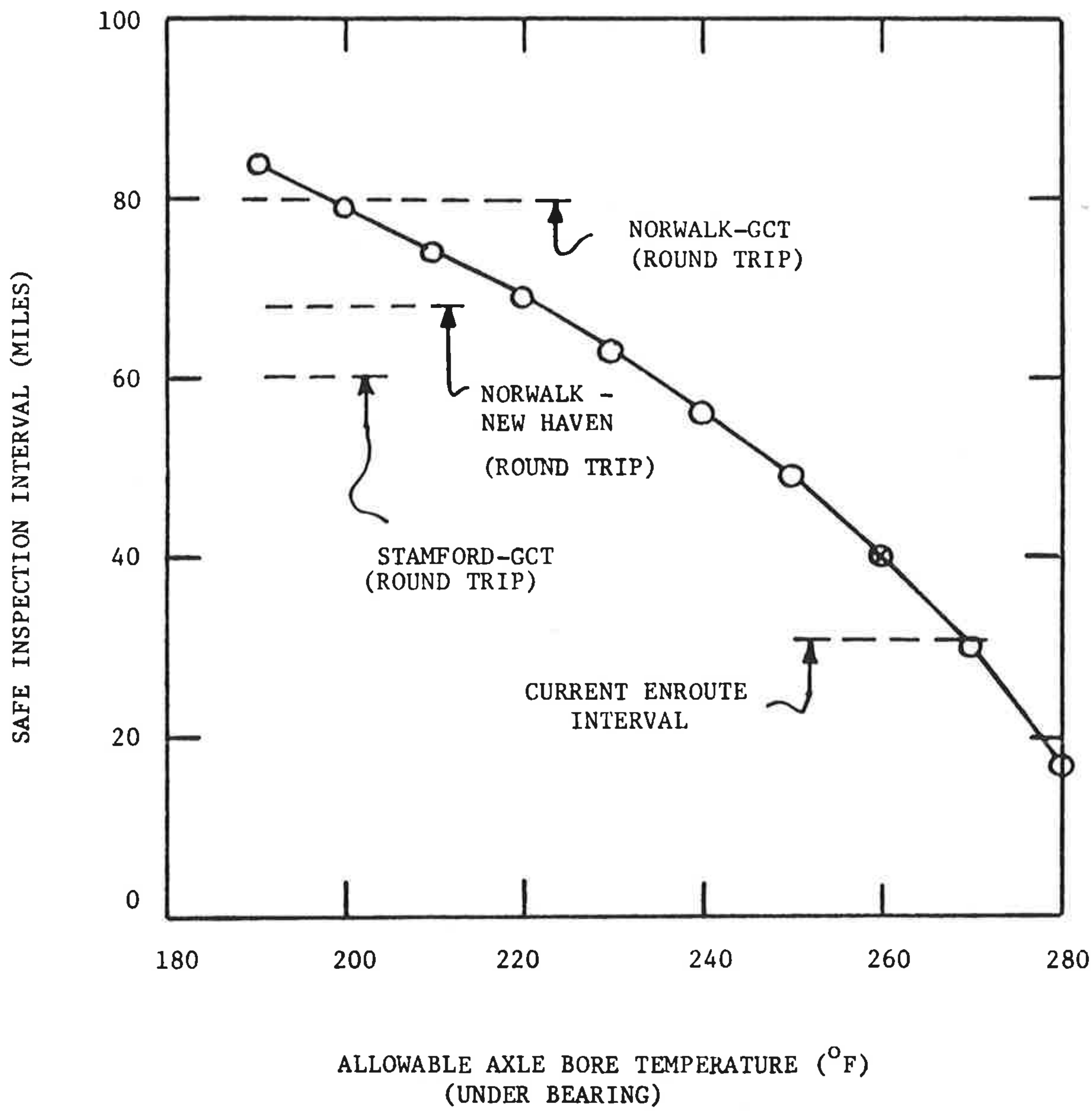
In the absence of any other external observations (feel, smell), measuring temperature at the axle end, e.g., by watching for indications of melting from a 200 F melt stick, is a misleading procedure. Calculations indicate that the end of the axle will remain close to ambient temperature (within 35 F), and will not exceed 200 F until well toward the end of a thermal runaway.

"Feeling" the axle end is better than the 200 F melt stick procedure, but is still not reliable enough. The descriptive term "feeling" is not accurate because the effectiveness of the procedure depends on having the palm of the hand over the axle bore so that the palm receives a radiation input from the hot zone under the bearing. The questions of reliability involve the effective radiant input for 250 F, the inspection geometry, and the variability to be expected in sensitivity of the human palm.

To be reliable for detecting incipient failures on 30 F days the current melt-stick procedure would require an 85 F rather than a 200 F melt stick. This approach is of doubtful value, however, since it would create 100-percent false alarm rate on warmer (60 F) days.

A better approach is to base the inspections on temperature sensing closer to the bearing. A hand held digital thermometer probe (pyrometer) has been adapted to allow measuring the temperature on the inner surface of the hollow axle opposite the bearing seat. This approach overcomes the objections to the use of a melt stick on the end of the axle, but is limited to use with a hollow axle. The durability of this device in rain or snow and interference with pellets already inserted in the axle will have to be resolved. Another option would be to use an extended melt stick which will reach inside the axle bore directly underneath the bearing.

Utilization of a positive indication of temperature in the axle bore under the bearing would allow the possibility of relaxing the inspection interval. The safe inspection interval was calculated as a function of the allowable axle bore temperature under the bearing to pass inspection (See Figure 6-2). Pellets with the appropriate melt temperature could be used or the actual temperature could be measured with a pyrometer for this inspection approach. If 190 F under the bearing is used as the criterion for taking a vehicle out of service, Figure 6-2 shows that an 80-mile interval between inspections would be safe. Additionally, use of the existing 213 F pellet in the bore will allow a 66 mile interval (with some margin).



BEARING TEMP. = BORE TEMP + 10°F

FIGURE 6-2. SAFE INSPECTION INTERVAL VERSUS ALLOWABLE AXLE BORE TEMPERATURE

The advantages of this approach are illustrated in Figure 6-3 which compares the current inspection points with either a 190 F or 213 F detection criterion. Stamford train inspections could be reduced to one inspection per round trip. This could be done by inspecting the 213 F pellet at Stamford, thus avoiding any delays since the trains normally lay over at that terminal. Similarly, the New Haven Train inspections could be reduced to one inspection at Norwalk for each one-way trip using a 190 F pyrometer criterion.

6.3 Implications for Automated Wayside Inspection

Infrared (IR) radiometers are used to monitor bearing temperature at automated wayside inspection stations for freight service. Standardized systems are commercially available and are well suited to the task of detecting overheats in outboard bearings. These systems have been extensively applied to mainline freight service, where the benefits of reduced derailment rate outweigh the capital cost of the systems. These wayside "hot box" detectors are generally sited with spacings ranging from 20 to 35 miles.

The application of wayside hot box detectors to inboard bearings is currently experimental. A unit similar to the freight hot box detector would be a possible configuration, i.e., hardware located to the field side of the track with separate IR radiometers viewing the axle/wheel-hub and the wheel-rim areas. The purpose of looking toward the wheel rim as well as the hub is to be able to discriminate between overheats caused by dragging brakes and overheats caused by failing bearings.

The inboard configuration poses an unsolved technical problem, viz: the wheel hub temperatures associated with normally operating bearings, overlubricated bearings which are mechanically normal, and failing bearings may not differ enough to permit reliable detection of a failing bearing without the penalty of a high false-alarm rate. The heat transfer calculations mentioned in Section 5 were extended to consider this question, with the following results:

- The difference in temperature at the end of the axle for a normal versus a failing bearing was estimated to be about 15 - 40 F for hollow axles depending on the ambient temperature.
- The difference for an overlubricated versus a failing bearing was estimated to be about 10 - 30 F.
- It is questionable whether the field-side IR detector would be able to "see" even the modest temperature differences quoted above. To be workable, such detectors may have to view the wheel hub area as well as the axle end. The wheel hub will be cooler than the axle end, and would thus tend to reduce the measurable temperature differences below the values that could be measured from a narrowly focused spot.

An alternative detection scheme has been proposed to give the IR detector direct access to a high-temperature view. In this approach the "wayside" hardware is actually located between the crossties within the rail gauge, and a single IR detector looks upward at the underside of the bearing housing. If such a system could look directly at the bearing cup, it would be able to measure temperature close to the conditions of the rolling components of the

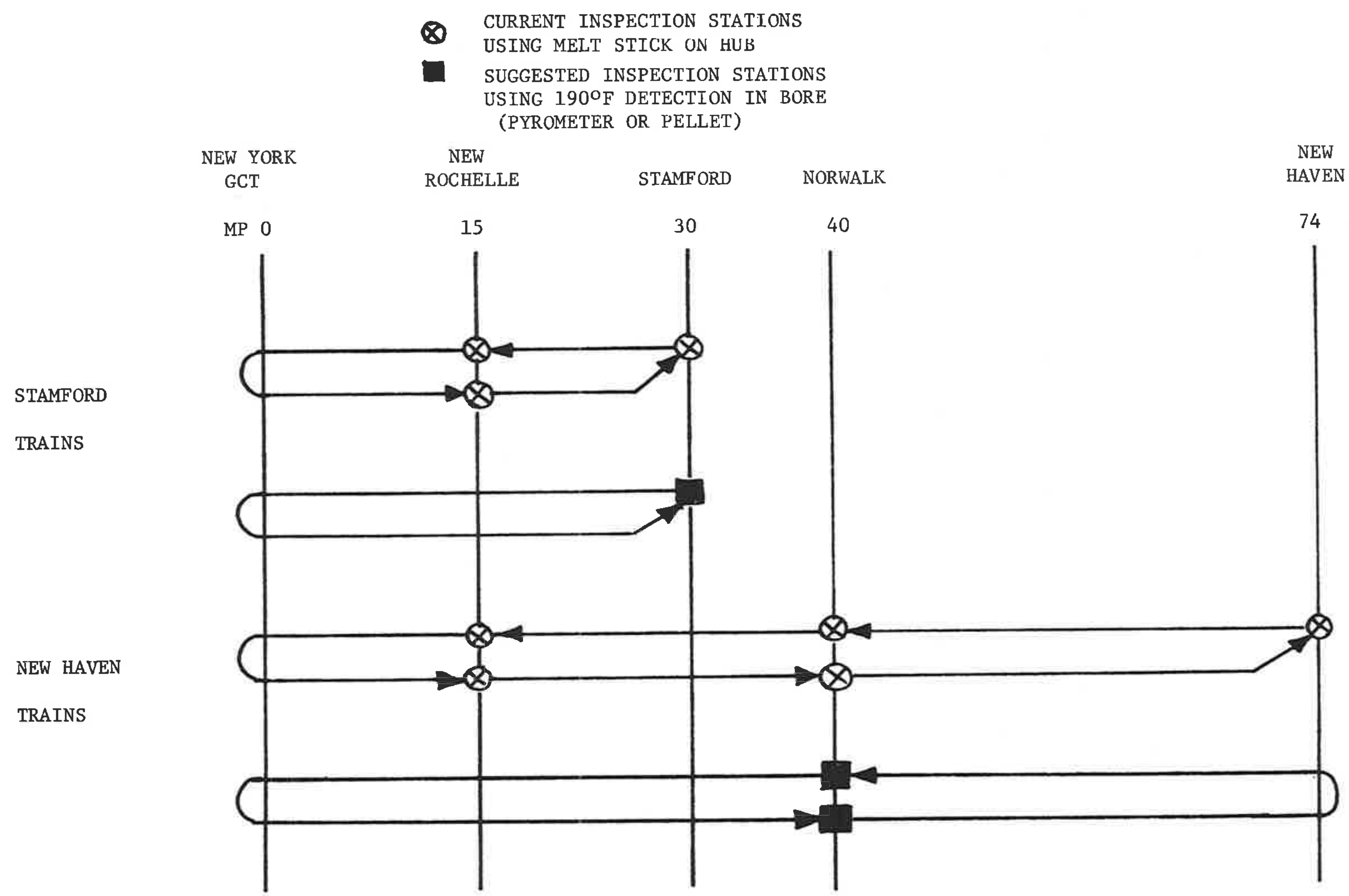


FIGURE 6-3. INSPECTION STATIONS

bearing and could easily discriminate between failing bearings (250 F) and normal or overlubricated bearings (170 to 190 F). In the case of a Brenco bearing, the present housing design may allow such access to the cup. The Timken bearing housing is closed, but modification appears reasonable. However, the practicality of such modification has not been investigated.

Detection system suppliers have been reluctant to develop a within-gauge/upward-looking system because of the extensive hardware changes it requires, vis-a-vis the field-side IR detector. The upward-looking system also has the following practical liabilities:

- o Susceptibility of the hardware to be damaged by dragging vehicle equipment.
- o Susceptibility of the system to false alarms from drive motor heat, hot water tanks, etc.

6.4 On-board Heat Detection Systems

There are two basic methods for onboard bearing overheat detection that are available and could be used on the M-2 and similar vehicles. The first is under trial by AMTRAK. This system measures bearing temperature using thermocouples on each bearing adapter. The data is collected for each bearing and an alarm is given for an overheat condition. Current reports indicate a false alarm rate that has resulted in shutting the system off on many cars. Expectations are, however, that the method can be made to work with an acceptable false alarm rate. The principal disadvantage of such a system for limited-mileage service application is cost since it has a high per vehicle installation cost which could easily exceed wayside system costs for commuter vehicle applications.

The second method for onboard overheat detection, currently in use by RoadRailer, employs a fusible plug that melts at a pre-set temperature. Melting of the plug releases brakeline air, thus activating the brakes. This feature would involve a source of potential brake reliability reduction. Cost of this method is estimated to be considerably less than the thermocouple system discussed above.

A related set of onboard heat detection methods involves the sensing of an overheated adapter resulting in the activation of a secondary detection system. These secondary systems may then be detected by train crew or by wayside personnel. This group includes smoke bombs, temperature indicators and odor. Timken manufactures pyrotechnical devices (including smoke bombs) for this purpose. The Chicago Transit Authority is using inexpensive plastic thermal sensors (similar to poultry cooking timers) adapted to the axle to sense a condition of overheat. The trucks in use on AMTRAK, PATCO, and M-1 cars clamp the adapter in a rubber shock isolation ring. It has been reported that the smell of hot rubber has resulted in the detection of several bearing overheat conditions. The CTA wires pieces of rubber tire to their adapters for just this reason.

In summary, there are several methods are available that will improve the probability of hot bearing detection. The two most promising onboard methods to either alert the train crew or stop the train have not been fully refined to reduce false alarm rate. Further, these methods are relatively expensive.

6.5 Evaluation of Solid Axle Retrofit Program

The failure process as described in Section 5 indicates that the hollow axle has characteristics that contribute to possible axle/bearing failure in several ways. Replacement of the hollow axles with solid axles in the M-2 fleet will provide the following improvements:

- The end-clamping force for nominal conditions was calculated to be 10.8 tons for the solid axle as compared to 4.2 tons for the hollow axle because of the increased longitudinal stiffness.

- The solid axle curvature under bending load was calculated to be about 15 percent less for solid axles.

- Axle twisting which is only a secondary contributor in the failure process is nonetheless reduced by using a solid axle.

- The solid axle is unlikely to enter service with end clearance on the bearing stack, even given the ranges of wheel pressing and spiking forces likely to be achieved in shop practice. However, the hollow axle with 3.5-inch bore is expected to have such clearance for about half the expected force combinations.

The contributions of these factors are sufficient to support the decision to change all hollow axles in the M-2 fleet to solid axles. However, the retrofit does not guarantee against failure if not accompanied by proper maintenance and assembly procedures. Although the hollow axle does provide convenient access for temperature measurements in the vicinity of the bearing seat, inspection convenience is not an appropriate reason for continuing the use of hollow axles on the M-2 fleet.

7.0 OPERATIONAL FLEET ASSESSMENT

About 2,400 passenger vehicles at properties throughout the United States have axle/bearing design configurations similar to the M-2 and could be expected to have similar assembly, inspection, and maintenance practices. An operational fleet assessment was therefore conducted to determine whether the axle/bearing problems experienced on the M-2 vehicles might extend to other passenger vehicles in service. The Task Force collected and evaluated data through on-site visits, formal requests, and telephone contacts. The data collected also served as a key input to the Team 1 analyses to test the consistency of the failure theories with the observed fleet behavior.

Passenger vehicles which have experienced severe axle/bearing problems include not only vehicles with hollow axles and inboard bearings but also vehicles with solid axles and outboard bearings. However, the incidence of burnoffs which occurred during revenue operation (four on the M-2 cars and two on the Arrow II cars) heightened concern for those vehicles configured with inboard bearings and hollow axles. Hence, the Team 2 assessment focused on inboard-bearing/hollow-axle vehicles. Table 7-1 identifies the fleets which were assessed and summarizes pertinent quantitative data. The information collected was evaluated in the following five categories:

- o Failure observations (Section 7.1).
- o Vehicle design parameters (Section 7.2).
- o Operating conditions (Section 7.3).
- o Maintenance and assembly practices (Section 7.4).
- o Inspection practices and intervals (Section 7.5).

Section 7.6 summarizes the results of the evaluation.

7.1 Evidence of Axle/Bearing Problems

Table 7-2 provides the pertinent details of the reported burnoff and near-burnoff events in the M-2 and Arrow II fleets.* It is significant that four of these failures occurred within 100,000 miles after rework and reassembly, and that three of the four had bearings reworked and reassembled at the same shop. It is reasonable, therefore to conjecture that these four burnoffs conformed with the faster path to failure which can be produced by improper assembly (see Section 5.3.1). Conversely, the remaining failures appear to conform to the slower failure path (mileage in excess of 170,000 miles) associated with design, operational, and maintenance factors (see Section 5.3.2).

*The Task Force is also aware of the fact that bearing seizures have occurred in the M-1 fleet, but was unable to obtain quantitative data on these incidents.

TABLE 7-1. DESIGN CHARACTERISTICS OF RAIL PASSENGER VEHICLES WITH HOLLOW AXLES & INBOARD BEARINGS

LOCATION/ VEHICLE	# of Cars	Date In-Service	Truck Type	Weight (k lbs) Min - Max	Max Speeds (mph)	Bearing Fit (In) Min - Max	Axle Diam at Bearing (In)	Axle Bore (In)	Bearing Rework Interval (Miles)	Reported Burnoffs
San Francisco										
BART	450-RT	1972-75	RS	59.8- 93.3	80	0.0018 - .0045	6.1915 $^{+.000}_{-.001}$	2.5	500,000	0
Chicago										
CTA (Boeing)	200-RT	1976-78	RS	50.5- 73.8	55	0.0020 - .0030	5.1905-5.1915	2.312	500,000	0
Atlanta										
MARTA	120-RT	1979	RS	76.0-114.8	70	- N O T A V A I	L A B L E -	2.5	200,000	0
N.Y./Conn.										
MTA M-1	407-CR	1970-71	RS	92.0-113.0	80	Timken-0.0015-0.0045 SKF-0.0017-0.0047	5.6905/5.6915	2.25	300,000	0
MTA M-2	204-CR	1973-75	PD	124.0-145.0	75	.0025-.0045	6.1905/6.1915	3.5	500,000	4
New Jersey										
ARROW II	70-CR	1974-75	PD	122.5-140.0	100	.0025-.0045	6.1905/6.1915	3.5	150,000	2
ARROW III	230-CR	1977-82	PD	130.1-150.0	100	.0025-.0045	6.1905/6.1915	3.5	150,000	0
PATCO	121-RT	1969&1980	RS	79.5-110.5	75	.0025-.0045	5.6905/5.6915	2.5	300,000	0
Philadelphia										
SEPTA										
Silverliner 1V	230-CR	1974-77	PD	128.3-150.0	85	.0025 - .0045	6.1905/6.1915	3.5	85,500	0
Washington, D.C.										
WMATA	300-RT	1976	RS	72.0-106.1	75	.0015 - .0045	5.6905/5.6915	2.5	275,000	0

RT = Rapid Transit
CR = Commuter Rail

PD = Pedestal or Equalizer Truck
RS = Rubber Shock Ring Truck

TABLE 7-2. BURNOFF INCIDENT SUMMARY

VEHICLE	ACCIDENT DATE	LOCATION APPLIED AT:	DATE	FROM INSTALLATION MONTHS	& MILES	BEARING MANUFACTURER	BURNOFF LOCATION	COMMENT
1. NEW JERSEY ARROW II #537	7/19/1977	Manufacturer (G.E.)	9/74	34	180,000	Timken	R-4/G	Wheel Off
2. MTA (M-2) #8663	6/21/1977	Manufacturer (G.E.)	1973	60	285,000	Brenco	L-2/NG	Severe Overheat
3. MTA (M-2) #8804	7/22/1977	Manufacturer (G.E.)	1974	36	171,000	Timken	L-3/G	Wheel Off
4. MTA (M-2) #8500	8/2/1977	Manufacturer (G.E.)	1972	60	285,000	Brenco	L-1/G	Wheel Off
5. MTA (M-2) #8703	8/3/1977	Manufacturer (G.E.)	1975	36	171,000	Timken	L-2/G	Wheel "Sagged"
6. NEW JERSEY ARROW II #564	9/29/1980	Wilmington	3/17/80	6	24,000	Timken	R-3/NG	Wheel Off
7. MTA (M-2) #8468	4/28/81	Reading	5/80	11	52,000	Timken	R-2/G	Wheel Off
8. MTA (M-2) #8720	2/4/82	Manufacturer (G.E.)	Not. Avail.		Est. 500,000+	Brenco	R-4/G	Hot Journal
9. MTA (M-2) #8440	2/23/82	Reading	5/80	21	100,000	Timken	R-2/G	Wheel "Sagged"
10. MTA (M-2) #8819	4/4/82	Reading	11/80	16	76,000	Brenco	L-2/NG	Wheel Off
11. MTA (M-2) #8542	11/3/82	Harmon	8/29/79	39	185,000	Timken	R-4/G	Wheel Off

L = Left Side of Car; 1-4 = Axle Number; NG = Non-gear Side; G = Gear Side; G.E. = General Electric

All properties contacted had experienced the symptoms of fretting, overheating, and in some cases, deep scoring or grooving of the bearing journal. These same symptoms were present regardless of whether the axle was hollow or solid, or whether the bearing was inboard or outboard. However, the severity of the symptoms observed at any of the other properties was much less than what has been observed in the M-2 fleet. For example, the Silverliner IV is quite similar to the M-2 in gross weight, axle/bearing design, and operation. However, very few Silverliner IV axles are rejected for loss of cone fit during rework operations, while about 40 percent of the M-2 axles are rejected.

Conrail recently inspected about 200 M-2 hollow axles and selected 60 of them for detailed examination. Of these 60 axles, 33 had bearings with excessive running lateral and axles with significant grooving and loss of cone fit. The measured running lateral was as large as 0.290 inch (compare with field condemning limit of 0.030 inch), and grooves were observed to have reduced the bearing journal diameter by as much as 0.181 inch under the cone. (In the latter case the cone bore had also worn to an increase of 0.015 inch on the diameter, so that the total gap was 0.196 inch.)

The foregoing observations suggest that axles/bearings should be expected to deteriorate at varying rates in service. It also appears that the other properties have been able to control the deterioration at the "symptom" level.

7.2 Vehicle Design Parameters

Section 5 pointed out that axle flexibility is an important factor affecting axle/bearing longevity. Therefore, the other fleets were compared with the M-2 fleet by making calculations similar to those mentioned in Section 5. Table 7-3 summarizes the comparative results for two quantities:

- o Percentage of proper assemblies expected to have critical end clearance.

- o Fretting motion amplitude.

Critical end clearance is defined as a gap between the bearing stack and the locating ring or the wheel hub of at least 0.0005 inch but less than 0.002 inch. A gap of this size is considered to be undetectable but large enough to permit fretting motions in service. The figures in Table 7-3 represent the percentage of critical press operations expected from assumed uniform distributions of wheel pressing and spiking forces within their shop practice ranges, with no correlation between the two forces. The ratio of shaded area to the area of the "shop control" rectangle in Figure 5-2 is an example of a percentage in accord with the foregoing definition. A design with a high percentage cannot be effectively assembled by realistic shop practice.

Fretting motion amplitude is defined as the maximum relative motion expected between the cone of an unclamped bearing and the axle. The maximum motion occurs near the edges of the cone, and depends on the weight of the car. Data on axle dimensions and car weights appear in Table 7-1. To obtain consistent results, fretting motion amplitude was calculated using the fully seated weight for each car.

TABLE 7-3. FLEET ASSESSMENT OF AXLE FLEXIBILITY EFFECTS

AXLE	FLEET	PERCENT CRITICAL	FRETTING MOTION AMPLITUDE (MICROMETERS)
HOLLOW-F/3.5" BORE	M-2 (ORIG) ARROW II ARROW III SILVERLINER IV	54	12 - 14
SOLID-F	M-2 (MOD)	5	10 - 12
HOLLOW-E/2.5" BORE	BART MARTA PATCO WMATA	39	8
HOLLOW-E/2.25" BORE	M-1 (ORIG)	39	10
SOLID-E	M-1 (MOD)	10	9
HOLLOW-D/2.3" BORE	CTA	55	6

It is evident from Table 7-3 that the group of fleets equipped with the hollow-F/3.5" bore axle combine the highest risk of critical end clearance with the highest fretting motion amplitude. The other fleets are at less risk in at least one of these categories.

Another significant design factor is the L-10 life of the bearing. L-10 life depends on car weight and bearing size, and is defined as the mileage at which ten percent of a bearing population would be expected to have developed condemnable fatigue defects such as cone bore growth or spalling of the rolling components. Roller bearing suppliers follow the practice of calculating L-10 life for each application as a guide for establishing the bearing rework interval. An alternative way to assess this factor is to compare the car weight with the rated bearing load. Each size bearing has a specific rated load, and L-10 life decreases as the ratio of car weight to rated load increases.

Although roller bearing fatigue does not appear to cause the axle/bearing failures considered in this study, L-10 life is still a significant factor because it may be related to the duration of the roller overload and damage phase discussed in the Section 5 description of the failure progression and because of its influence on bearing rebuild intervals. Table 7-4 summarizes the Task Force calculation of L-10 life for each property. The results separate the fleets into two groups: one with L-10 lives near 500,000 miles and one with L-10 lives between one and two million miles. These calculations were made with the formula that has been established by the Anti-Friction Bearing Manufacturers Association. The calculated lives are based on full seated car loads and an "application factor" of 2.0 to account for dynamic effects on the static car load.

The final significant design factor involves the truck configuration. The M-2, Arrow II, Arrow III, and Silverliner IV employ a pedestal-type truck (also called an equalizer-beam truck) which captures the axle bearings in a metal bearing adapter housing riding under the pedestal portion of the equalizer beam. The housing is a dry fit on the bearing, and this interface appears to be highly resistant to relative motion. Thus, bearing seizures on pedestal trucks usually result in motion between the bearing and the axle, with the consequence being burnoff. Conversely, PATCO, BART, and the M-1 employ a shock-ring truck which captures the axle bearings within a rubber shock-isolation ring rigidly clamped to the truck frame. In this case, the interface between the bearing and the shock ring appears to have less resistance to motion than the interface between a seized bearing and the axle. Thus, seizures on this type of truck have less severe consequences, viz: the bearing and axle turn in the housing, the frictional heat burns the shock ring, and the burning rubber provides a smoke/odor signal.

7.3 Operating Conditions

Consideration was given to the fact that external influences, apart from the vehicle design, can adversely affect vehicle performance and mechanical degradation rates in a failure process. Therefore, the properties were surveyed regarding their track conditions and operational practices.

TABLE 7-4. L-10 LIVES OF HOLLOW-AXLE CARS

CAR	L-10 (MILES) AF = 2
M-2	460,000
SILVERLINER IV	540,000
M-1	540,000
ARROW III	630,000
ARROW II	670,000
PATCO (LINDENWALD)	730,000
PATCO (BUDD)	830,000
WMATA	1,080,000
MARTA	1,530,000
BART	2,090,000
CTA (BUDD)	2,480,000
CTA (BOEING)	3,090,000

Regarding track conditions, Conrail reported that the M-2 fleet operates over 71 percent welded-rail track. The percentage of welded rail is about the same on the SEPTA system, while virtually the entire BART system is welded. Variations in observed track conditions are not large enough to cause significant deviations from the measured axle dynamic load factors found in the September 1982 M-2 operational test.

Two observations were made regarding operational practices. First, it was noted that M-2 vehicles are not normally looped or turned at the end of a run. The effect of this practice is to bias the wheel/rail interactions and to cause relatively rapid wheel-flange wear on one side of the vehicle. The M-2 appears to be unique in this respect, since the vehicles on other properties are normally reversed either procedurally or as a result of variations in route assignments. However, the available data did not reveal any correlation between lack of reversal and axle/bearing failures.

Second, it was noted that many of the fleets studied operate at relatively high speeds (70 to 100 mph). Specifically, the M-1, the Arrows, Silverliner IV, BART, and PATCO are in this category, as well as the M-2 cars prior to the November 11, 1982 speed restriction. Thus, operating speed does not appear to be a unique discriminant for axle/bearing failures. Speed is an important factor, however, in considering the risk of failing axle/bearing systems proceeding to seizure. It is noted that all reported burnoffs or severe overheats occurred at or immediately after traveling at speeds of 70 mph.

7.4 Maintenance and Assembly Procedures

Axle and bearing maintenance and assembly procedures vary throughout the rail industry. While the bearing manufacturers provide recommended procedures for each application, the only industry-wide standards are contained in the Wheel and Axle Manual and Roller Bearing Manual published by the Association of American Railroads (AAR). These standards were initially developed for freight cars operating in interchange service. Because these vehicles all have outboard bearings, which must be removed in order to remove a wheel, the AAR standards do not specifically address the question of interval for bearing removal. Also, the AAR standards do not address the specific inboard-bearing design present in the passenger vehicles under discussion. However, the AAR standards have been applied in varying degrees to both urban transit and intercity passenger rail operations.

The following sections present the pertinent findings on the maintenance and assembly practices which were observed at the transit and intercity rail properties visited.

7.4.1 Bearing Maintenance Procedures

The bearing maintenance intervals, policies, and procedures were found to vary among the properties. CTA reworks bearings at 500,000-mile intervals. Conrail's past policy for the M-2 fleet was not to rework bearings until 500,000 miles. However, the Reading Shop reworked some M-2 bearings at every wheel change, while in some other shops bearings were inadvertently allowed to reach 700,000 miles before rework. Conrail's current policy is to rework bearings at every wheel change. Other properties rework their bearings at much shorter intervals (see Table 7-1).

At several properties such as Conrail, PATCO, and CTA the bearings are disassembled and rebuilt in-house. Bearing rework is subcontracted at other properties such as BART and MARTA. The shop procedures differed in the level of detail, record-keeping requirements, and quality control applied. For example, BART had very detailed shop procedures and kept press records for every component pressed onto an axle. The engineering department was responsible for preparing shop manuals and for independent quality control, with ultimate authority over shop and inspection procedures. This extensive approach was not evident at other properties.

Lubrication policies and procedures varied significantly among the properties. Some used "No-Field-Lubrication" (NFL) bearings, which are designed to run without the need for added lubrication between bearing rework. The properties which applied lubrication in the field varied significantly in terms of the interval between application and the amount applied. However, there did not appear to be any correlation between lubrication policy and axle/bearing problems.

7.4.2 Assembly Procedures

Assembly procedures varied considerably among the properties. Several properties such as PATCO and MARTA measured each axle to insure that its bearing journals were within specification for the proper interference fit. Two properties measured both the journal and the cone bore. Others made no measurements prior to assembling axles and bearings.

All properties relied on the magnitude of the pressing and spiking forces to insure that the bearing had been expanded radially and clamped laterally. However, the magnitudes varied and at one property the wheel press force recorder appeared to be out of calibration.

Practices for lubricating the interference fit varied from castor or mineral oil (nominal practice) to linseed oil (acceptable) and linseed oil with white lead (prohibited under bearings). One property pressed on bearings dry. This practice is probably not damaging for new bearings, which generally have phosphate-coated bores, but is extremely risky in operations where reworked bearings are being pressed.

7.5 Inspection Intervals and Practices

Several types of inspection procedures were observed during the field visits. For clarity of presentation, the material is grouped into four categories:

- o Field inspection.
- o Light maintenance inspection.
- o Wheel shop inspection.
- o Special inspections.

The section on field inspection deals with practices normally performed in the course of train operations. Light maintenance inspection includes those practices requiring a car shop to perform the inspection. Wheel shop inspection refers to those practices performed principally during or related to bearing rebuild and wheel mounting. The section on special inspections deals with unique practices not covered elsewhere.

7.5.1 Field Inspection

All properties employ periodic field inspections as the principal check on in-service bearing condition. These inspections are typically performed on a daily basis, and involve a walk-around to detect missing parts, wheel discoloration, leaking grease, or other visually evident trouble indications.

Inboard bearings present a special problem for daily inspections, however, because of the difficulty of visual access.

7.5.2 Light Maintenance Inspection

The FRA requires periodic inspection of self-propelled passenger rail equipment. The car must be "shopped" and placed over a pit for a complete undercarriage inspection, including inspection of the bearings for grease leakage or defective seals.

The FRA requirement specifies a maximum inspection interval of 90 days. SEPTA and AMTRAK inspect at shorter intervals (30 or 45 days).

7.5.3 Wheel Shop Inspection

Most shops visited followed AAR and/or car/bearing supplier recommendations. However, new axles and bearings were not always measured (see Section 7.4).

Some concern has been reported regarding the relative accuracy of micrometers versus snap gauges used to make the measurements. The accuracy issue relates to used axles, where the wheel seat or bearing journal is only slightly fretted or grooved and the ability of the measurement technique to discriminate between acceptable and condemnable axles is in question. The issue is also of concern because most properties do not match-mark parts for reassembly.

Inspection record-keeping and part traceability varied considerably. BART had the most comprehensive system for keeping records. BART's quality control and automated maintenance information system provides both traceability and reliability of data not available elsewhere.

The reported inspection results indicated that none of the other properties have experienced axle rejection rates comparable to the M-2. The inspection intervals varied from 80,000 to over 500,000 miles. It is noteworthy that the Silverliner IV and the Arrow cars (similar to the M-2 in design and weight) have inspection intervals significantly shorter than past practice in the M-2 fleet. Specifically, the Silverliner IV axles and bearings are inspected at 80,000 to 100,000-mile intervals (governed by a more rapid wheel wear-out rate) and the Arrow cars at 150,000 miles. Past policy for the M-2 fleet was 500,000 miles, and even this interval was exceeded in some cases. It should be noted that shorter inspection intervals may reduce the rate of axle/bearing failures by removing failing articles from the fleet before they have progressed beyond the mechanical degradation phase discussed in Section 5.3.

7.5.4 Special Inspections

BART and PATCO routinely "spin-test" their wheelsets during the light maintenance inspection. The spin test involves disconnecting the drive motor and using an external power source to rotate the axle. The purpose of the spin test is to detect abnormally noisy bearings, which may reflect a condition of spalling or other damage to the internal components.

Temperature-sensitive pellets have been inserted in the bores of M-2, Arrow, and Silverliner IV hollow axles. The pellets are visually inspected at car layover points, and any axle containing a melted pellet is immediately removed from service for wheel shop inspection. A limited pellet program was started on the M-2 fleet in 1981, using pellets that melted at 190 F, but this detection temperature resulted in a large number of false alarms. Conrail expanded the pellet program, with a two-level detection approach using 213 F and 238 F pellets, shortly after the April, 1982 wheel burnoff. The program was also instituted in the Arrow and Silverliner IV cars at that time. The pellet inspection program has since been strengthened by tightening the removal criteria, and was made a part of the November 11, 1982 agreement.

Conrail initiated and the MTA has continued special inspections for excess running lateral. This inspection program supplements the M-2 retrofit activities. At present, the excess running lateral check is being performed on a schedule which places the highest priority on the highest-mileage hollow axles still in service in the M-2 fleet. Preliminary data developed from the excess lateral program indicates that the priority is justified.

7.6 Summary

From field observations it would appear that none of the properties visited have experienced journal wear or bearing failures at rates comparable to the M-2. This may be partly accounted for by differences in vehicle weight and conservatism of axle/bearing design.

However, even vehicles with designs and weights similar to the M-2 (the Arrow II and Silverliner IV) have not experienced the M-2 degradation and failure rates. The Arrow II fleet has had two burnoffs versus nine burnoffs and near-burnoffs for the M-2. The Arrow II pellet-melt rate is not known but its axle rejection rate is lower than the M-2 rate. The Silverliner IV has had no burnoffs, no pellet melts, and very few axle rejections.

It is believed that these differences arise in part from the differences in bearing inspection intervals and in part from the fact the the Silverliner IV fleet has less accumulated mileage than the M-2 fleet. The relevant inspection intervals are: 80,000 to 100,000 miles (Silverliner IV); 150,000 miles (Arrow II); and 500,000 miles (M-2, pre-1981). The average fleet accumulated mileages as of January 1983 are approximately: 270,000 miles (Silverliner IV); 400,000 miles (Arrow II); and 460,000 miles (M-2).

Except for the special case of low mileage in the Silverliner IV fleet, the significant parameters which have emerged from the operational fleet assessment are vehicle speed and weight, wheel shop inspection interval for bearings, bearing rated load, and axle flexibility. Bearing rated load and axle flexibility have the most influence on mechanical degradation, which has been discussed in Section 5. The remaining parameters provide a means for comparing the fleets with each other in terms of expectations of final failure.

Figure 7-1 presents such a comparison, in which the product of vehicle speed and weight is plotted against inspection interval. The product of speed and weight is proportional to heat generation rate in the bearing, and is thus an important factor affecting the risk of bearing seizure. The plot separates the fleets into two major groups: the M-2, Arrow II, and Arrow III, for which a significant failure rate either has occurred or can be expected; and the other passenger fleets, for which significant failure rates are not expected. Based on this observation, as well as the design similarities with the M-2 cars, the Arrow cars are a continuing cause of some concern.

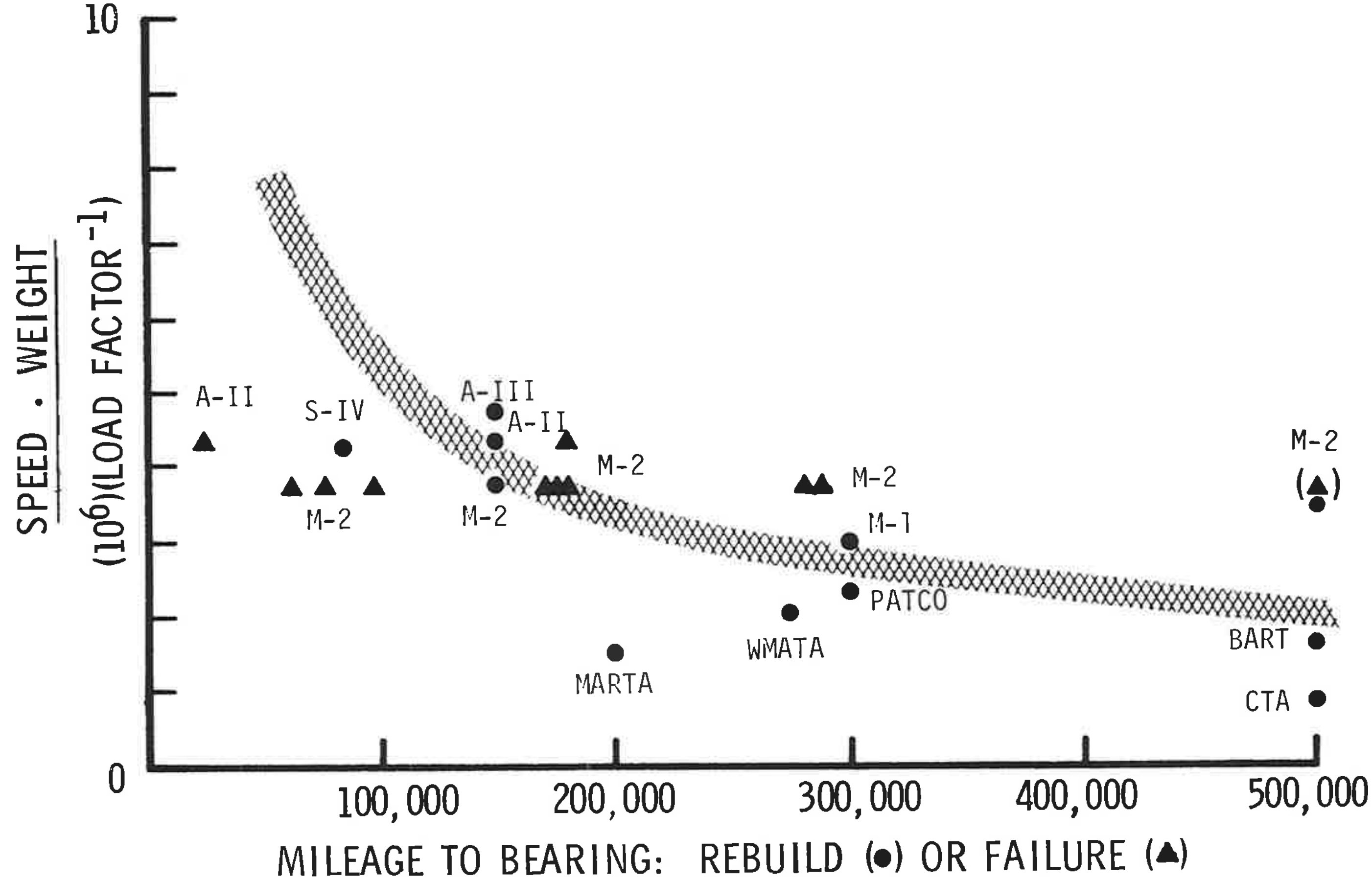


FIGURE 7-1. OPERATIONAL FLEET ASSESSMENT

8.0 CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

The following conclusions are based on a combination of site visits, data collection, and engineering analyses over 60 days. It must be noted that in several instances the Task Force was presented with contradictory information and statements which were not possible to reconcile in the time available. Further, the analyses have not been subjected to verification tests. Based on the data and information available, however, the conclusions are felt to be valid and supportable.

The conclusions relating to the failure mechanism of the M-2 axle/bearing system are:

1. The axle/bearing failure process that, if undetected, leads to the loss of a wheel on M-2 vehicles can be caused initially by improper assembly of the bearing or by inherent design characteristics of the axle/bearing configuration and assembly specifications. Either assembly or design factors could initiate the failure mechanism or these factors could combine to cause the problem.

2. Improper assembly procedures can cause initiation of the failure process if there is excess clearance at the end of the bearing stack and/or the interference fit of the bearing is less than specified.

3. The design configuration has characteristics that contribute to initiation of the failure process. The location of the bearing inboard from the wheel places the bearing cone on the portion of the axle subject to curvature and it limits the ability of the assembly to provide the proper clamping force on the bearing. The hollow axle, as compared to a solid axle, allows increased curvature and thus contributes to decreased clamping force.

4. Past practices had allowed M-2 wheels to be changed out without changing bearings, thus accumulating excessive mileage on journals and bearings between inspections.

5. Recent improvements in quality control and emphasis on critical steps in the axle/bearing maintenance and assembly procedures on the M-2 fleet have improved the margin of safety.

6. Prescribed assembly and quality-check procedures cannot provide positive assurance that the current M-2 hollow-axle assembly has effectively restrained the bearing against fretting under normal service loads.

The conclusions relating to the countermeasures that are being employed in the operation and retrofit of the M-2 cars are:

1. The 55 mph speed limit on the M-2 fleet is considered appropriate as an interim safety measure.

2. Thermal runaway that begins just after the last en route inspection before terminal layover may result in a burnoff before the first en route inspection on the next run. Therefore, en route inspection must be supplemented by pellet inspection of layover cars.

3. The 30-mile interval for intermediate inspection for excess heat provides a safe margin for detection of incipient failures but requires that the inspection procedure used can reliably detect bearings that have reached a temperature of 250 degrees Fahrenheit.

4. The detection required in item (3) can be achieved by measuring the temperature in the axle bore beneath the journal. Procedures which depend on contact with the axle/hub interface (melt stick or touching) are unreliable.

5. Utilization of a positive indication of bearing temperature would allow the possibility of relaxing the inspection interval.

6. Either an automated wayside heat detector or onboard detection could be used to replace manual measurements. However, of the possible systems reviewed, there are none that appear to solve the problem without additional development, test and evaluation.

7. The possibility exists that en route inspections might be replaced by periodic shop inspections to reliably detect loss of cone fit and bearing spread.

8. Retrofit of the M-2 fleet with solid axles is a sound decision. However, the retrofit does not guarantee against failure if not accompanied by proper assembly and maintenance procedures.

The conclusions relating to the possibility of the failure mechanism affecting other rail passenger vehicles with similar axle/bearing configurations are:

1. Inboard bearing configurations with hollow axles should not cause concern in general.

2. The New Jersey Arrow cars have a bearing configuration similar to the M-2 cars and operate with similar weights and higher speed. However, the Arrow bearings are regularly rebuilt at shorter mileage intervals.

3. The mileage interval between bearing inspection and reassembly is a critical issue for all passenger rail vehicles. It should not exceed the bearing manufacturer's recommendation in any case. Risk of failure can be reduced by increasing the frequency of bearing inspection and rebuild.

4. Quality control in the assembly process should be sufficient to prevent fleetwide incidents, but cannot guarantee that a rare failure will not occur. Therefore, appropriate inspections at appropriate intervals can further reduce the risk that improper bearing assembly will lead to an axle/bearing failure.

8.2 Recommendations

Based on the foregoing conclusions, the following recommendations are offered. They include some actions that should be taken immediately and some which, although less urgent, will lead to safer rail service in the future. The short term recommendations are:

1. Continue current 55 mph speed restrictions, and the daily initial-run axle pellet inspection for the M-2 fleet per the November 11, 1982 agreement.

2. Expand the initial daily pellet inspection to cover all layover trains.

3. Use of a device such as a hand-held pyrometer to measure temperature in the axle bore which, when combined with more frequent pellet checks, can provide a more accurate and reliable inspection technique while also providing a more flexible schedule of en route inspections.

4. Continue the solid axle retrofit on the M-2 vehicle. Proper assembly of the bearings and wheels on the axle is essential and must be closely monitored.

5. Notify all operators of passenger vehicle fleets of the importance of not exceeding the bearing manufacturer's recommendation for the mileage interval between bearing inspection and reassembly.

6. New Jersey Transit should continue observance of the 150,000 mile bearing rebuild interval on Arrow Cars and insure proper bearing assembly and quality control procedures.

The recommendations which should lead to safer operation in the future are:

1. Develop a training program for FRA Safety Inspectors to focus on inspection, maintenance and assembly practices for bearings and axles in particular and shop practices in general.

2. Urge the passenger train operators (through AAR and APTA) to adopt more uniform inspection, maintenance and assembly procedures for bearings and axles.

3. Urge the industry to develop and test automated wayside or onboard detection devices for overheated inboard bearings.

4. Contact European and Japanese operators of high speed trains to obtain data on failures and relevant inspection, maintenance and assembly procedures for rail passenger train axles and bearings.

5. Hold a briefing for industry on the results of this Task Force investigation.

APPENDIX A

TASK FORCE BIOGRAPHICAL INFORMATION

TSC PERSONNEL

Robert J. Ravera, Deputy Director
Transportation Systems Center

Dr. Ravera was awarded a PhD in Applied Mechanics by Lehigh University in 1967. His specialties include structural analysis and dynamics. He taught Mechanical Engineering at Lehigh University and also held a number of engineering research and management positions before joining the Department of Transportation in 1975. For 2 years Dr. Ravera was Director of DOT's Office of University Research and as such functioned as the primary contact between the Department and the nation's academic community. He is a member of several professional societies including the American Society of Mechanical Engineers. Dr. Ravera has been awarded the departmental Silver Medal for Meritorious Achievement and Bronze Medal for Superior Achievement and was the recipient of the Apollo (NASA) Achievement Award for work involving spacecraft dynamics and control.

George W. Neat, Deputy Associate Director
Office of Technology Applications

Mr. Neat was awarded an Engineers Degree in Aeronautics and Astronautics by MIT in 1971 and he received an M.S. degree in Mechanical Engineering from the University of Washington in 1961. During the past 12 years at TSC his responsibilities have included test, evaluation and equipment development for urban rail transit vehicles, manager of the Urban Mass Transportation Rail Systems Technology Program, Executive Assistant to the Director and Chief of the Office of Program Development. Prior to joining DOT, Mr. Neat was a Systems Engineer for NASA's Electronic Research Center and he worked for the Boeing Company for ten years as a Control Systems Engineer. Mr. Neat is past chairman of the IEEE Land Transportation Committee and is currently a member of the Association of American Railroads Track Train Dynamics Steering Committee. He received the Department of Transportation Superior Achievement Award (Bronze Medal) in 1981.

Pin Tong, Chief
Structures and Dynamics Division

Dr. Pin Tong received his PhD degree in Aeronautics and Mathematics from the California Institute of Technology in 1966. He has 20 years of experience in the analysis and design of mechanical systems. During the past 8 years at the Transportation Systems Center (TSC), he has been responsible for directing and conducting analytical and experimental studies and designing mechanical components of transportation vehicle systems. While at the Massachusetts Institute of Technology from 1966 to 1974, he conducted and led research and taught various undergraduate and graduate courses related to structural analysis and design.

Dr. Tong is the author of numerous papers on structural analysis and a book on the finite element method. He is a recipient of the Von Karman Award in 1974 by TRE, Inc., for outstanding contributions to structural and material technology, and of the DOT Meritorious Achievement Award (Silver Medal) in 1977.

H. David Reed, Chief
Track Safety Research Division

Mr. Reed received a B.S. degree in Engineeering from the University of Wyoming in 1964. He currently heads the Track Safety Division at TSC. The program, sponsored by the FRA, concentrates on the assesssment of track related derailments and is directed at the creation of an equitable set of performance based track specifications. He is a recipient of the Department's Bronze Medal for Superior Achievement. He has also been responsible in his 12 years at TSC for the analysis and definitions of systems to control and improve transportation operations in the various modes. Prior to joining DOT Mr. Reed was a mission controller at the NASA Manned Spacecraft Center in Houston. He actively participated in the control of Apollo flights and helped develop the pin-point landing techniques used during lunar touchdown. As a result of his role in the Apollo 13 mission he was awarded the NASA Special Achievement Award and was a joint recipient of the Presidential Medal of Freedom.

Oscar Orringer
Senior Mechanical Engineer

Dr. Orringer received his ScD degree from the Massachusetts Institute of Technology in 1970. He leads the Center's work on failure investigations involving fatigue and other mechanical/structural failures in transportation equipment fleets. Current and recent responsibilities include railroad passenger vehicle axles and disc brake fretting failures, assessment of weld fatigue failures and modifications in the Grumman Flexible 870 city bus fleet, and management of revenue and test track experiments to determine the rates of growth of fatigue cracks in rails. Applicable experience prior to entry into government service includes membership on U.S. Air Force committees for structural integrity review of the C-5A, C/KC-135, and F-4E/F air frames and failure investigation of pitch control machinery in U. S. Navy ship propellers. Dr. Orringer has taught systems engineering design and integration at the Massachusetts Institute of Technology. As an officer on active duty with the U. S. Army Corps of Engineers, he was responsible for earthwork and paving design and construction. He received the DOT Superior Achievement Award (Bronze Medal) in 1981.

Herbert Weinstock
Senior Mechanical Engineer

Dr. Weinstock received his ScD degree in 1968 and has 25 years of experience in the analysis, design and testing of mechanical systems operating in severe dynamic environments. At the Transportation Systems Center, he has been responsible for analytical and experimental studies of vehicle, track and roadway dynamic interaction as related to safety and performance of transportation vehicles and guideways. While at the MIT Instrumentation Laboratory (now Draper Laboratory) from 1960 to 1965 and at the NASA Electronics Research Center from 1965 to 1970, he was responsible for design and development of advanced navigation and flight control instrumentation and for the design and development of equipment for severe dynamic load environments. Dr. Weinstock is registered as a Professional Engineer in the Commonwealth of Massachusetts. He is a recipient of the Department of Transportation's Superior Achievement Award (Bronze Medal) and, in 1982, the Meritorious Achievement Award (Silver Medal).

Adelbert L. Lavery, Chief
Safety and Security Division

Mr. Lavery received a B.S. degree in Electrical Engineering from Rensselaer Polytechnic Institute in 1959. He currently heads the Safety and Security Program at TSC which is applying technological solutions to contemporary problems in transportation safety and security areas. During his 12 years at TSC, he has also been responsible for several program areas relating to safety and improved operations in air, rail and automotive modes. Prior to joining TSC, Mr. Lavery was actively involved in the electro-optics and medical instrumentation fields. He is a member of several societies including the American Society for Nondestructive Testing and the Locomotive Maintenance Officers Association. Mr. Lavery has been awarded the Bronze Medal for Superior Achievement for his work in track inspection.

William T. Hathaway
Mechanical Engineer

Mr. Hathaway holds an M.S. degree in Mechanical Engineering from Northeastern University. He has for the past 7 years served as a Senior Project Engineer at TSC and in this capacity has directed several studies of transportation safety. His most recent assignments have been in the area of transportation safety where he has been responsible for the development and direction of projects in fire safety, emergency preparedness, failure analysis and risk assessment. Prior to joining TSC, Mr. Hathaway served as a mechanical and marine engineer with the Boston Naval Shipyard. In this position he was responsible for the design and testing of main propulsion machinery and auxiliary machinery systems. A member of several technical and professional societies, Mr. Hathaway is also a Registered Professional Engineer in the Commonwealth of Massachusetts. He has authored several papers and technical reports on his work and in 1980 was awarded the Bronze Medal for Superior Achievement by the Urban Mass Transportation Administrator.