AN ASSESSMENT OF NEEDS AND OPPORTUNITIES

FREIGHT CAR TRUCKS FOR THE 21ST CENTURY

A TECHNOLOGY TRANSFER FORUM



PRESENTATIONS



September 7-9, 1994

The Levis Faculty Center
919 West Illinois
University of Illinois at Urbana/Champaign

Association of American Railroads

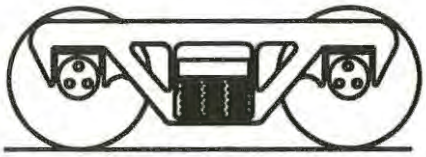
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K. L. Hawthorne, AAR

INTRODUCTION

R. R. NEWMAN

BURLINGTON NORTHERN RAILROAD

AAR TECHNOLOGY TRANSFER FORUM
University of Illinois
Urbana-Champaign
September 7, 1994

AN ASSESSMENT OF NEEDS AND OPPORTUNITIES:

FREIGHT CAR TRUCKS FOR THE 21ST CENTURY

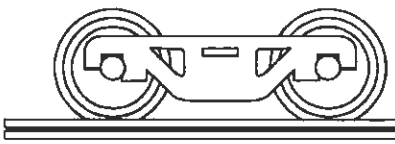
KEYNOTE ADDRESS

by Ronald R. Newman
Chief Engineer Research and Development
Burlington Northern Railroad

I would like to express my personal gratitude, and that of the Burlington Northern Railroad and the AAR's Research Committee (of which I am a member) to the University of Illinois at Urbana-Champaign, and Professor Baremberg, for having had the opportunity to present the keynote address to the Technology Transfer Forum on *Freight Car Trucks for the 21st Century*. I also want to thank both the AAR and the University of Illinois for being instrumental in accelerating rail industry technology.

Ronald R. Newman

September, 1994



BACKGROUND: *Where Trucks Came From*

THE FIRST TRUCKS

The conventional three-piece truck is a time-honored workhorse. It has served the rail industry since the late 1800's and has proven to be a widely used and well engineered system. The first freight car trucks, as shown in Figure 1, were based on road wagons. They had four wheels attached to fixed axles. The principal construction materials were wooden beams. Later, the wooden beams were trussed for added strength, and some trucks were actually built from iron bars.

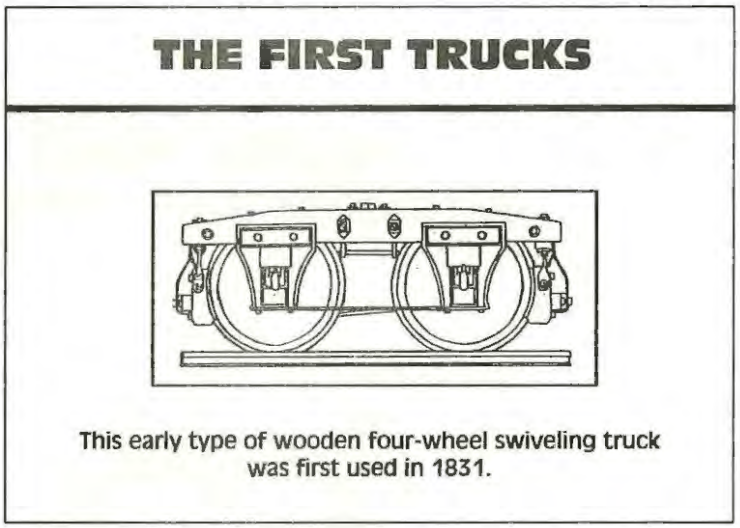


Figure 1

In an 1831 Baltimore and Ohio Railroad report, the chief engineer called attention to the ride damage to a car and recommended that, *"...a number of cars be furnished with springs in order to test their advantageous use on such cars."* As a result of his observations, the *diamond* or arch-bar truck was introduced in the latter part of the nineteenth century. Its name was derived from its side frames, which were characteristically manufactured from bar iron in a bowed, or arched design. The arch-bar truck, shown in Figure 2, has since been called the "father" of the modern four-wheel truck design.

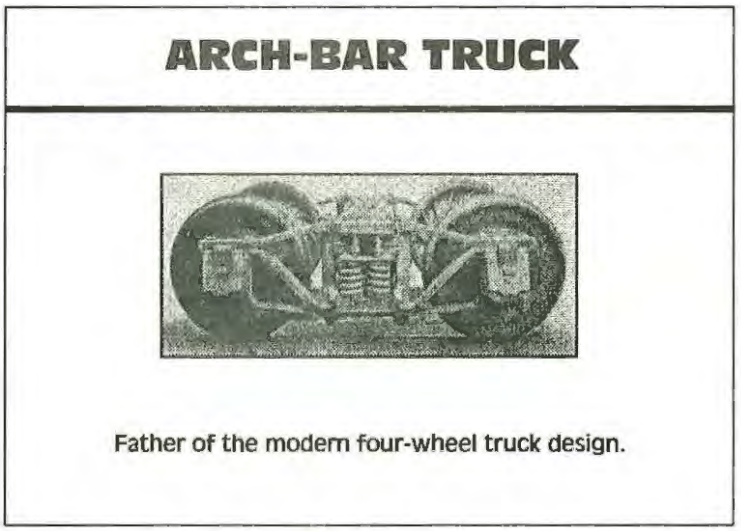


Figure 2

NOT AS SIMPLE AS THEY SEEM

Addressing the many issues surrounding freight car trucks is like opening Pandora's box. When we first look at a conventional three-piece truck, it looks relatively uncomplicated and straightforward: a truck bolster, centerplate, two side frames, wheel and axle sets, and some individual components such as springs, wear plates, friction shoes, gibs, bearing adapters and other related stability and dampening devices. As shown in Figure 3, the three-piece truck appears and has proven to be simply constructed and easily assembled and disassembled in the field and in railway shops.

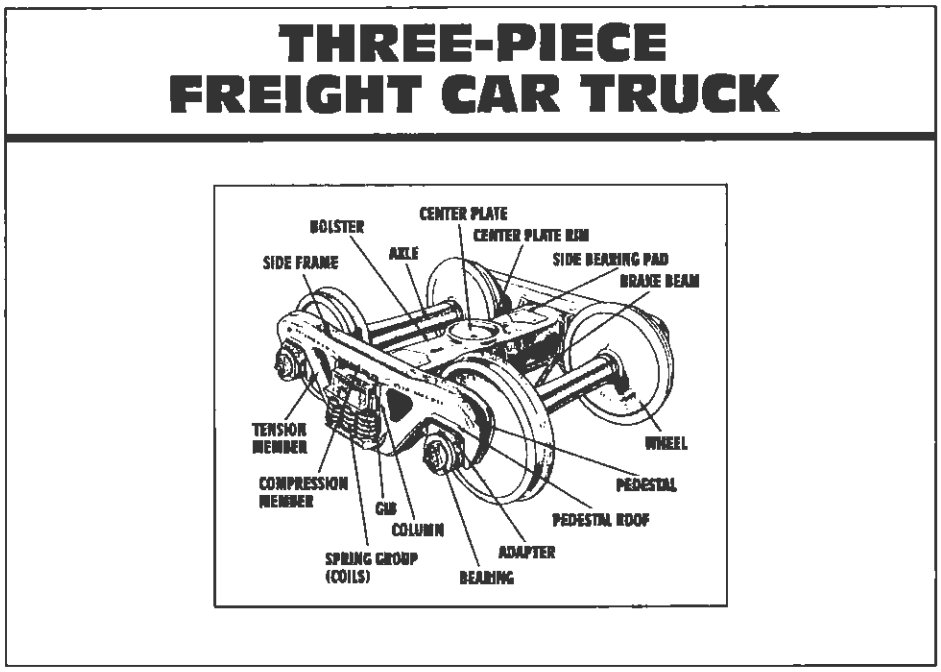
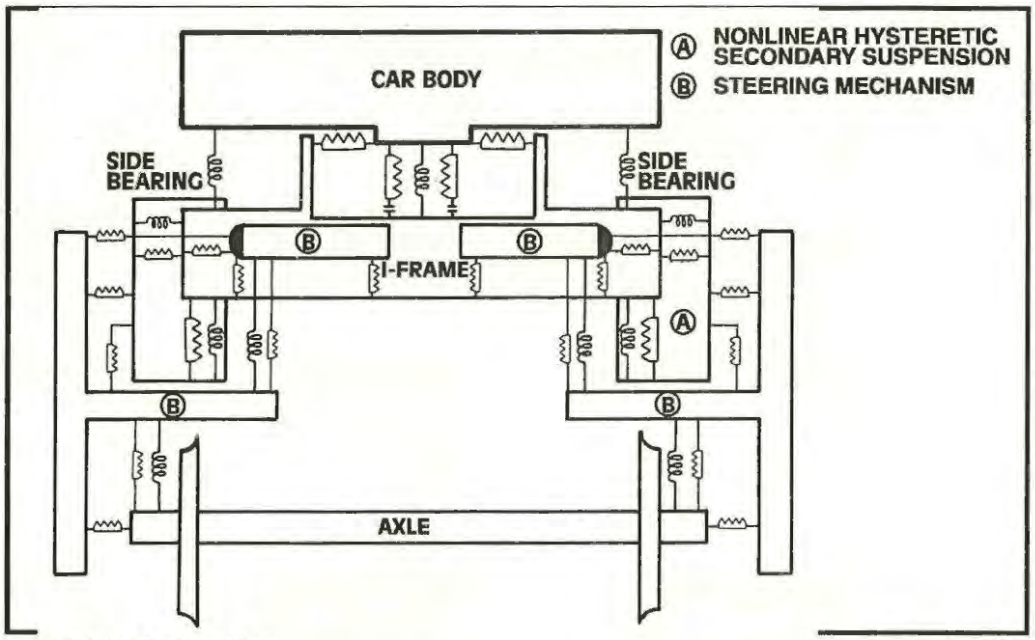


Figure 3

However, when we closely examine the three-piece truck's ride environment and its actions within a train, and when we model it analytically, a different picture emerges. We see that what at first appeared to be simple, is, in reality, a complex engineering system coupled to a car body that is required to interface with the track and with many other variables.

COMPLEX MODELING

In Figure 4 you see (from a typical NUCARS analysis) that the simulation model for an advanced design three-piece steerable railcar truck may consist of as many as 23 rigid bodies, 134 degrees of freedom, and 186 linear and nonlinear connection elements, including springs, gaps, friction elements, and viscous dampening.



- **23 rigid bodies**
- **134 degrees of freedom**
- **186 linear and non-linear connection elements including histeretic springs, gaps, friction elements, and viscous damping**

Figure 4

We also observe, as shown in the *Key Performance Attributes and Design Parameters* table below, that there is a long list of key performance attributes and design parameters that must be considered to prevent undesirable "truck hunting," to improve wheel and truck wear characteristics in curving and in twist and roll. Trucks are an engineering *system* that must be designed to interface with the wheelsets, track, and carbody.

KEY PERFORMANCE ATTRIBUTES AND DESIGN PARAMETERS

HUNTING	CURVING	TWIST & ROLL
PERFORMANCE ATTRIBUTES	**PERFORMANCE ATTRIBUTES**	**PERFORMANCE ATTRIBUTES**
Carbody lateral acceleration at centerplates	Wheel-to-rail resistance	Peak-to-peak carbody roll angle
Axle lateral displacement	Vertical wheel load	Vertical wheel load
	Total truck lateral wheel-to-rail force	
DESIGN PARAMETERS	**DESIGN PARAMETERS**	**DESIGN PARAMETERS**
Centerplate yaw friction	Centerplate yaw friction	Suspension stiffness
Side bearing yaw friction	Side bearing yaw friction	Suspension hysteresis
Carbody-to-bolster torsional stiffness	Carbody-to-bolster torsional stiffness	Carbody length
Carbody-to-bolster yaw damping	axle misalignment	Centerplate wear liner stiffness
Truck "shear" stiffness	Carbody torsional stiffness	Carbody torsional stiffness
Truck "bending" stiffness	Side bearing vertical stiffness	Side bearing vertical stiffness
Suspension stiffness	Bolster torsional stiffness	Bolster torsional stiffness
Suspension hysteresis		
Wheel/rail profile		
Bearing adapter yaw friction		

RECENT AND CURRENT RESEARCH: *What's Happening With Trucks Today*

OVERVIEW

I want to highlight a few ideas that I hope will serve as a backdrop to the twelve detailed presentations and discussions that comprise the working sessions of this Technology Transfer Forum. I hope that these ideas will guide you in formulating meaningful thoughts and conclusions about the information presented at this Forum, and that you will take these thoughts and conclusions home with you as action items for your own organization.

The working sessions focus on *Recent and Current Research* (Sessions 1 and 2), *Best Practices* (Session 3) and finally, *Future Directions* (Session 4). Whether you are a university, a railroad, a railroad supplier, or a researcher, the information and knowledge you acquire during this forum may be the catalyst that starts your company on a road to new concepts in quality maintenance practices and innovative truck designs.

THE PAST AND THE PENALTIES

The simplicity of the three-piece truck's design and its relatively low capital cost have been its strong suits. The three-piece truck has gained favor because it is a reasonable performer even on rough track, and it has a robust construction and it's easy to maintain.

However, in today's competitive marketplace there are many subtle areas related to truck performance that must be carefully monitored and managed effectively to keep them from accruing *astronomical* cost penalties to the rail industry each year. Subtle items such as:

- Maintenance costs
- Fuel consumption
- Track wear
- Harsh ride environment
- Derailment potential

These cost penalties can be avoided through the use of improved materials, quality maintenance practices, and innovative approaches in truck designs. Avoiding these costs and their related lading damages may become a primary factor in achieving a more customer-responsive, competitive rail industry in the Twenty-first Century.

THE "MAKERS AND SHAKERS" OF TRUCK TECHNOLOGY

Now I want to go through—flash through—a number of innovations in truck technology that have occurred in our industry since the 1970's. The truck and component manufacturers, railroads and research organizations that I am going to highlight have been the "makers and shakers" over the past 20 years in our industry for truck research and development. Their efforts have yielded significant results, and they are expected to continue their leadership roles well into the Twenty-first Century.

In 1971, the Association of American Railroads, the Transport Development Agency of Canada, the Railway Products Institute, and the Federal Railroad Administration began a multi-year research program called the *National Research Program on Track-Train Dynamics*. This three-part effort focused on the development of train handling and train makeup guidelines, and on improvements in derailment prevention (such as truck hunting), truck suspension and carbody fatigue enhancements, and reliance on the Pueblo Transportation Test Center as a primary facility for conducting industry railcar truck tests. Some of the results of this research yielded the reports shown in Figure 5.

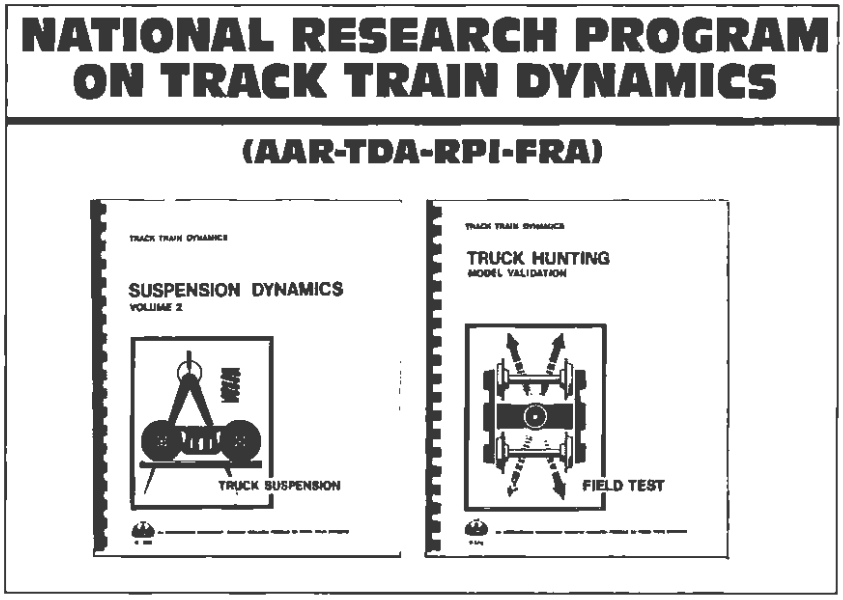


Figure 5

Out of these efforts came a myriad of mathematical models, one of which is shown in Figure 6. For the very first time in the rail industry, models were developed and used that defined highly complex relationships for freight car trucks and track interaction, and that later have led to many improved designs.

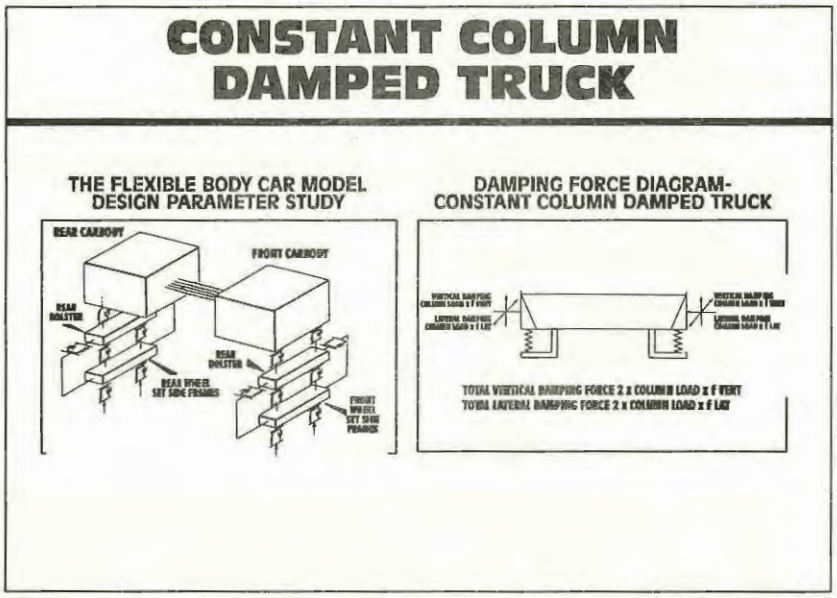


Figure 6

During this same period in the 1970's, the Federal Railroad Administration initiated an extensive truck system research effort known as *The Freight Car Truck Design Optimization Project (TDOP) (see Figure 7).* From this project came an excellent characterization and understanding of the existing railcar environment, and identification of future testing and design needs, particularly from a safety perspective.



Figure 7

RAPID TECHNOLOGY CHANGE

Over the past decade, however, the concept of defining the existing railcar environment has become a moving target due to the rapid technology changes in railcars. Consider, for example, the development of:

- Multiplatform spine cars
- Doublestack container cars
- RoadRailers
- Autostack cars

Also consider the many notable component achievements by truck manufacturers and component suppliers such as:

- Radial steering trucks
- Braced trucks
- Single axle trucks
- Forced steering trucks
- Other dynamic stabilizing devices

The major component suppliers have also provided enhancements that have included trucks with either constant or variable damping, larger shoe face and slope areas on their control elements with corresponding changes in side frame column wear plate dimensions, different friction shoe materials, and optimized truck component tolerances.

TRUCK SUPPLIERS

Figure 8 shows one of several major truck systems and companies that exemplify state-of-the-art truck design and manufacturing. Figure 8 shows American Steel Foundrie's *Super Service Ride Control (SSRC)* truck. This truck has been an excellent performer used in a variety of services on many railroads. ASF's *Ridemaster* truck also meets AAR specifications for controlling harmonic rock without expensive hydraulics.



Figure 8

ASF also has adopted the *AR-1 Steering Device.* It incorporates two C-shaped steering arms which provide steering in curves and eliminate undesirable truck hunting. The AR-1, shown in Figure 9, can be retrofitted to many existing three-piece trucks.

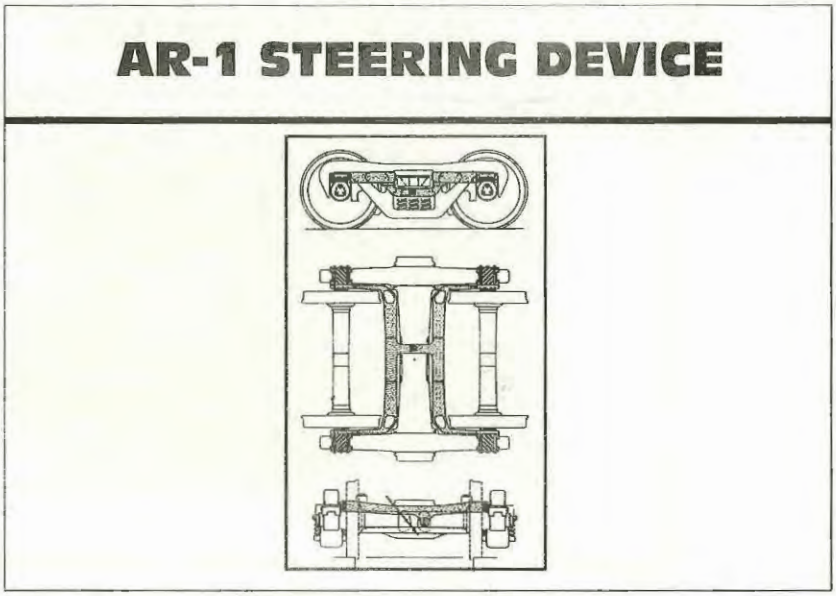


Figure 9

Standard Car Truck Company has demonstrated a product line of reliable performance trucks which include the *Barber S-2* (shown in Figure 10) and HD (heavy duty) Stabilized Trucks.



Figure 10

Standard Car Truck Company also has a long line of innovative radial trucks, such as the *Barber Radial* shown in Figure 11.



Figure 11

And, Standard Car Truck Company also offers frame-braced trucks for increased truck hunting stability and enhanced curving performance, such as the *Can-AM-Barber* frame-braced truck shown in Figure 12.

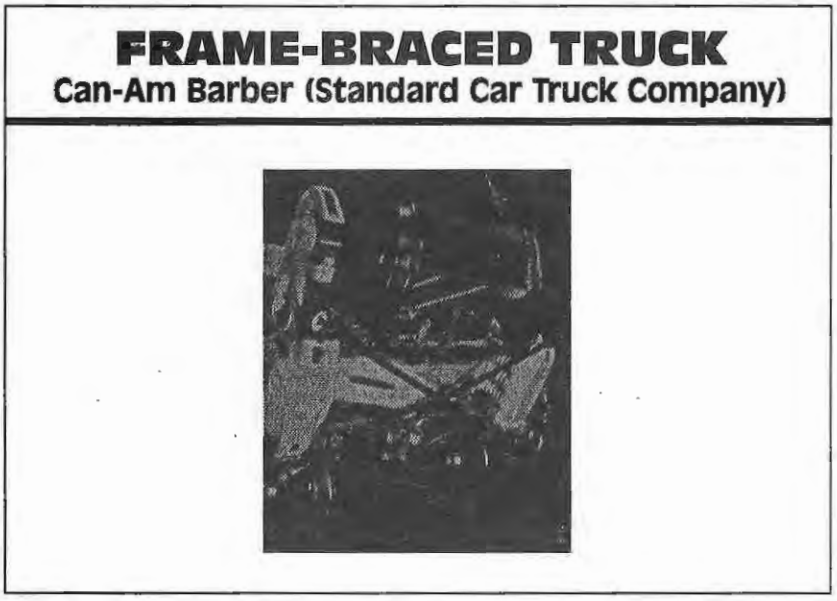


Figure 12

Standard Car Truck Company, as Barber Resco, recently led the development of a unique *stabilizer frame* retrofit design for automotive and intermodal equipment, an example of which is shown in Figure 13.



Figure 13

Standard Car Truck Company has also fostered the development of centerplate extension pads, such as the *C-PEP* shown in Figure 14, for truck hunting stability.



Figure 14

National Casting's *Super C-1 Wedgelock* and their *Swing-Motion* trucks are shown in Figure 15. Both of these designs focus on reductions in truck skewing, adverse rock and roll, and an increase in critical truck hunting speed.

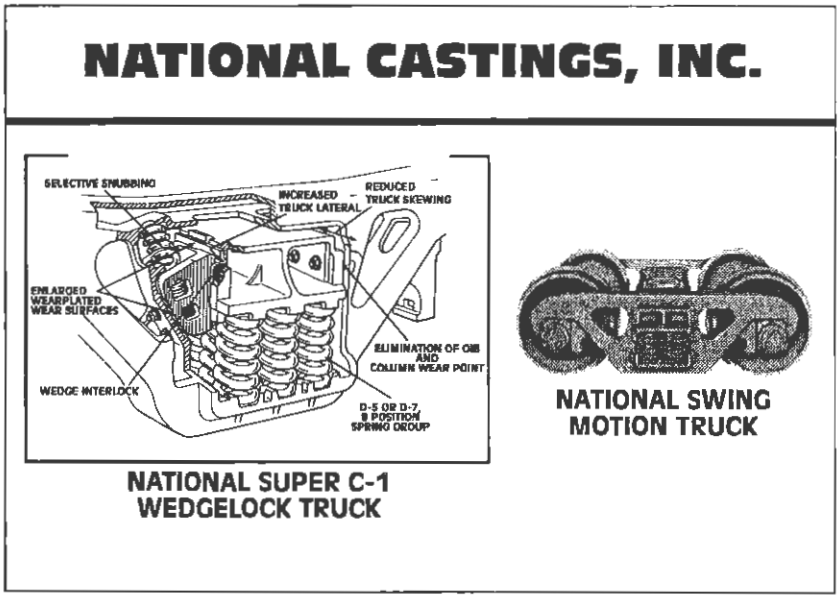


Figure 15

The Swing-Motion truck, shown again in Figure 16, is designed for superior performance for higher speed intermodal and automotive service.

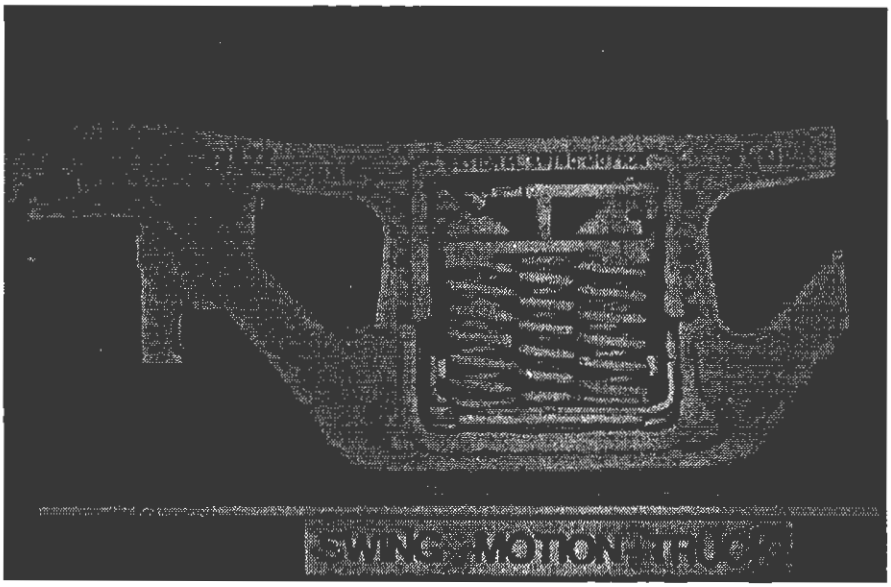


Figure 16

National has also been leading the effort to develop a single-axle truck called the *Unitruck*, shown in Figure 17. These trucks have already been placed under a number of intermodal platform cars in revenue service. National is currently working on two-axle and three-axle versions using the same suspension systems.

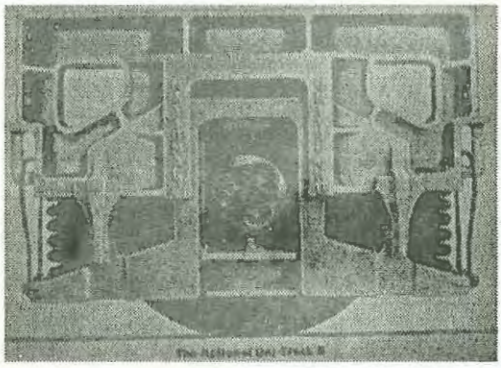


Figure 17

Buckeye Steel Castings Company has developed a new series of *XC-R* trucks for multipurpose 70-ton and 125-ton rated freight service. These trucks, shown in Figure 18, have better truck warp stiffness characteristics, and are equipped with wide biased wedges for increased truck squaring and enhanced wear benefits.



Figure 18

Finally, Westinghouse Electric Corporation (Marine Division) has developed a radically different truck concept which has been designed to resolve the dichotomy between *optimizing for a good curving truck* and *avoiding truck hunting*. Through the use of a *forced steering* truck and carbody link as shown in Figure 19, the design allows for the natural action of the wheels to yaw, but in a controlled manner.

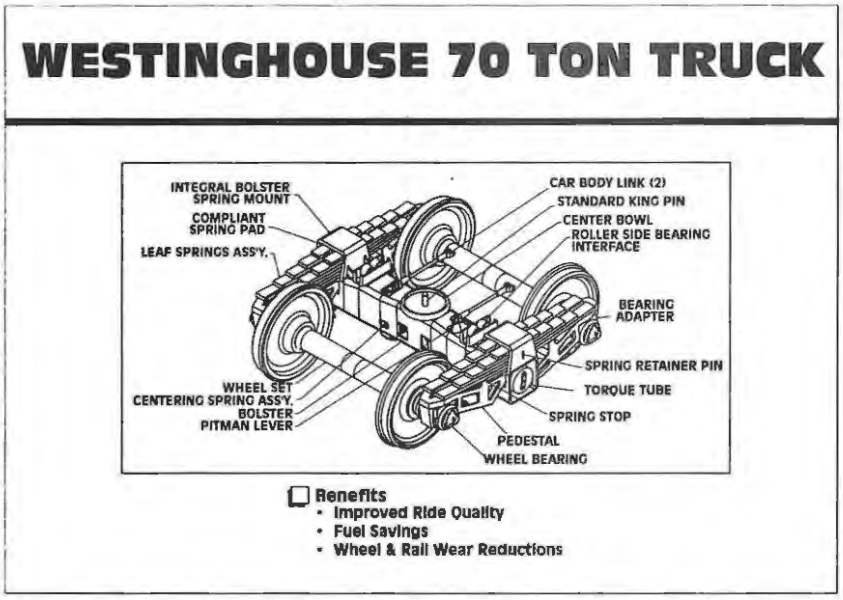


Figure 19

By interconnecting steering arms from the axles to the carbody (see Figure 20), the Westinghouse truck is much stiffer in the tangent-straight line direction. However, the natural generation of steering moments caused by the wheel-to-rail geometry allows the axles to yaw into optimum alignment in curves.

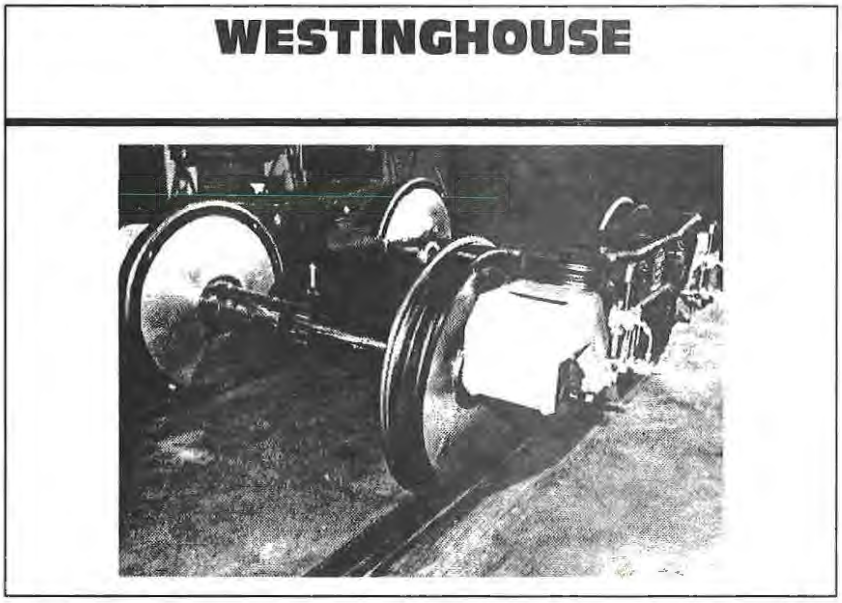


Figure 20

COMPONENT SUPPLIERS

In addition to truck manufactures, there have also been a number of component suppliers in the rail industry that have been instrumental in accelerating innovative improvements for truck stability and enhanced wear performance.

A. Stucki Company has dedicated much of its research effort toward solving car/truck dynamics and associated wear problems. The significant contributions coming from that effort have been:

- Hydraulic dampers for lateral and vertical control
- Constant contact side bearings for cost-effective hunting control
- Elastomeric backed friction wedges for truck wear control

Figure 21 shows a conventional truck equipped with A. Stucki's dynamic control systems.

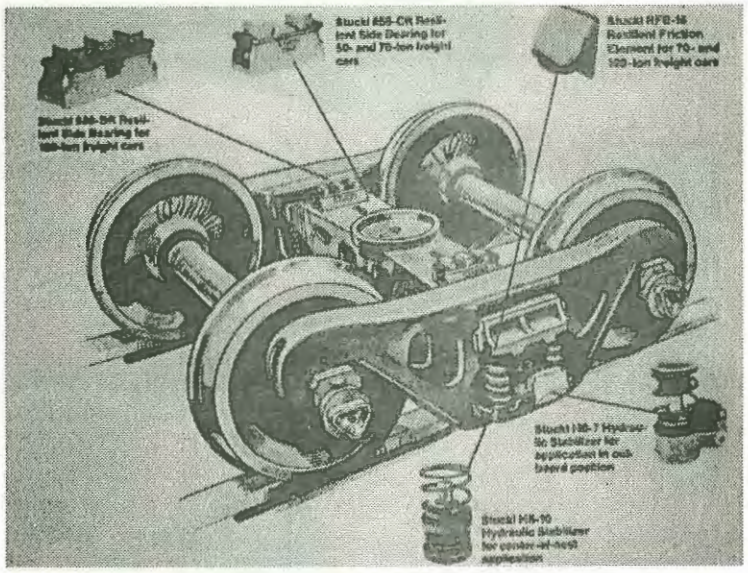


Figure 21

Ongoing research has also enabled A. Stucki to develop hydraulic stabilizers for various trucks with a design target of providing over one million maintenance-free service miles. Figure 22 shows the latest version *HS-7* and *HS-7-100* hydraulic stabilizers.

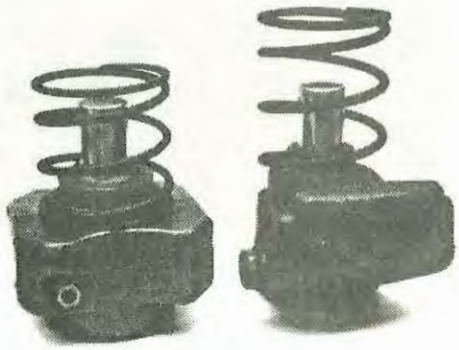


Figure 22

The newest generation of constant contact side bearings are used to prevent undesired truck hunting. The ISB dual action series introduced in 1990 brings technology full circle by combining the advantages of roller side bearings and constant contact side bearings into one design, as shown in Figure 23.

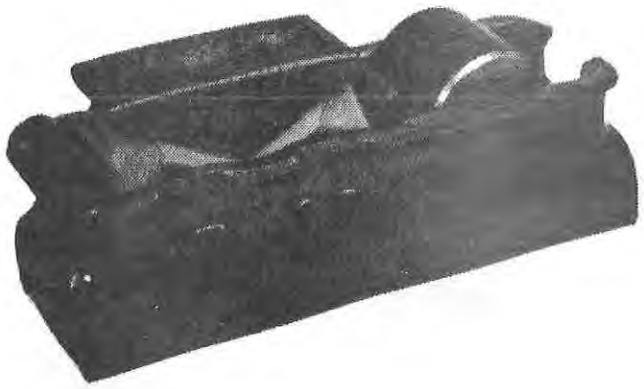


Figure 23

Resilient friction shoe elements, such as A. Stucki's RFE-16 shown in Figure 24, have also proven to be a cost effective, maintenance saving innovation for conventional three-piece trucks.

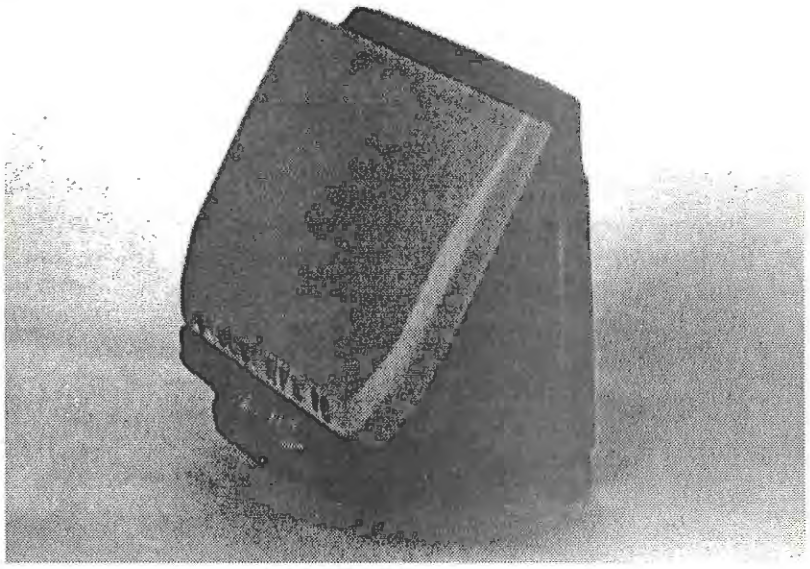


Figure 24

As early as 1944, side bearings produced by W.H. Miner were placed on over 500,000 railcars. More recently, Miner Enterprises has been successful with metal capped constant contact side bearings, as shown in Figure 25. Miner started retrofitting with *TecsPak* kits in 1981, leading to complete constant contact side bearings with cast iron housings in 1983.



Figure 25

Miner's roller assisted constant contact side bearings, as shown in Figure 26, are primarily used on articulated 125-ton doublestack cars to limit the buildup of truck turning moments during curving.



Figure 26

Miner is presently introducing a new line of long travel and roller-assist models (shown in Figure 27) to advance the performance and productivity of railcars.



Figure 27

Long travel service constant contact side bearings (shown in Figure 28) were introduced by Miner in 1988 to allow increased torsional compliance between adjacent carbody units of articulated trucks and to improve L/V ratios in curving.



Figure 28

TESTING AND DESIGN ACCOMPLISHMENTS BY COMPANIES OTHER THAN RAILROADS

Another significant player in industry research, in addition to that of the truck manufacturers, has been the leading edge technology testing and designs undertaken by TTX Company (see Figure 29). Beginning in 1992, all autorack conversions and maintenance at TTX has included the application of premium truck assemblies. This is done to improve the ride quality, minimize lateral movement, and reduce truck hunting at higher speeds. In essence, TTX is targeting to meet their own customers' needs (the railroads) as well as the needs of *their* customers: the rail shippers. By 1998, all autorack cars in TTX's approximate 37,000 car-fleet are expected to be fitted with premium trucks.

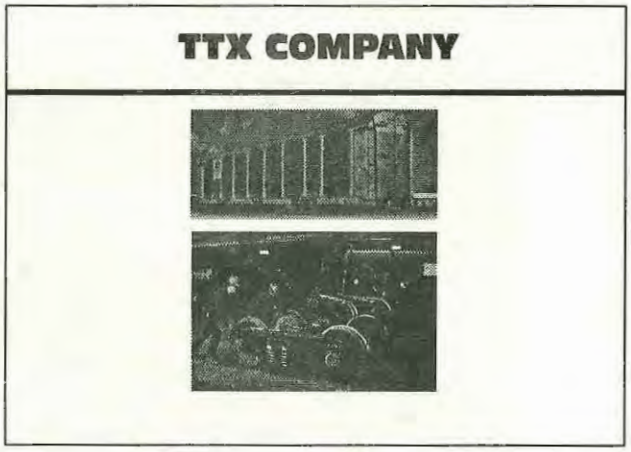


Figure 29

Recently, TTX Company has fostered many new car types and their associated components and trucks. TTX, of course, has been assisted in these development efforts by another set of real shakers and makers in the industry: the car builders (see Figure 30). The car builders include organizations such as Gunderson, Trinity and others. These companies provide the ever important car designs, which must successfully interface with the truck systems.

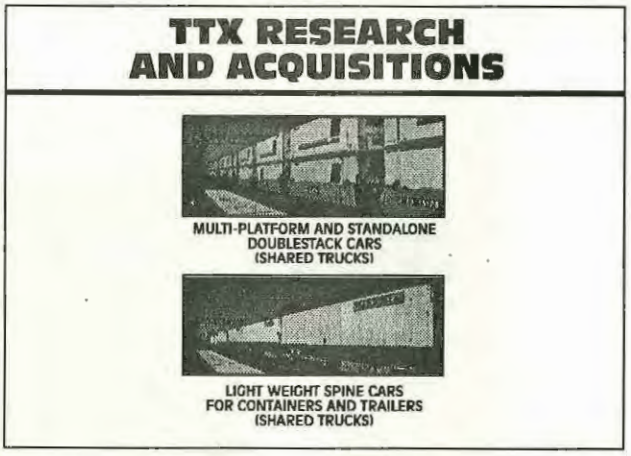


Figure 30

For example, shown on the left side of Figure 30 is the acquisition of new classes of articulated doublestack container-well cars, designed with increased capacity and shared trucks between platforms. Including the first *standalone* high capacity doublestack. Also, as shown on the right side of Figure 30, the first-time use of lightweight, multiplatform spine cars for carrying both containers and trailers, and an all-purpose version that can carry both. TTX has also performed extensive research and testing in evaluating single-axle trucks for intermodal rail cars.

As shown in Figure 31, TTX has recently been developing a new truck, the *TI-7*, which uses a parabolic leaf spring to serve both as the structural truck bolster and as the spring. The objective of this design is to significantly reduce lateral accelerations at 70 mph or greater.

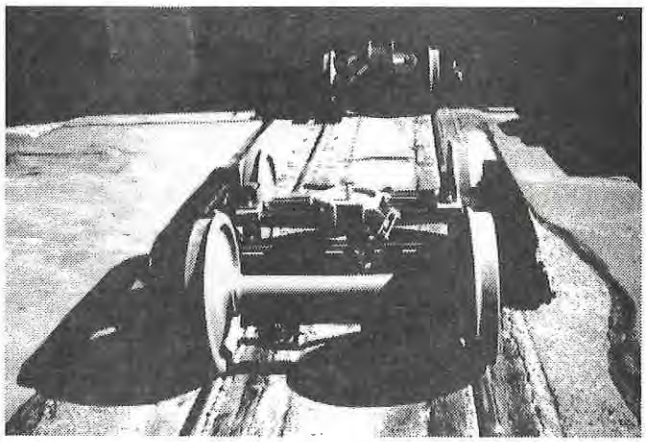


Figure 31

TTX has also pioneered the application of hydraulic yaw dampers (see Figure 32) to conventional three-piece trucks as a means to control truck hunting without generating excessive curving resistance. These yaw dampners have been applied to a significant number of autorack cars and 53-foot all-purpose spine cars.

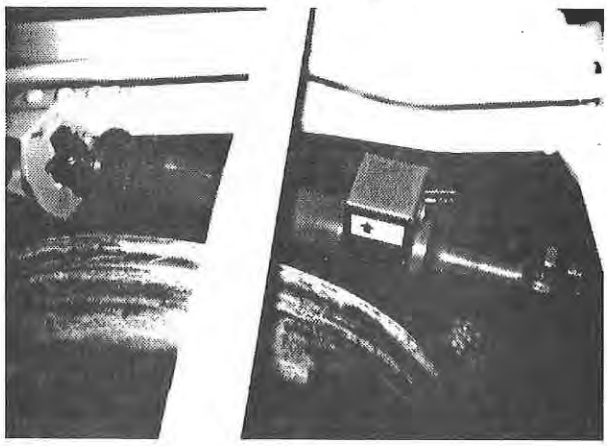


Figure 32

TESTING BY THE RAILROADS

Individual railroads have also performed extensive truck wear measurement tests over the years, and supported many truck ride environment tests. As indicated by the truck test reports shown in Figure 33, Burlington Northern, through a variety of individually sponsored truck tests and research, is one of the leading railroad users of Pueblo facilities and personnel. CP Rail is one of the foremost railroad leaders for the development and application of braced trucks that minimize wheel flange angles of attack and prevent truck warp. Norfolk Southern, Union Pacific, the Santa Fe, CSX, Conrail and a number of other railroads have also advanced truck testing and specific designs in recent years.

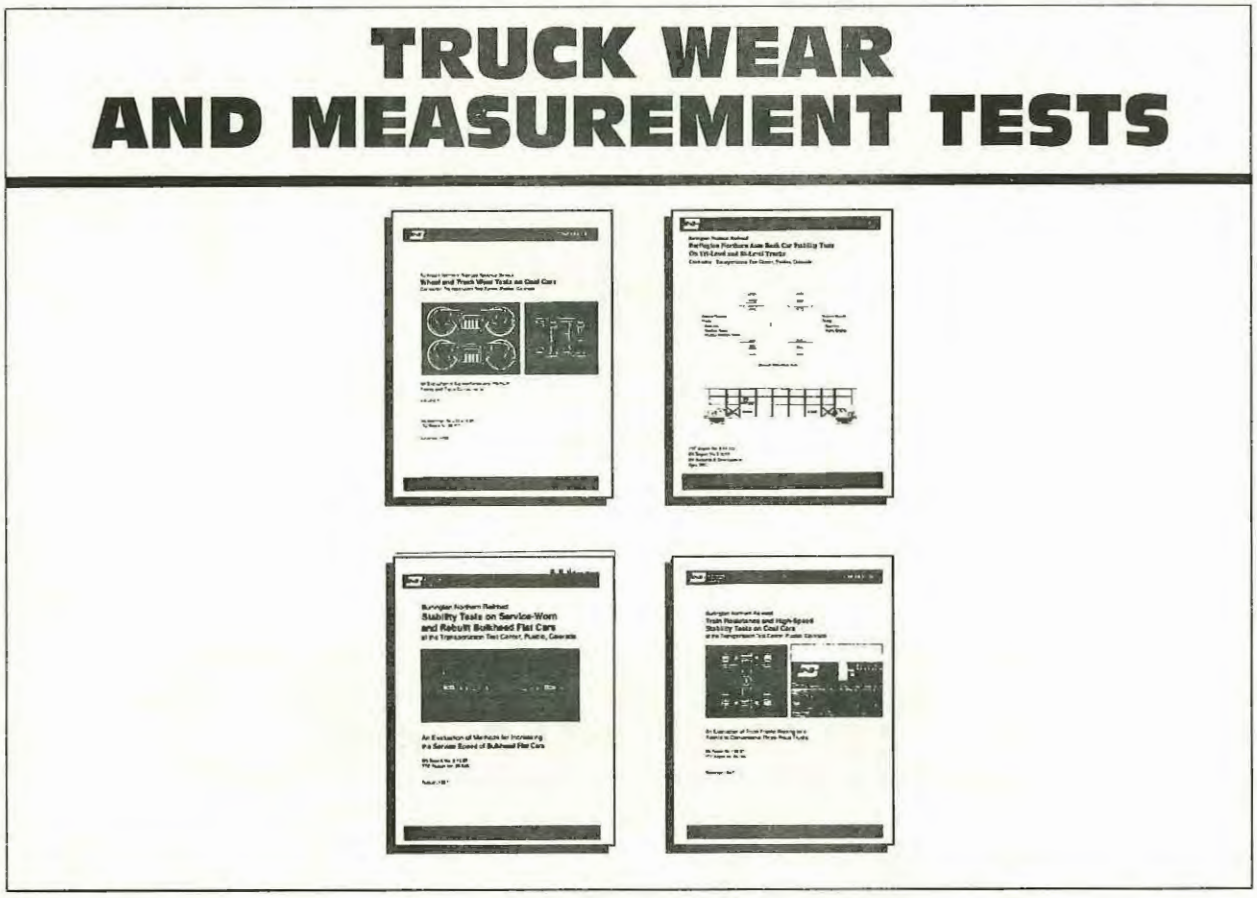


Figure 33

As the friction shoe wedge of a conventional three-piece truck rises, the available travel can also be restricted due to friction wedge lock-up and binding, as shown in Figures 34 and 35.

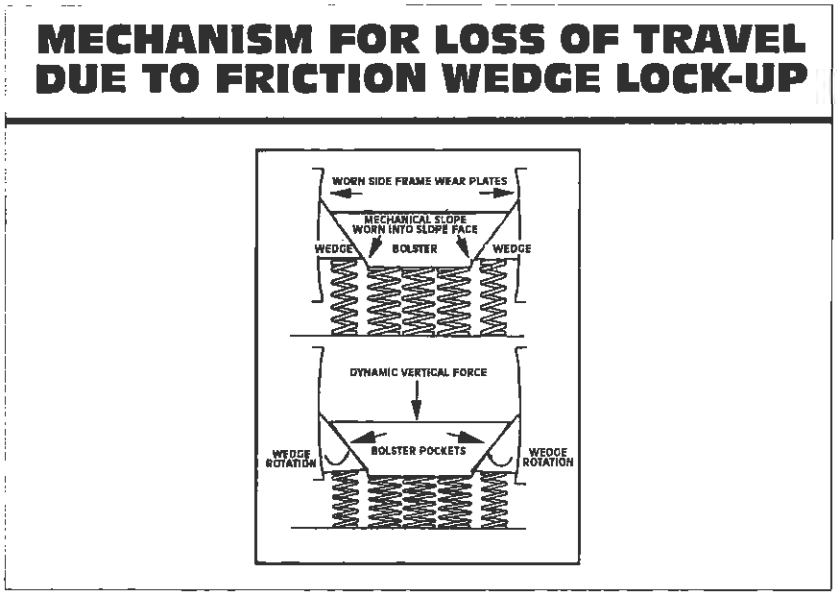


Figure 34

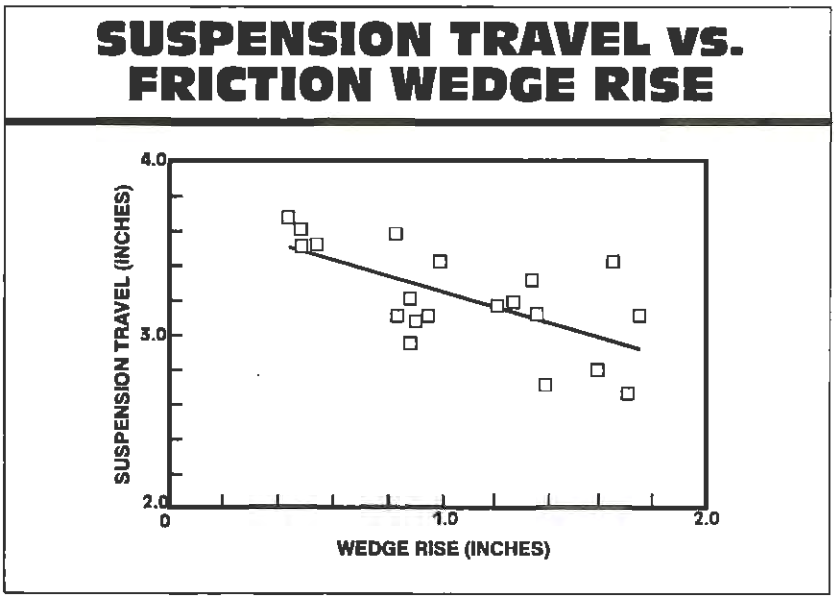


Figure 35

SKEWED TRUCKS

Skewed trucks with axle misalignments can lead to accelerated wear and derailment. And, a poorly aligned truck also consumes much more energy (fuel), which is an often overlooked, but very important economic consideration. It has been demonstrated in past AAR tests that a loaded car running on tangent track experiences energy losses that can lead to a a serious increase in overall train resistance on Class 2 and 3 track—up to 1.19 pounds per ton of additional rolling resistance.

TRUCK WEIGHT REDUCTION

It has also been estimated that an average 4000 pounds per car can be removed from trucks with the use of lightweight composite materials. This reduction can lead to a significant 230 tons or greater total weight reduction per aluminum coal train set, or a 10-20 percent reduction in total unit coal train tare weight (aluminum cars).

AEG Westinghouse Transport System in Europe, supported by the German Federal Ministry, has produced prototype fiber composite truck frames as recently as 1990. These trucks, some of which are shown in Figure 36, have performed well in ongoing intercity passenger train service.

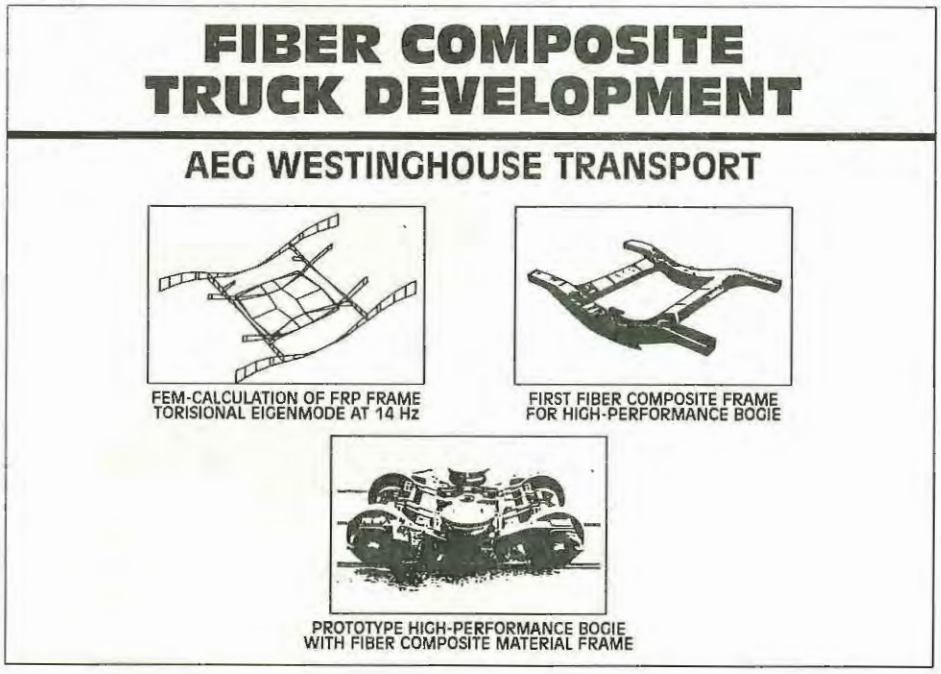


Figure 36

PREVENTION OF DOUBLESTACK RAIL ROLLOVER DERAILMENTS

There is also new information and solutions resulting from recent research by the rail industry to prevent rail-rollover truck related derailments from occurring with 125-ton doublestack cars. During the period of 1988 through 1992, the industry experienced 22 unexplained derailments in curves. These derailments took place on a variety of railroads in which primarily the low rail in curves was overturned.

INDUSTRY TASK FORCE

After an extensive three-year research effort by an industry task force, an 18-member team called the *Doublestack Dynamics Task Force (DDTF)* (see Figure 37) determined that truck warping of the supplying manufacturer's three-piece trucks was a primary factor leading to derailment of the 125-ton doublestacks. Most, if not all of the Doublestack Dynamics Task Force team members, shown in Figure 38, are represented at this truck technology forum.

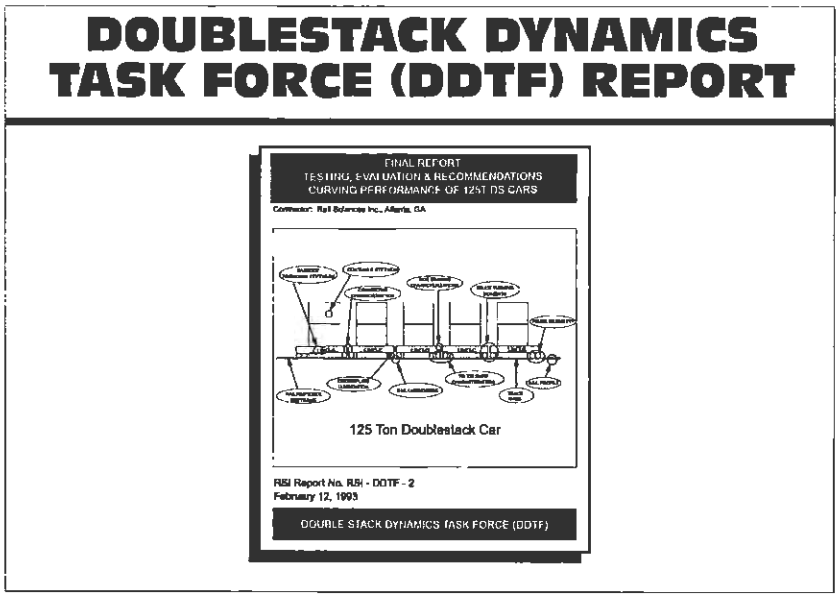


Figure 37

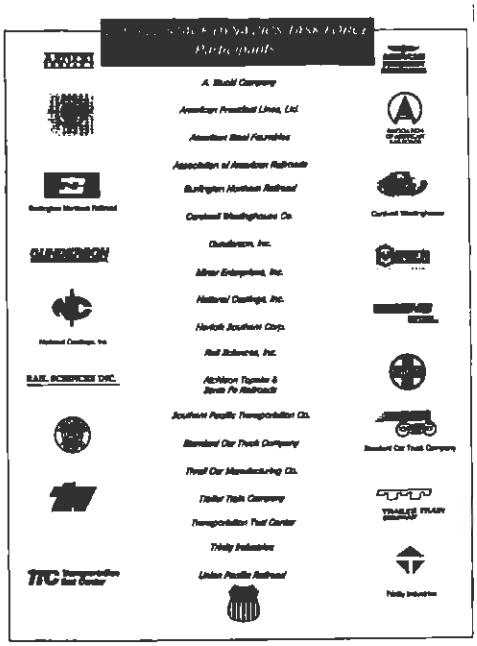


Figure 38

As shown in Figure 39, and as verified by the extensive mathematical modeling, the testing at Pueblo, the testing in manufacturer's fixtures, and the testing in revenue service, warped trucks that produce high gage spreading forces on both wheelsets versus normally curving trucks have been identified as a *primary factor* leading to the risk of derailment.

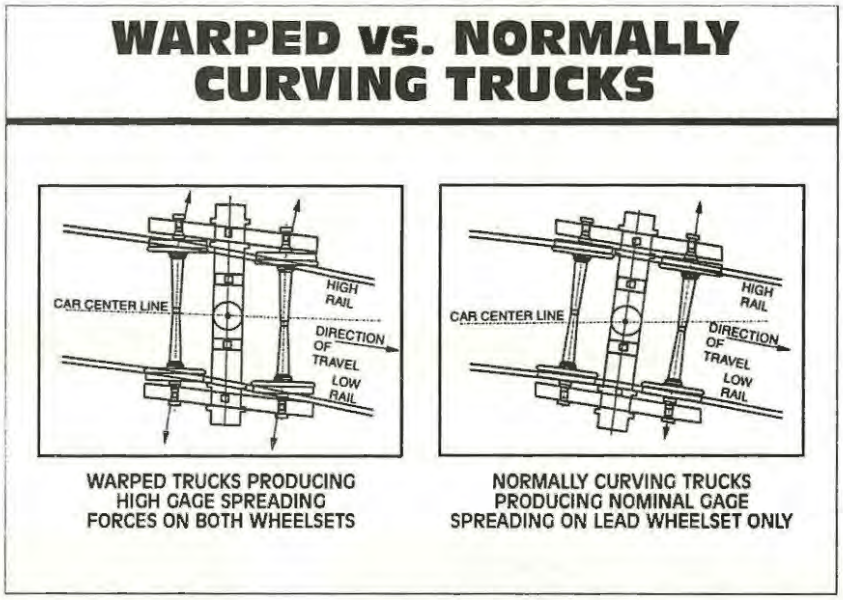


Figure 39

DERAILMENT PREVENTION

The truck-related recommendations coming from the Doublestack Dynamics Task Force included:

- Removal of resilient adapter pads (VIP pads)
- Annual lubrication of truck centerplates
- Enhanced constant contact side bearings
- Truck warp restraint

Implementing these recommendations, along with certain track and rail lubrication enhancements, has effectively eliminated the derailment problem with 125-ton doublestacks. This major dilemma which challenged one of our industry's major growth business lines was quickly and effectively resolved through the collective engineering and research efforts of the 18-member industry team. This is evidence that collective efforts, such as those forming in this Forum today, do breed success.

CHALLENGE OF THE FUTURE: *Where Trucks Are Going*

REVOLUTION, NOT EVOLUTION!

As an industry, in recent years we have rightfully steered research efforts toward two primary goals: to examine the overall problems associated with 100-ton three-piece truck designs, and to recommend maintenance guidelines and operating practices that will improve and optimize the standard three-piece truck's performance. However, we need to be challenged today to a higher objective—one that may lead us as an industry to revolutionary, not evolutionary, changes in truck designs, maintenance practices, cost-effectiveness, safety, ride quality and customer service.

We should be taking the *clean sheet* approach that EMD did a few years ago when it analyzed, tested, and produced the radically new *breakthrough technology* for AC traction and radial steering for locomotive trucks, now called the RADIALAC (see Figure 40). Railroads, including Burlington Northern, are now moving forward with rapid deployment of such locomotive trucks—evidence that breakthrough technology can occur.

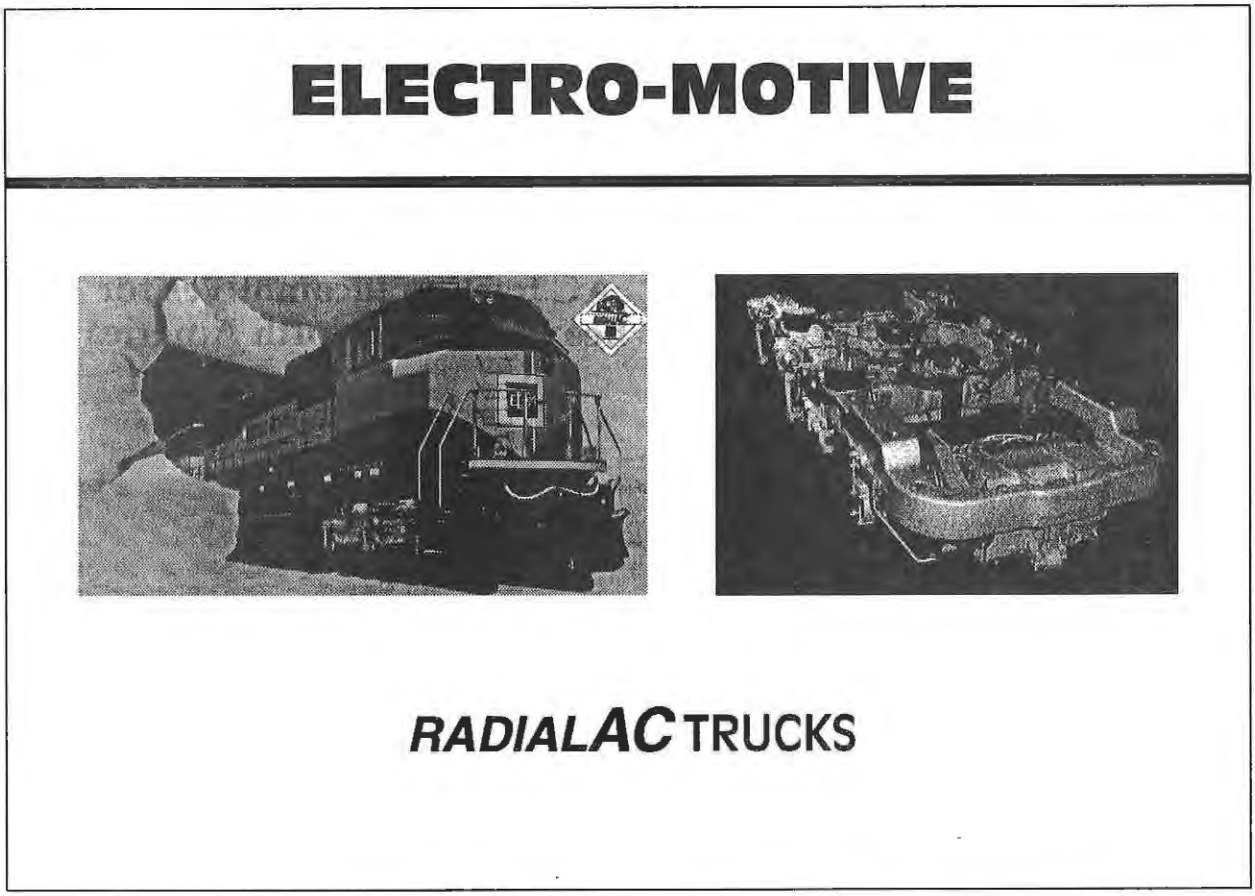


Figure 40

THE CHALLENGE

In conclusion, on behalf of AAR's Research Committee and its Chairman Carl Taylor, I want to leave you with the following challenge. Further fine tuning of the conventional three-piece truck to accommodate higher speeds and greater car capacities is still possible; to accomplish these goals, the long-term R&D objectives for trucks must be to:

- Develop a totally new and improved rail car suspension system
- Significantly reduce truck weight through the use of alternative materials
- Extend truck endurance to meet the cost competition and our customers' ride quality requirements for the Twenty-first Century.

With over 2.4 million freight car trucks in our industry, and over 45,000 new trucks acquired by railroads and others in the first six months of 1994, we'd better move quickly!



GENERAL OVERVIEW OF RELATED AAR RESEARCH TOPICS

A. J. REINSCHMIDT & A. J. PETERS

ASSOCIATION OF AMERICAN RAILROADS

AAR RESEARCH & TEST DEPARTMENT COMMITTEE ORGANIZATION

- AAR Board Committee on R&D
- Research Committee
- Steering Committees
- Technical Committees



AAR RESEARCH & TEST ACTIVITIES

- Strategic Research
- Strategic Technical Support
- Technology Scanning
- Technical and Engineering Support

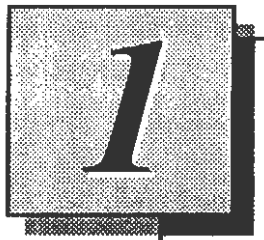
STRATEGIC INFRASTRUCTURE RESEARCH

Objective

- Provide a more reliable, cost-effective infrastructure for the North American Railroad industry.



HIGH PRIORITY STRATEGIC RESEARCH



WHEEL/RAIL LUBRICATION

- Film Deposition System
- Improved Lubricants and Tribometer

HIGH PRIORITY STRATEGIC RESEARCH

2 ADVANCED FREIGHT CAR TRUCK

- Design
- Wayside Defect Detection

HIGH PRIORITY STRATEGIC RESEARCH

3 VEHICLE/TRACK PERFORMANCE STANDARDS

- Track Strength/Geometry
- NUCARS



HIGH PRIORITY STRATEGIC RESEARCH

4

HEAVY AXLE LOAD IMPLEMENTATION

- FAST Operation
- Track Substructure Assessment
- HAL Revenue-Service Evaluation



HIGH PRIORITY STRATEGIC RESEARCH

5 WHEEL/RAIL PROFILE OPTIMIZATION

- Wheel/Rail Profile Maintenance
- Flange-Climb Derailment Investigation



HIGH PRIORITY STRATEGIC RESEARCH

6 VEHICLE DEFECT PREVENTION/ DETECTION

- Review of Bearing Reconditioning Standards (286,000 lb. vs. 263,000 lb. GRL)
- Bearing Inspection Strategy, Including Nondestructive Testing/Inspection



HIGH PRIORITY STRATEGIC RESEARCH

7 CUSTOMER-FOCUSED TRAIN SYSTEMS

HIGH PRIORITY STRATEGIC RESEARCH

8 ADVANCED TRAIN BRAKING SYSTEMS

- Advanced Train Braking Systems
- Advanced Coupling Systems Including Air/Electric Lines

HIGH PRIORITY STRATEGIC RESEARCH

9

SPECIAL TRACKWORK

- Current Designs and Maintenance Procedures
- Advanced Designs and Materials



HIGH PRIORITY STRATEGIC RESEARCH

10

BRIDGES

- Seismic Performance of Steel Bridges
- Steel Bridge Life Extension
- Timber Bridge Life Extension



HIGH PRIORITY STRATEGIC RESEARCH

11 TRACK COMPONENTS

- Alternative Crosstie Treatments
- High-Performance Rail Steels
- In-Track Rail Welding



HIGH PRIORITY STRATEGIC RESEARCH

12 TRACK OCCUPANCY IMPROVEMENTS

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HIGH PRIORITY STRATEGIC RESEARCH

13 ERGONOMICS IMPROVEMENTS IN EQUIPMENT DESIGN



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HIGH PRIORITY STRATEGIC RESEARCH

14 ENVIRONMENTAL PROTECTION

- Prevention
- Remediation



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TRUCK-RELATED MECHANICAL RESEARCH

A. J. Peters
Association of American Railroads

An Assessment of Needs and Opportunities - Freight Car Trucks For The 21st Century A Technology Transfer Forum

September 7-9, 1994
Levis Faculty Center
University of Illinois at Urbana-Champaign

ADVANTAGES OF THE THREE-PIECE TRUCK

- **Ease of assembly**
- **Simplicity**
- **Maintainability - abundance of parts**
- **Versatility**
- **Reliability and dependability**
- **Load equilization**
- **Economics**

DISADVANTAGES OF THE THREE-PEICE TRUCK

With Increased Wear

- **Reliability and dependability goes down**
- **Ride quality deteriorates**
- **Increased wheel wear/reduced bearing life**
- **Reduced warp stiffness/damping**
  - **Low rail gage spreading derailment**
  - **Loss of high speed stability**
- **Brake rigging malfunctions**

TRUCK-RELATED MECHANICAL RESEARCH PROJECTS

- **Condition-related performance of the three-piece truck**
- **Corrective measures for performance deficiencies**
  - **Center bowl lubrication**
  - **Wheel profile/rail grinding studies**
  - **Wheel sliding (with and without braking)**



TRUCK-RELATED MECHANICAL RESEARCH PROJECTS

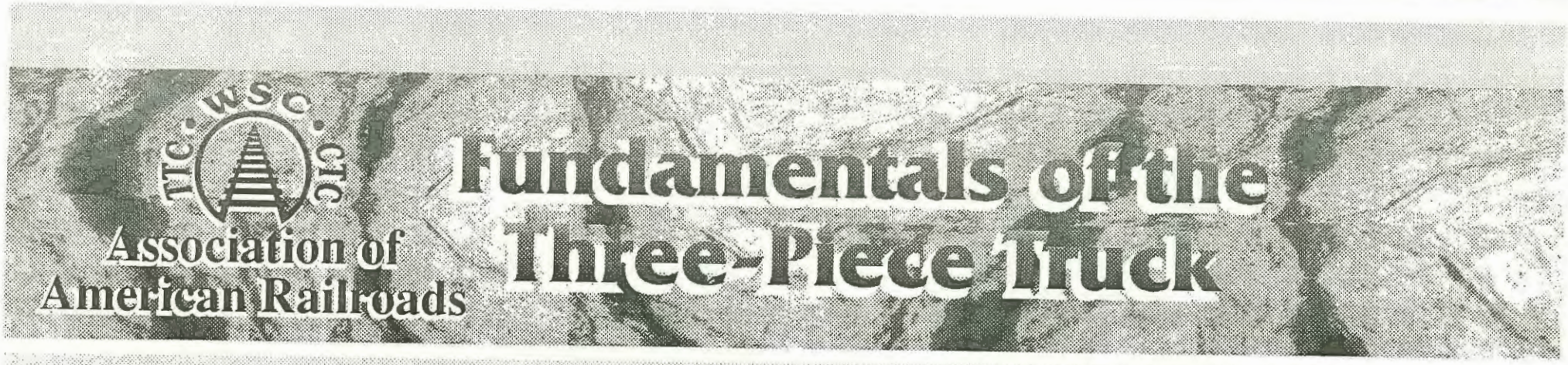
- **Performance requirements for new and improved applications**
  - **HAL truck program**
  - **Ride quality measurements**
  - **New bearing performance specification**
  - **Disk/tread braking**
- **Economic analysis**



FUNDAMENTALS OF THE THREE-PIECE TRUCK

S. P. SINGH

ASSOCIATION OF AMERICAN RAILROADS



Sōm P. Singh

An assessment of needs and opportunities--
freight car trucks for the 21st Century

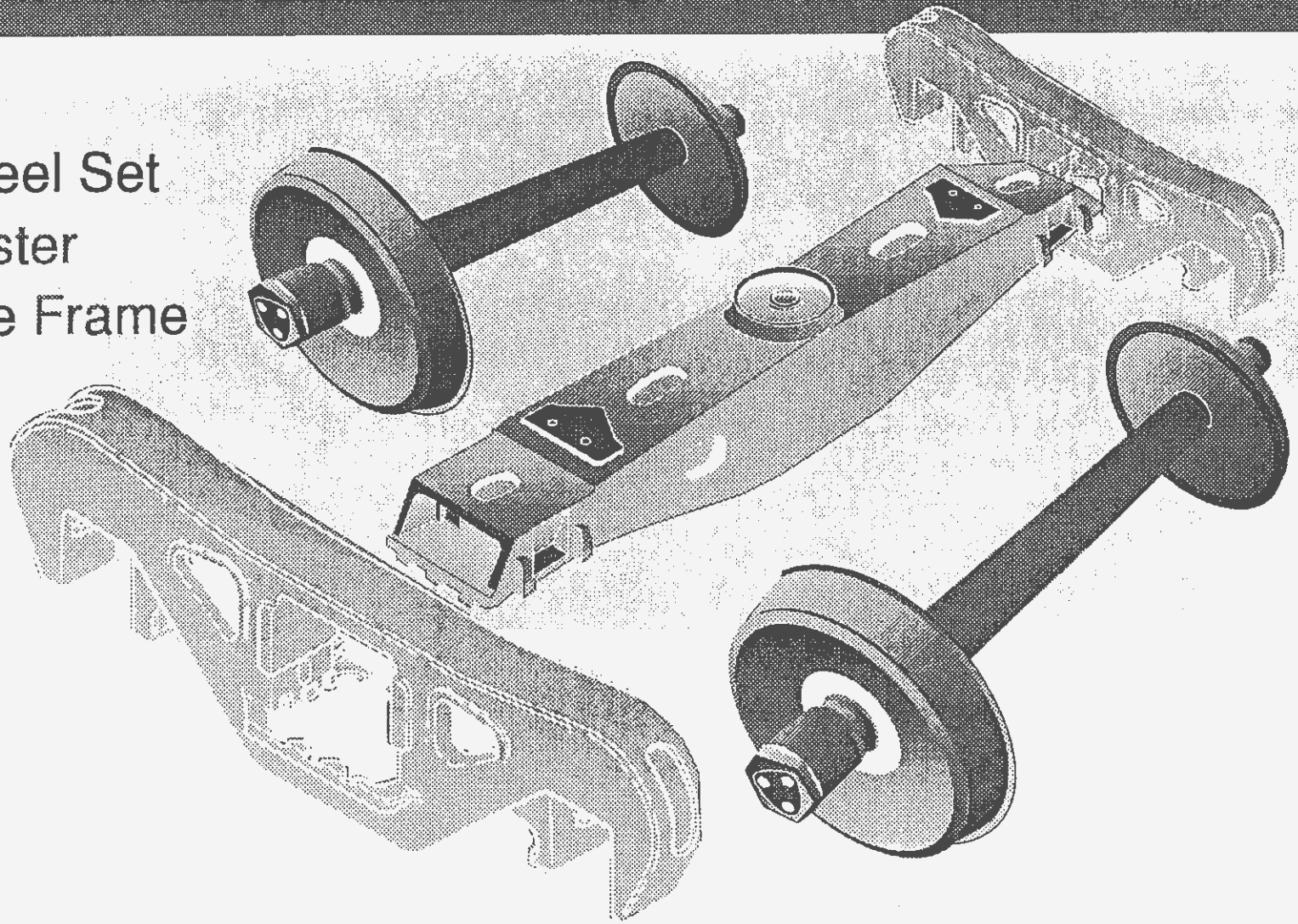
A Technology Transfer Forum

September 7-9, 1994

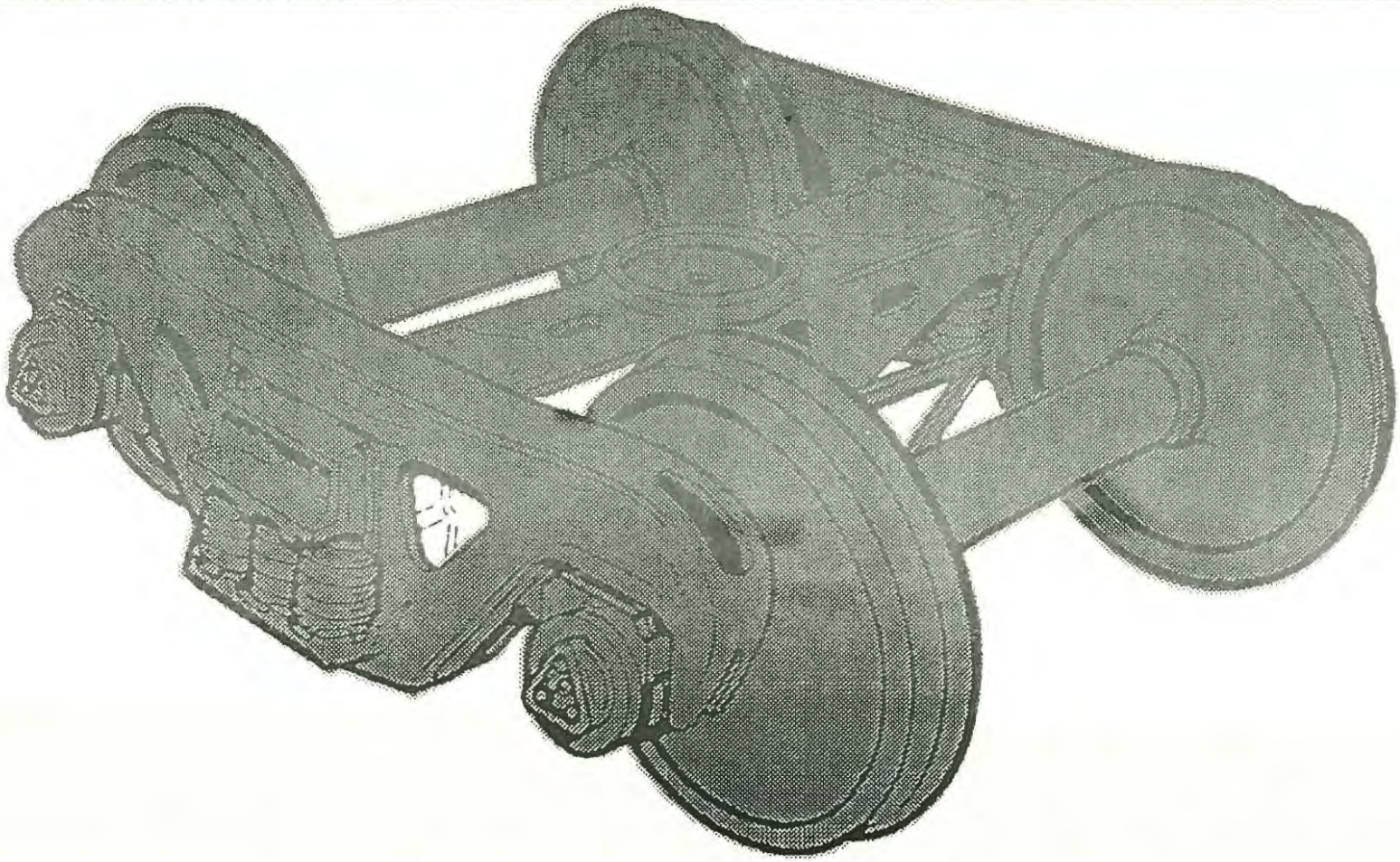
Levis Faculty Center, University of Illinois at Urbana-Champaign

Fundamentals of the Three-Piece Truck

1. Wheel Set
2. Bolster
3. Side Frame



Fundamentals of the Three-Piece Truck



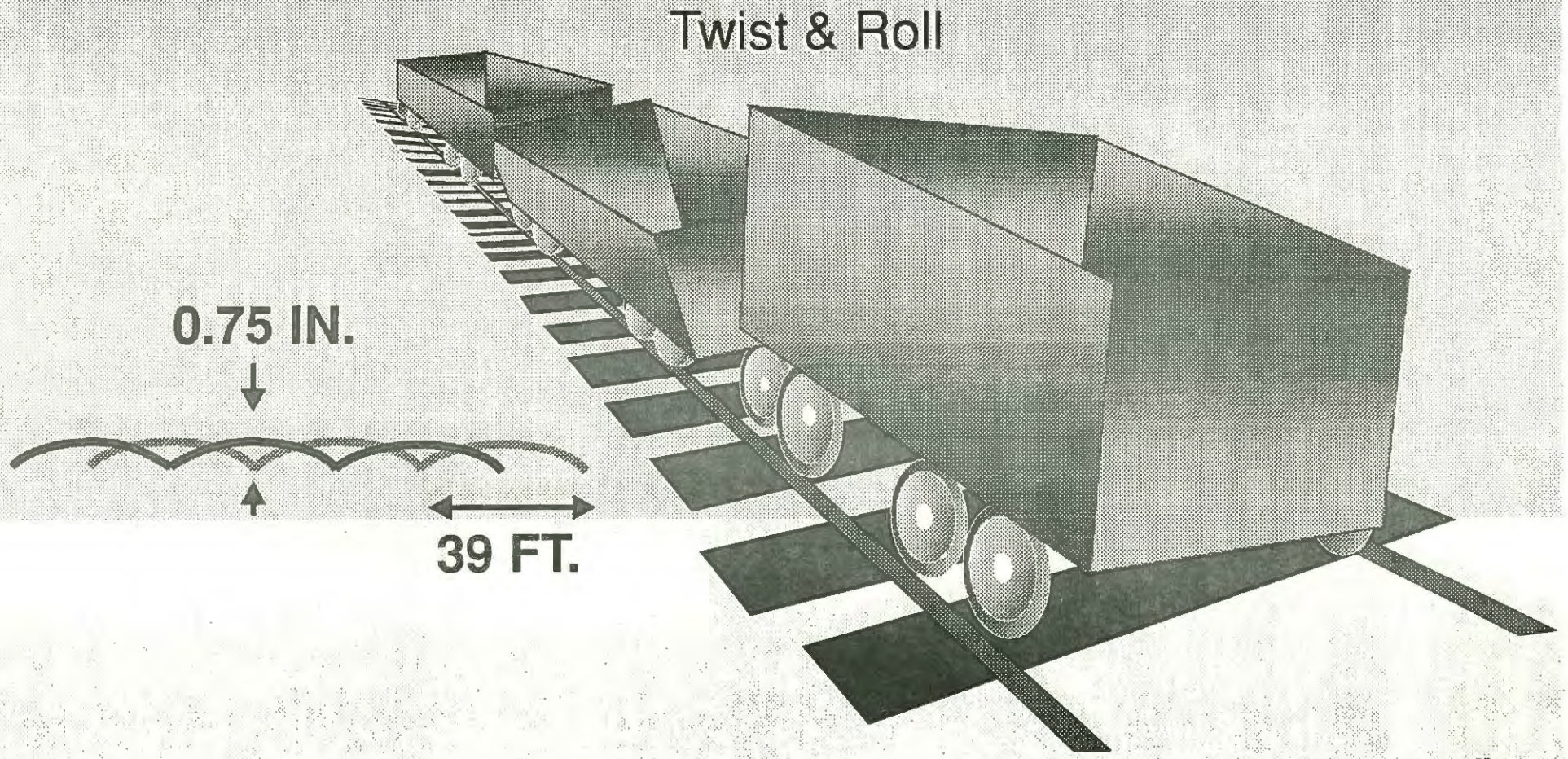
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Fundamentals of the Three-Piece Truck

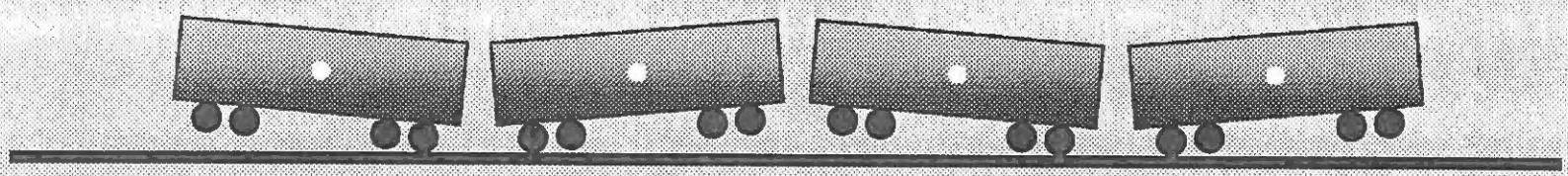
Functions of the Three-Piece Truck

- Carry the Vertical Load
- Provide Guidance

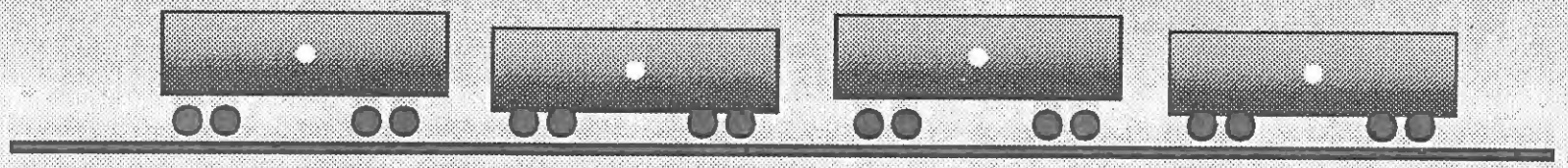
Isolate Lading from the Track Geometry Deviations by Attenuating Dynamic Response of the Car



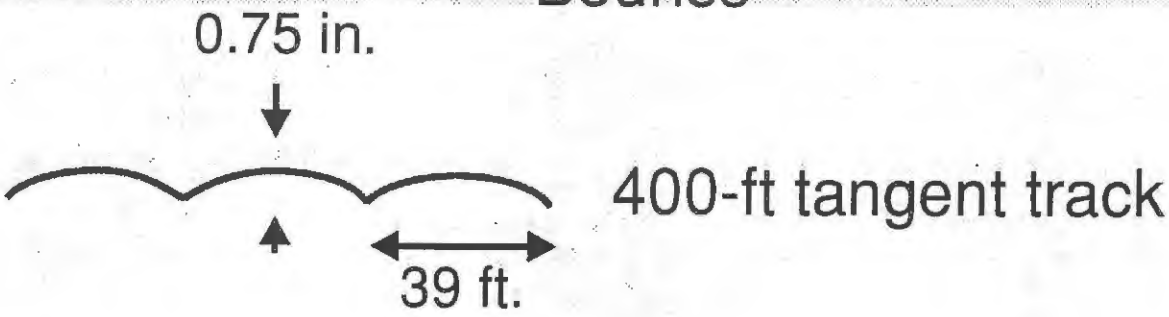
Isolate Lading from the Track Geometry Deviations by Attenuating Dynamic Response of the Car



Pitch

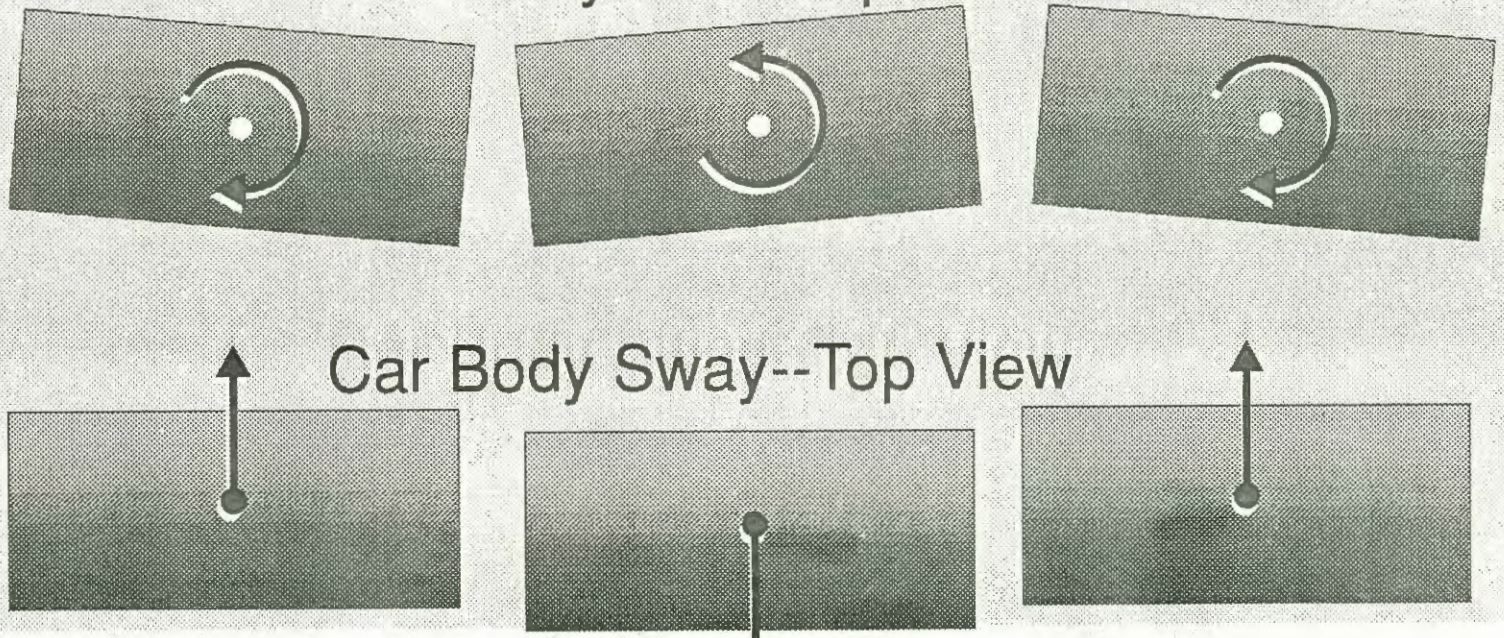


Bounce



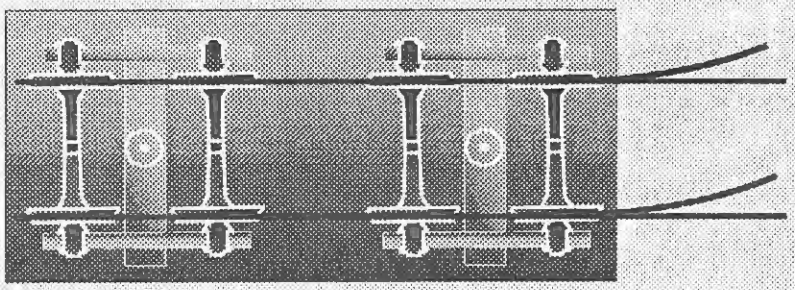
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Isolate Lading from the Track Geometry Deviations by Attenuating Dynamic Response of the Car

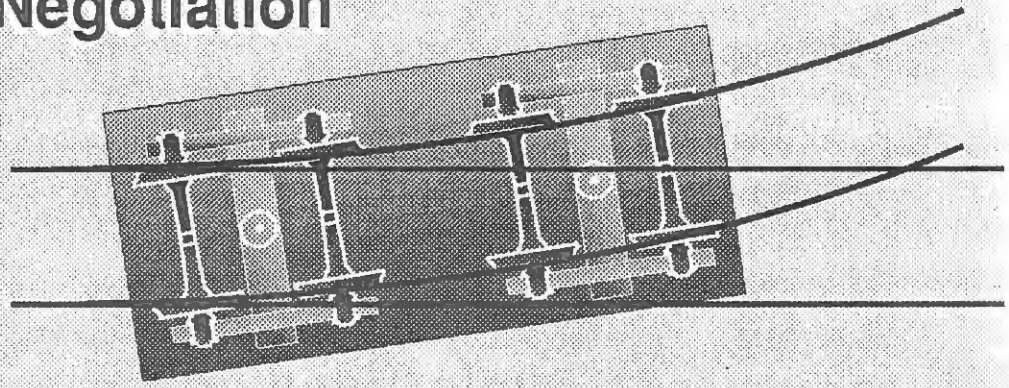


Guide the Car through Curves and Turnouts

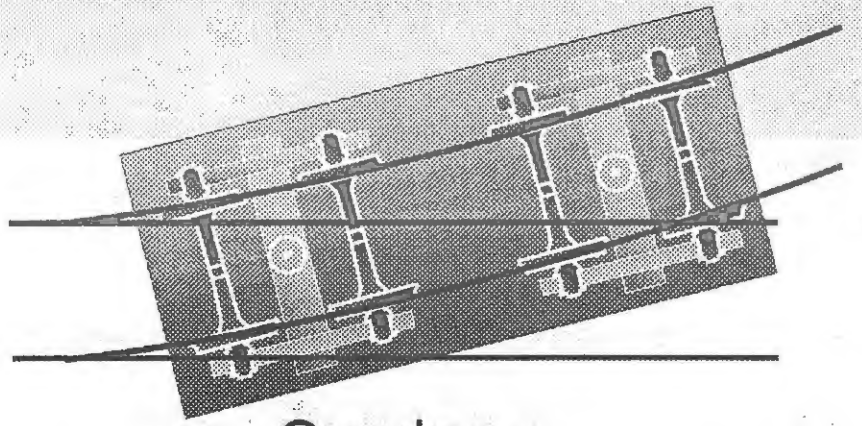
Turnout Negotiation



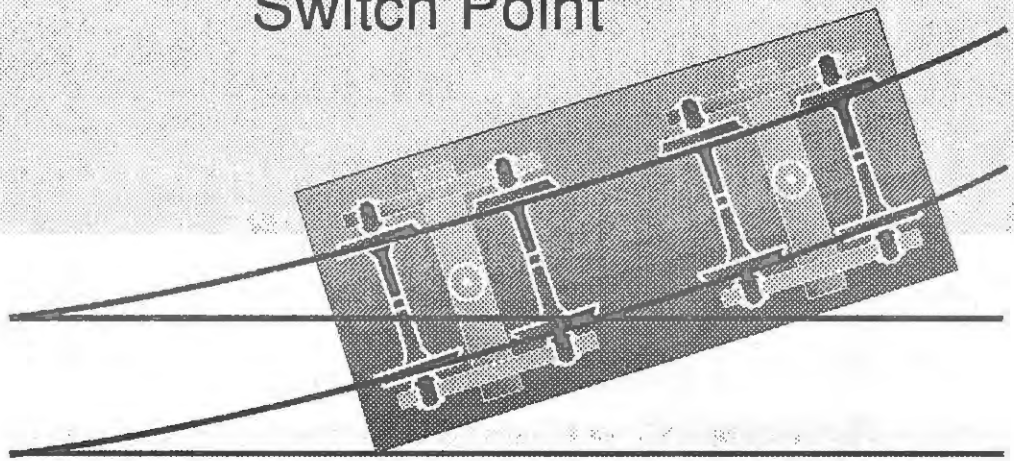
Entry



Switch Point



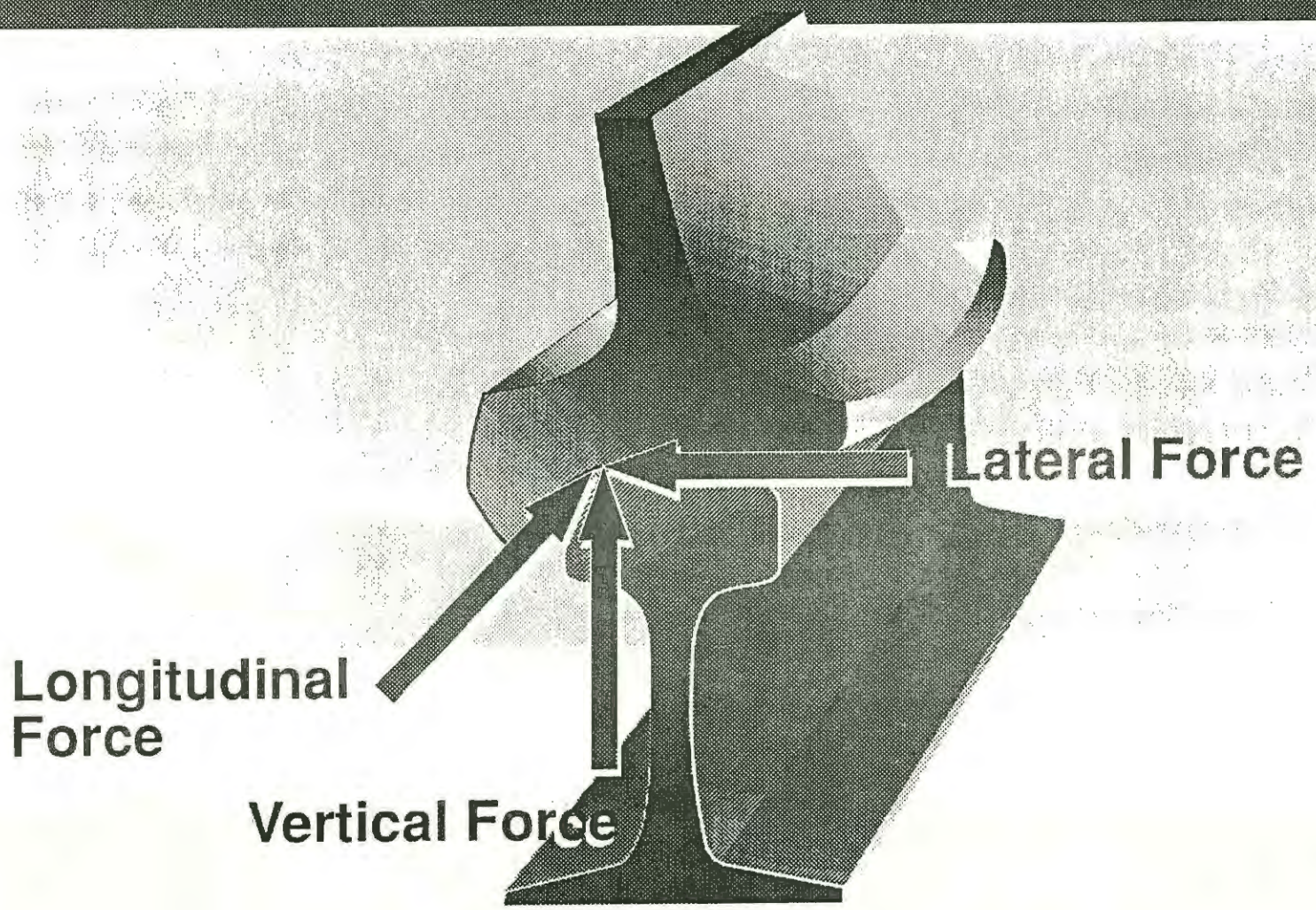
Curving



Frog/Guard Rail

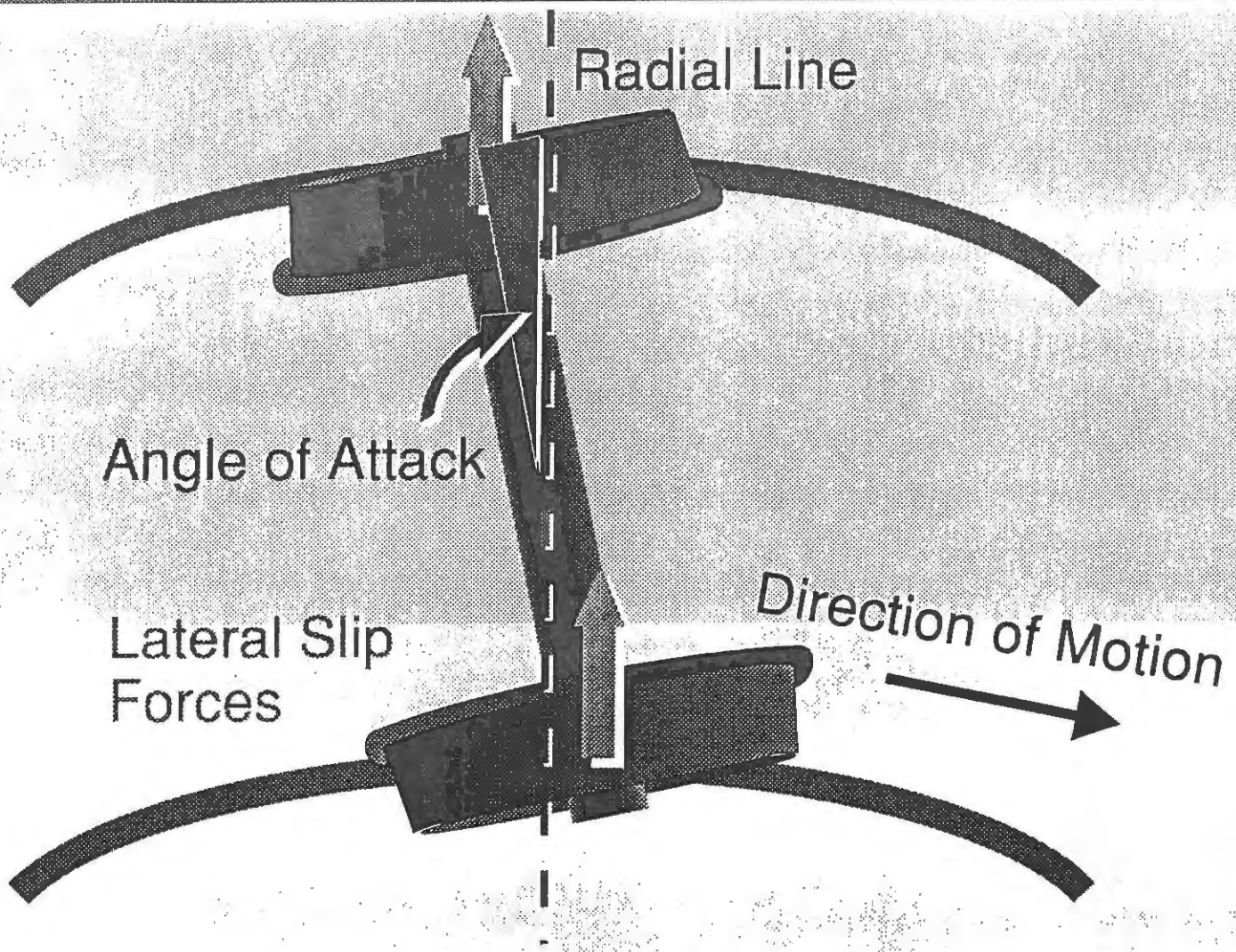
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Wheel / Rail Forces



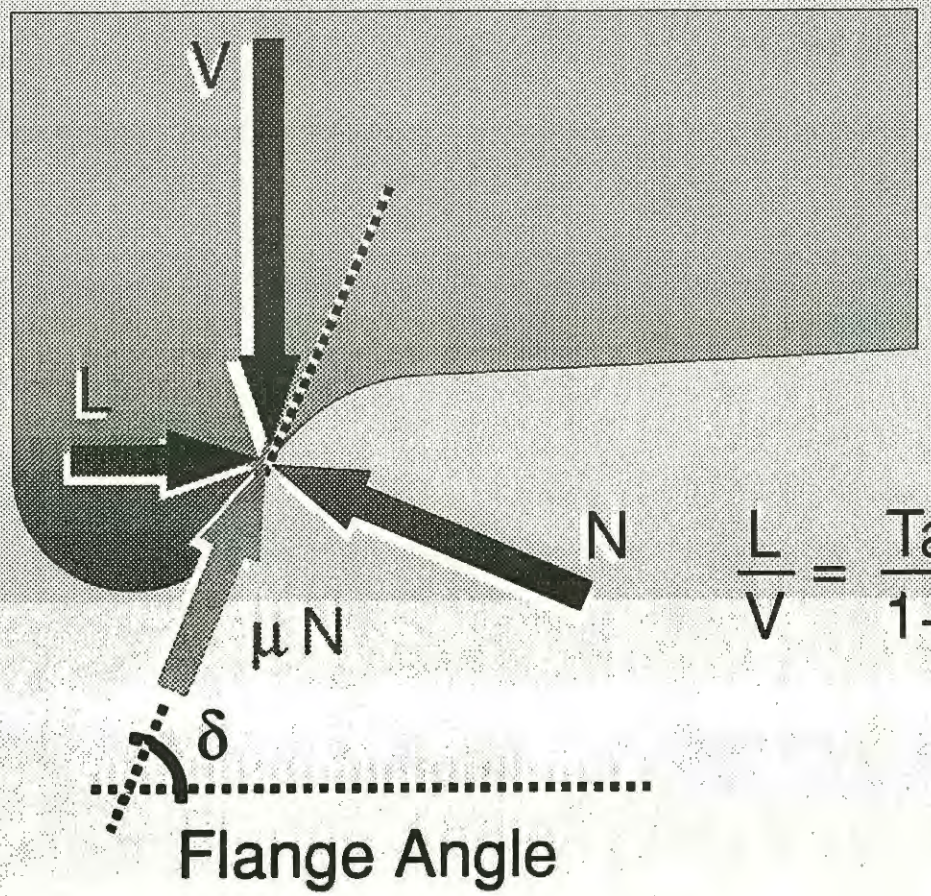
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Lateral Wheel Set Forces



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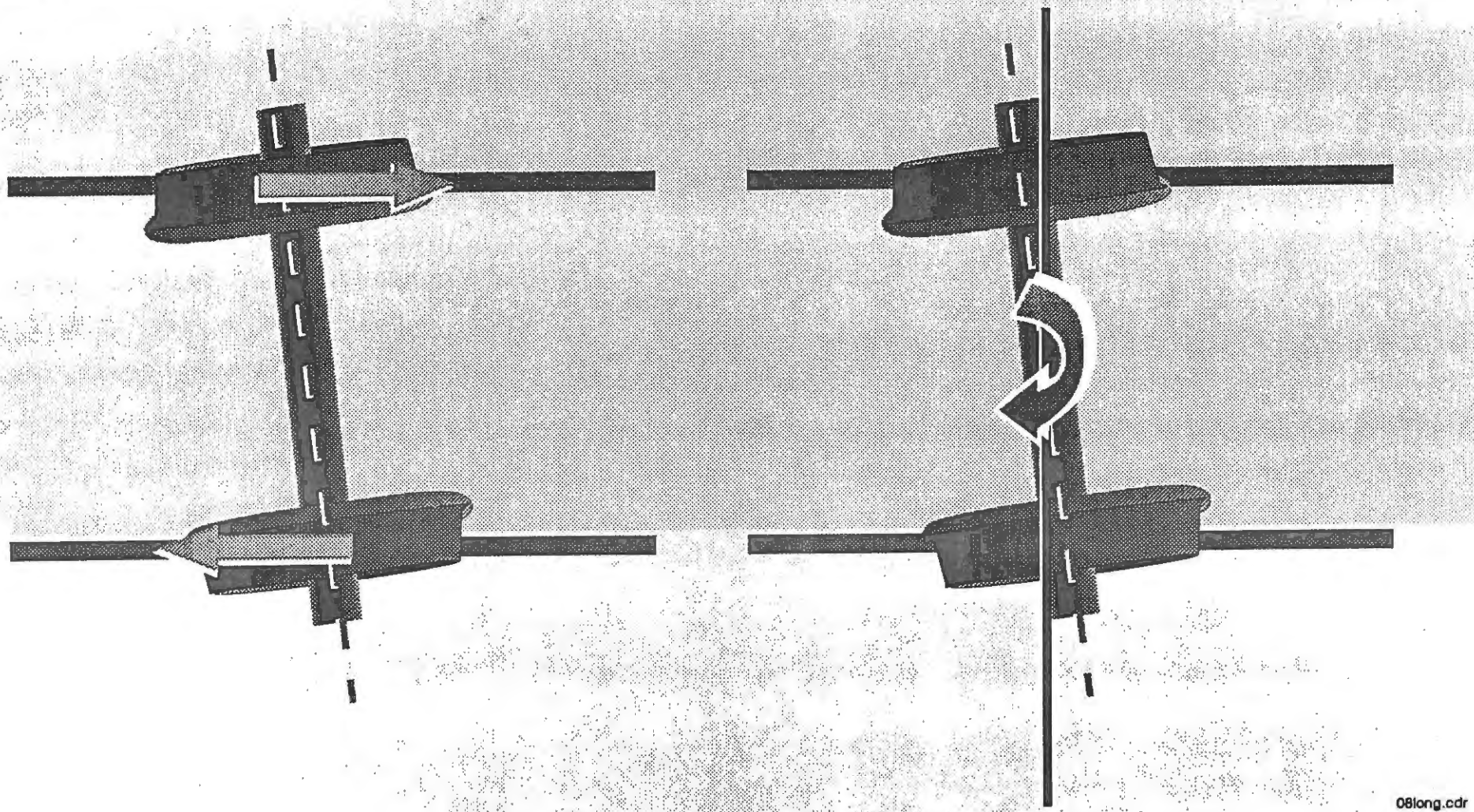
Wheel Forces



$$\frac{L}{V} = \frac{\mathrm{Tan}(\delta - \mu)}{1 + \mu\,\mathrm{Tan}(\delta)}$$

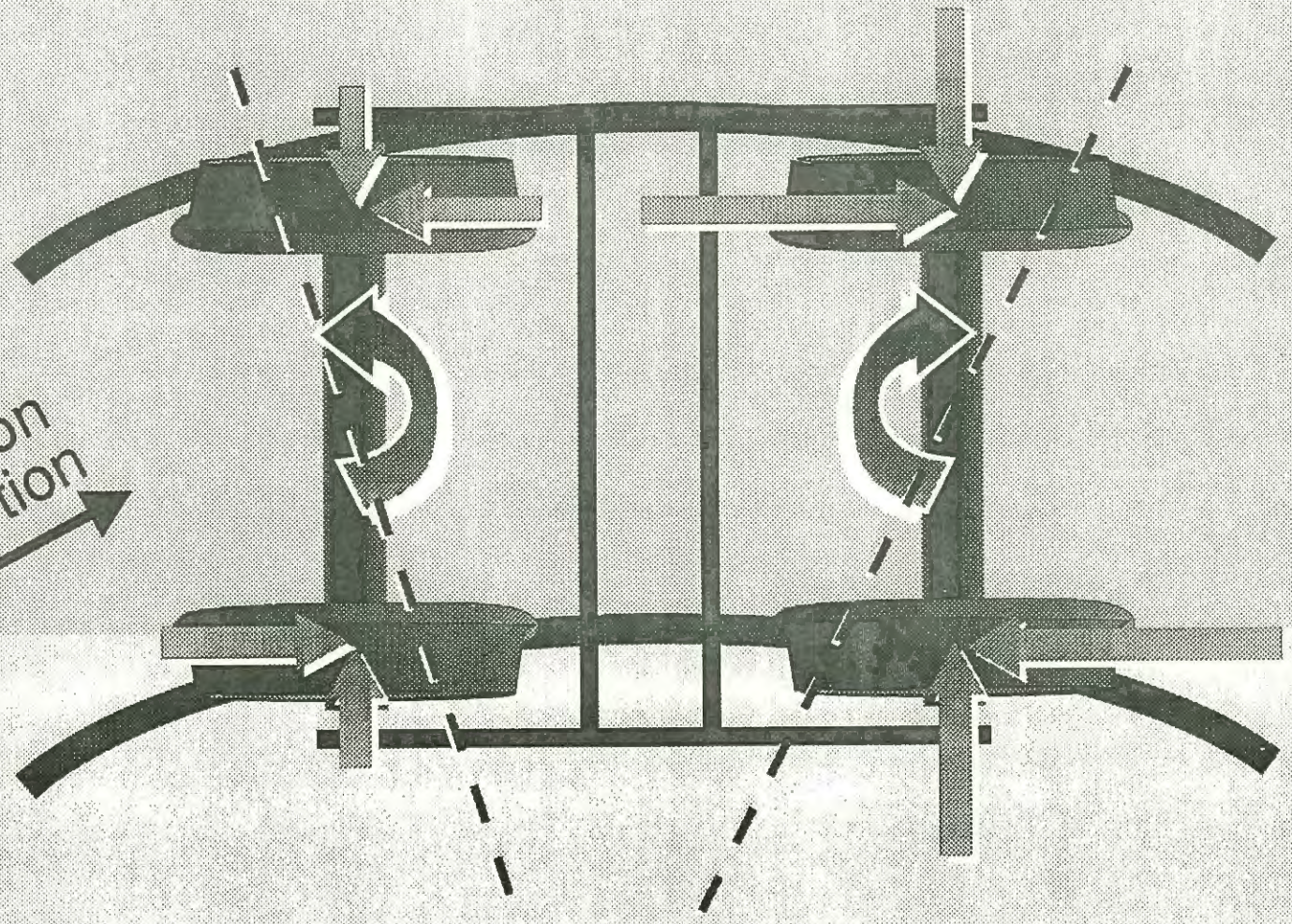
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Longitudinal Wheel Set Forces



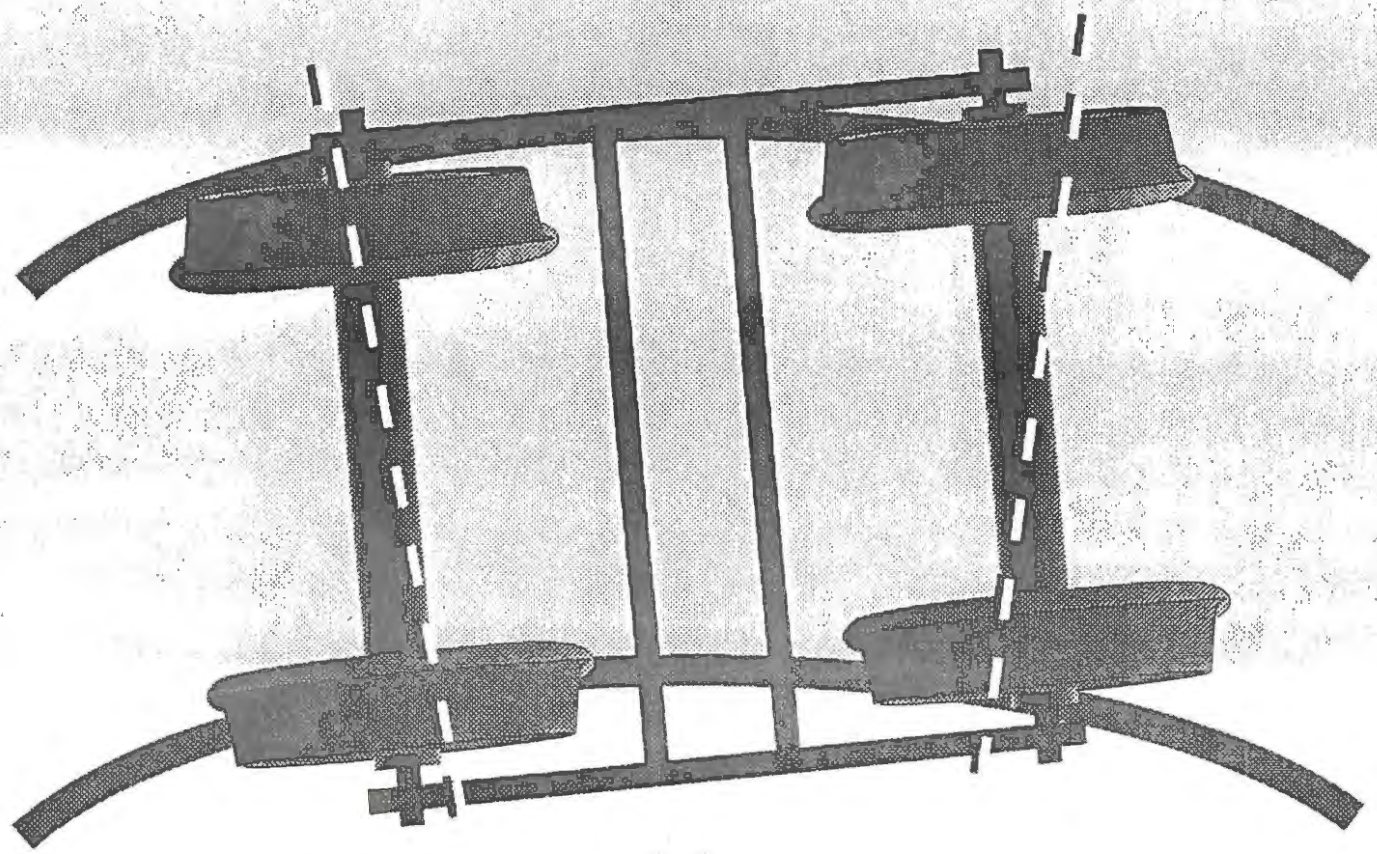
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Truck Curving Forces



12truck.cdr

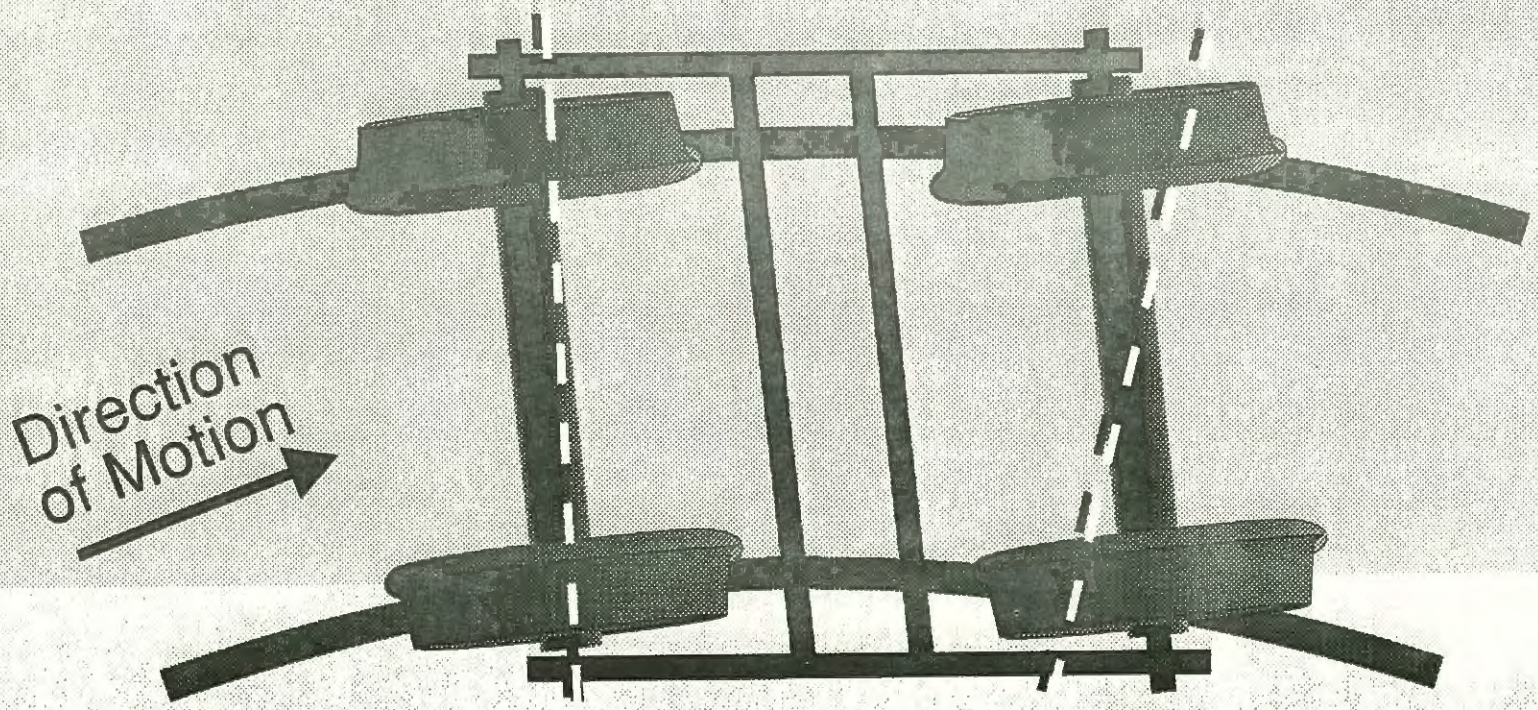
Curving Position--without Truck Warp



Direction of Motion

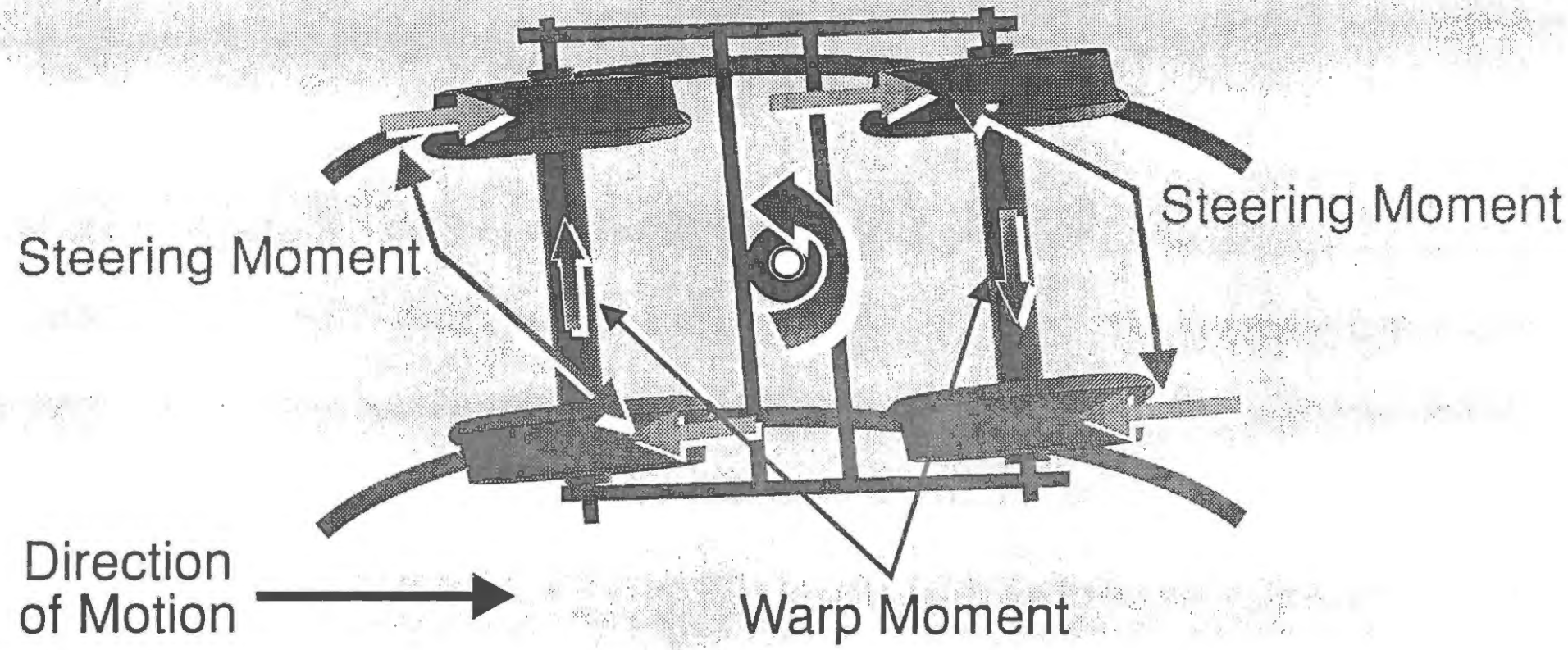
13wowarp.cdr

Curving Position--Truck Warped



14warp.cdr

Forces & Moments on a Warped Truck

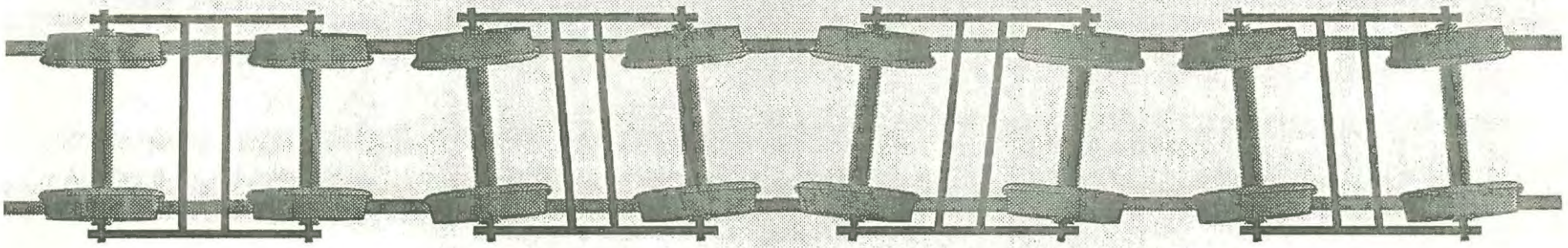


Turning Moment = Steering Moment + Warp Moment

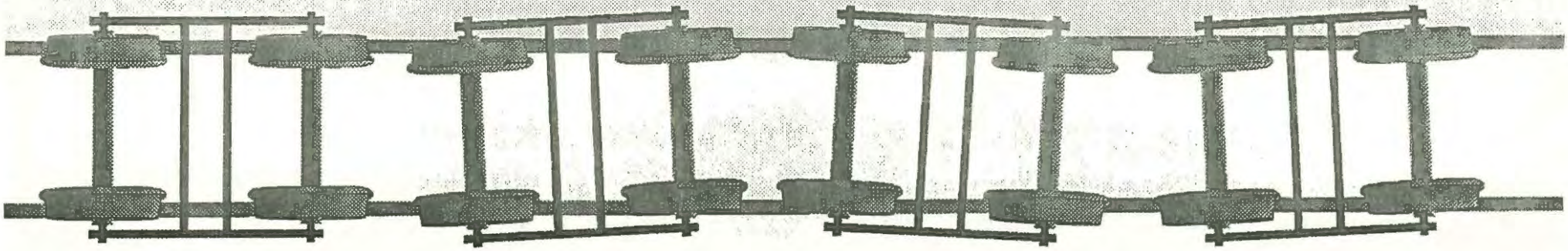
15warped.cdr

Truck Hunting

With Warp



Without Warp



16w&wowp.dr

Other Truck Performance Issues

Turning Moment

- Center Bowl Lubrication
- Liner
- Edge Contact
- Side Bearing Preload

Other Truck Performance Issues (continued)

Steering Characteristics

- Wheel / Rail Profiles
- Single Point vs 2-Point Contact
- Lubrication

Other Truck Performance Issues (continued)

Warp Characteristics

- Side Frame / Bolster Interface
- Bearing Adapter Condition
- Wedge Rise
- Load & Control Coil Conditions

20other.cdr

NEW CAR SYNDROME REVISITED

L. A. McLEAN

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Family Lines Rail System,
Jacksonville, FL
Mem. ASME

H. B. Weber
Engineering Manager - Trucks,
National Castings Division,
Midland-Ross Corporation,
Cleveland, OH

New Car Syndrome—The Derailment Tendency of New Freight Cars

An investigation of a series of derailments involving new coal hopper cars led to the discovery of certain characteristics of friction snubbing, casting tolerances and other factors which contribute to derailments, a condition termed as "new car syndrome."

BACKGROUND

Following delivery of a group of 1,000 new 100 ton hopper cars, a series of 14 derailments occurred over a period of several months. In each case it was found that the first car to derail had been in service less than two months. Because the derailments generally occurred on curves with the train speed close to the critical for rock and roll, the first assessment was that these were typical rock and roll derailments. However, other trains operated over the same derailment sites without problem. Moreover, a group of 450 identical cars received from the same carbuilder some two years previous to the subject cars had been operating satisfactorily without derailment problems.

In reviewing past derailment problems, it was recalled that as far back as 1962, 100-ton cars had been derailing on the L&N immediately after delivery from the manufacturer. Subsequent investigations led to considerable test work of such cars on the Frankfort, Kentucky test track.[1] It was these tests that defined the rock-and-roll problem and brought about the development of a variety of devices to stabilize cars at critical speed. Because of the mountainous terrain through which some parts of the railroad operate, L&N has many sharp curves with super elevation in the track. To protect its equipment, L&N invested large sums of money for hydraulic supplemental snubbing.

Nevertheless, the record shows that despite these measures, there have been sporatic cases where certain lots of new cars have had tendencies to derail shortly after being delivered. One such experience which vividly illustrates the case in point involves a group of 4,000 cu. ft. covered hopper cars which caused over ten derailments on L&N. These new SCL cars were en route from the carbuilder's plant to the phosphate fields in Florida when the derailments occurred.

Because the cars were equipped with only double coil side springs on the snubber blocks and had no supplemental snubbing, the L&N set out to prove that the lack of adequate rock and roll dampening caused the derailments. By the time arrangements to test the cars at Frankfort, Kentucky were completed, the cars had accumulated about six months' service. The testing of both loaded and empty cars did not produce even a single wheel lift.

A similar experience occurred with a group of L&N 70 ton coal hopper cars on which rock and roll is not considered to be a problem. Experiments with constant contact side bearings appeared to treat the problem, but it was actually service time which was the final cure.

NEW CAR SYNDROME

The mysterious tendency for derailment of new cars has often been referred to by railroad men in the field as "new car syndrome". The extraordinary cost of the damage caused by these most recent derailments dictated that a solution be found to the problem. It simply could no longer be assumed that only rock and roll action was responsible for derailment of new cars especially since the cars were already equipped with premium hydraulic devices.

[1] The Frankfort test track is a portion of a lightly used branch line. The area in which tests were conducted contained a 9^{0} curve with 6" super elevation. A special high guardrail was installed next to the low rail to prevent derailment when test conditions caused large wheel lifts. The test track is no longer used for testing.

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Copies will be available until August 1, 1982.

Field Inspections

Additional insight into the problem was gained from field inspections and particularly from observing the cars operating. On one significant observation from the ground as a train rounded a curve at slow speed it was noted that the wheels on the outer rail in a curve literally rode the wheel flanges and the side of the rail. The wheels were galling heavily along a band on the flanges. The base of the rail in the area was liberally sprinkled with steel filings. The performance of the trucks under one of the subject cars was also observed from a caboose as the train traversed the area of previous derailments. Throughout the trip, only one end of the bolster actually appeared to be moving up and down and it did so with a popping noise. The opposite end appeared to be motionless, indicating an abnormality in the operation of the truck.

Following this, a full scale investigation of each of the truck components was made by L&N. The dimensional differences discovered led to the laboratory test program at the National Castings Division of Midland-Ross Corporation Technical Center.

Details of Trucks

The cars were all equipped with Barber S2C trucks with double coil friction snubber springs. The spring grouping was the standard AAR arrangement with 6 D-5 outer coils, 6 D-6 inner coils and 4 D-6-A inner-inner coils. Center plate bowl diameter is 14" and brake arrangement is a rod-through design with truck dead levers connected to the center sills. Wheels are AAR standard 1-W 36" diameter with AAR 1:20 tread contour. Stucki HS7-100 hydraulic snubbers provide the supplemental snubbing. Although the truck castings were furnished by three manufacturers, the side frames and bolsters were assembled together in matched sets of the same manufacturer's castings.

Summary of Derailment Factors

Simultaneously with the test program, the overall operating conditions relating to the problem were examined and a list of factors which could contribute to derailment of the hopper cars was developed.

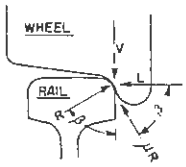
Each factor was analyzed to determine its potential contribution to vertical wheel unloading, a condition which can lead to wheel climb or wheel lift and subsequent derailment. The estimated contribution to wheel unloading is shown in Table 1.

Table 1 Summary of Derailment Factors Contributing to Vertical Wheel Unloading

DERAILMENT FACTORS	Vertical Wheel Unloading, %
1. Worn vs. New Wheel Profile	* 0 to 38
2. Worn vs. New Rail Head Profile	?
3. 3" Superelevation	* 16
4. 2" Dip at Low Rail	** 10
5. Friction Forces of Truck Friction Mechanism	**16 to 24
6. Hydraulic Snubber Forces	**16 to 33
7. Critical Speed - Car Roll (Dynamic Augment)	?
8. Longitudinal Train Action Force	?

* Calculated
** Test

Item No. 1, (Table 1) Worn vs. New Wheel Profile. The comparison is based on the fact that new wheel flanges have a relatively low slope angle compared to worn wheels which have a steep (almost vertical) slope due to wear. Therefore, a new wheel can climb the rail much easier, i.e., with less vertical unloading (or less lateral force per a given vertical load) compared to a worn wheel. A theoretical analysis of flange climbing based on L/V ratios (Fig. 1) shows that a worn wheel can tolerate up to 38% more vertical unloading before climbing takes place compared to a new wheel. Therefore, one method of reducing the likelihood of derailment would be to equip cars with wheels manufactured with a contour close to that of a worn wheel. Such a new wheel contour with a flange angle or slope of 78^{0} has already been proposed.



$\frac{L}{V} = \frac{\tan\beta - \mu}{1 + \tan\beta \cdot \mu}$ WHERE $\mu = .3$ (NADAL'S FORMULA-REF: AAR REPORT MR 435)

THEREFORE:

NEW WHEEL ($\beta = 69^{\circ}$)	PROPOSED NEW WHEEL ($\beta = 78^{\circ}$)	WORN WHEEL ($\beta = 81^{\circ}$)
$\frac{L}{V_1} = 1.29$	$\frac{L}{V_2} = 1.83$	$\frac{L}{V_3} = 2.08$

TO DETERMINE VERTICAL LOADS, LET L=1,

THEN $V = \frac{L}{\text{RATIO}} = \frac{1}{\text{RATIO}}$

EXAMPLE: $V_1 = 1/1.29$, $V_3 = 1/2.08$

THEN, SETTING $V_1 = 100\%$ AS BASE LINE AND $V_3 = X\%$,

$X = \frac{1/2.08}{1/1.29} \times 100 = \frac{1.29 \times 100}{2.08} = 62\% = V_3$

WHEEL TYPE	L/V RATIO	%V	PERMISSIBLE % OF VERTICAL WHEEL UNLOADING TO WHEEL CLIMB
NEW AAR	1.29	100	0
PROP. 78°	1.83	71	29
WORN	2.08	62	38

FIG. 1 L/V RATIO FOR IMPENDING WHEEL FLANGE CLIMB

Item No. 2 (Table 1) Worn vs. New Rail Head Profile. The calculations for Item No. 1 above were made for new rail. Considering that much of the rail in service has a curve-worn profile compounds the problem and makes the need for wheels with higher slope angle an even more important factor. Figure 2 illustrates profiles of heavily worn rail which were found during an NTSB investigation of a serious derailment on a major railroad and exemplifies rail which cars may be required to operate on.

Item No. 3 (Table 1) 3" Superelevation. Figure 3 shows the vertical forces acting on a wheel set under static condition on level track. The percent of wheel unloading for Item 3, i.e., 16%, was calculated from car and track geometry for a curve having a moderate superelevation of 3", the superelevation existing on most derailment sites.

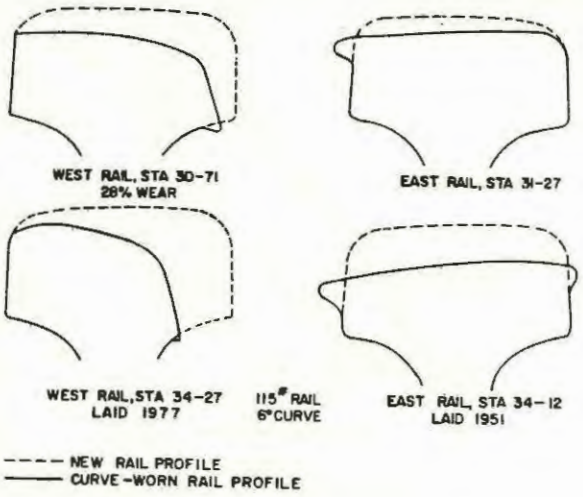
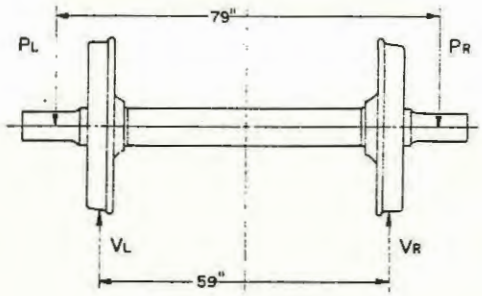


FIG. 2 PROFILE OF CURVE-WORN RAIL



UNDER STATIC CONDITIONS ON LEVEL TRACK EACH WHEEL HAS THE SAME VERTICAL LOAD:

$$V_L = V_R = P_L = P_R$$

UNDER DYNAMIC CONDITIONS ON SUPERELEVATED TRACK LOAD INCREASES ON ONE WHEEL AND DECREASES ON THE OTHER:

$$V_L > V_R$$

FIG. 3 VERTICAL FORCES ACTING ON WHEEL-AXLE SET.

Item No. 4 (Table 1) 2" Dip at Low Rail. This represents the condition of one wheel on the low rail dropping into a 2" dip resulting in a wheel unloading on the high, outside rail.

Item No. 5 (Table 1) Friction Forces of Truck Friction Mechanism. This value represents the effect of the Coulomb friction damping adding vertical forces on the compression stroke and subtracting vertical forces on the extension stroke of the truck spring groups. The 8% range, i.e., 16% to 24%, is a normal variation that can be expected within the tolerance limits of the truck components interfacing each other.

Item No. 6 (Table 1) Hydraulic Snubber Forces. As in the case of Coulomb damping, hydraulic dampers add vertical forces during the compression stroke of the load springs. (This applies to loaded cars only in case of the HS-7-100 for subject cars.) The addition of vertical forces during the compression stroke on one side of the truck causes a reduction of vertical forces on the other side of the truck, i.e., wheel unloading. The hydraulic snubber used on the cars investigated is valved and installed so that it does not subtract vertical forces during the extension stroke. The range of 16% to 33% is based on test results, in which case the higher percentage represents high closing velocities not normally experienced in service. The 16% rating is considered valid based on the 28,000 lbs. to 30,000 lbs. blow-off force of the hydraulic snubber.

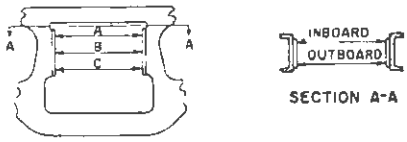
Items No. 7 & 8 (Table 1) Critical Speed of Car Roll and Longitudinal Train Action Forces. It is known that these factors can have an effect on wheel unloading and derailment tendency. For example, most of the derailments occurred with loaded cars operating near the critical speed for car roll on curves. However, these factors were not included in the investigation and their relative contribution to the problem are unknown.

TRUCK INVESTIGATION

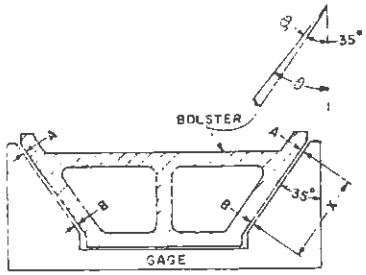
In the experience of the L&N and SCL, all of the potential derailment factors were essentially the same on all 100-Ton coal hopper cars except for the abnormality in operation observed on some trucks manufactured by National, which made the friction forces of the friction mechanism a suspect. Therefore, the main thrust of the investigation was placed on investigating the difference of performance characteristics between trucks having ASF and National side frames and bolsters.

Truck Measurements

The ASF and National trucks were disassembled at the Technical Center and detailed measurements of bolster ends and side frame columns were taken at locations shown in Figure 4. Dimensions were, for the most part, within AAR tolerances and there appeared to be no significant difference between the trucks. However, bolster wedge pocket angles were near the minimum tolerance (34 1/2°) on National bolsters and near the maximum (35½°) for ASF. Also, side frame column openings tended to be narrower at the bottom on National side frames and wider at the bottom on ASF frames.



SIDE FRAME COLUMN SPACING
TOE-IN = A-C



BOLSTER WEDGE POCKET ANGLE
$\theta = 35° + \delta$, where $\tan \delta = \frac{B-A}{X}$

FIG. 4 TRUCK MEASUREMENT LOCATIONS

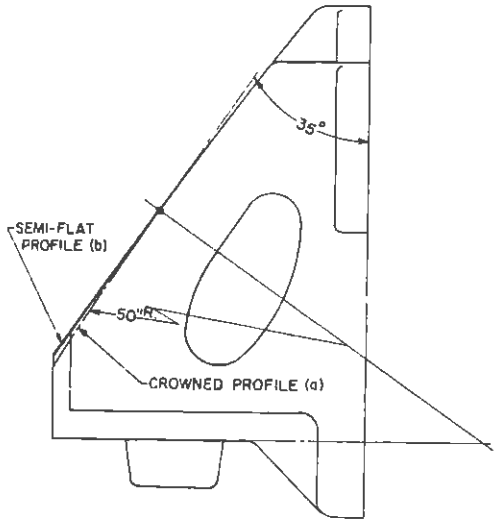


FIG. 5 BARBER FRICTION WEDGE DESIGNS
a) ORIGINAL 50" RADIUS CROWN
b) NEW SEMI-FLAT DESIGN INTRODUCED JAN., 1980.

The Barber friction wedges, which are furnished by Standard Car, were also measured and the angle between vertical and sloped surfaces was between 35 1/2° and 36 1/2° compared to 35° normal. These wedges were found to be of two designs - the original design with a full 50" radius crown on the rear sloped surface and a new design with the bottom half of the sloped surface being flat as shown in Figure 5. This new semi-flat design was introduced by Standard Car in January, 1980 to alleviate non-related problems experienced with some bolsters and wedges in other service. The two designs had been randomly mixed during assembly of the trucks. Neither the railroad nor the truck manufacturers had been previously aware of the design change. A summary of the measurements is given in Table 2.

Laboratory Static Tests

Static vertical calibration tests were conducted on as received ASF and National trucks in a 1,200,000 lbs. universal static test machine. Increasing load was applied to a level of 240,000 lbs. which is theoretically beyond solid capacity of the spring groups. Force-travel closure and release curves were automatically plotted during the tests. All tests were repeated at least three times to ensure repeatability of results.

The National truck of Car #550052 exhibited a "stick-slip" characteristic during these tests. The friction wedges would stick to the column wear plates or hang-up while the force built up; then they would suddenly break loose with a popping noise, moving down slightly with the bolster; then they would stick again, break loose, etc. The ASF truck of Car #550267, in contrast, had a relatively smooth closure with little sticking. A stick-slip characteristic is common for friction damping systems being tested at low machine speed (static). However, it was much more pronounced in the National truck, resulting in approximately 50% more friction damping than the nominal design value of 27,000 lbs. as theoretically estimated by Standard Car. Damping force in the ASF truck was close to the nominal value. Force-travel curves shown in Figure 6 illustrate the difference in truck characteristics. Note that the stick-slip action has the appearance of chatter in the force-travel curve.

TABLE 2

SUMMARY OF MEASUREMENTS

CAR NO.	MFG.	DERAILED (YES/NO)	SIDE FRAME SERIAL NO.	SIDE FRAME TOE-IN (+) TOE-OUT (-) IN INCHES	BOLSTER SERIAL NO.	BOLSTER ANGLE IN DEGREES BR END.	BOLSTER ANGLE IN DEGREES OPPOSIDE END	FRICTION WEDGE WEDGE IDENT.	FRICTION WEDGE DESIGN	FRICTION WEDGE ANGLE IN DEGREES
550052	National	No	3837	-.015	4422	34.6	35.3	N-1	Crown	36.0
								N-2	Crown	36.0
			3895	+.062		34.6	35.3	N-5	Crown	35.5
								N-6	Flat	35.8
550267	ASF	No	7189	-.037	3520	35.2	35.2	A-1	Crown	35.5
								A-2	Crown	35.5
			7258	-.049		35.2	35.2	A-3	Flat	36.2
								A-4	Flat	36.0
550003	National	Yes	3455	+.078	4149	34.7	35.1	NE	Flat	36.0
								SE	Flat	36.0
			3425	+.094		34.7	35.1	NW	Crown	35.5
								SW	Crown	35.5
550945	National	Yes	4779	+.047	5451	34.6	35.1	SE	Flat	36.0
								SW	Flat	36.0
			4831	+.022		34.6	35.1	NE	Flat	35.7
								NW	Flat	36.0

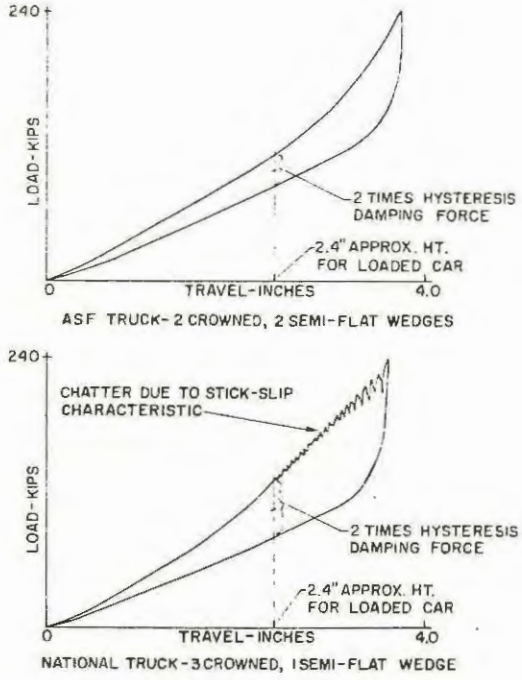


FIG. 6 STATIC CALIBRATION TESTS-TRUCK CHARACTERISTICS

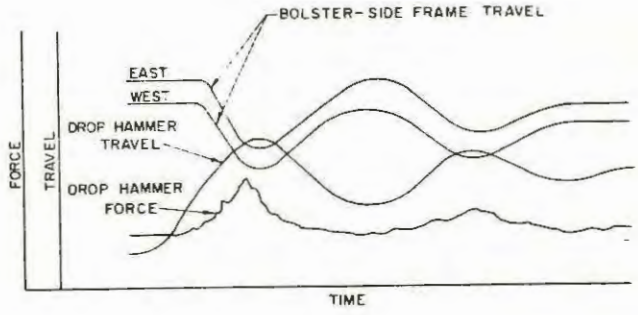


FIG. 7 DYNAMIC CALIBRATION TESTS-SAMPLE OSCILLOGRAPH RECORD

The stick-slip and higher damping characteristic was the only apparent difference found between the ASF and National trucks. Such a condition can result in a wheel load increase on one side of the car and a decrease on the other side while traveling on superelevated curved track, over low rail joints, etc. Therefore, it was considered probable that the stick-slip characteristic could cause increased derailment tendency. Accordingly, additional tests were conducted to determine the cause of the stick-slip condition, and hopefully, a means of minimizing it. ASF components, i.e., side frames or bolster, were selectively and individually substituted in the National truck to determine the effect of dimensional variations in these parts. Different friction wedge combinations were used in the truck to determine the effect of original (full crown) vs. new (semi-flat) designs. Also, other factors were explored including effect of having single side springs, wedge, lubrication, etc.

A sample of static test results illustrating some side frame-bolster wedge combinations tested is given in Table 3. An analysis of the test data and the truck measurements showed that stick-slip and high damping force occur when the following dimensional conditions are present in the side frame-bolster-friction wedge interface:

a. Friction wedges of new design (bottom portion flat) and/or with larger than nominal wedge angle.
b. Bolster wedge pocket angle smaller than nominal (to minimum tolerance).
c. Side frame columns narrower at bottom than at top (toward minus tolerance).

The presence of all three conditions made the stick-slip and high damping more pronounced. Opposite dimensional conditions tend to alleviate the stick-slip and high damping tendency.

Laboratory Dynamic Tests

Dynamic vertical calibration tests were conducted on ASF and National trucks in a 27,000 lb. drop hammer test machine. Force vs. time and travel vs. time traces were generated by an oscillograph during the tests. An example of typical oscillograph records is shown in Figure 7.

TABLE 3

SAMPLE OF STATIC TEST RESULTS (NOT DERAILED TRUCKS). EFFECTS OF TOE-IN VS TOE-OUT; WEDGES FLAT VS CROWNED, BOLSTER POCKET ANGLE PLUS VS MINUS.

CAR Nº	TRUCK MFG'R	S. F. TOE IN+ TOE OUT-	BOLSTER POCK. ∠	WEDGE DESIGN	LOG SHEET	CYCLE	HYST. DAMPING FORCE (KIPS) AT 2.4" STATIC DEFLECTION 0 20 40 60	CHATTER (KIPS) MAX. PEAK TO PEAK 0 20
550267	ASF	-.086	35.2	2 CROWNED 2 FLAT	56 & 57	1 2 3 4 5		
550052	NAT'L	+.046	34.6 TO 35.3	3 CROWNED 1 FLAT	27 & 28	1 2 3 4 5		
550267	ASF	-.086	35.2	FLAT	58 & 59	1 2 3 4 5		
550052	NAT'L	+.046	34.6 TO 35.3	FLAT	29 & 30	1 2 3 4 5		
550267	ASF	-.086	35.2	CROWNED	60 & 61	1 2 3 4 5		
550052	NAT'L	+.046	34.6 TO 35.3	CROWNED	31 & 32	1 2 3 4 5		
550052	NAT'L	+.046	34.6 TO 35.3	3 CROWNED 1 FLAT	27 & 28	1 2 3 4 5		
550267	ASF	-.086	35.2	3 CROWNED 1 FLAT	23 & 24	1 2 3 4 5		

The National truck provided approximately 25% more damping force than the ASF truck, indicating that the stick-slip effect might still be present dynamically. However, the damping force range was much smaller than in static tests and actual sticking was not evident.

Velocity of the spring group travel was in the range of 30 to 50 in/sec well above that needed to actuate the hydraulic snubbers. The snubbers provided a large amount of damping force, i.e., about 25,000 lbs. to 50,000 lbs. per unit more than the truck friction mechanism. This can result in a 16% to 33% increase in vertical wheel unloading. According to Stucki Co., the dynamic test velocity was too high to accurately estimate damping force in service. Therefore, the actual wheel unloading is probably closer to 16%.

Standard Car Truck Company conducted separate tests of one hydraulic snubber on an MTS machine at velocities of 2 to 9 in/sec which is closer to actual service operating velocities. These tests showed that the snubber acts as a very stiff spring for a time period of 0.05 sec. in the early stage of closure, increasing the spring group stiffness by a substantial amount during this time period. This effect, while limited to 0.05 sec., is significant because this represents about 1/8 of a wheel revolution at 20 mph and would add to derailment potential.

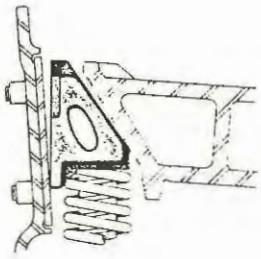


FIG. 8A

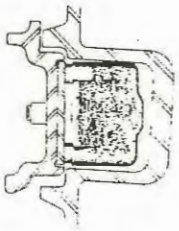


FIG. 8B

Stick-Slip Explained

A detailed analysis of the effect of conditions related to the stick-slip mechanism and high damping is beyond the scope of this paper. However, a basic understanding can be obtained by examining the side frame-bolster-friction wedge interface. First, the combination of adverse friction wedge and wedge pocket angles and/or use of semi-flat wedges results in these two components contacting only at the bottom edge of the wedge sloped surface (Figure 8(a). This can cause a high concentrated force between bolster, wedge and side frame at the bottom of the wedge. Secondly, when side frame columns are narrow at the bottom, the bolster and wedges are forced to occupy a narrower space as they move downward under increased load. In order to do this, the wedges must move down and inward (towards each other) in the wedge pockets against the wedge spring force and against the frictional resistance between bottom edge of the wedge and sloped surface of the wedge pocket. When all of these dimensional conditions exist, forces between friction wedge and side frame column can become so high that the wedges will not move down until the vertical load builds up high enough to overcome the wedge locked-up condition. The moment the frictional resistance is overcome, the wedge suddenly breaks loose and is forced down until the load is relieved. Then the wedge sticks again until the next force build up. Although not duplicated in the laboratory, it is known that trucks take a parallelogram shape in curves. This can further bind the friction wedge in the bolster pocket as shown in Figure 8 (b).

The dimensional variations as discussed above are within permissible tolerance ranges as established by Standard Car Truck Co., and can be expected to produce a 50% variation in friction damping as well as the 16% to 24% variation in wheel unloading as found in the tests.

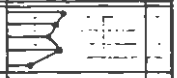
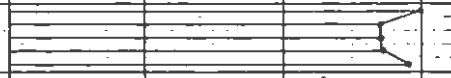
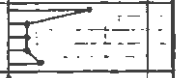
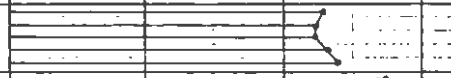
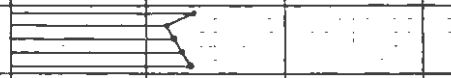
Validation of Initial Tests on Derailed Trucks

Compared to actual service, the static tests have certain limitations. It is recognized that the slow machine speed did not realistically represent service conditions and caused scoring of the vertical surface of the friction wedges, whereas wedges in service normally acquire a smooth, highly polished surface. Also, since the trucks tested were not involved in a derailment, their characteristics might be different from those which derailed. To compensate for these shortcomings, two types of verification tests were conducted. First, dynamic drop hammer tests of the same trucks were conducted to verify the basic findings of the static tests. Second, additional trucks which had derailed, were furnished by the L&N and new unscored friction wedges were obtained from Standard Car Truck Company. Additional static tests which were conducted with these trucks to obtain the performance characteristics of trucks which derailed confirmed expected improvements when specific truck modifications were made.

A summary of the static tests on trucks which derailed is shown in Table 4. Note that the truck from Car No. 550945 which had derailed had all semi-flat wedges as received and the damping force was approximately 60,000 lbs. There was also a large amount of chatter (stick-slip). As soon as these were replaced with fully crowned wedges, damping force dropped into the nominal range of 27,000 lbs. and chatter was eliminated. This extremely large difference was not observed with the truck from Car No. 550003. This was believed to be due to the larger side frame column toe-in at the bottom on the second truck. Therefore, the effect of toe-in was examined next. In Table 4 it can be seen that the damping force is from 44,000 to 59,000 lbs. with

TABLE 4

SAMPLE OF STATIC TEST RESULTS (DERAILED TRUCKS) EFFECTS OF WEDGES FLAT VS. CROWNED, TOE-IN VS. TOE-OUT

CAR No.	TRUCK MFG'R	S. F. TOE IN + TOE OUT -	BOLSTER POCK. #	WEDGE DESIGN	LOG SHEET	CYCLE	HYST. DAMPING FORCE (KIPS) AT 2.4" STATIC DEFLECTION 0 20 40 60	CHATTER (KIPS) MAX. PEAK TO PEAK 0 20
550945	NAT'L	+.070	34.6 TO 35.1	4 FLAT	601 & 602	1 2 3 4 5		
550945	NAT'L	+.070	34.6 TO 35.1	CROWNED	605 & 606	1 2 3 4 5		
550003	NAT'L	+.172	34.7 TO 35.1	2 CROWNED 2 FLAT	607 & 608	1 2 3 4 5		
550003	NAT'L	+.172	34.7 TO 35.1	CROWNED	611 & 612	1 2 3 4 5		
550003	NAT'L	+.172	34.7 TO 35.1	4 FLAT	613 & 614	1 2 3 4 5		
	NAT'L	+.070	34.7 TO 35.1	CROWNED	605 & 606	1 2 3 4 5		
	NAT'L	+.109	34.7 TO 35.1	CROWNED	615 & 616	1 2 3 4 5		
	NAT'L	+.155	34.7 TO 35.1	CROWNED	617 & 618	1 2 3 4 5		
550003	NAT'L	+.172	34.7 TO 35.1	CROWNED	611 & 612	1 2 3 4 5		

0.172" total toe-in for the truck from Car No. 550003. This damping force could be reduced to approximately 27,000 lbs. with total toe-in of 0.109" or less.

TEST RESULTS

Results of these tests confirmed that trucks which derailed have a larger amount of damping force and that this force can be reduced by using fully crowned friction wedges and side frames with minimal column toe-in at the bottom. It was apparent that use of crowned wedges reduces damping force to the nominal level when side frame column toe-in is sufficiently small. However, the force is not reduced sufficiently when large toe-in is present. Finally, a series of tests were conducted with side frame columns modified to provide different amounts of toe-in in order to develop the performance curve shown in Figure 9. In this figure the points shown as ⊕ represent average damping force for three test runs and the solid line is the resultant curve for tests with frames and bolsters from derailed trucks. Additional points have been included for other tests in order to verify the curve and extend it over a wider range (dashed portion). The following damping force levels are designated:

Nominal Force = 27,000 lbs.
Force Limit = 40,000 lbs. This limit was established on the basis that normal tolerance variations can and do permit this force level to exist in service. Such force levels have not caused a problem in the past.
Lowest Force Recorded for Derailed Truck = 54,000 lbs.
Highest Force Recorded for Derailed Truck = 64,000 lbs.

Note that the 40,000 lb. force limit is well below the lowest force recorded for a derailed truck (54,000 lbs.). Therefore, if the 40,000 lb. limit is not exceeded this represents a reasonably safe force level. Based on this, the solid line curve intersects the force limit line at a side frame column toe-in of 0.15" (total per truck), indicating that any smaller amount of toe-in will result in a satisfactory damping force level. Accordingly, the next smallest fractional dimension, i.e., 1/8" or 0.125" was selected as the maximum total truck toe-in (two side frames) which should be permitted. It was further decided as an arbitrary limit that no individual side frame should exceed 3/32" or 0.09375" toe-in even when total truck toe-in is 1/8" or less.

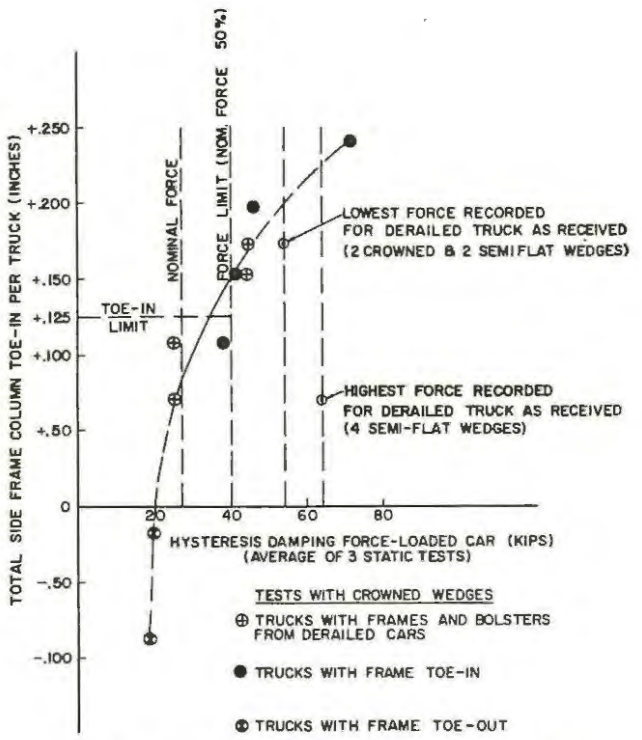


FIG. 9 SIDE FRAME COLUMN TOE-IN VS DAMPING FORCE PER TRUCK

Results Related to Other Factors

A number of other static tests were conducted with the following results:

Tests with outer friction wedge springs only (inner springs removed) to reduce the amount of friction caused no significant reduction in stick-slip or damping force.

Lubrication of the vertical friction wedge and column wear plate surfaces eliminated stick-slip and reduced the damping force to well below the design level. The amount of damping force was considered to be too low for safe operation. Lubrication of the sloped surfaces of friction wedges and bolster wedge pockets reduced stick-slip but had no appreciable effect on damping forces.

Tests simulating a 2" low rail joint showed that 10% vertical wheel unloading can be expected.

Removing the hydraulic snubber from the spring group had no effect on stick-slip and damping force because the machine speed was too slow to activate this velocity-dependent device.

CONCLUSIONS

In order to significantly reduce the "New Car Derailment Syndrome" the following steps should be taken:

(1) The slope of the wedge pocket is critical and should have a manufacturing tolerance of $+1^{0}$, -0^{0} to the designer's specified slope angle.

(2) Fully crowned friction wedges assure contact at the tangency point of the friction wedge pocket midway up the slope and provide additional tolerances of pocket variations. The wedge angle should also be toleranced at $+0^{0}$ -1^{0}.

(3) Excessive side frame column toe-in is critical and an additional tolerance should be applied to current AAR specifications controlling the distance between column wear plates. Based on the tests performed, it is recommended that maximum toe-in be limiated to 3/32" and maximum toe-out be limited to 1/8".

(4) To aid the break-in period of the wedge-bolster interface, apply dry lubricant to the sides and sloping surfaces of the friction wedges when first assembled in the truck taking care not to contaminate the front surface which contacts the column wear plate.

(5) To reduce occurrence of impending wheel climb, utilize improved wheel contours which reduce lateral forces and increase the L/V ratio that new cars can tolerate.

The above recommendations were implemented on 1300 new 100-Ton coal cars received subsequent to the investigation (approximately 1000 cars have modified Heumann profiles similar to CN design). After 9 months of service not a single derailment has occurred on these cars or any of the cars which were subject of this paper and which were also retrofitted to the above criteria except for wheel profiles.

ACKNOWLEDGMENTS

The authors wish to thank the managements of both Midland-Ross Corporation and Family Lines Rail System (Division of CSX) for the permission to release this information in the interest of advancing the art of truck design. Also, the contributions in the study which were furnished by Mr. Robert Bullock of Standard Car Truck Co., Mr. Donald Wiebe of A. Stucki Co. and Mr. Charles Spencer of American Steel Foundries are greatly appreciated.

REFERENCES

1 National Transportation Safety Board - Railroad Accident Report, NTSB-RAR-81-1 dated February 3, 1981, titled Report of Accident on ICG Muldraugh, Ky., on July 26, 1980.

2 AAR Research Dept., Report No. MR-435 Considerations for the Design of Long Cars for Improving Curve Negotiability, November, 1966.

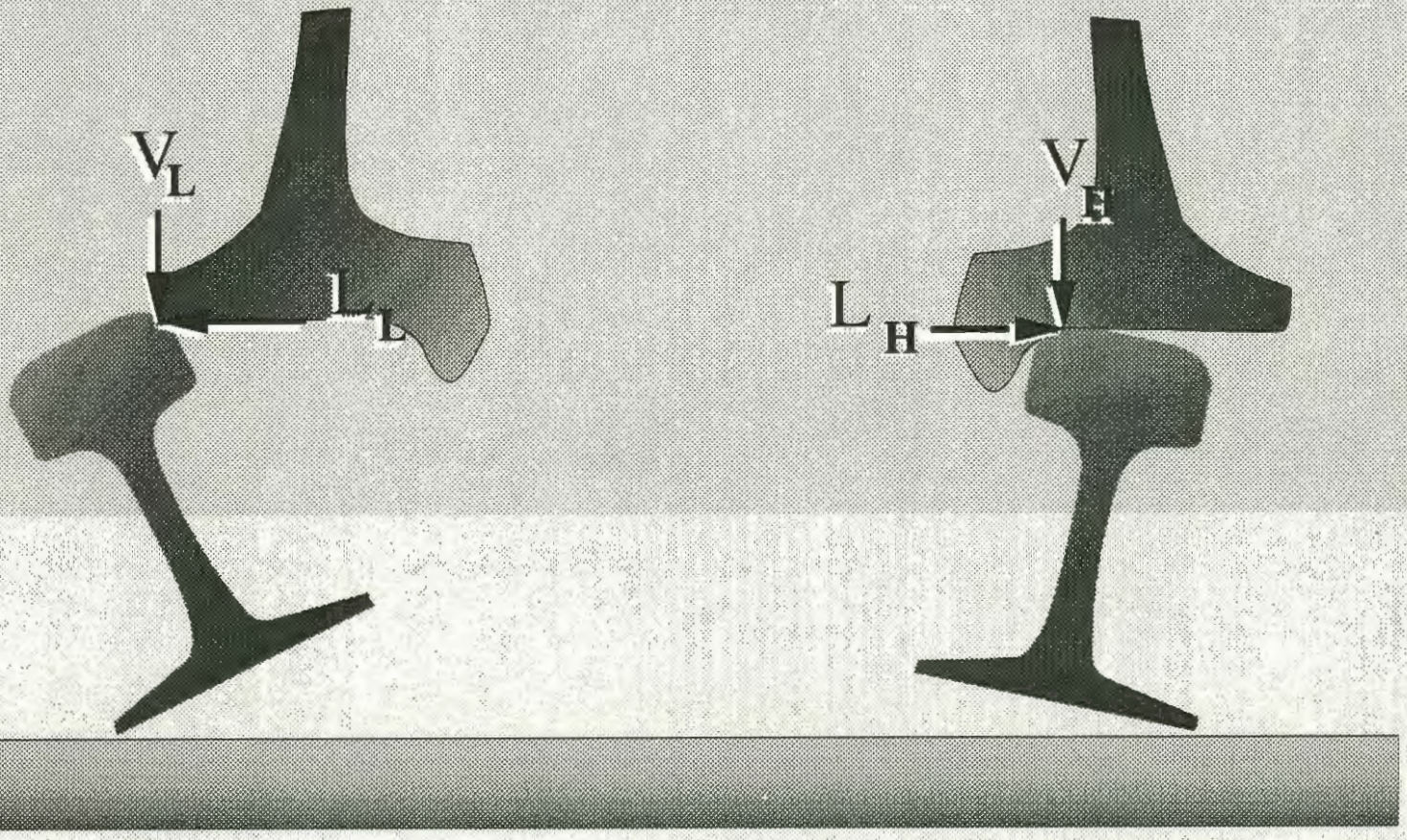
3 Marcotte, P. P., Mathewson, K. J. R., Caldwell W. N., "Improved Wheel Tread Profiles for Heavy Freight Vehicles," ASME 80 RT-3.

LOW RAIL ROLLOVER DERAILMENTS

S. E. MACE

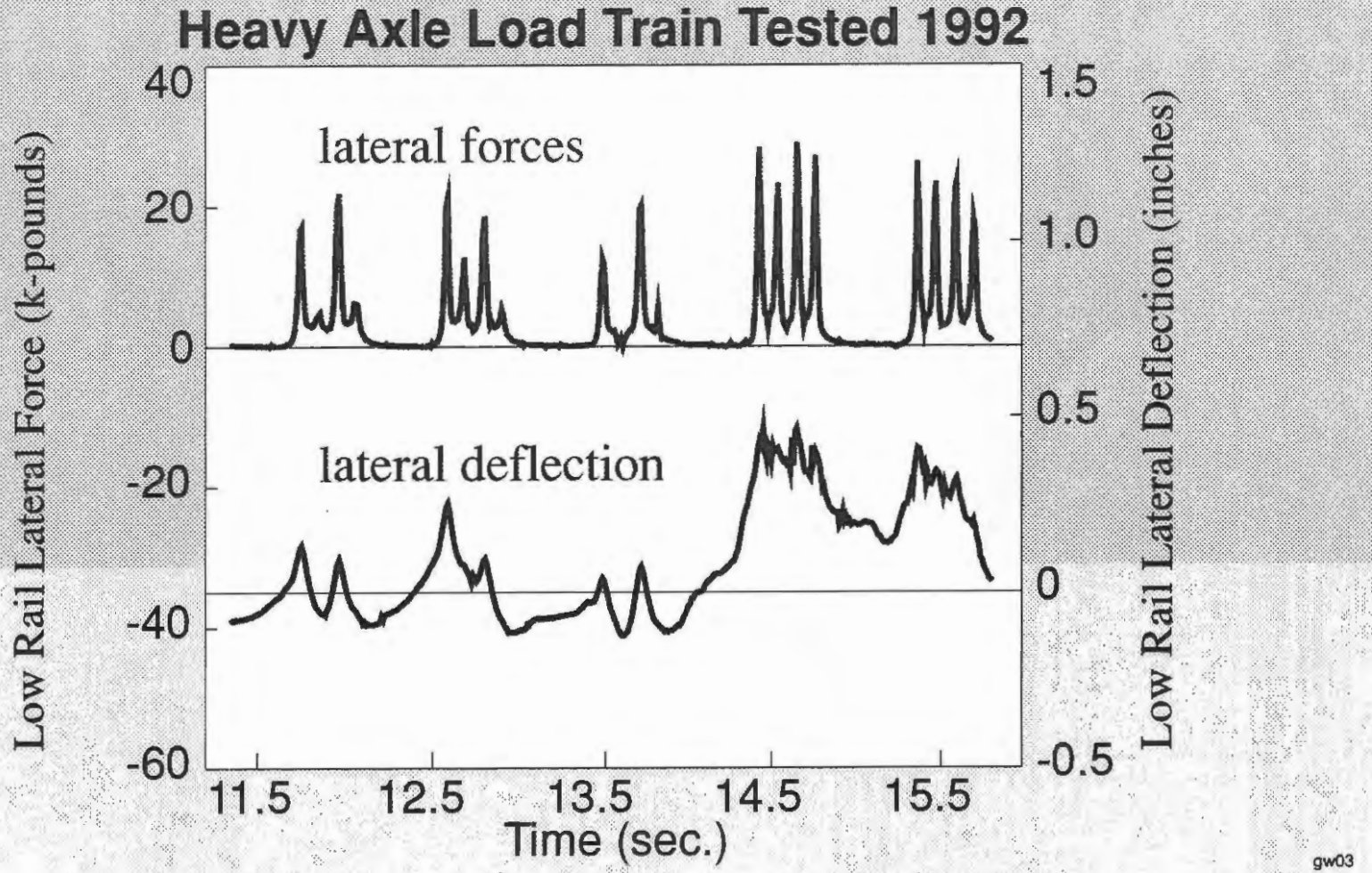
ASSOCIATION OF AMERICAN RAILROADS

Rail Rollover Derailments

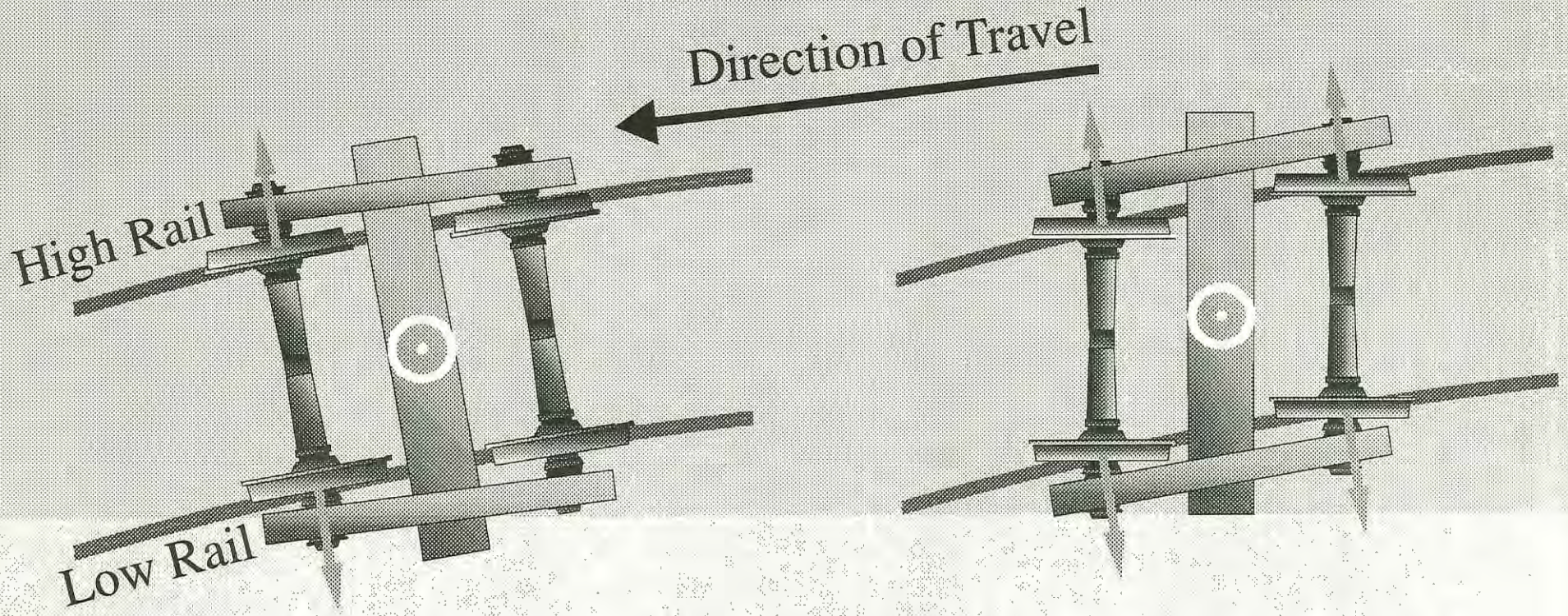


gw01titl

Effect of Adjacent Trucks



Three-Piece Truck Curving

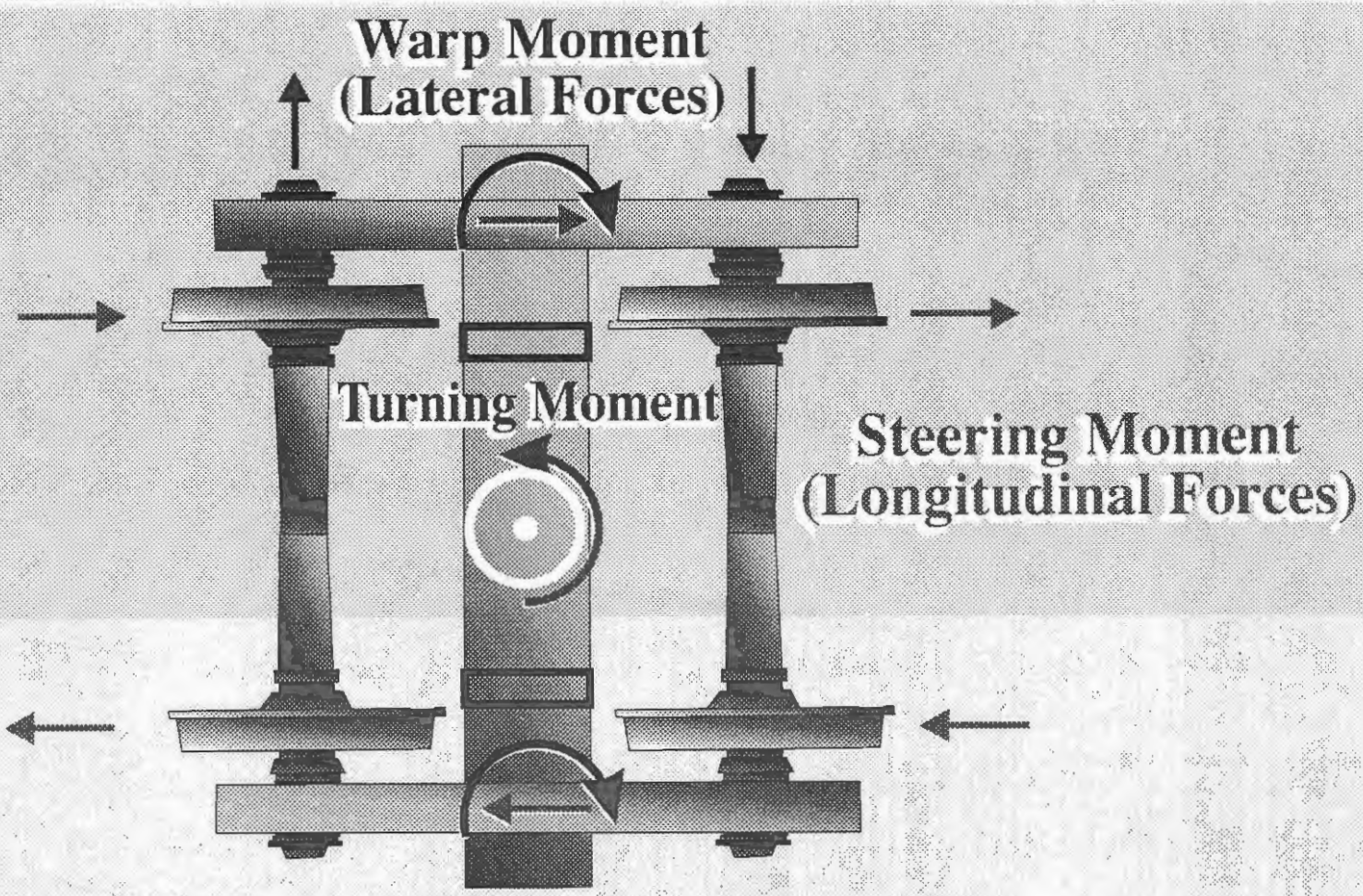


Turned Truck

Warped Truck

GW02STEE

Truck Moments



Warp Moment = Turning Moment + Steering Moment

gw05

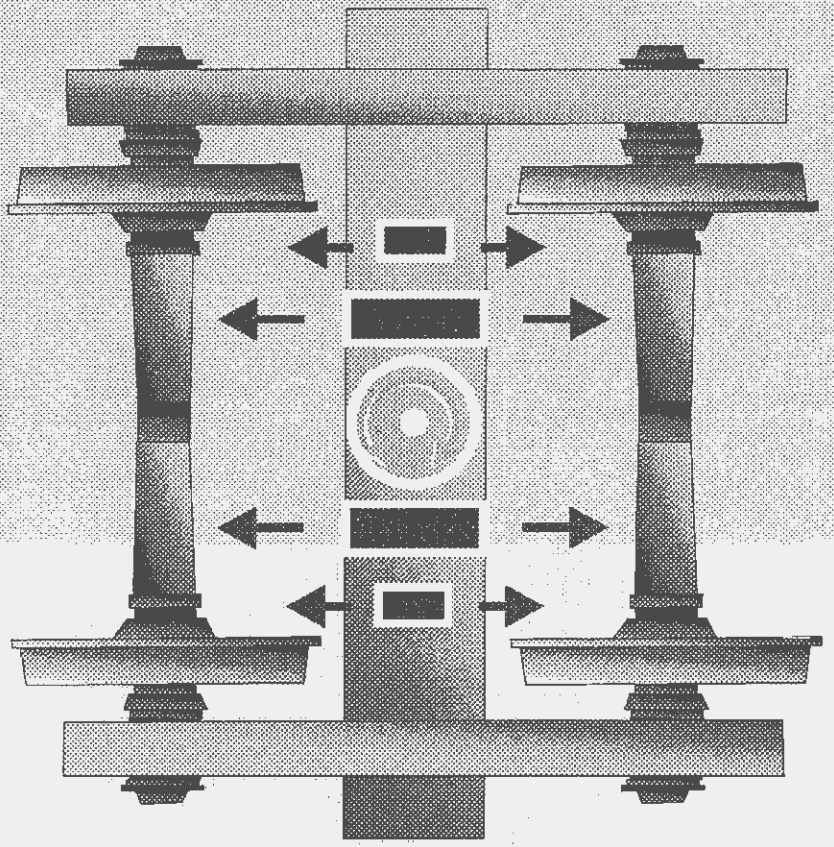
Three-Piece Warp Restraint

- Typical warp restraint: 120,000 - 250,000 in-lb.
- Most of warp restraint from bearing adapter/side frame friction
- Some warp restraint from friction wedges
- Bearing adapter pads may reduce truck warp restraint
- Modifications to snubbers may change warp restraint

gw04.cdr

Warp Moment = Turning Moment + Steering Moment

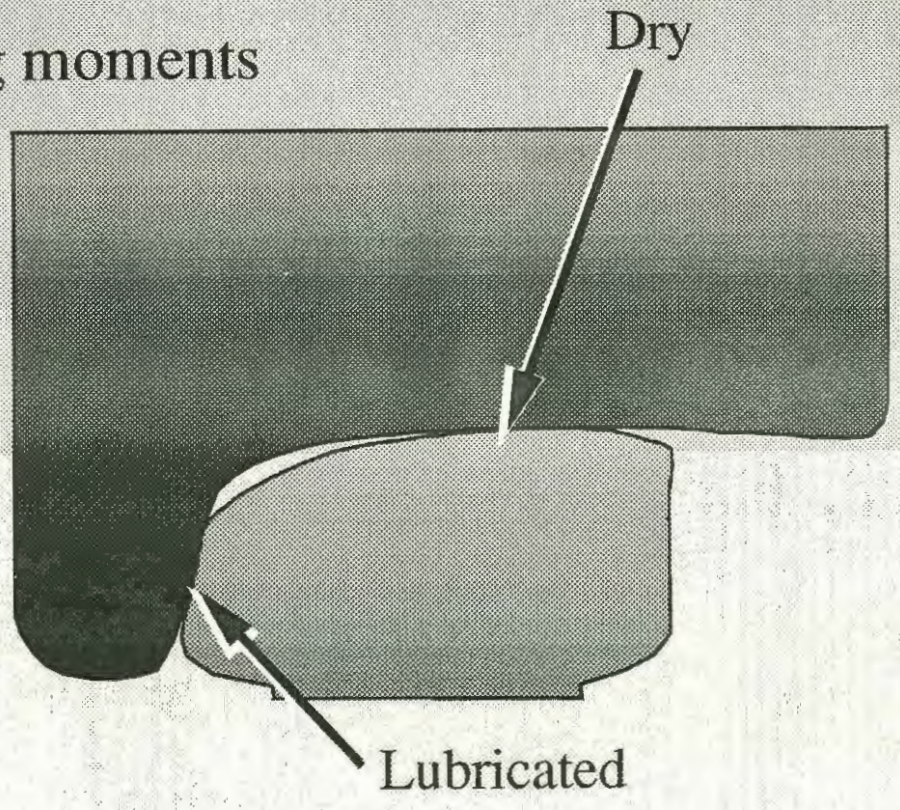
- Turning moment generated by:
  - Center bowl friction
  - CPEP friction
  - Side bearing friction
- Turning moment often exceeds 600,000 inch pounds for dry center bowls



gw06

Warp Moment = Turning Moment + Steering Moment

- Steering moment generated by longitudinal wheel / rail forces
- Factors which affect steering moments
  - Wheel / rail friction
  - Wheel / rail profiles
- Reversed steering moments can exceed 300,000 inch-pounds



gw07

Truck Modifications to Reduce Warping

- Increased truck warp restraint
  - Frame bracing
  - Special low-clearance bearing adapters
- Reduced turning moment
  - Center bowl lubricants / liners
  - Special side bearings
- Increase steering moment
  - Wheel / rail profiles--AAR wheel / rail profile project
  - Track lubrication

gw08.cdr

New Truck Design Specifications for Rail Rollover Derailment Prevention

- Minimum truck warp restraint > 600,000 inch-pounds
- Enhanced steering
- Reduced truck / carbody friction

gw09.cdr

BRAKING ISSUES

F. G. CARLSON

ASSOCIATION OF AMERICAN RAILROADS

BRAKING ISSUES IN TRUCK DESIGN

F.G. Carlson

G.F. Carpenter



Association of American Railroads

B2069.52

CHEEK-DISC BRAKE ASSEMBLY

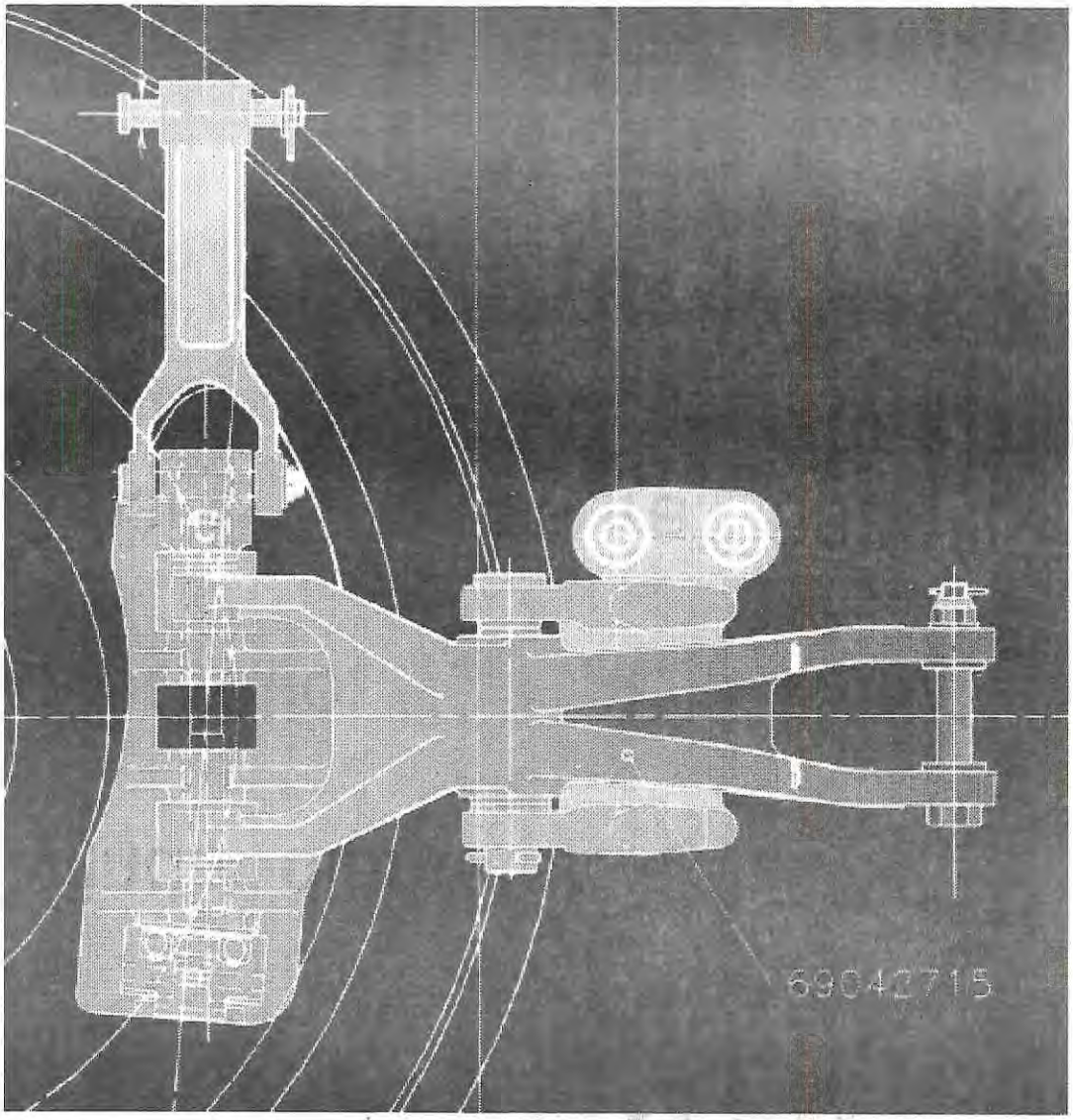
Purchased from SAB-WABCO

- This design is used in European freight service
- 2-wear flat plate wheel - 37.5 in diameter

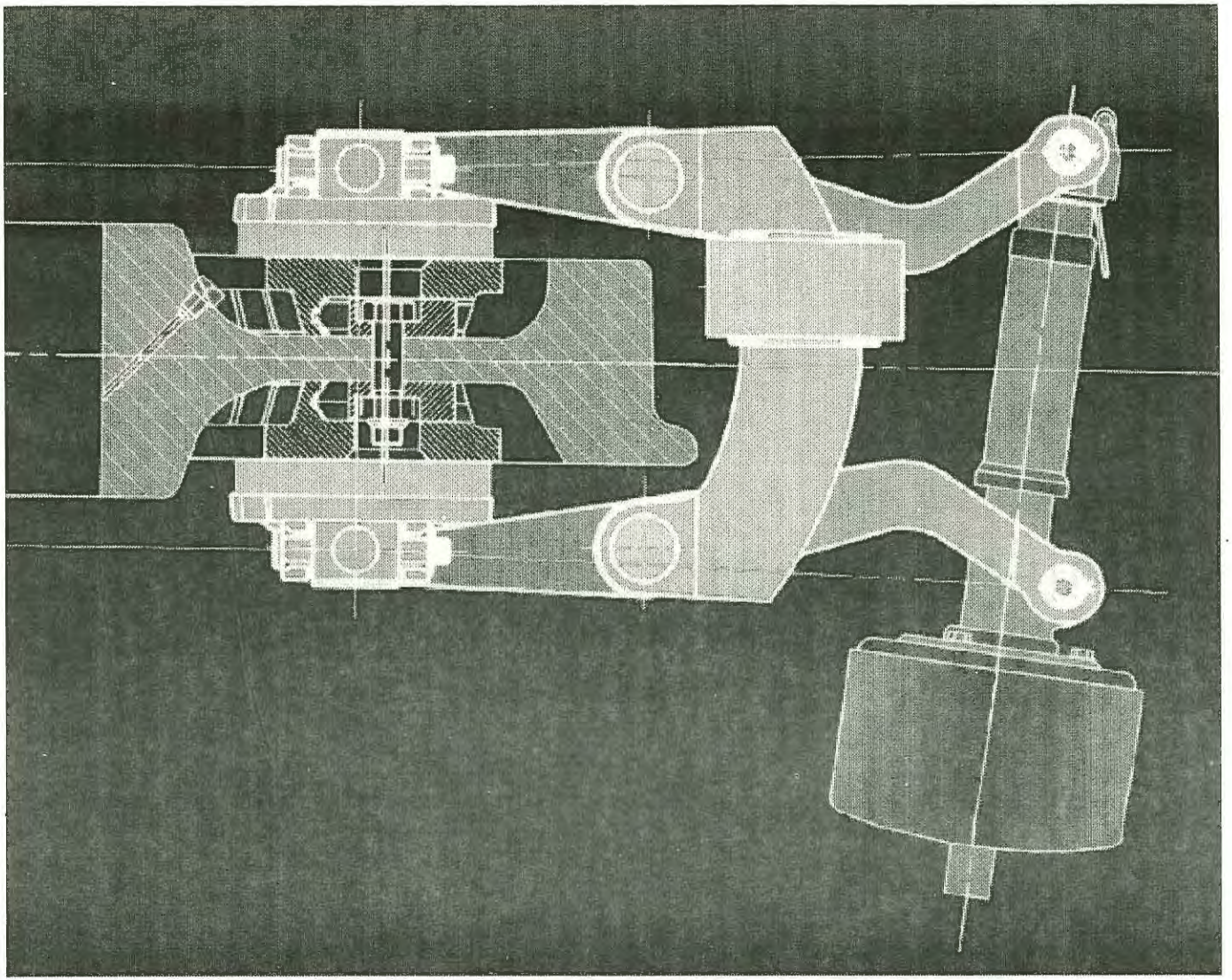
ACTUATOR

- Cylinder diameter: 7.74"
- Lever ratio: 1.46/1
- 93 psi B.C. pressure: 6388 lbs. clamping force





B2069.55



B2069.56

THREE TEST SERIES WERE CONDUCTED

- Constant force grade braking at 20 and 40 m.p.h.
- Stops from 80, 60, 40 and 20 m.p.h.
- Static holding tests



B2069.57

EACH SERIES WAS CONDUCTED IN THREE STAGES

1. Used only the disc to find forces which compared with previous tread brake tests

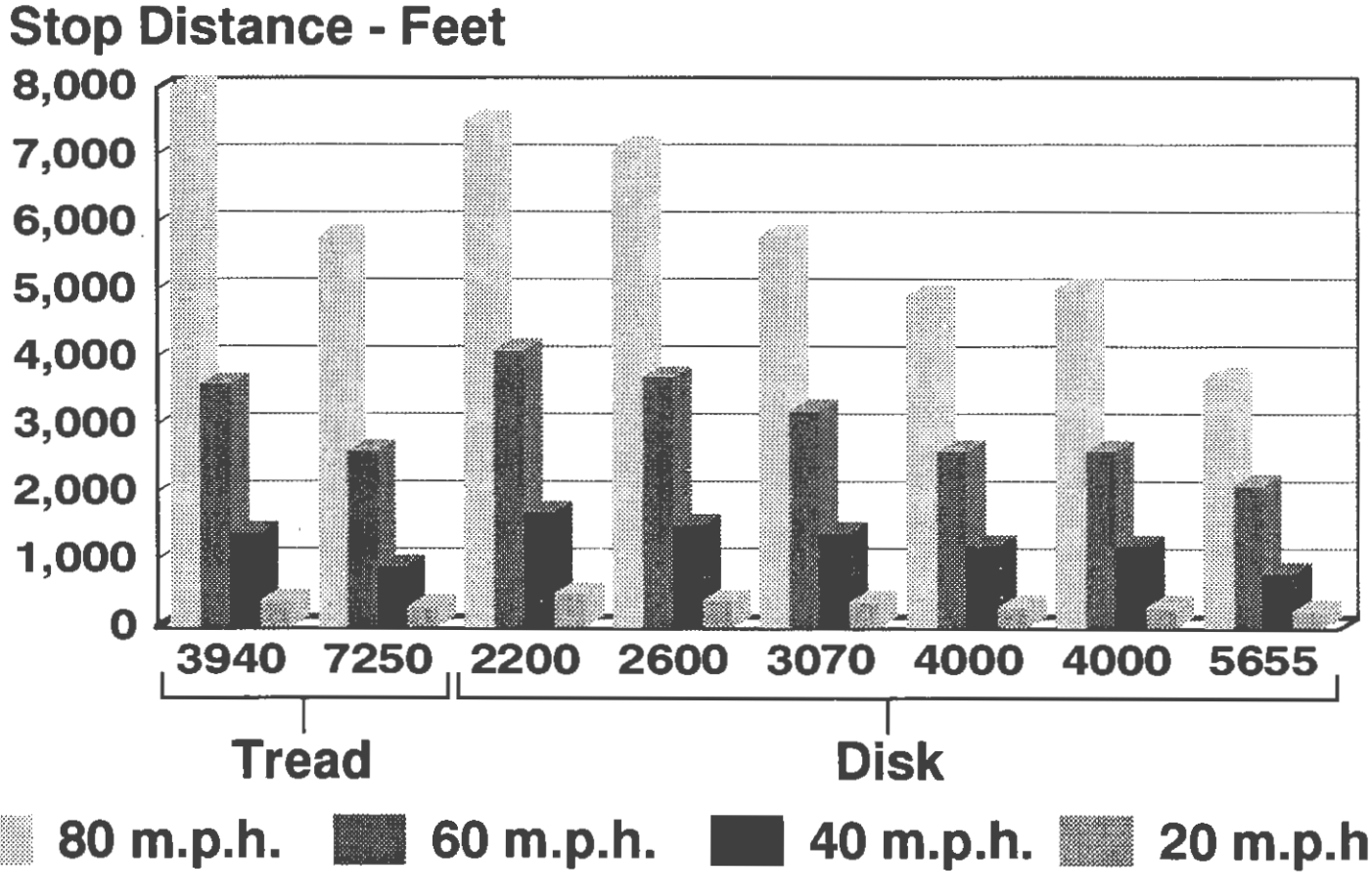
2. Used both disc and tread with disc/tread ratios of 1/1, 2/1, 3/1 and 4/1

3. Used 2/1 ratios while increasing the overall braking force



DISK/TREAD BRAKE

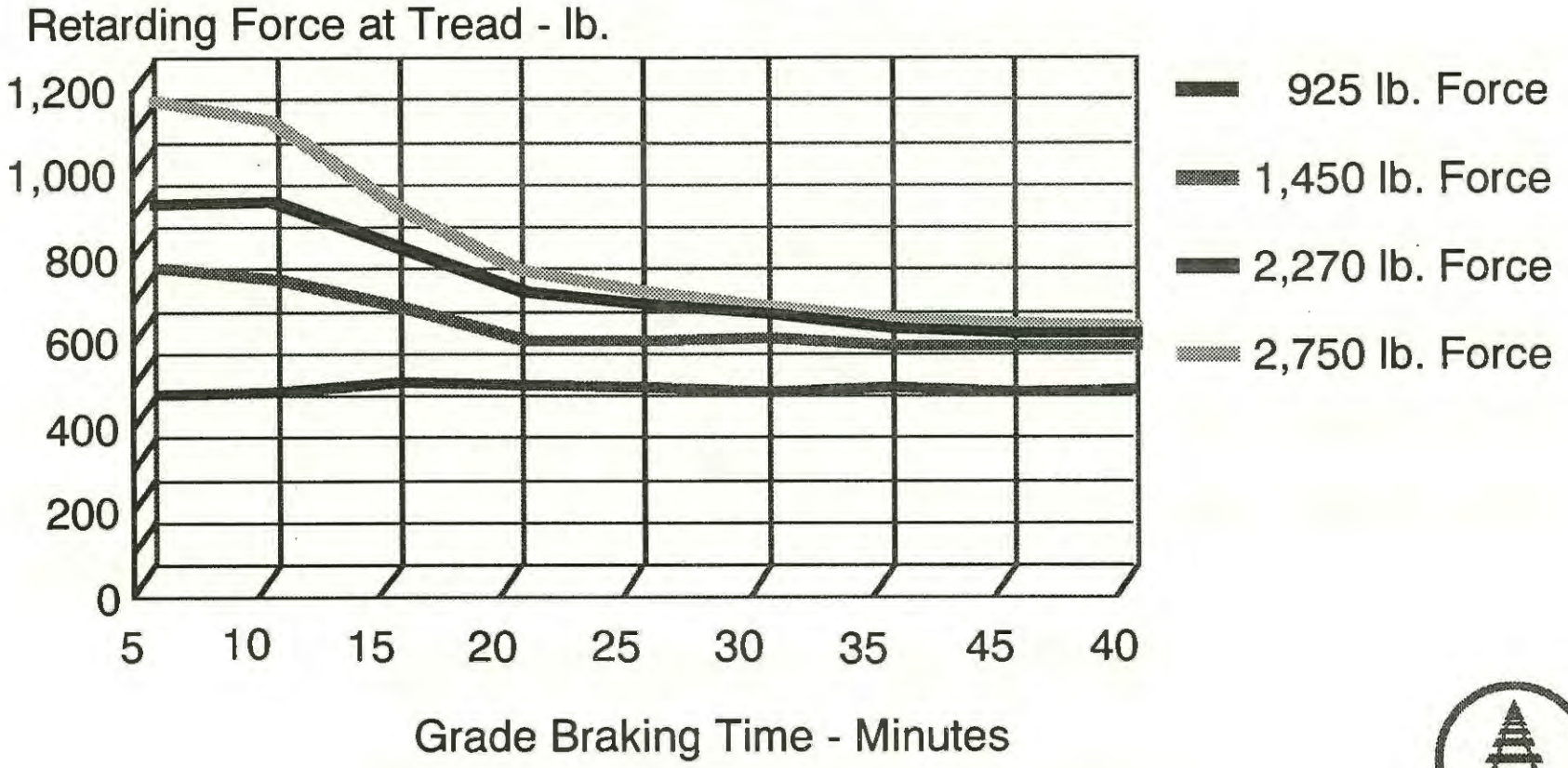
Reference Tread and Disk Only



B2069.62

TREAD BRAKE R-844

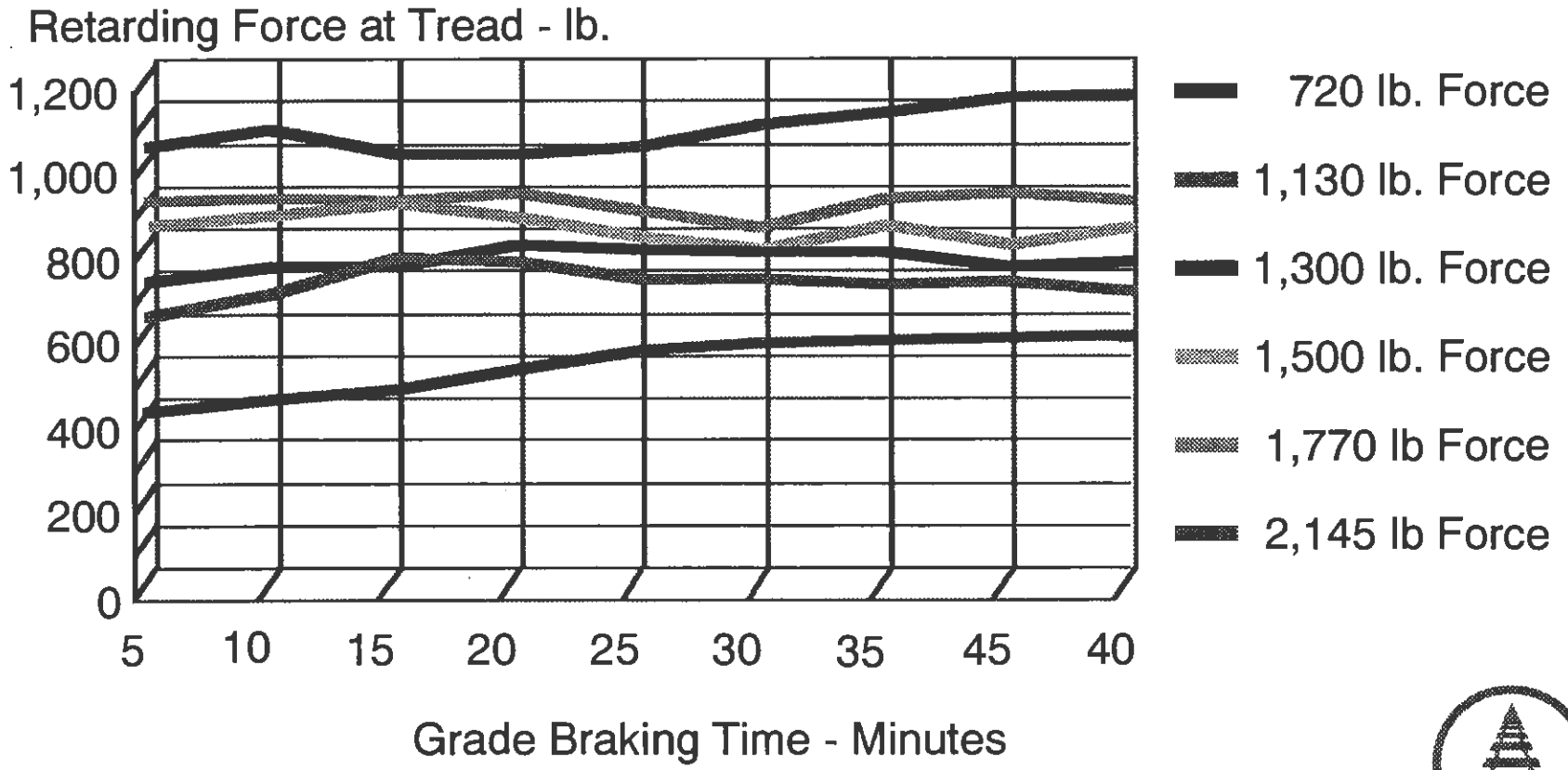
20 m.p.h.



B2069.66

DISK BRAKE ONLY

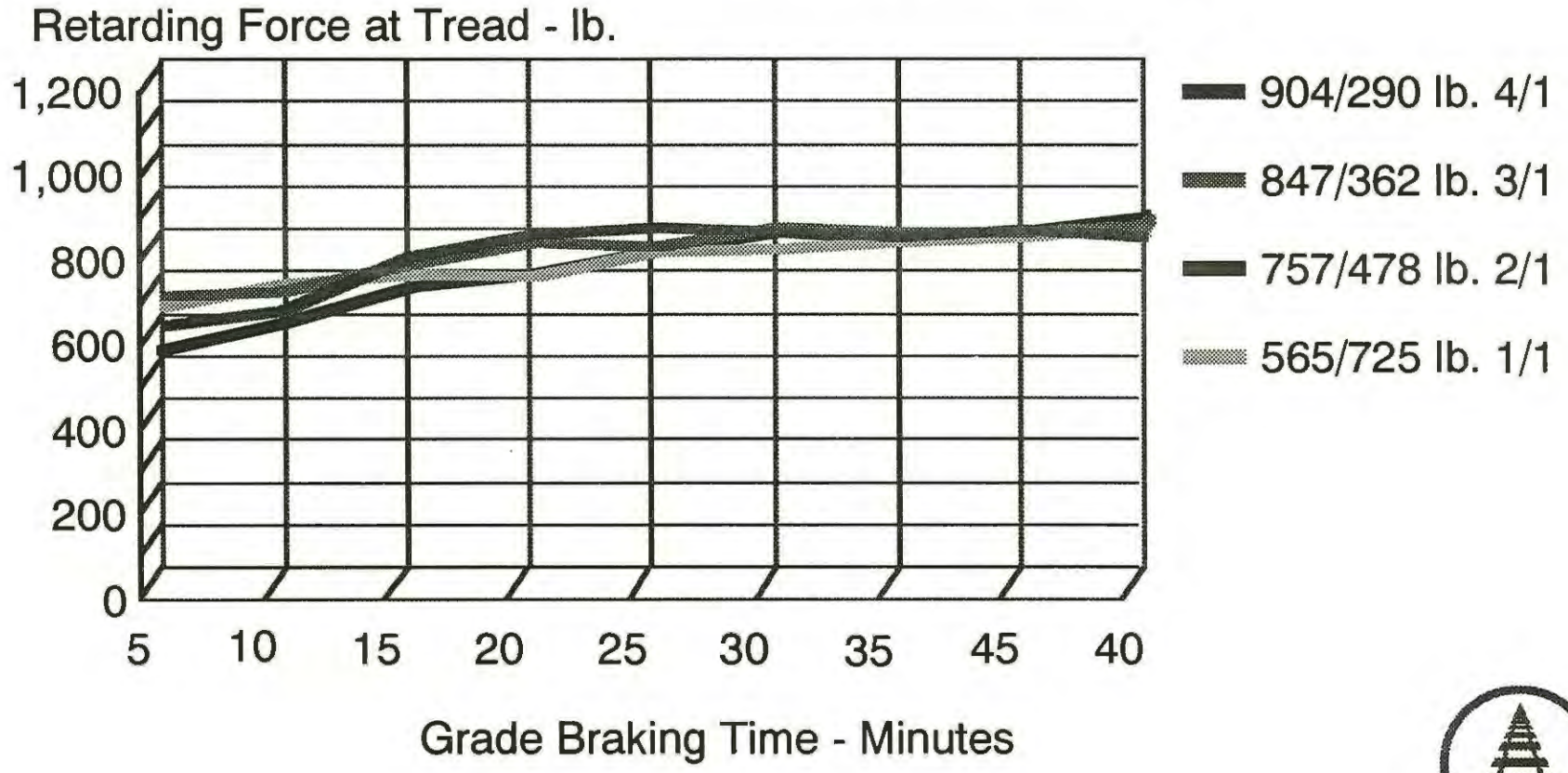
Second Series - 20 m.p.h.



B2069.65

DISK/TREAD BRAKE

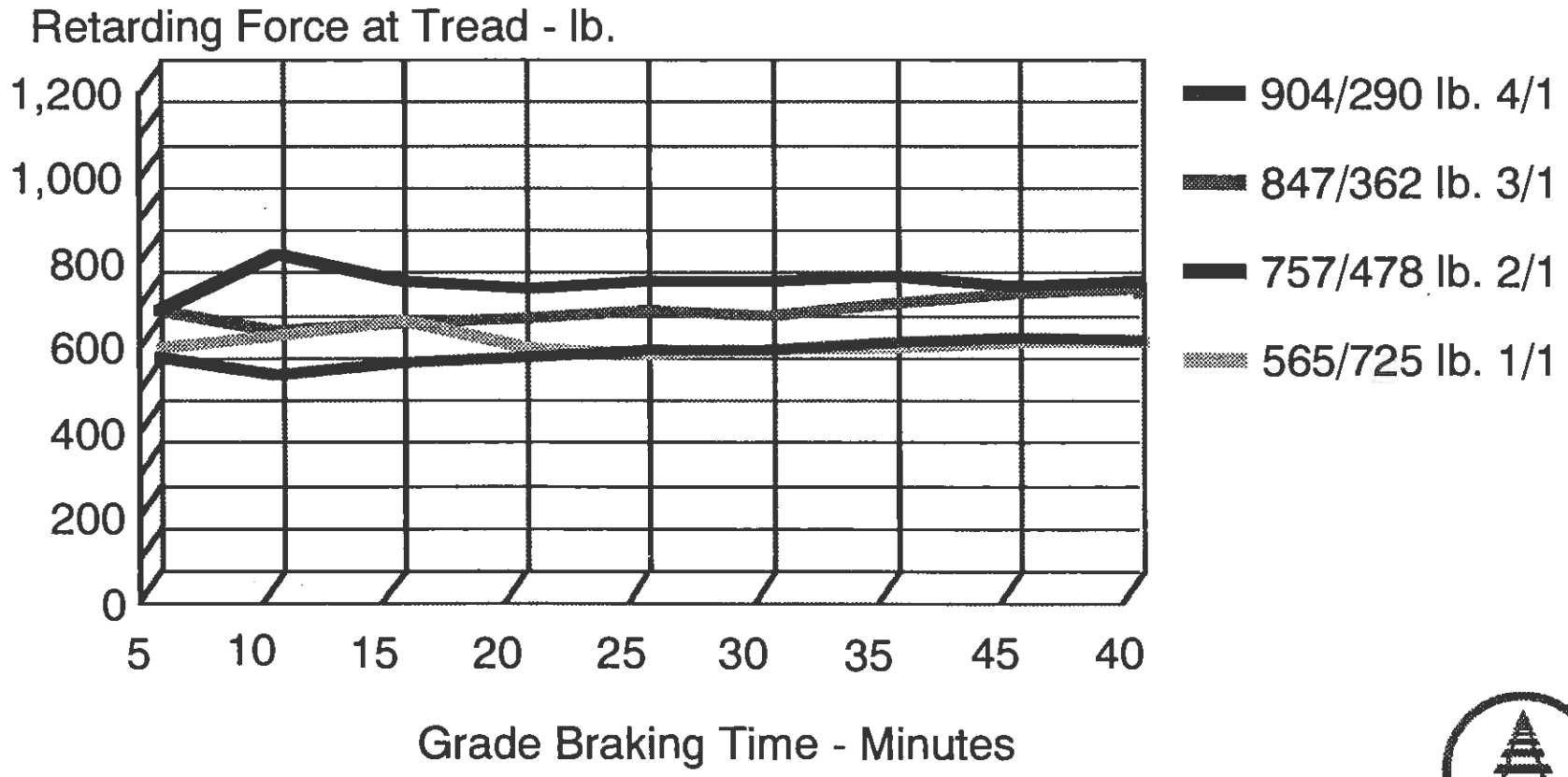
High Force Ratio Series 20 m.p.h.



B2069.63

DISK/TREAD BRAKE

High Force Ratio Series 40 m.p.h.



B2069.64

- 105 loaded 125-ton cars
- Seven 200-ton 6-axle units
- No dynamic or independent brake
- Total train weight = 17,938 tons
- 90 psi B.P. pressure - DB60 or ABDX
- 2 degree constant curvature

% Grade	Init. Speed	B.C.P. to Hold Speed	Horsepower into (Tread/Disc)	B.P. Redctn. to Hold Speed	Emerg. Stop Dist.	Peak H.P. in Emer. (Tread/Disc)
-1.5	40	18.5	19.3/38.8	7.4	--	--
-2.0	40	23.2	24.2/48.4	9.3	--	--
-2.0	50	23.7	29.7/59.4	9.5	2490	80.5/161.1
-3.0	30	34.9	28.9/57.8	14.0	1081	51.9/103.7

B2069.60

- 25 400-ton double stacks
- Seven 200-ton 6-axle units
- No dynamic or independent brake
- Total train weight = 10,000 tons
- 90 psi B.P. pressure - DB60 or ABDX
- Cars loaded to 90% rail load

% Grade	Init. Speed	B.C.P. to Hold Speed	Horsepower into (Tread/Disc)	B.P. Redctn. to Hold Speed	Emerg. Stop Dist.	Peak H.P. in Emer. (Tread/Disc)
-2.62	60	27.5	33.3/66.6	11.0	--	--
-2.54	50	26.8	27.9/55.8	10.7	2330	67.2/134.4
Level	90	F.S.	96.5/193.1	--	8715*	--
Level	90	F.S.	102.0/203.9	--	5733*	--

B2069.61

**Full service stops - conventional and electric brake*

CONCLUSIONS

In the Design of Any New Trucks:

- Be aware of possible brake beam binding problems at the end extension/unit guide interface
- Leave room for a cheek/disc brake installation. We may not need it now, but probably will in the future

SERVICE WORN TRUCK TESTS

C. L. URBAN

ASSOCIATION OF AMERICAN RAILROADS

Subject: Overview of the Truck Maintenance Program Presented at the Technology Transfer Forum on Freight Car Trucks for the 21st Century, September 7-9, 1994, The Levis Faculty Center, University of Illinois at Urbana/Champaign

I would like to thank all those who attended the Forum. The turnout was excellent. I hope that everyone benefited from the Forum. Also, I would like to thank those who provided feed-back on my presentations. Thanks to one and all.

This is a brief summary of the results that were presented at the Truck Forum concentrating on the five service worn car sets of trucks acquired from a member road. The trucks were obtained and studied for a number of reasons; 1) to further the understanding of how trucks wear in revenue service, 2) to provide a comparison of test results with the instrumented truck test; and 3) to help guide the recommendations governing when maintenance should be performed.

TRUCK SELECTION

The trucks were acquired in the fall of 1992. Truck selection was based on wedge rise measured in the field and general condition. Approximately 80 cars were inspected. The best cars/trucks for each wear state were selected from unit coal service. Wedge rise for the selected trucks were 0.5, 0.8, 1.0, 1.3 and 1.7 inches. It is hard to estimate when or if these trucks were ever rebuilt. Given the general condition, only the trucks with 0.5 inches of wedge rise appear to have been rebuilt.

It seems evident that when these trucks required repair, only the worn out friction wedges were replaced. This observation is based on wedge rise and general truck condition.

The following tests and studies were conducted on these trucks:

CHARACTERIZATION STUDIES

Mini Shaker Unit (MSU) characterization studies of vertical, lateral and warp stiffness and damping properties were measured at TTC from the fall of 1992 to the winter of 1994. The results obtained from the vertical, lateral and warp tests showed a direct correlation between wedge rise, damping and stiffness properties. The scatter in the data is more evident for loaded tests. Data scatter is less evident for the empty condition.

INSPECTIONS

Inspections were made on all 5 car sets of trucks. These inspections were made in an attempt to further document the state of wear and general condition of the trucks. The inspections showed a number of wear patterns developing within the bolster pockets and side frame wear plate areas. The wear patterns seen could cause serious performance problems.

TRACK TEST

Three states of truck wear were track tested over the Chapter XI twist and roll, pitch and bounce, and lateral stability (hunting). Limited curving, and yaw and sway testing were conducted on two states of wear. All track test results showed the Chapter XI perturbations were too sever to quantify the performance differences between these trucks, i.e. all trucks showed similar responses. This prompted a modeling study as the next logical step in the investigation.

However, the hunting tests showed measurable differences in performance for the trucks tested. A maximum speed of 45 miles per hour was obtained on the truck with "zero wedge rise" before exceeding the Chapter XI car body RMS for acceleration limits of 0.26. Performance deteriorated with the increase in wedge rise from 0.5, 1.5, and "no wedge" conditions. The maximum speed obtained prior to exceeding the Chapter XI criterion was 30 mph for the "no wedge" condition.

MODELING PREDICTIONS

Truck parameters, such as stiffness and damping, were obtained from the MSU characterization studies and were used in the assembly of NUCARS models to simulate these trucks. Predictions were made for all track tests conducted as well as empty twist and roll performance.

Modeling results showed the following:

Hunting results showed a slight improvement in performance using the AAR-1B profile. These results were obtained using lateral car body accelerometers.

Pitch and bounce simulations obtained with 0.6 inch amplitude perturbed inputs indicated that 0.0-0.4 wedge rise trucks passed. Pass/fail criteria is no suspension bottoming or not less than 10 percent static wheel loads encountered during the simulation. The simulations also showed that trucks with 0.75 inches of wedge rise passed the simulation for loss of travel due to wedge binding. The data indicates that maintenance should be performed by 0.4 inches of wedge rise.

Twist and roll loaded simulations over Chapter XI inputs with 0.75 inches of perturbation amplitudes showed that the trucks equipped with 0.0-0.35 inches of wedge rise passed. Here pass/fail is indicated by wheel loads not less than 10 percent of static or car body roll angles not exceeding 6 degrees peak-to-peak. Empty car performance simulations showed that 0.0-0.1 inches of wedge rise passed.

For the loaded car, maintenance should be performed prior to 0.4 inches of wedge rise. Failure to negotiate the test section in the empty condition indicates a need for a two stage secondary suspension.

Limited **curving** runs were performed and indicated that further simulations are required to form a conclusion. The trucks did not appear to warp significantly based on the lateral wheel/rail forces obtained from the instrumented wheel sets during the track test.

These results were obtained on a 7.5 degree curve with worn 136 pound rail and worn AAR-1B wheel profiles. Dry rail and lubricated tests were conducted. Dry rail conditions were performed with friction levels of 0.4 to 0.5 on the rail head and gage face. Lubricated rail conditions were conducted with friction levels of 0.4 to 0.5 on the rail heads and 0.15 to 0.20 on the gage face.

Yaw and sway modeling simulations have not been performed. Track tests showed similar performance for the two trucks tested up to the point when the tests were terminated due to lateral instability.

FINAL CONCLUSIONS

It is difficult to understand why the car/truck combinations did not pass a limited set of Chapter XI tests. Chapter XI is the new industry standard for desired performance criteria for new car/truck combinations. Possibly if a different car was used with a longer truck base, the performance of these trucks could have been quantified.

From all the tests conducted, it was concluded that wedge rise was not a true indicator of performance. To restate that conclusion a little differently; wedge rise is a good indicator of the general wear condition of the truck and can be used to guide the user in determining when maintenance should be preformed. However, it is not a good indicator of performance for the reasons stated in the presentations. This is primarily due to hidden wear problems seen in the secondary suspension system of trucks.

It is felt by those conducting the tests that the only maintenance performed on these trucks was friction wedge replacement rather than truck restoration. The friction wedge wear indicator is a convenient way of determining when to replace them due to wear. From measurements conducted at CALPO TTX's repair facility in Ontario California, the following relationship between mileage and wear was determined:

1.2 inches of wedge rise per million miles—this wear indicates that roughly 50 percent of the rise is due to wear in the bolster pocket area and 50 percent is in the side frame wear area. This wear rate was obtained on intermodal type of equipment. Wear rates more than likely depend on the type of service and route characteristics.

For the sake of argument, let us assume that the wedge contributes 50 percent of the wedge rise. In the presentation, it was noted that if the friction wedge wear tab had worn away, then roughly 0.4 inches of wedge rise would occur. There would also be an additional 0.4 inches of rise caused by the wear on the bolster slope and side frame wear plates. The total wedge rise would be approximately 0.8 inches. This level of wear is just beyond the manufacturers' limit of 0.75 inches of rise, indicating that the truck is in need of repair rather than just replacing the friction wedge.

It should be a further recommendation from the manufacturers' that, if new wedges are installed and the wedge rise is beyond the recommended values then the truck should be repaired. This would also ensure that the extreme wear patterns seen in these trucks does not develop. All of the necessary wedge rise measurements could be done in the shop while the friction wedges are

being replaced. The railroads should collectively adopt a rule for measuring friction wedge rise during wedge replacement.

Another possible procedure for checking when maintenance should be performed is to measure the difference in wedge rise before and after wedge replacement. A criterion could be developed for the difference in heights measured. Also, during this procedure the wedge pockets can be inspected for abnormal wear. A criterion or gage could also be developed to address pocket condition.

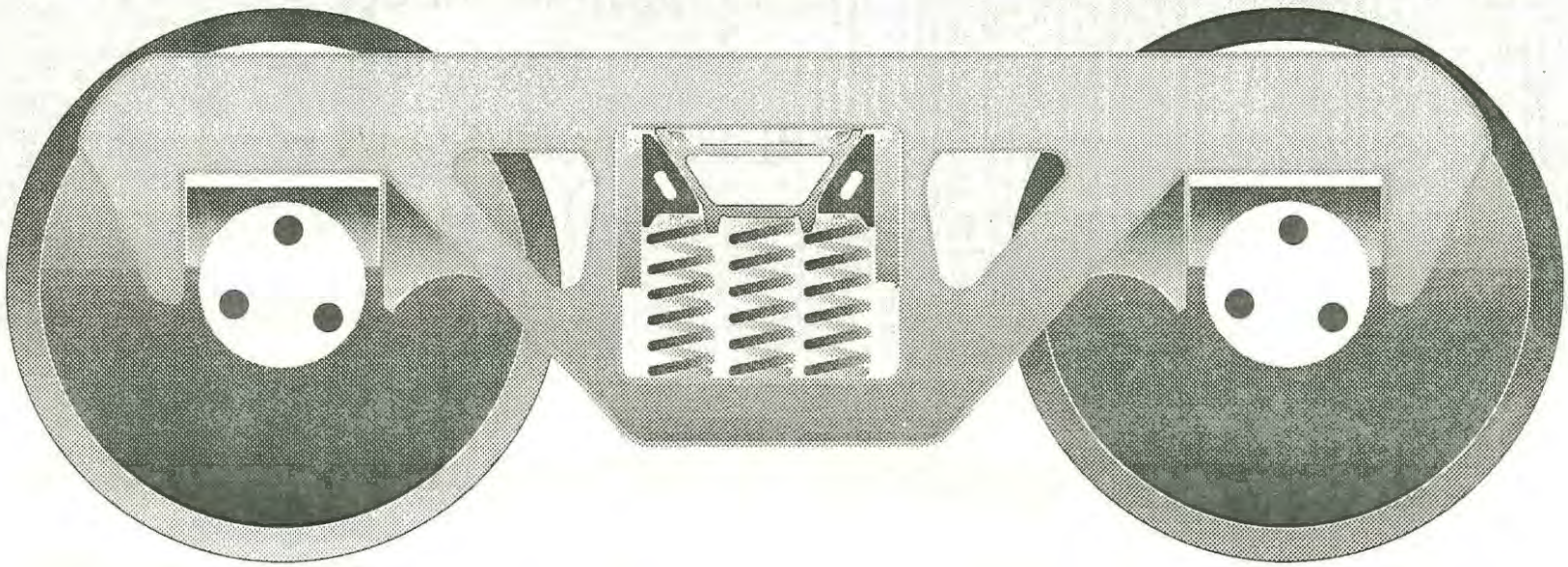
To maximize the investment of using bolster and side frame castings, the wedges should be the softest element in the truck. We can not loose site of the fact that the friction wedge is the main energy dissipation element in the truck. The only way to damp out unwanted oscillations is to dissipate heat via a hysteresis type of connection. If the wedge was thought of as expendable, rather than a permanent item, then the side frames and bolsters would not require repairs as frequently.

As a final note, bolt inspections were made on the trucks after returning to TTC. All the trucks were equipped with 3/4 inch bolts with lock washers to secure the side frame wear plates. All were initially loose and required re-torquing. Some required spot welding to ensure that the bolt did not interfere with the normal operation of the wedge.

Hopefully, this ties together some of the key concerns that were addressed during my presentations. Later this year a report will be completed detailing all the test results. Thank you.

C. L. Urban

Truck Maintenance Guidelines



TMG01

Outline

- Session 1 and 2 -- Current Research
  - Program 2.3.1

    History

    Current Work (Service-Worn Truck Test)
    - Mini Shaker Unit Test
    - Track Test
    - Modeling Predictions
    - Inspections
- Session 3 -- Best Practices
  - Applied Results to Solutions for Truck Related Problems

00outlin.cdr

Truck Maintenance Guidelines

Objective

- Determine Optimum Maintenance Procedure for Conventional Three-Piece Equipment

Purpose

- Reduce Maintenance and Degradation to both Vehicle and Track Structures

Truck Maintenance Guidelines

Method

- Establish Relationship between Mileage and Wear for Critical Suspension Components
- Determine Relationship between Wear and Performance
  - Modeling
  - Testing
- Simulate Wear
  - Bolster Face, Friction Wedge, Side Frame
  - Wear Plates and Gibs

tmg03b.cdr

Truck Maintenance Guidelines

August 1988 - March 1989

- 125-Ton Vertical Truck Investigation (Report 792)

November 1989 - June 1990

- UP Tank Car Derailment Investigation (Report 785)

November 1990 - October 1991

- Designed/Built/Tested Instrumented Truck to Simulate Wear

tmg04a.cdr

Truck Maintenance Guidelines

November 1990 - October 1991

- Designed and Built Instrumented Truck to Simulate Wear
  - Tri-axial Load Cells
  - Control Coil Load Cells
  - Adjustable Gibs and Wear Plates
  - Vertical, Lateral and Warp Displacements
- Track Test–Two Wear States Tested
  - Pitch and Bounce
  - Twist and Roll
  - Curving
- Results were Inconclusive

tmg05.cdr

Truck Maintenance Guidelines

November 1992 - November 1993

- Acquired/Characterized Five Service-Worn Trucks

February 1994 - April 1994

- Track Test on Three Wear States

May 1994 - July 1994

- Modeling Investigation

tmg04b.cdr

Truck Maintenance Guidelines

Service-Worn Truck Test

- Objective
  - Determine why the Instrumented Truck Tests were Inconclusive
- Methodology
  - Characterize a Number of Service-Worn Trucks of Various States of Wear and Compare these Results to those Obtained from the Instrumented Truck
  - Perform Track Test on a Number of Wear States to Establish a Relationship between Wear and Performance

tmg06.cdr

Truck Maintenance Guidelines

Service-Worn Truck Test

- Selected Five Service-Worn Car Sets from a Member Road. Friction Wedge Rises were: 0.5, 0.8, 1.0, 1.3, and 1.7 Inches
- Characterized Trucks on the Mini-Shaker Unit (MSU) for:
  - Vertical
  - Lateral
  - Warp

tmg07a.cdr

Truck Maintenance Guidelines

Service-Worn Truck Test

- Vehicle
  - 100-Ton Open Top Hopper
  - Truck Centers 40’ 6"
  - Roller Side Bearing
  - Lubed Center Plate

tmg08a.cdr

Truck Maintenance Guidelines

Service-Worn Truck Test

- Trucks (Service Worn and Instrumented)
  - 100-Ton Truck
  - 7 - D5 Outers
  - 6 - D5 Inners
  - Double-Side Coils

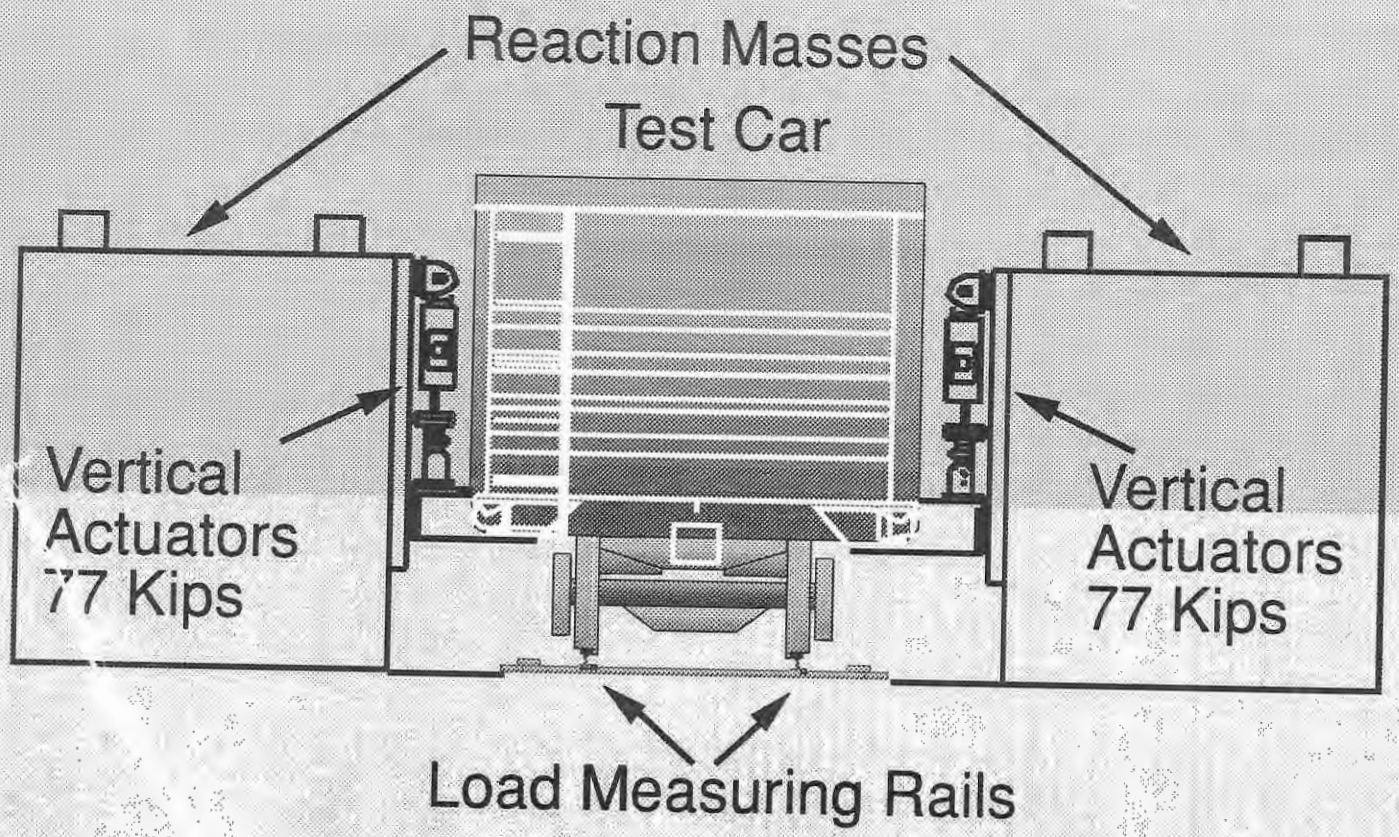
  } Newer Condition

  - Variable Damped
  - All Service-Worn Trucks Manufactured in 1980

tmg08b.cdr

MSU Vertical Test

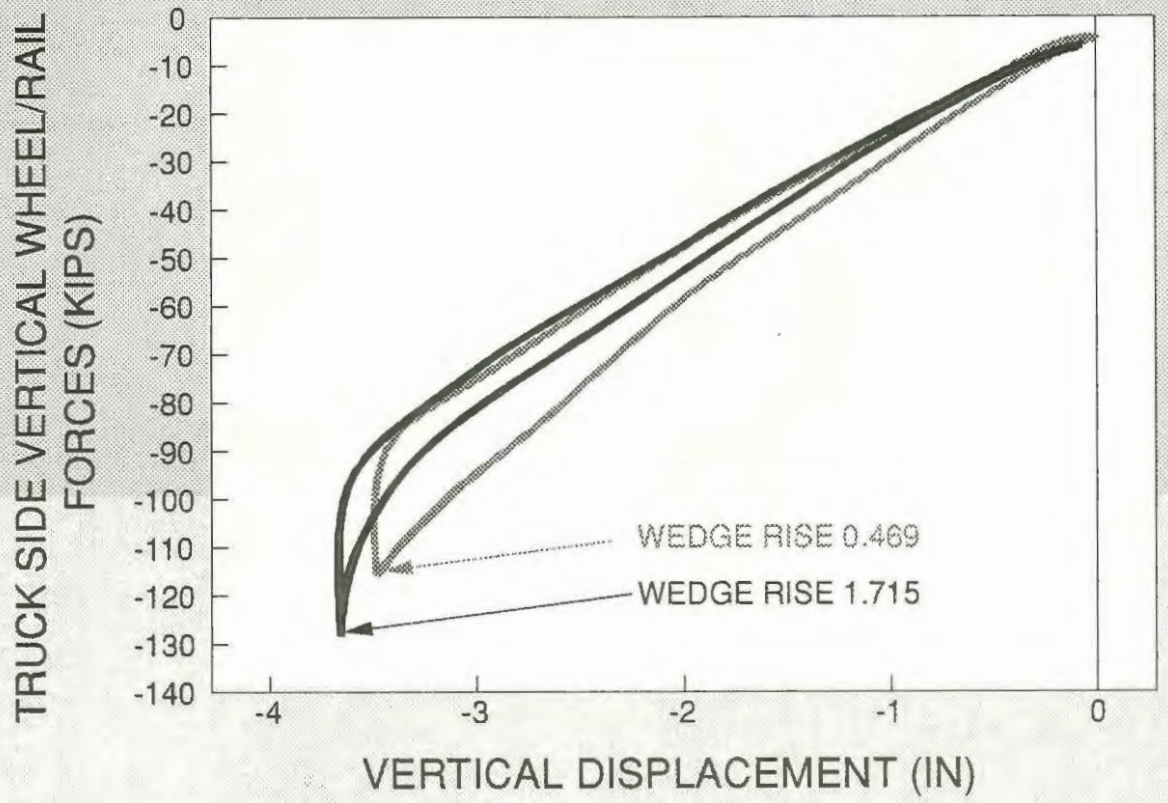
Vertical Suspension Characterization



HMSUVERT/MSU02

MSU Vertical Test

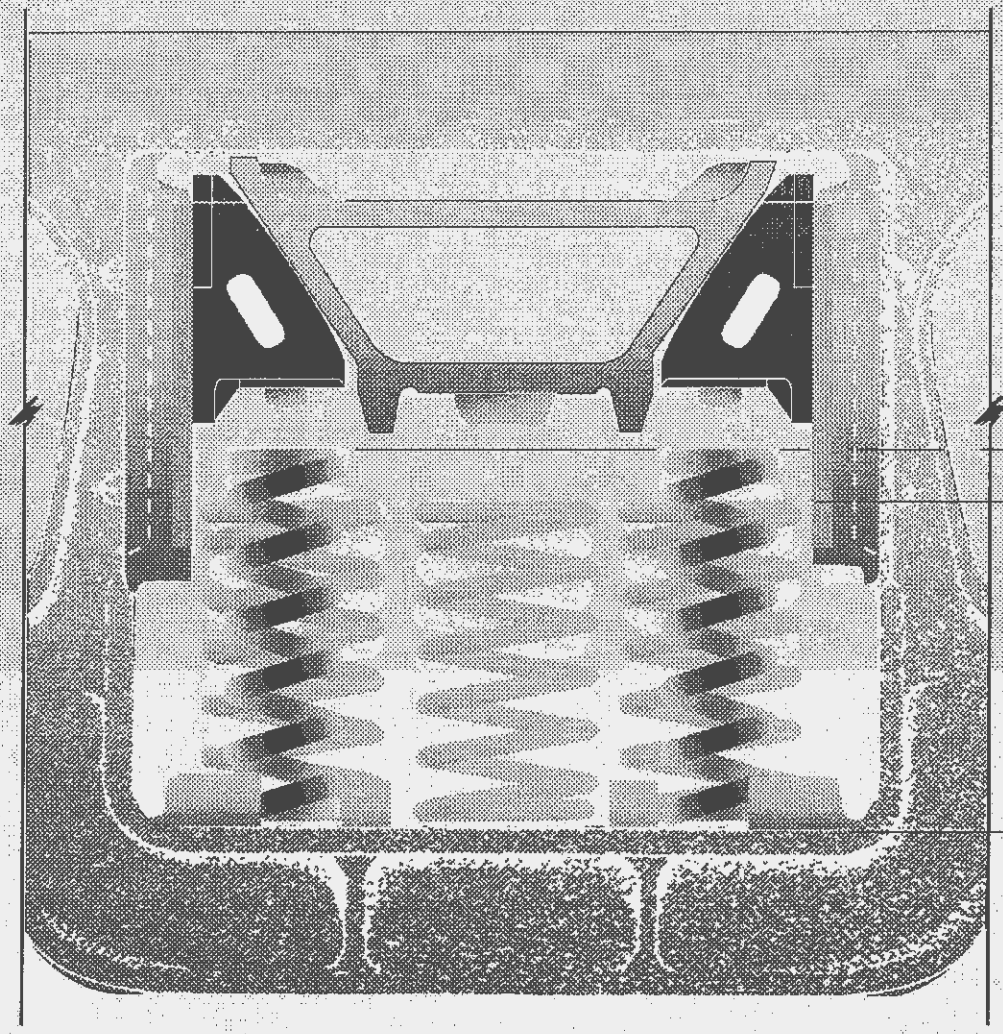
Comparison of Best and Worst Trucks



Note: Trucks Tested with New Coils in all Positions

HCOMPAR1\MSU03

Free Height of Main & Friction Wedge Springs



a - free height of main springs

b - free height of friction wedge springs

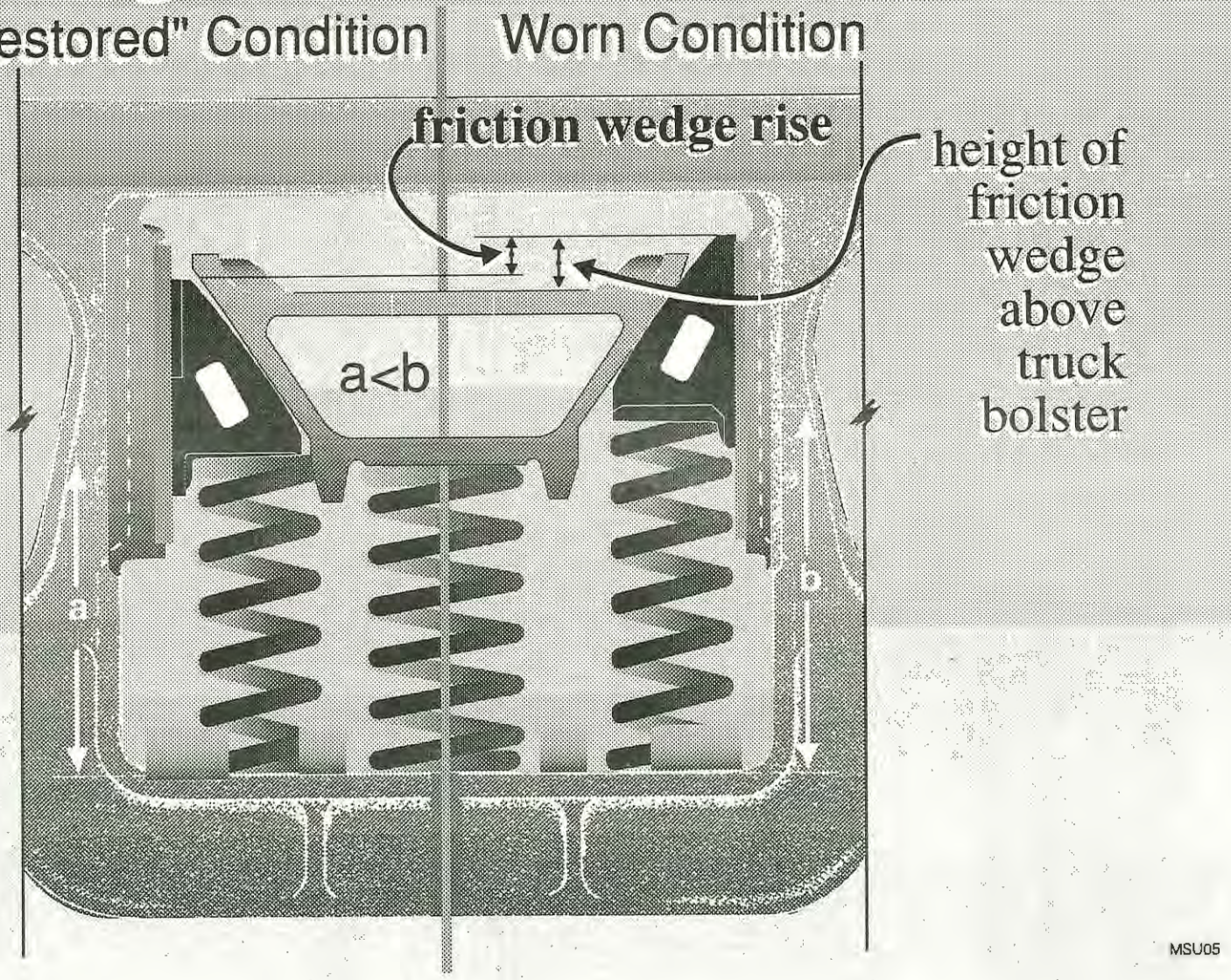
a

b

a < b

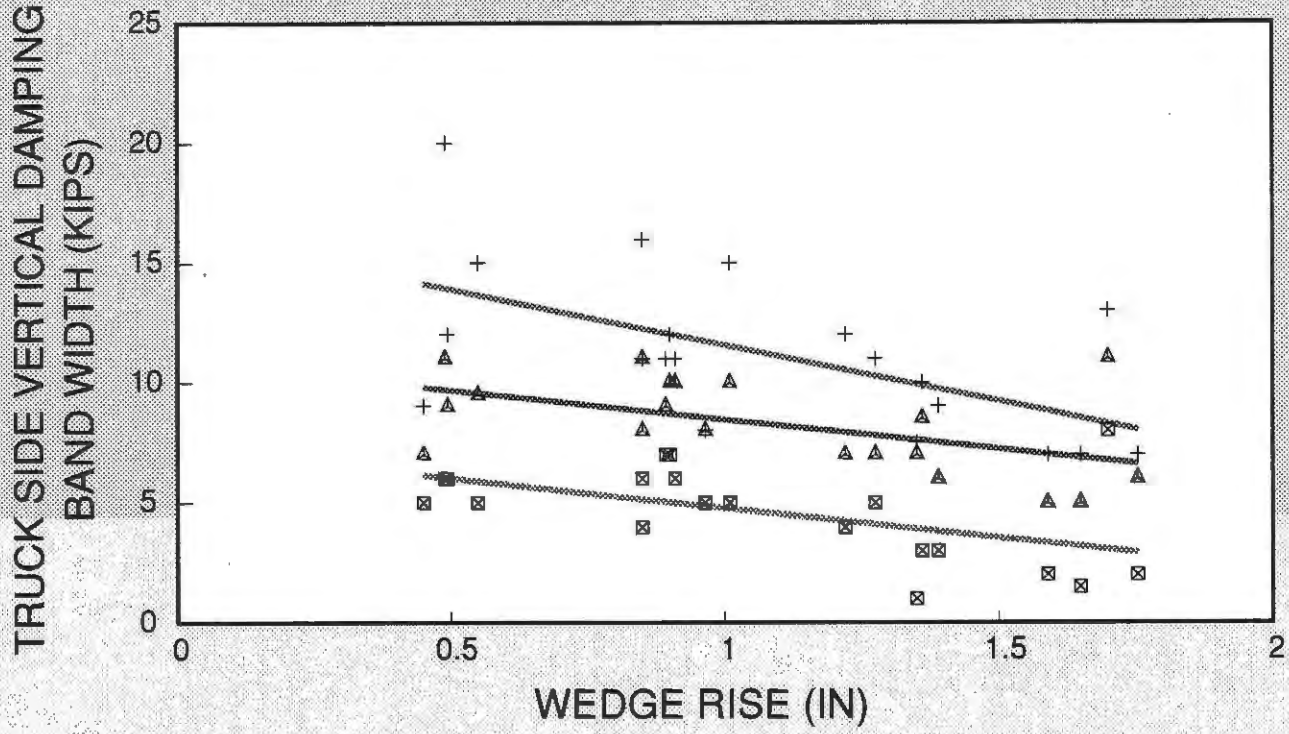
MSU04

Friction Wedge Rise in Truck Bolster Pocket



MSU Vertical Test

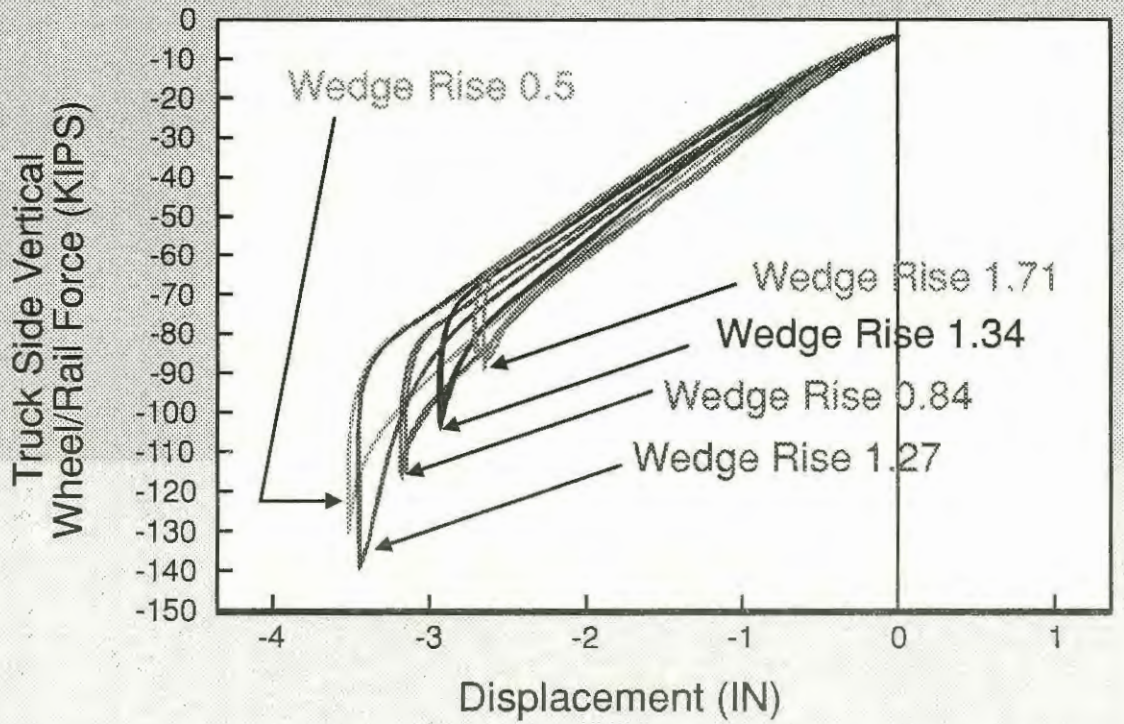
Vertical Damping vs Wedge Rise



HDAMPPOS/MSU06

MSU Vertical Test

Comparison of Suspension Travel

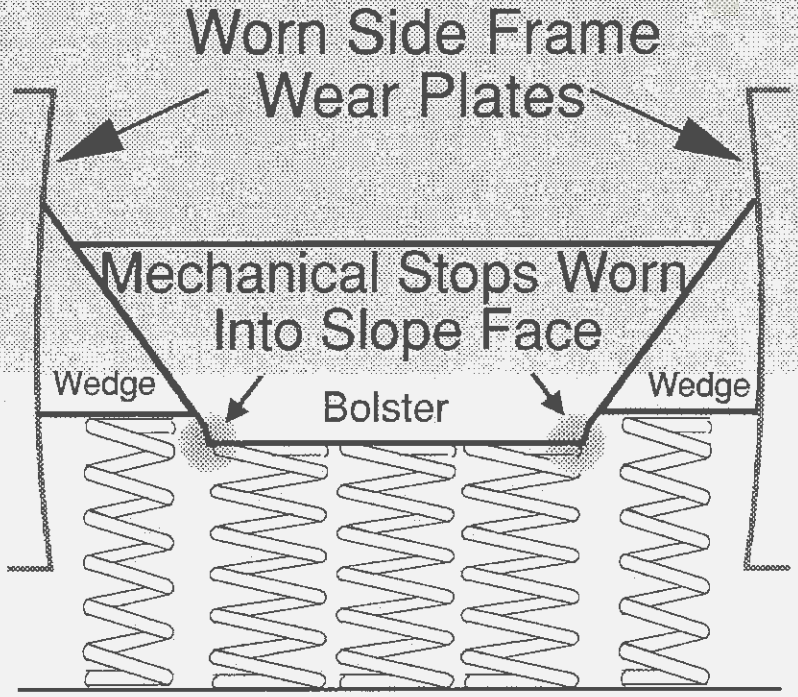


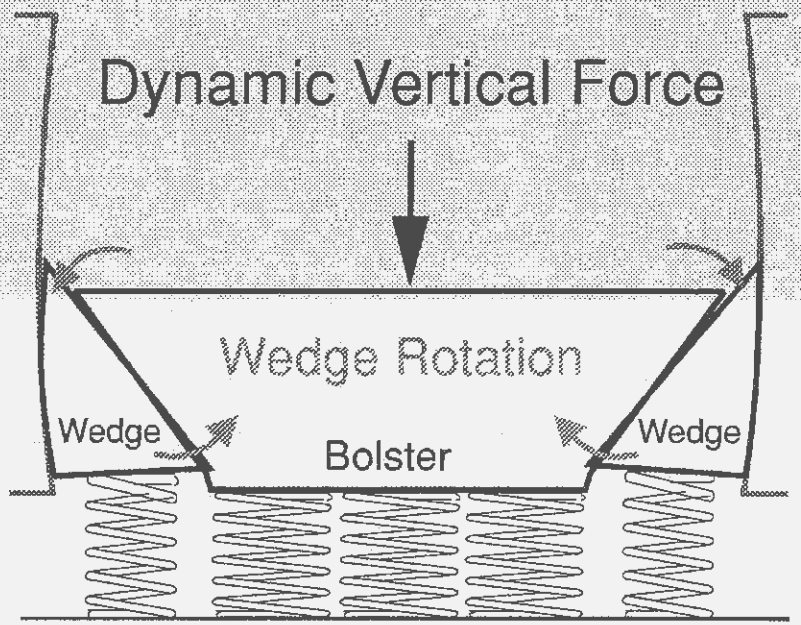
Note: Trucks Tested with New Coils in all Positions

HDAMPPOS/MSU07

MSU Vertical Test

Wedge Rotation in Worn Side Frame Wear Plates

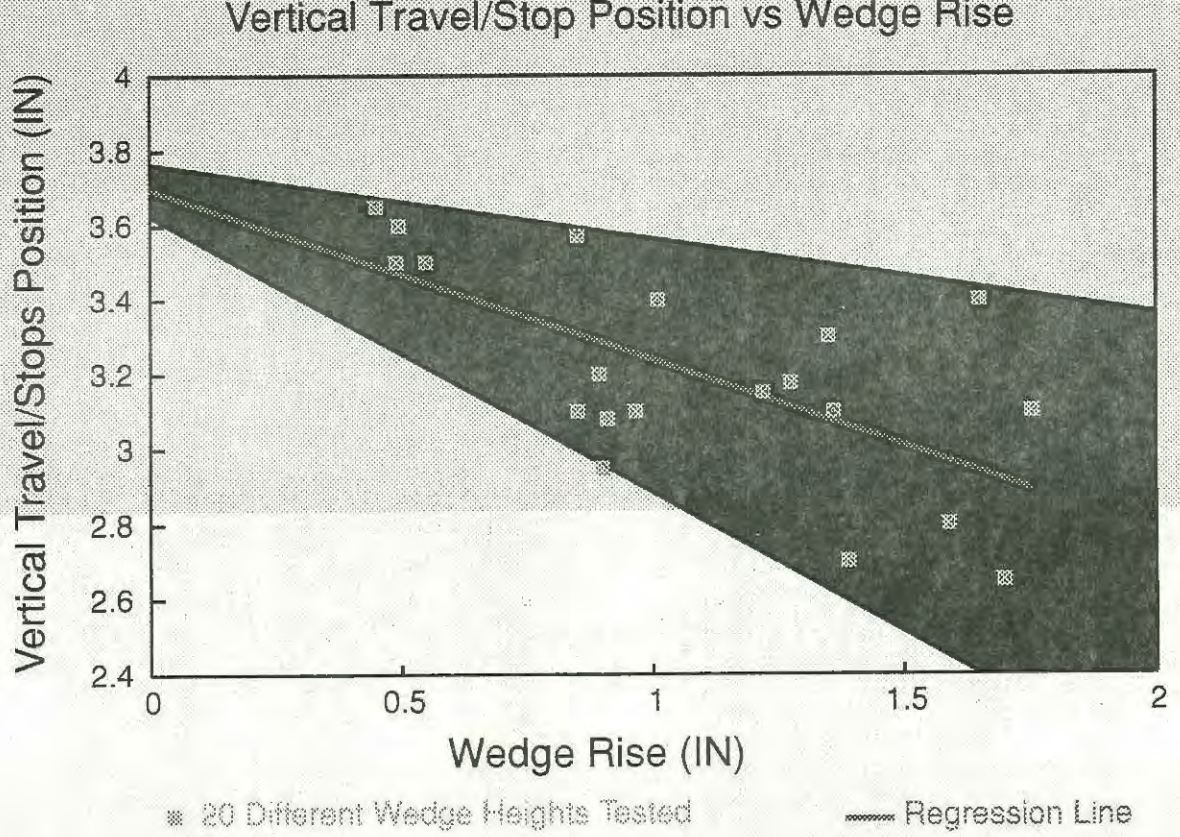




HEX-1A/MSU09

MSU Vertical Test

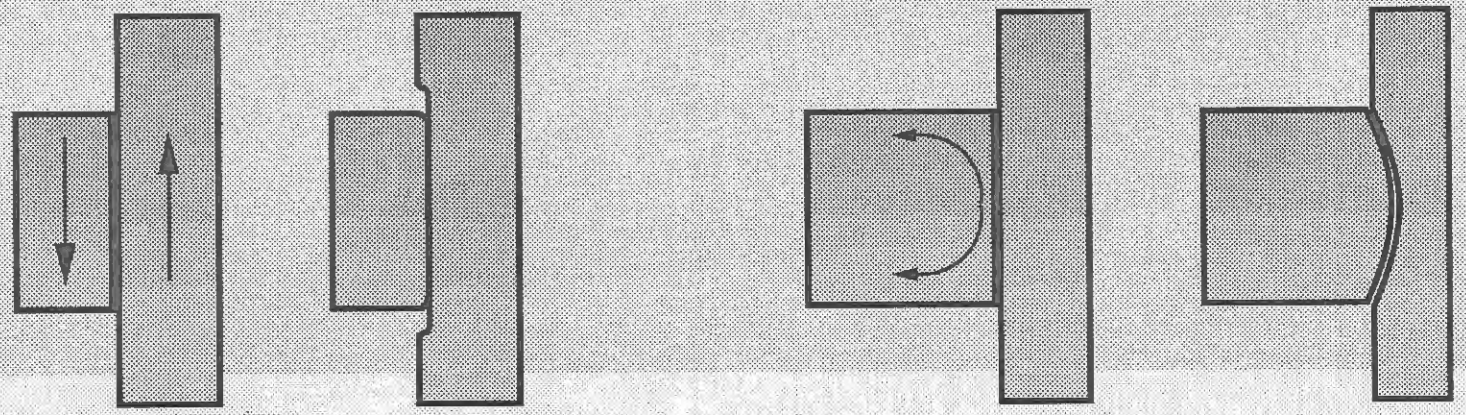
Loss of Travel Due to Wedge Binding



HDAMPPOS/MSU08

MSU Vertical Test

Friction Based Mechanical Damping Systems

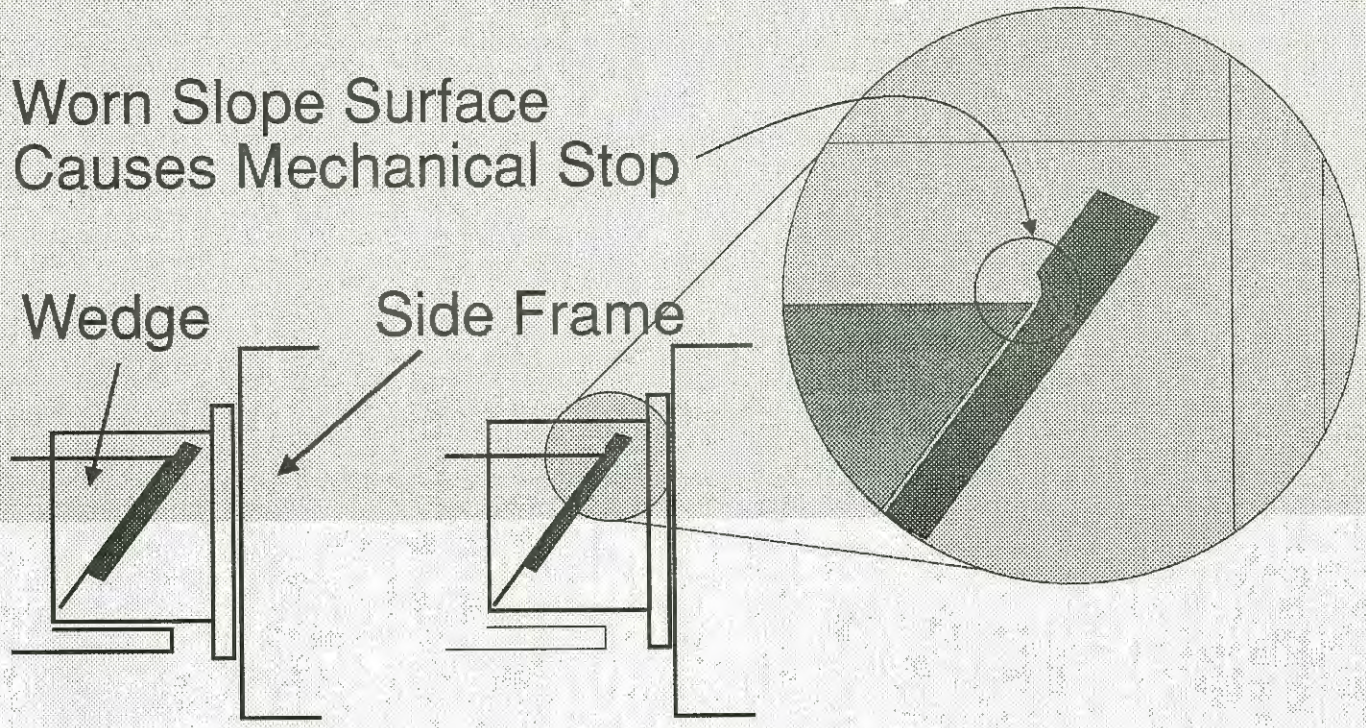


All Friction Systems Experience Localized
Wear Patterns Creating Mechanical Stops

MSU Vertical Test

Constant Damped Design

Worn Slope Surface
Causes Mechanical Stop



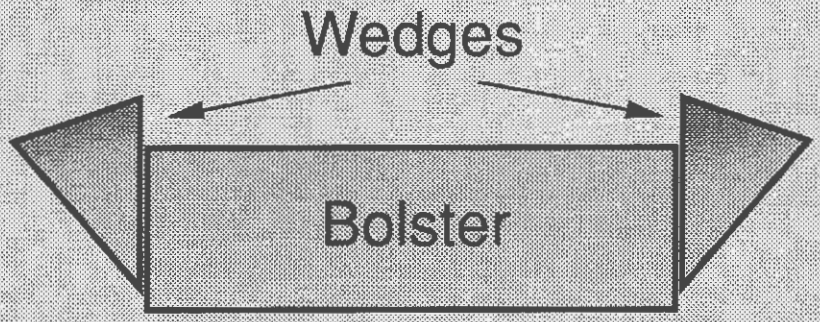
New Bolster Worn Bolster

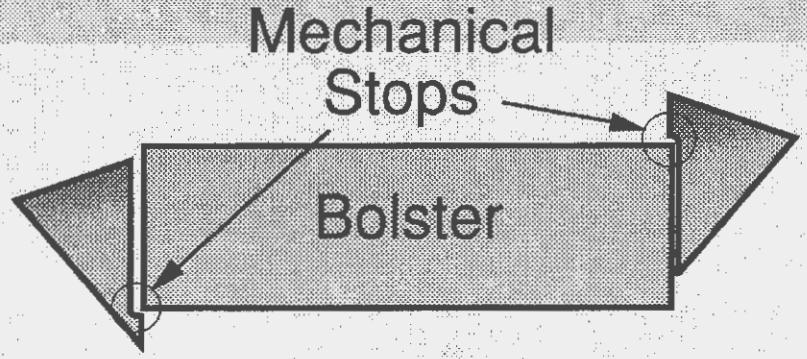
Note: Observations at CONRAIL Re-hab Shop

CON-WED/MSU11

MSU Vertical Test

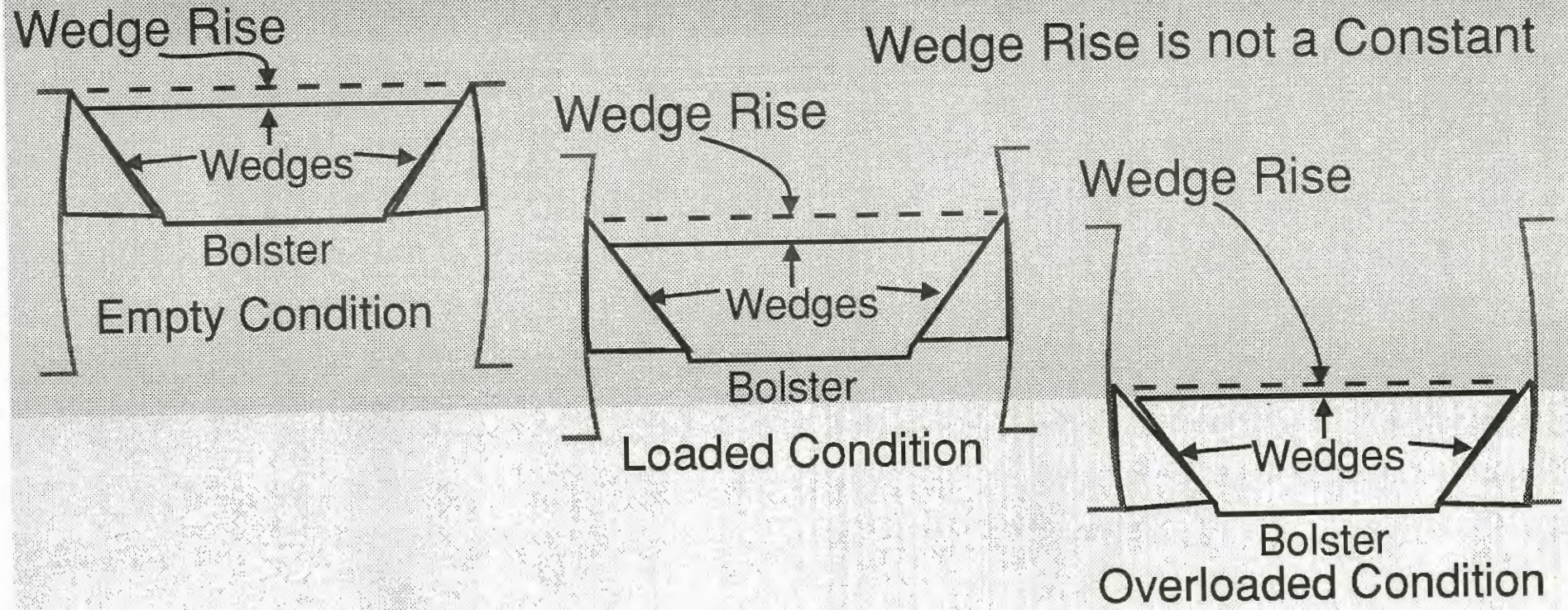
Other Friction Systems





MSU Vertical Test

Wedge Rise in Different Loaded Conditions

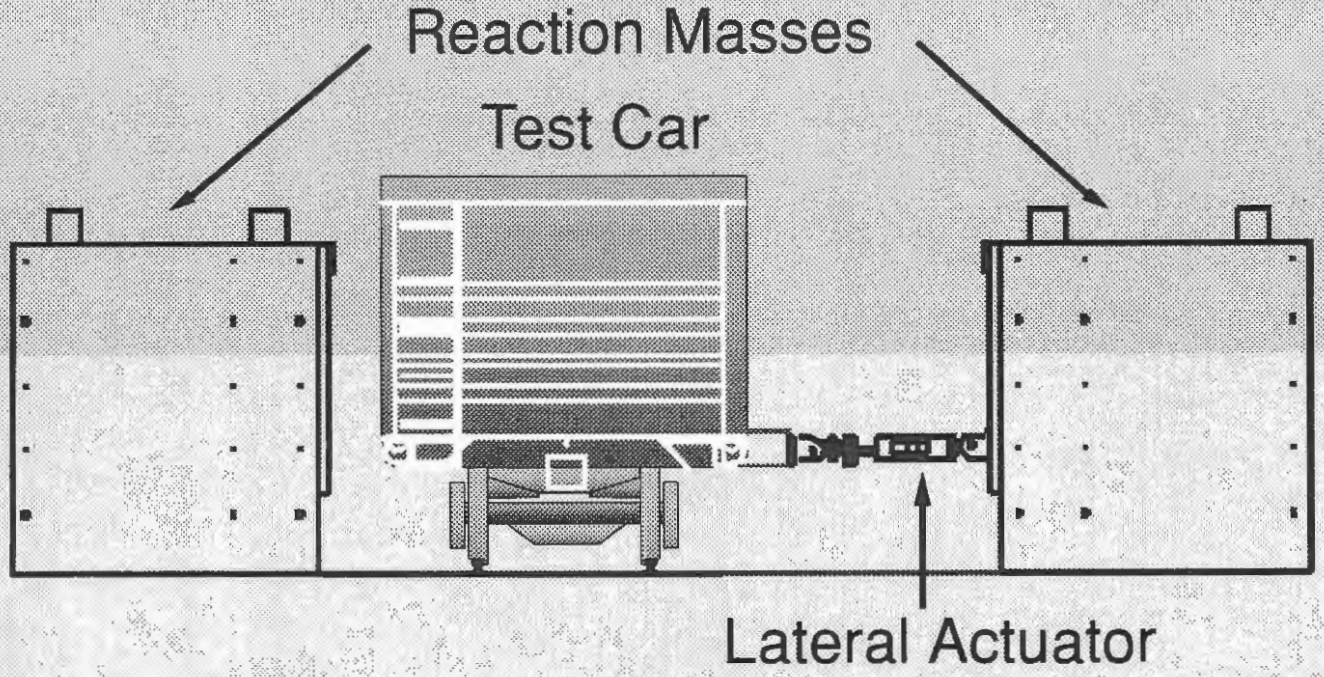


HRISE/MSU13

MSU Lateral Test

Mini-Shaker Test Unit

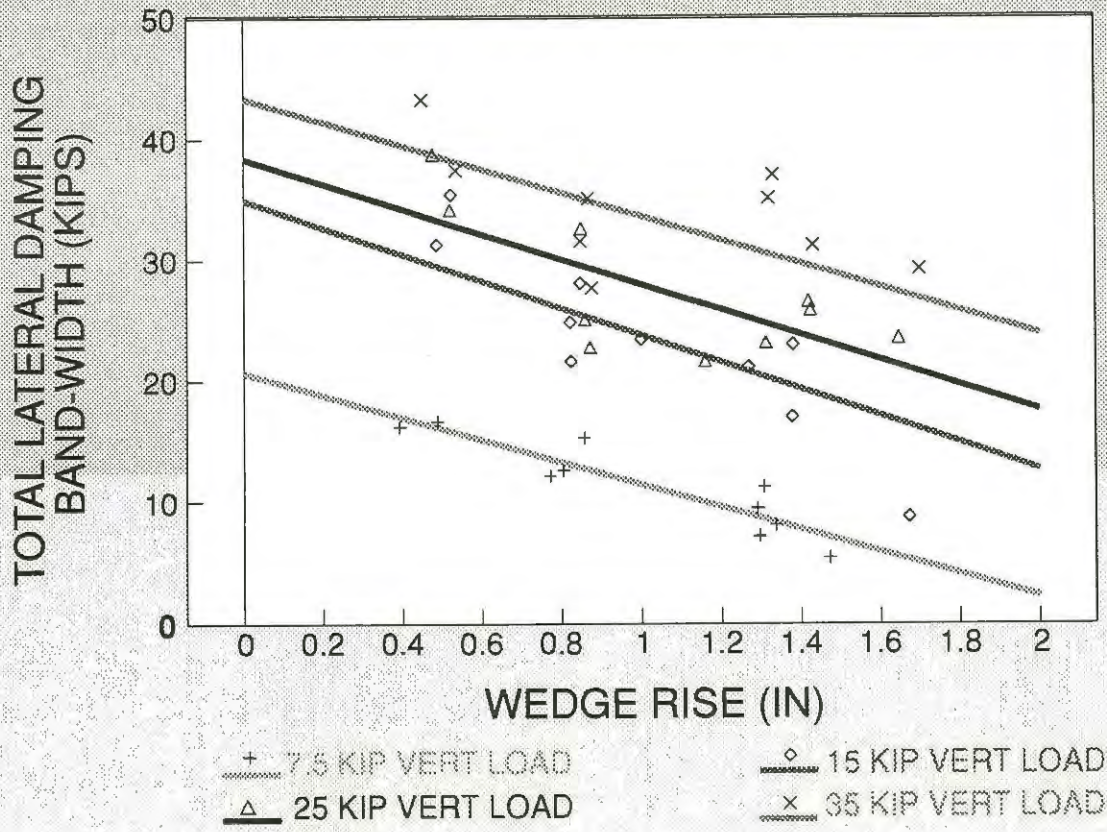
Lateral Suspension Characterization



HMSULAT/MSU14

MSU Lateral Test

Total Lateral Damping vs Wedge Rise



REGRESSION LINES

35 KIPS

25 KIPS

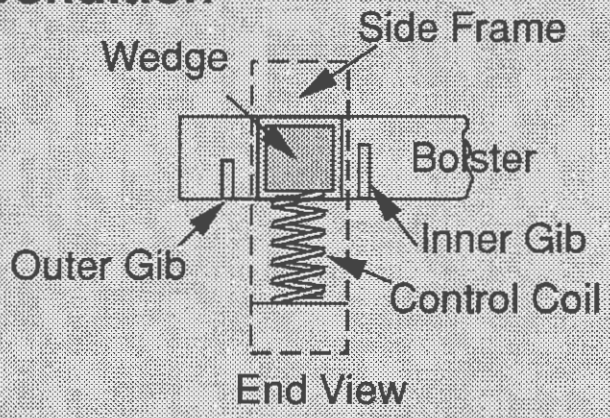
15 KIPS

7.5 KIPS

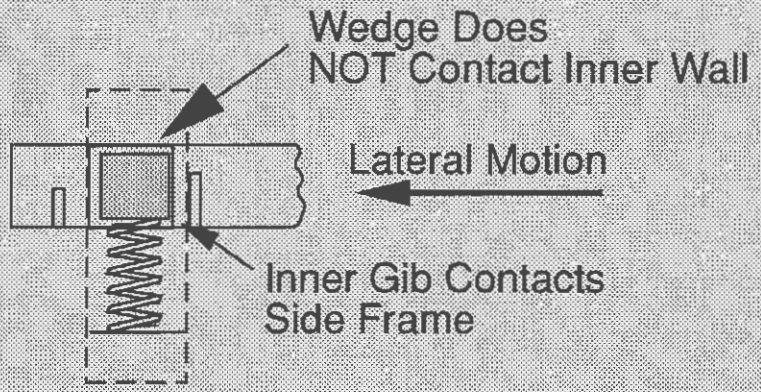
HFINAL/MSU16

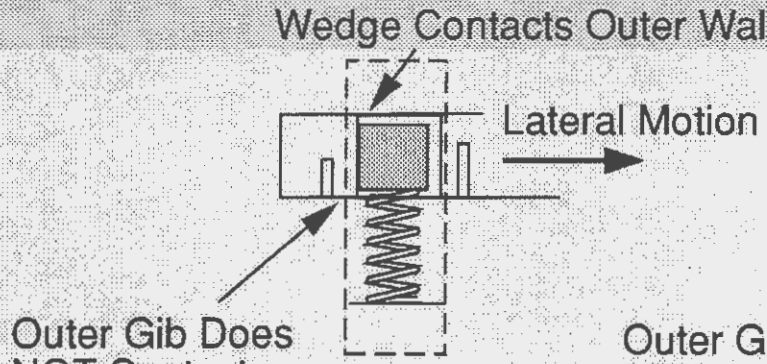
MSU Lateral Test

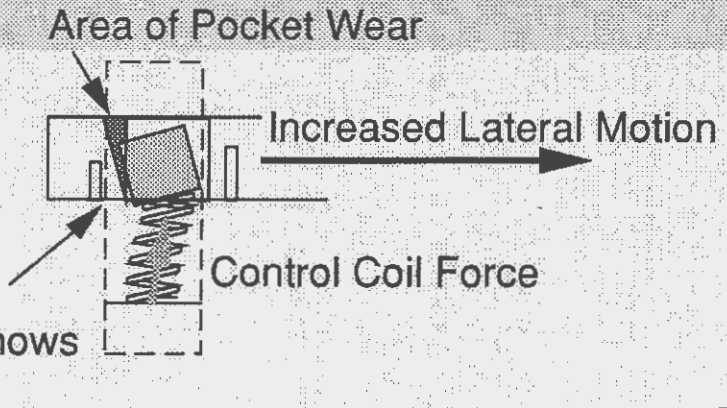
New Truck Condition



Area of Pocket Wear





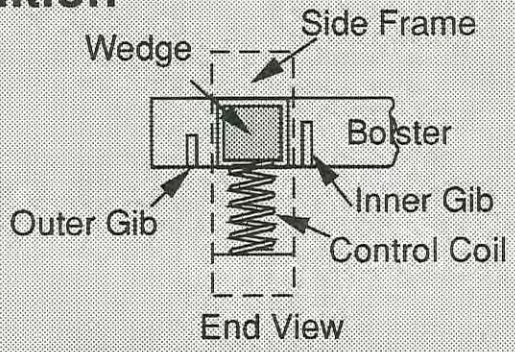


PocketW/MSU17

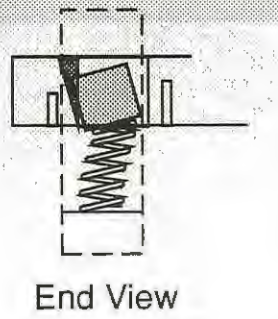
MSU Lateral Test

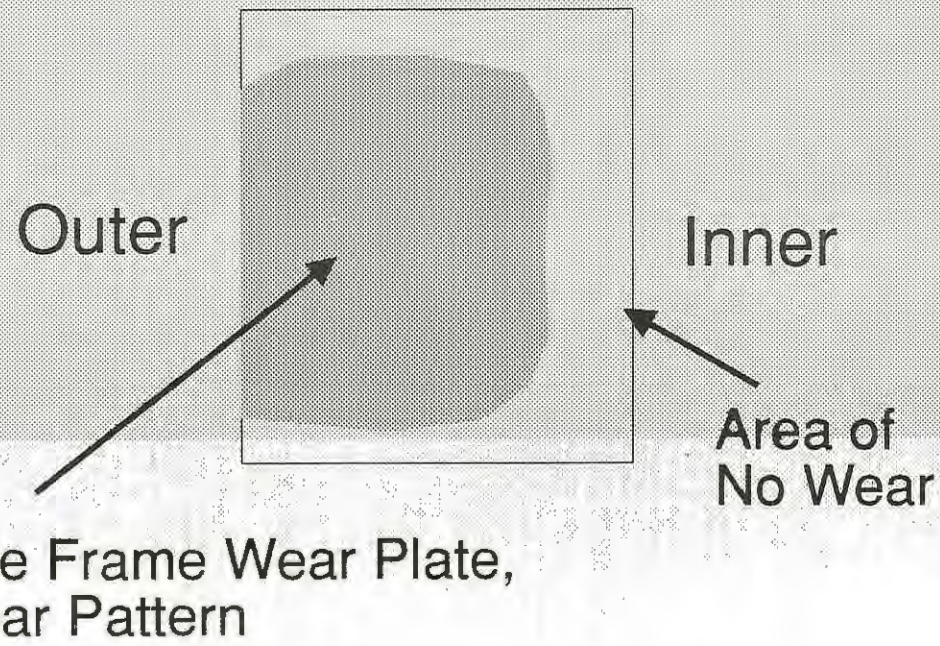
Side Frame Wear and Pocket Wear

New Truck Condition



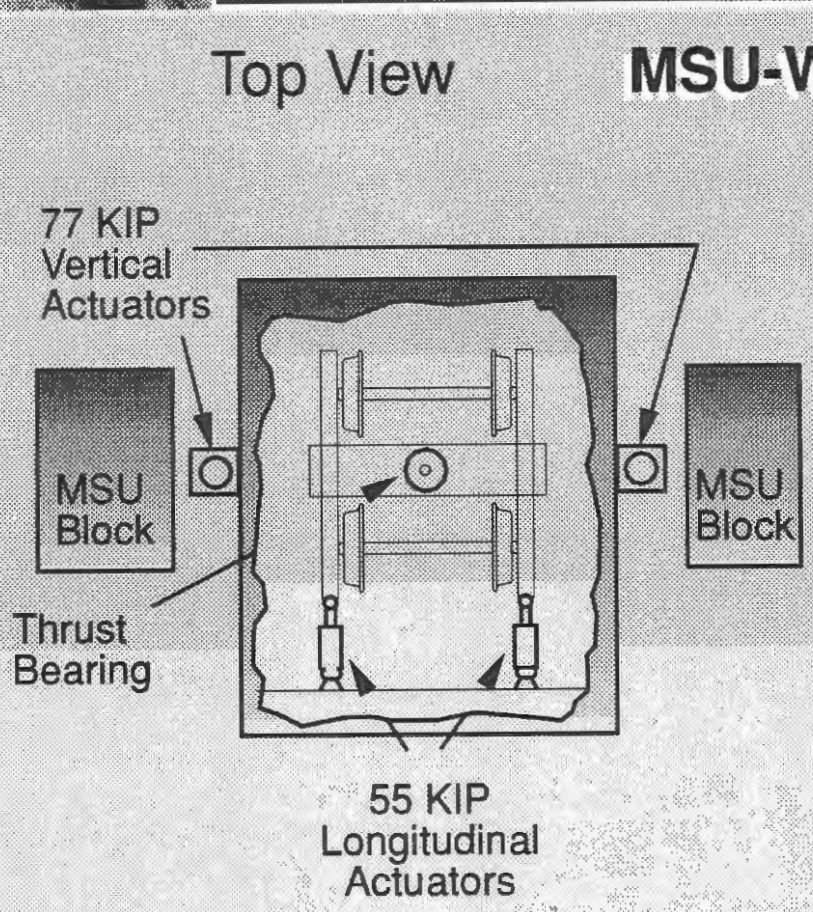
Worn Truck Condition

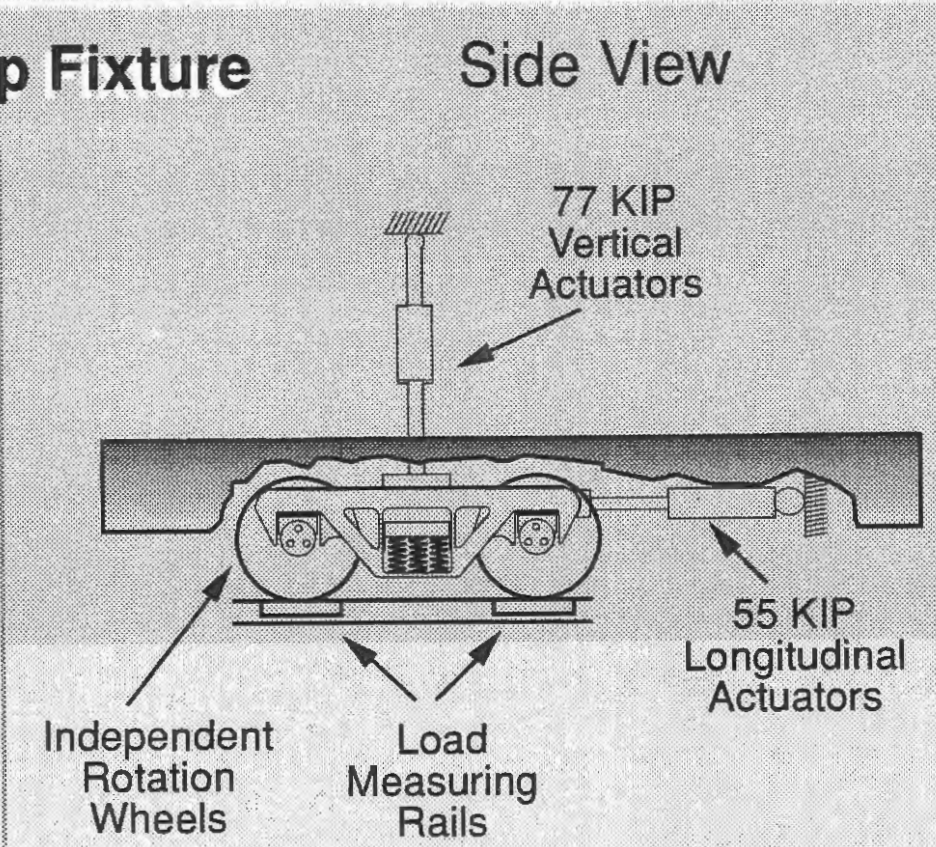




MSU Vertical Test

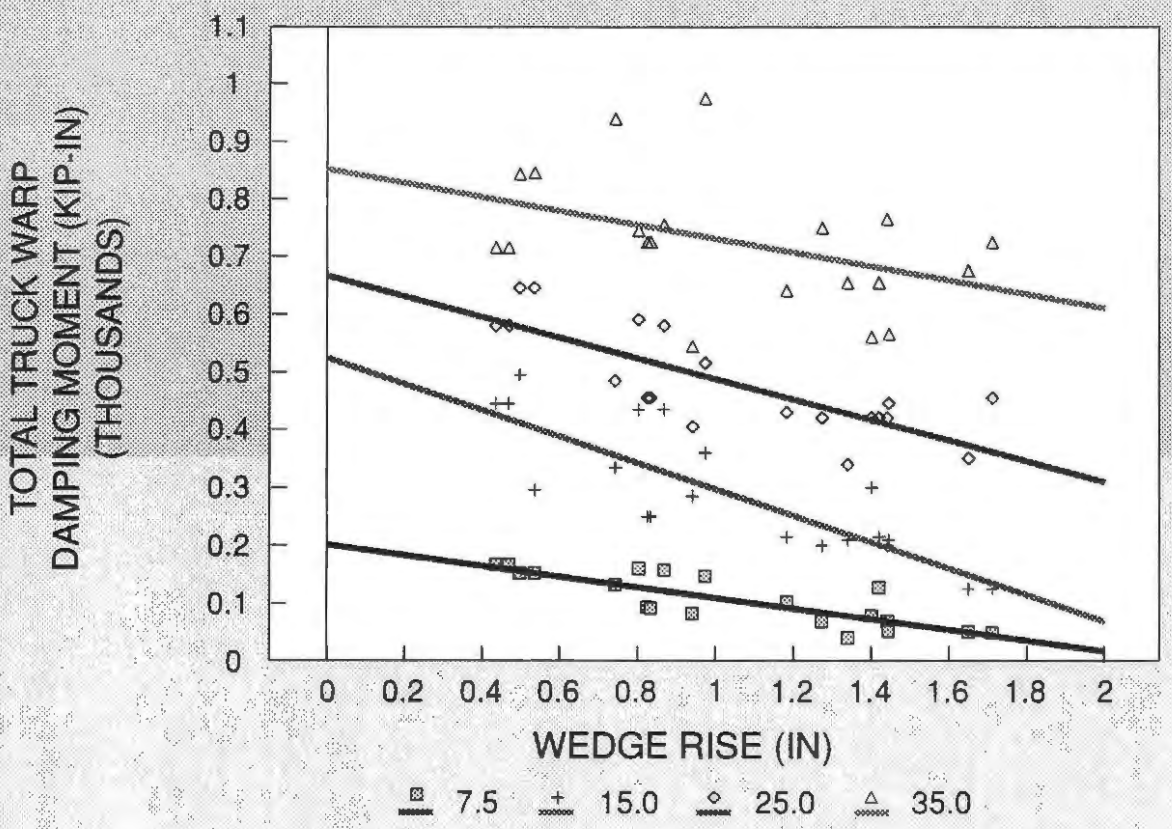
MSU-Warp Fixture





HWARPFIX/MSU19

MSU Warp Test



HCWWEDGE/MSU20

Service Worn Truck Test

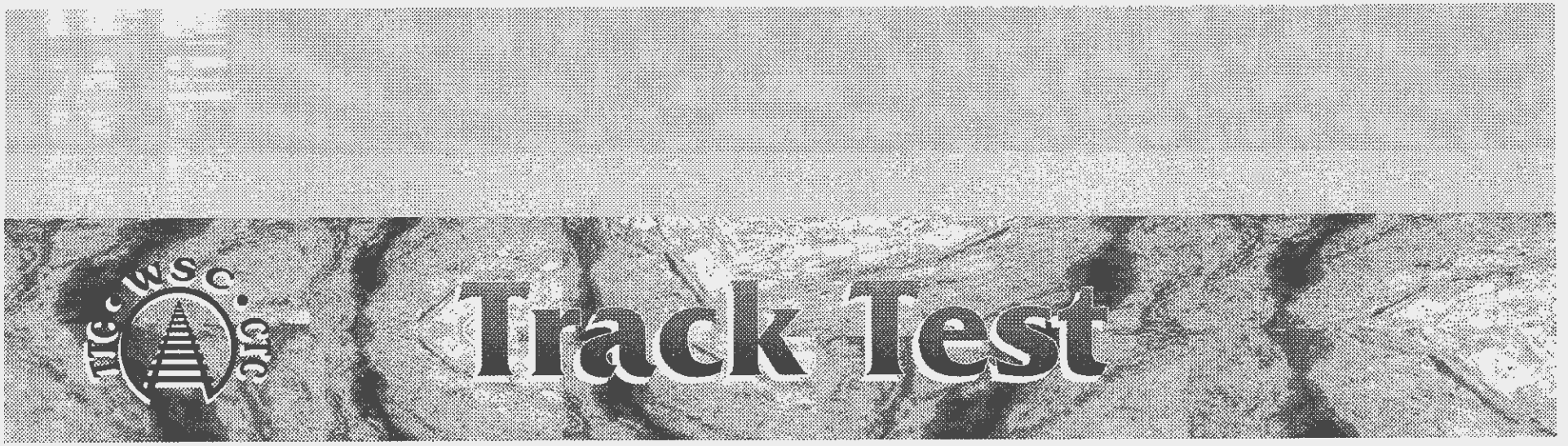
Conclusions--MSU Test

- Test showed relationship between wedge rise and loss of damping and change in vertical, lateral, and warp stiffness
- Test showed wedges lockup suspension due to clearance problems caused by wear

Service Worn Truck Test

Conclusions (continued)--MSU Test

- Identified Problems with Placement of Outer Bolster Gib Causing Pocket Wear Due to Combined Vertical and Lateral Truck Motions
- Instrumented Truck
  - New truck syndrome
  - Full stroke excitation galled surfaces



TT01.cdr

Track Test

Objective

- Establish a Relationship between Wear and Performance

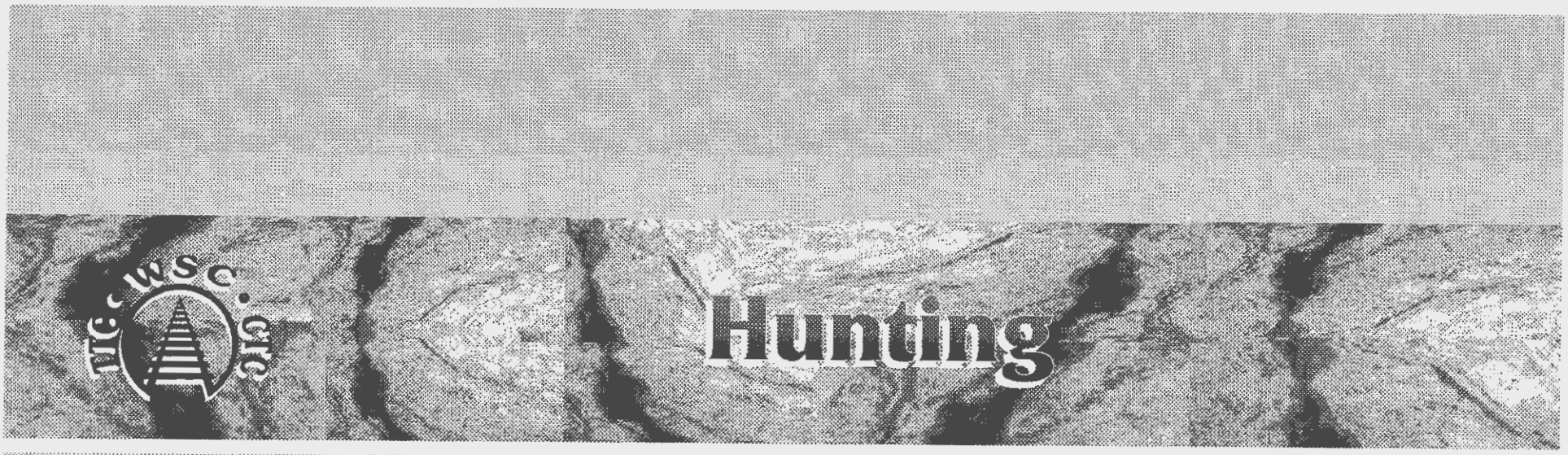
Method

- Test Three Wear States to Determine when Performance Deteriorates
- Wedge Rise 0, 0.4, 1.7 (0.4 shimmed to 0)
  - Hunting
  - Twist and Roll
  - Pitch and Bounce

} Chapter XI Perturbations

Note: All tests performed with TTC/FAST worn instrumented wheel sets

Tread -- Worn CN/AAR1B Shape Flange Angle -- Steep FAST Worn

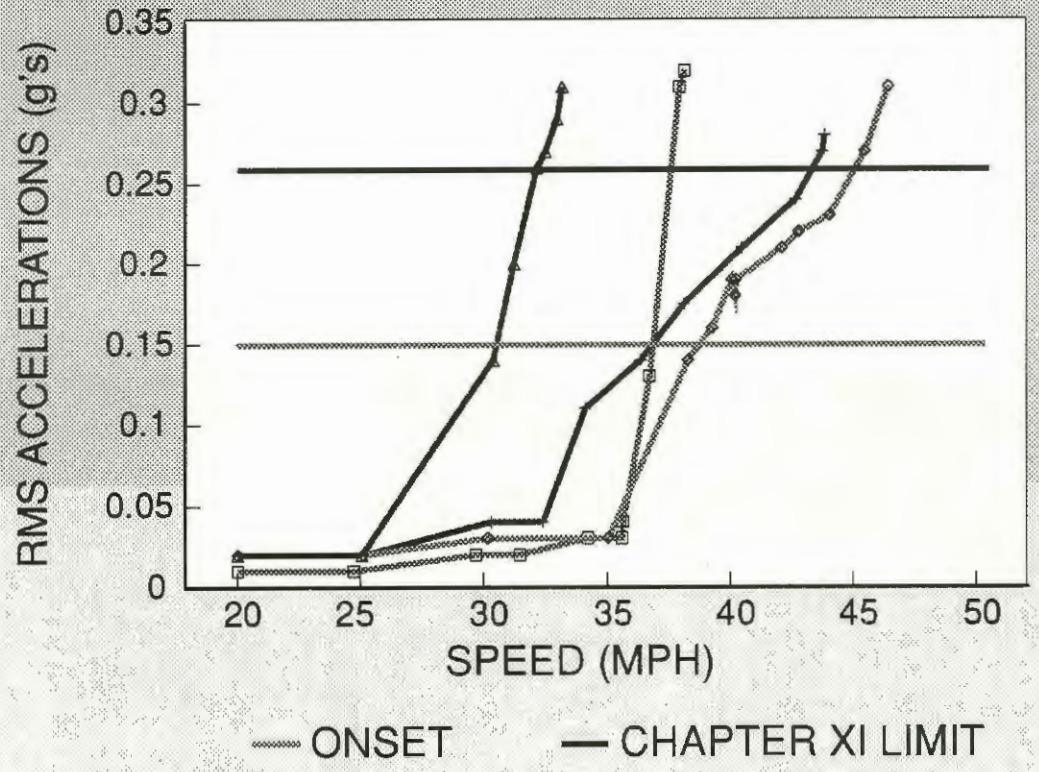


h01.cdr

Track Test

Hunting Track Test

COMPARISON OF CAR BODY LATERAL ACCELERATIONS, A-END



- WR = 1.5 - 1.7 mu = 0.43
- WR = 0.4 - 0.5: BL mu = 0.56
- WR = 0.4 - 0.5: AS mu = 0.58
- WR =2+ mu = 0.52

WR = "WEDGE RISE"
BL = B-END LEADING
AL = A-END LEADING
AS = A-END LEADING SHIMMED

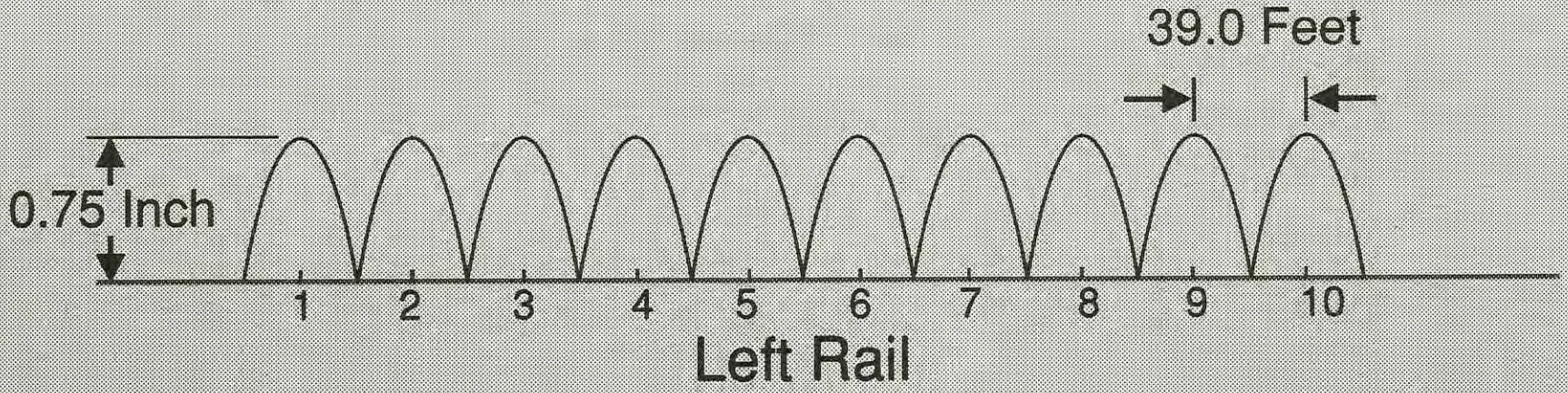
HHUNT.TT03

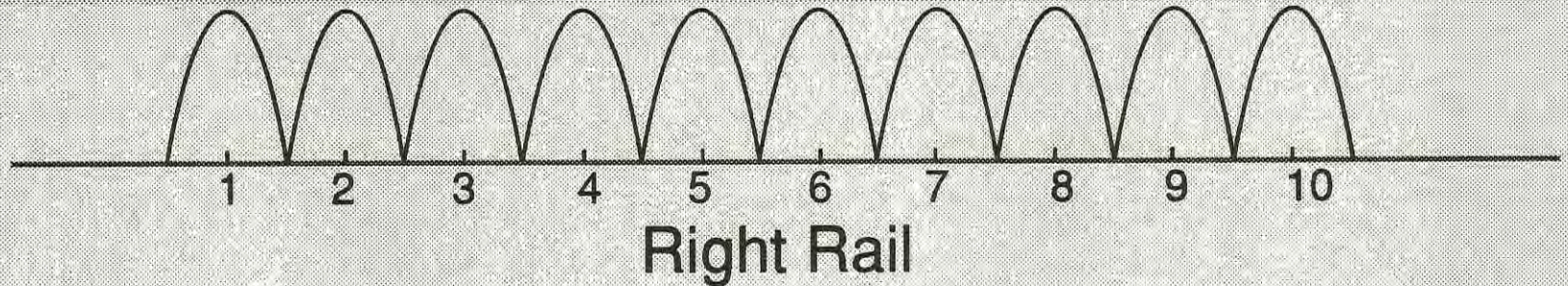


BP01.cdr

Track Test

Pitch and Bounce Test Section



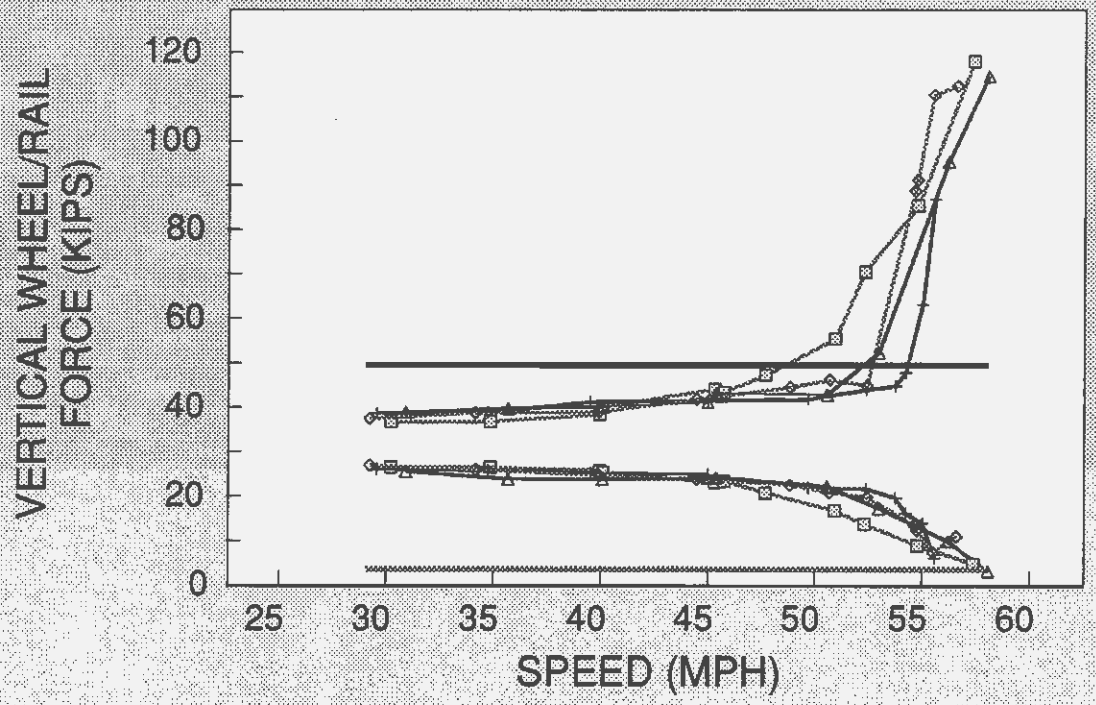


tt12

Track Test

Pitch and Bounce Track Test

COMPARISON OF VERTICAL RAIL FORCE, AXLE 1, RIGHT



WR = 1.5 - 1.7

WR = 0.4 - 0.5: BL

WR = 0.4 - 0.5: AL

WR = 0.0: AS

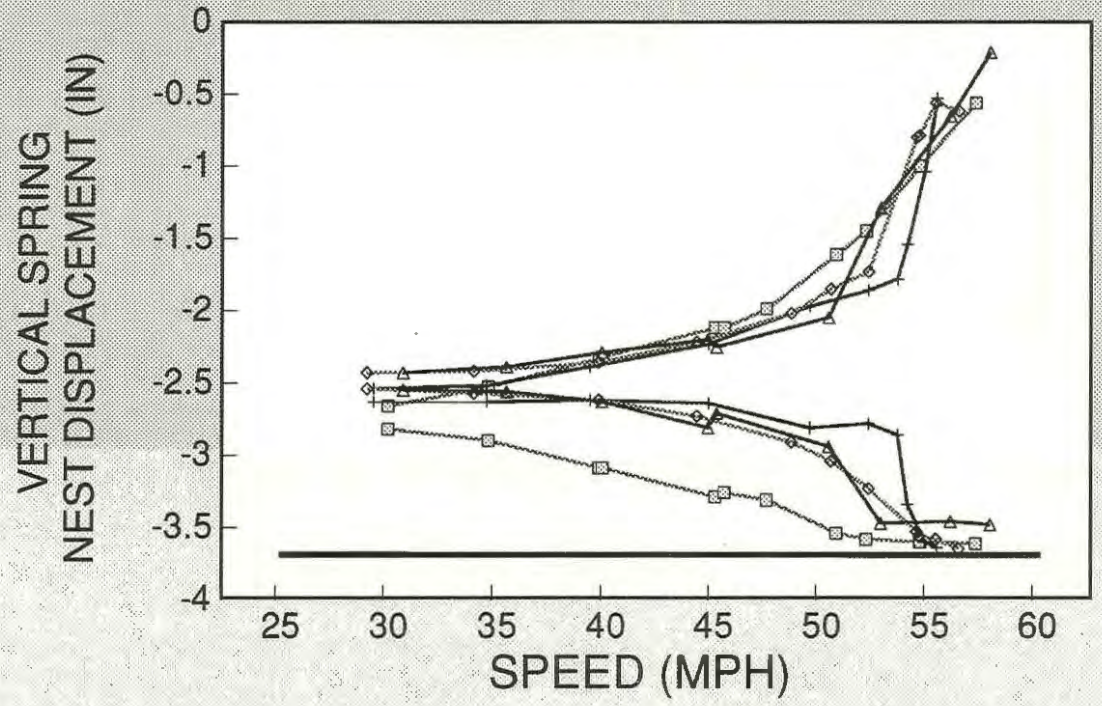
WR = "WEDGE RISE"
BL = B-END LEADING
AL = A-END LEADING
AS = A-END LEADING SHIMMED

—PROPOSED 1.5 DLF —— CHAPTER XI 10% MINIMUM

Track Test

Pitch and Bounce Track Test

COMPARE VERTICAL SPRING NEST DISPLACEMENT LEAD LEFT



WR = 1.5 - 1.7

WR = 0.4 - 0.5: BL

WR = 0.4 - 0.5: AL

WR = 0.0: AS

WR = "WEDGE RISE"

BL = B-END LEADING

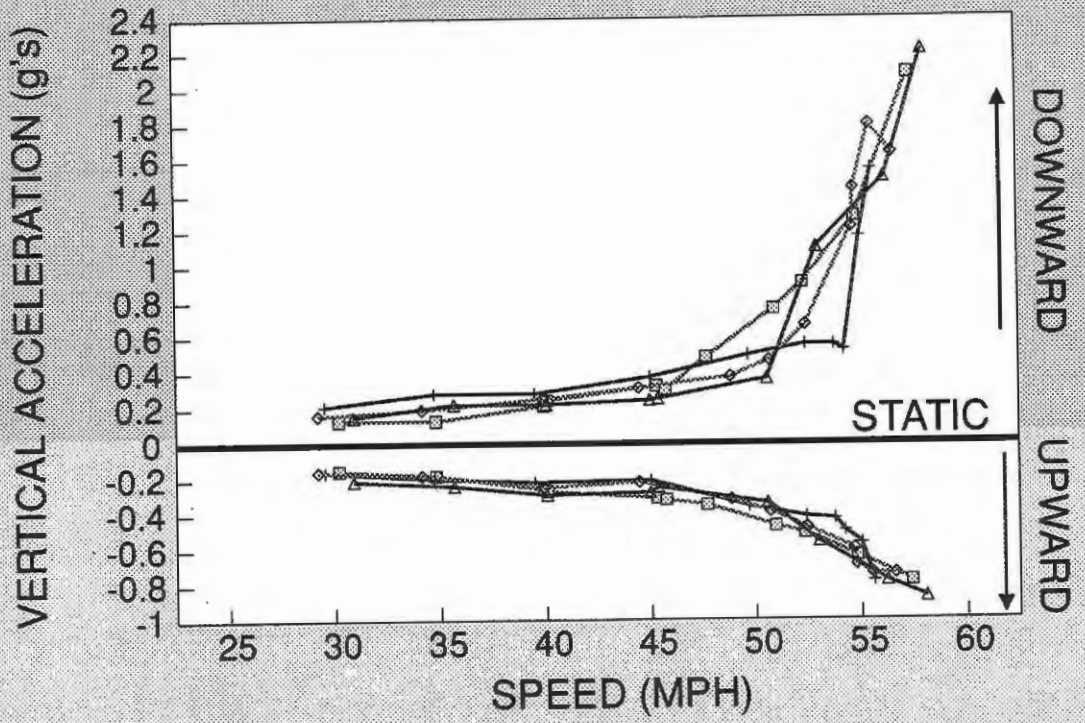
AL = A-END LEADING

AS = A-END LEADING, SHIMMED

Track Test

Pitch and Bounce Track Test

COMPARISON OF CAR BODY ACCELERATIONS, ACZ1



- WR = 1.5 - 1.7
- WR = 0.4 - 0.5: BL
- WR = 0.4 - 0.5: AL
- WR = 0.0: AS

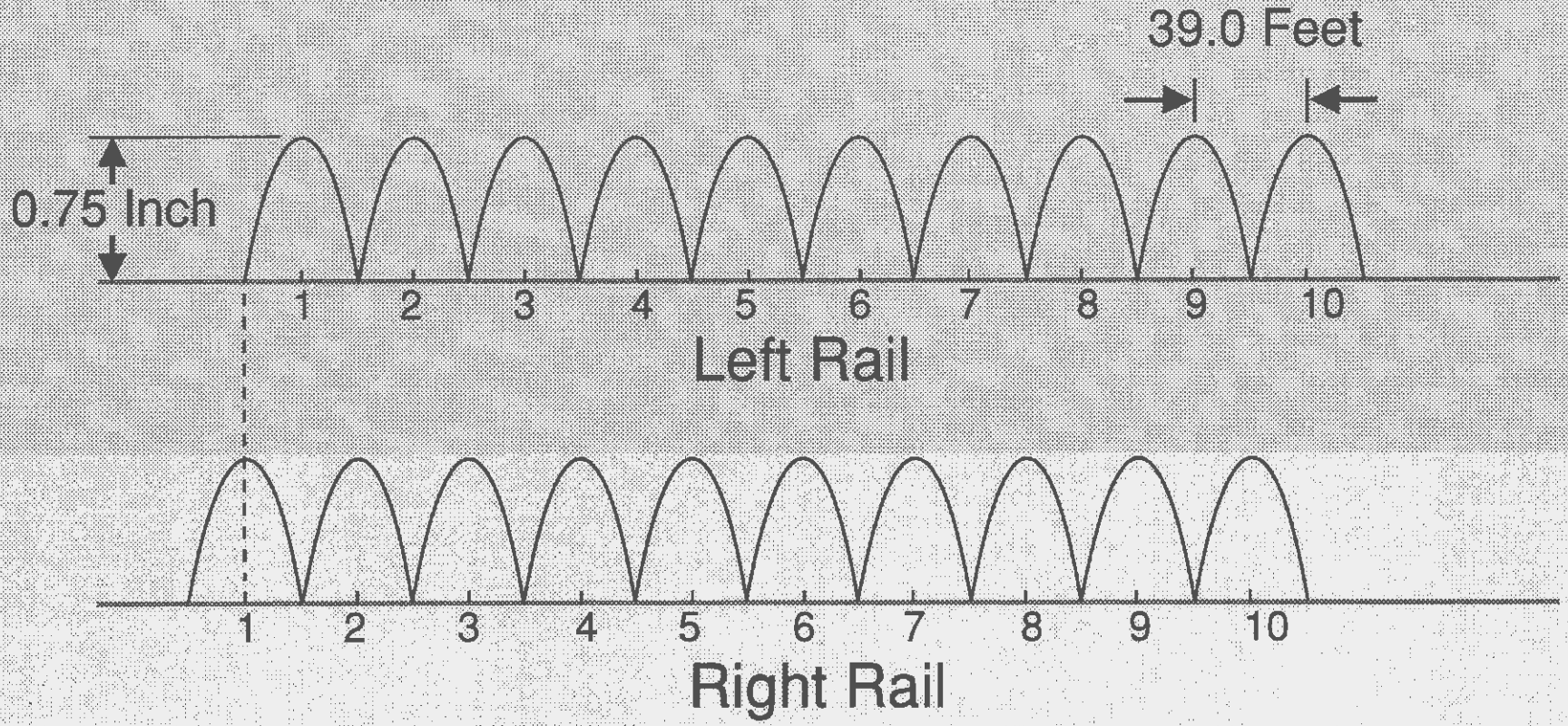
WR = "WEDGE RISE"
BL = B-END LEADING
AL = A-END LEADING
AS = A-END LEADING SHIMMED



MR01.cdr

Track Test

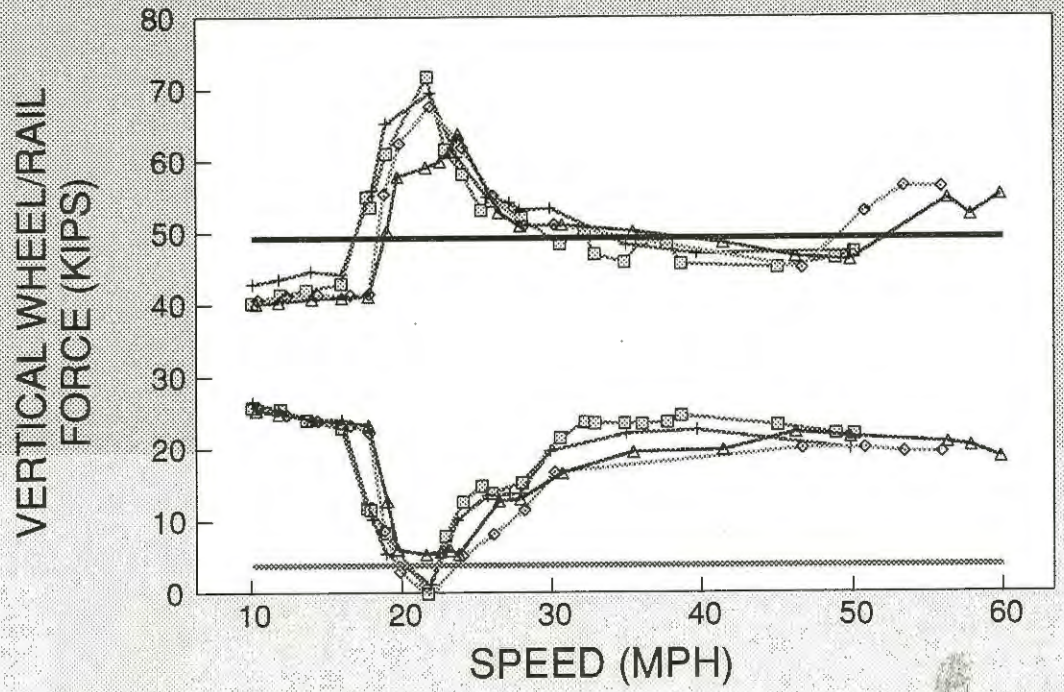
Twist and Roll Test Section



Track Test

Twist and Roll Track Test

COMPARISON OF VERTICAL FORCES, AXLE 1, LEFT



WR = 1.5 - 1.7

WR = 0.4 - 0.5: BL

WR = 0.4 - 0.5: AL

WR = 0.0: AS

WR = "WEDGE RISE"
BL = B-END LEADING
AL = A-END LEADING
AS = A-END LEADING SHIMMED

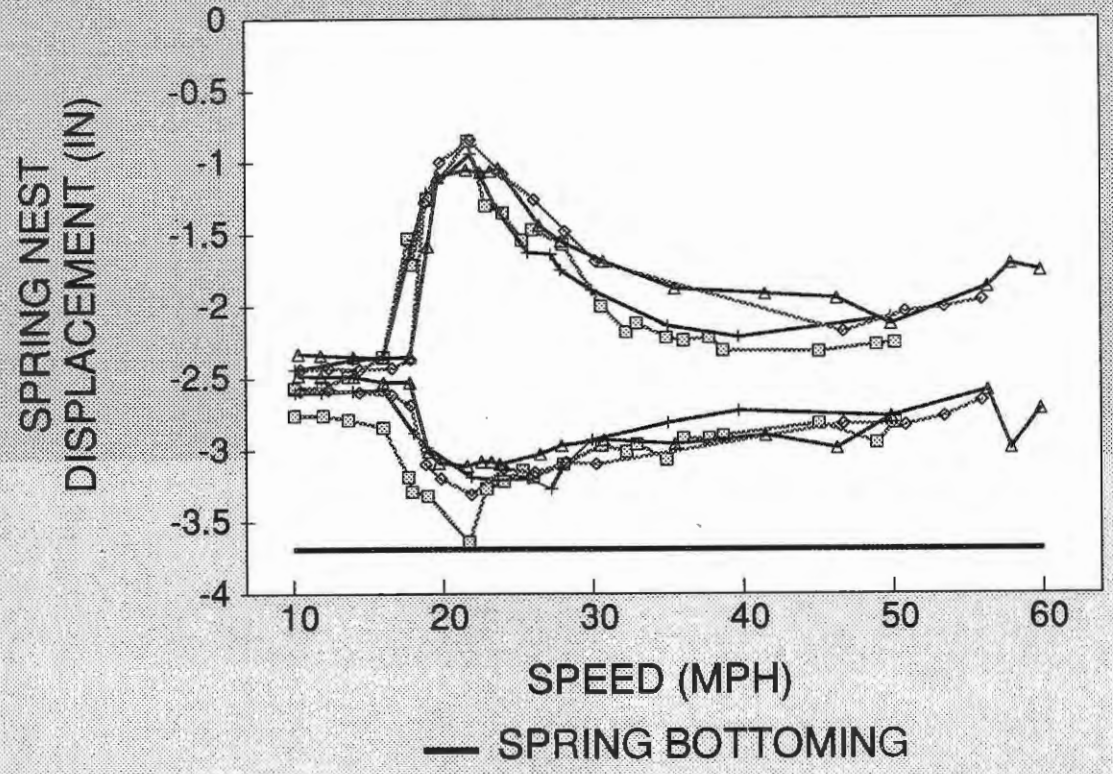
— PROPOSED 1.5 DLF — CHAPTER XI 10% MINIMUM

HT&RFORC/TT07

Track Test

Twist and Roll Track Test

COMPARISON OF VERTICAL SPRING NEST DISPLACEMENT, LEAD LEFT



WR = 1.5 - 1.7

WR = 0.4 - 0.5: BL

WR = 0.4 - 0.5: AL

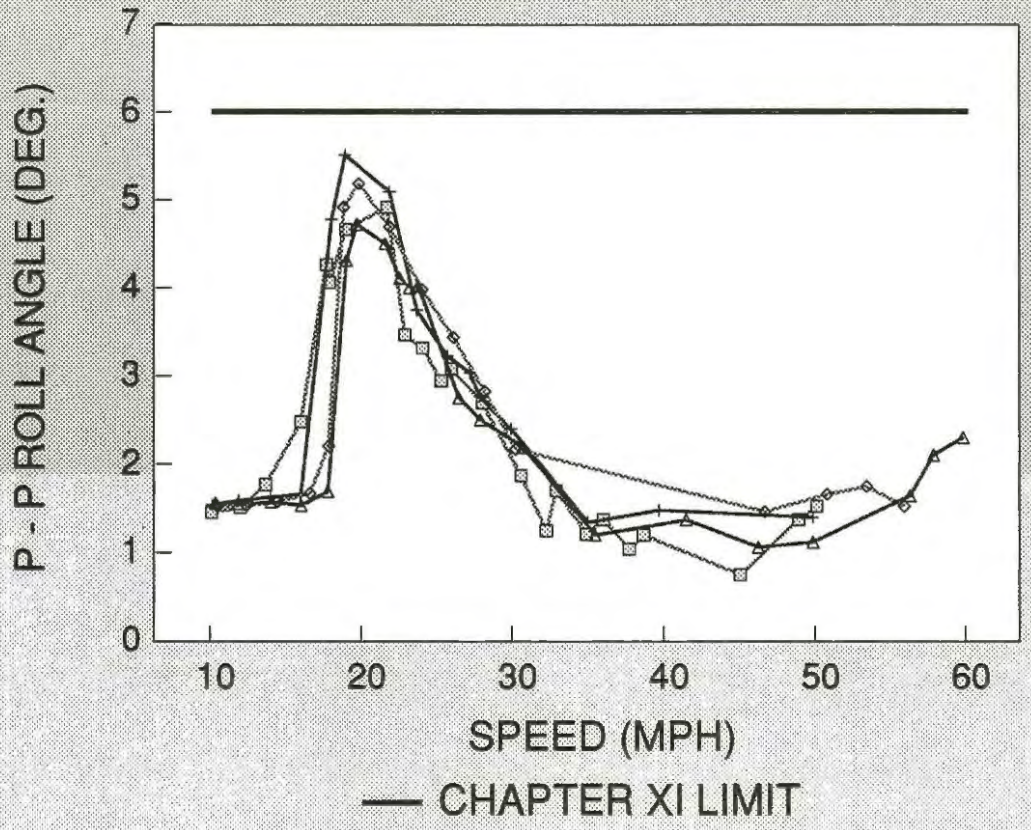
WR = 0.0: AS

WR = "WEDGE RISE"
BL = B-END LEADING
AL = A-END LEADING
AS = A-END LEADING SHIMMED

Track Test

Twist and Roll Track Test

COMPARISON PEAK-TO-PEAK ROLL ANGLE, A-END



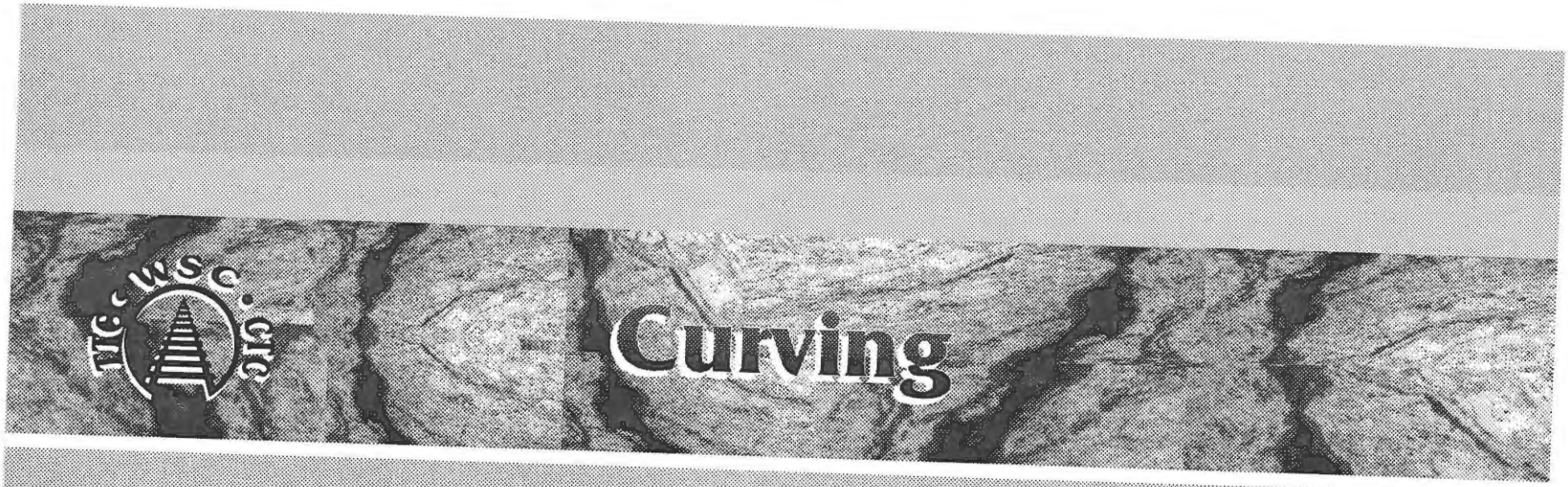
WR = 1.5 - 1.7

WR = 0.4 - 0.5: BL

WR = 0.4 - 0.5: AL

WR = 0.0: AS

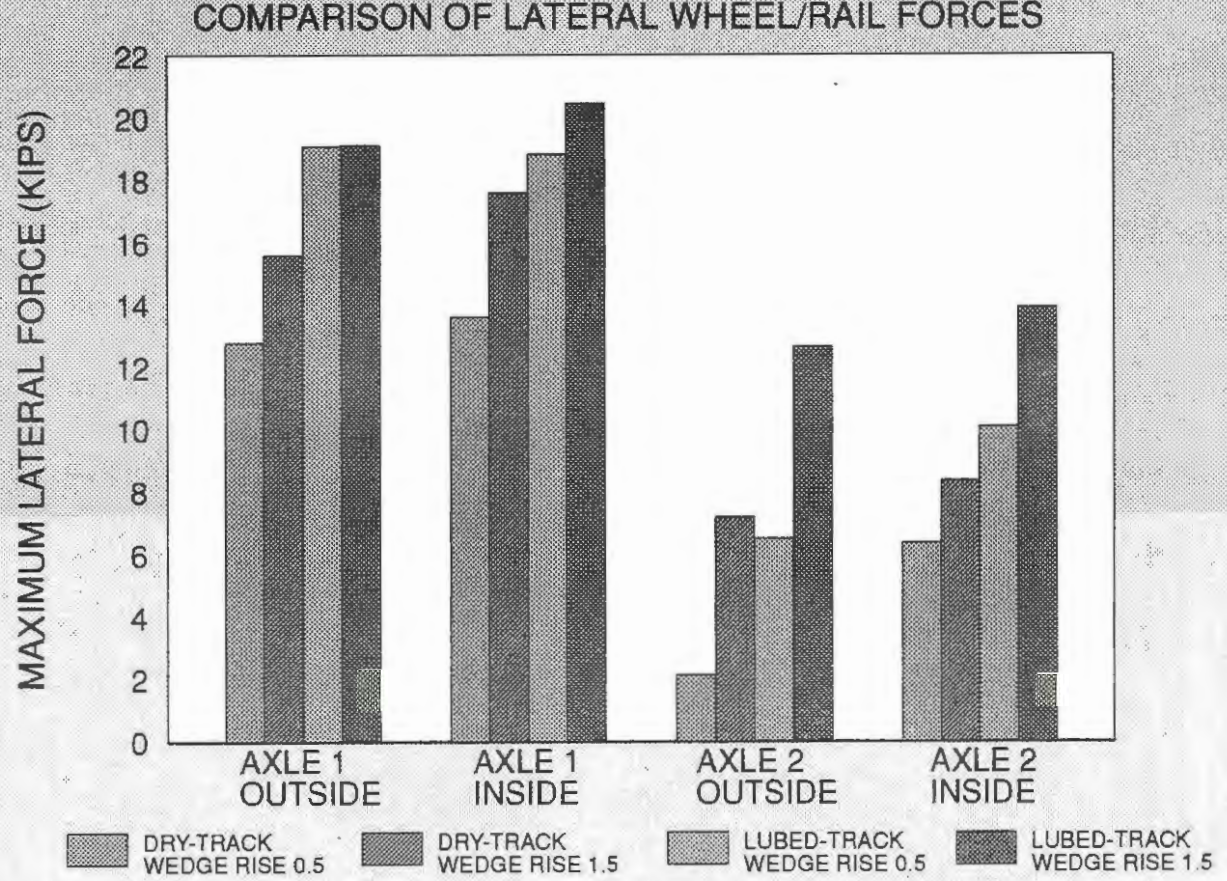
WR = "WEDGE RISE"
BL = B-END LEADING
AL = A-END LEADING
AS = A-END LEADING SHIMMED



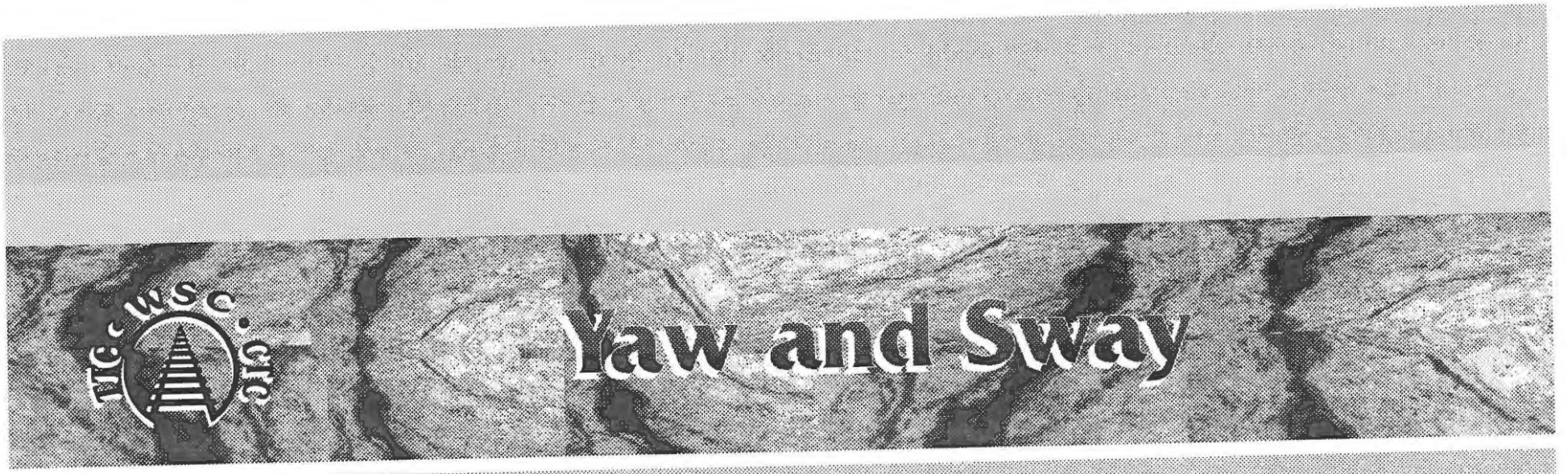
CRVINTRO

Track Test

Curving Track Test, 7.5 Degree Curve, 22 MPH



TT14

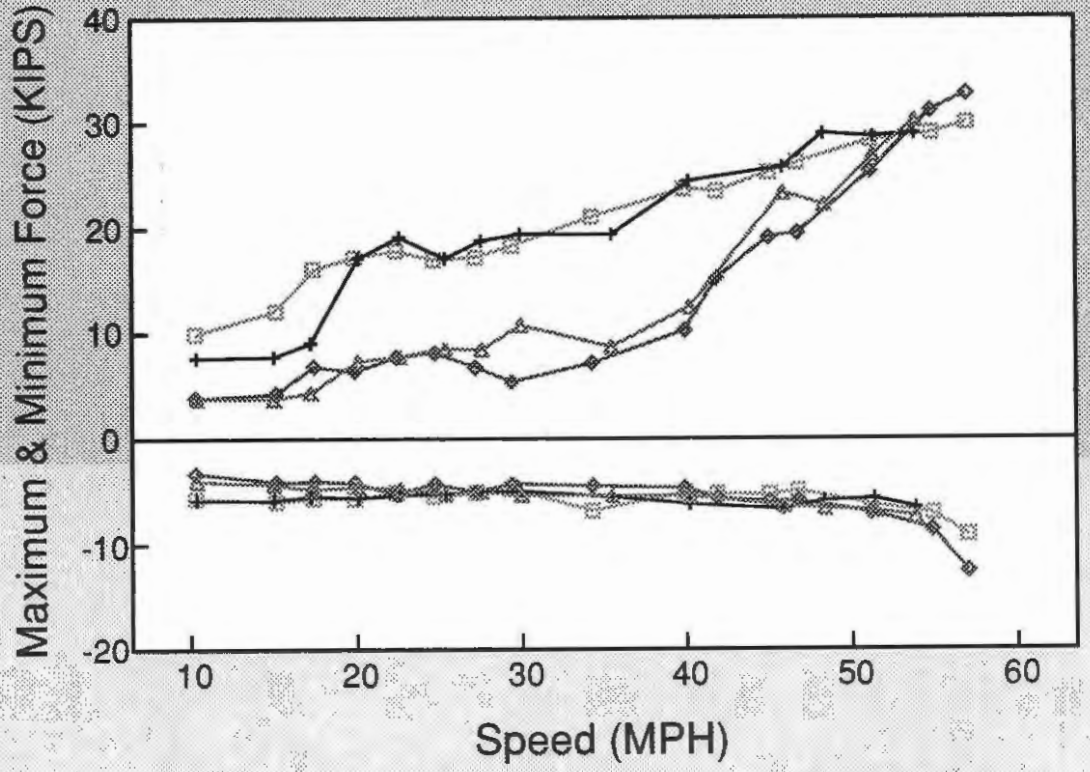


Y&S01.cdr

Track Test

Yaw and Sway Track Test

Comparison of Lateral Force, Axle 1 & 2

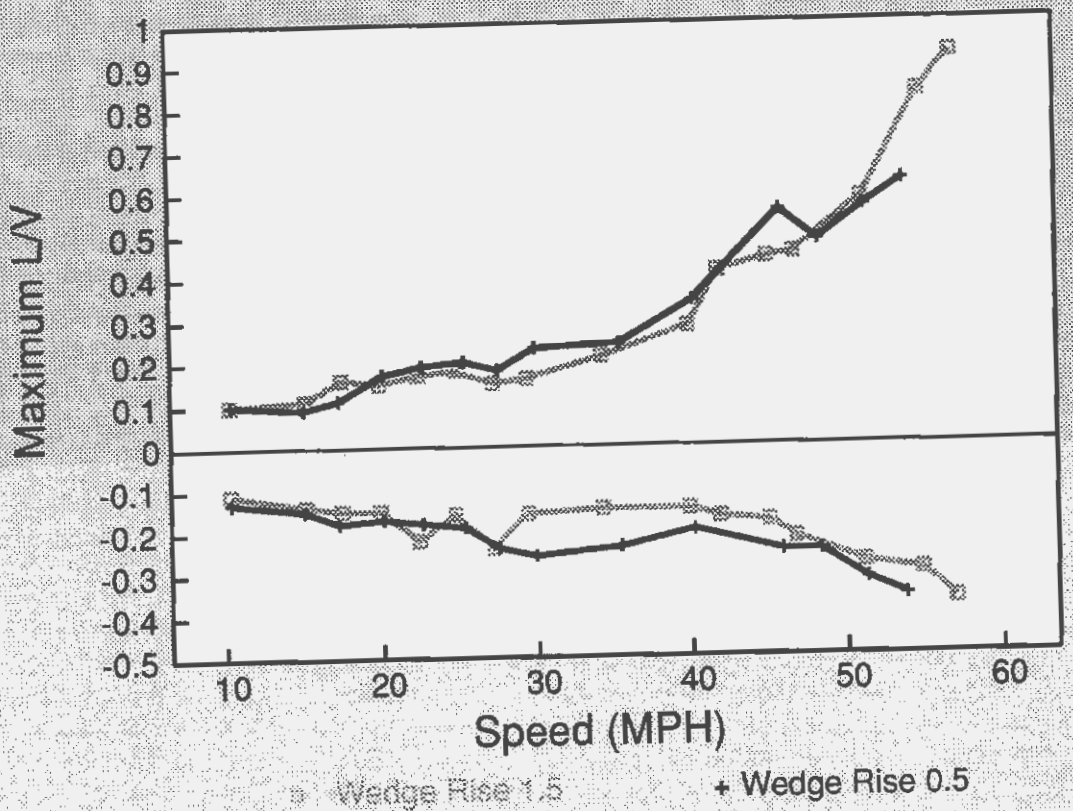


hsf1wear/Y&S02.cdr

Track Test

Yaw and Sway Track Test

Comparison of Maximum L/V, Axle 1

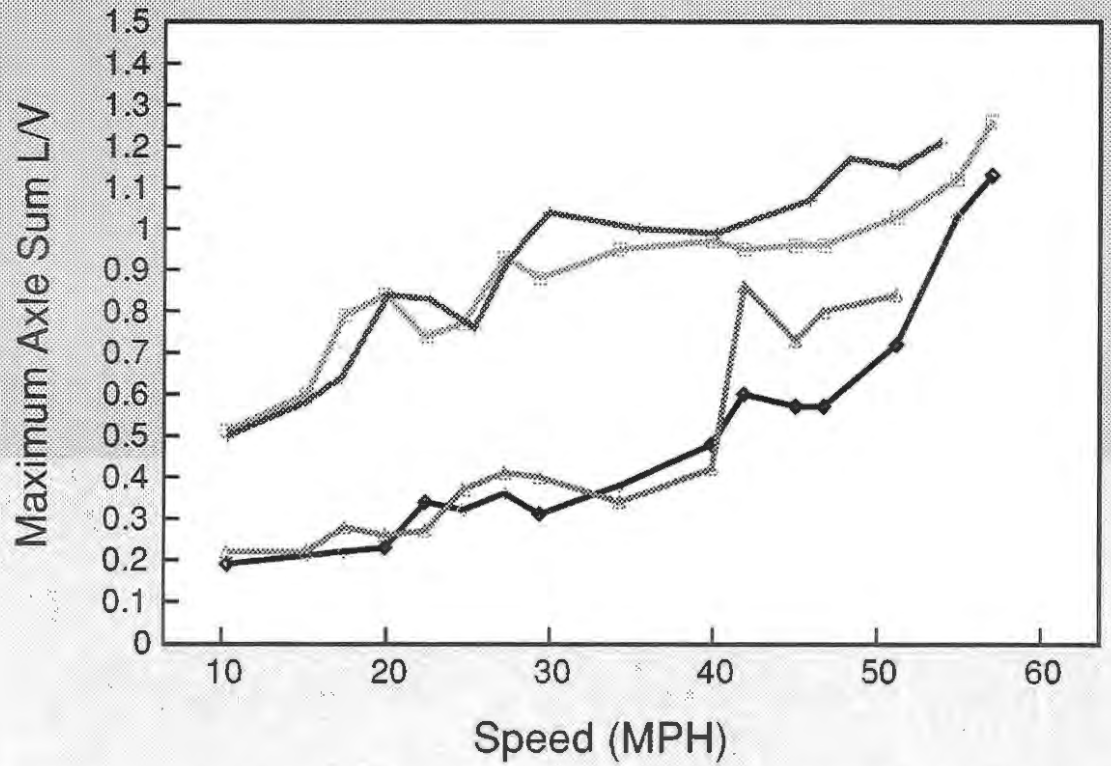


hsf1wear/Y&S03.cdr

Track Test

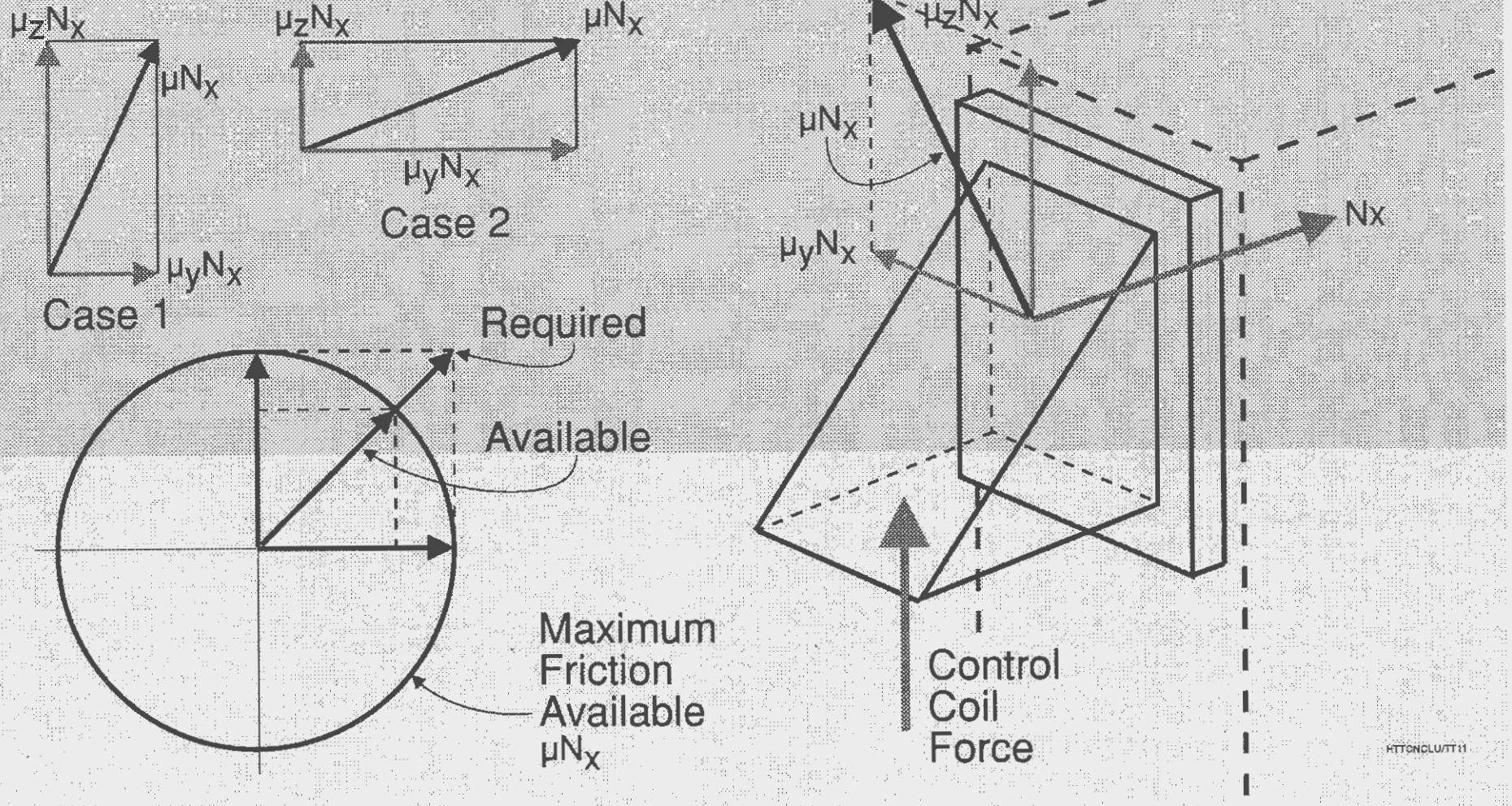
Yaw and Sway Track Test

Comparison of Maximum L/V, Axle Sum



hsf1wear/Y&S04.cdr

Track Test


$\mu_z N_x$
μN_x
$\mu_y N_x$
Case 1
$\mu_z N_x$
μN_x
$\mu_y N_x$
Case 2
Required
Available
Maximum
Friction
Available
μN_x
$\mu_z N_x$
μN_x
$\mu_y N_x$
Nx
Control
Coil
Force
HTTONCLU/TT11

Track Test

2.3.1 Truck Maintenance Guidelines

Track Test -- Conclusions (continued)

- Twist and Roll
  Same as Pitch and Bounce Except:
  Shimmed 0.4 Wedge Rise Truck (i.e. Zero Wedge Rise) Passed
- Curving and Yaw & Sway
  - No Measureable Difference in Performance
- Similar Results were Obtained with Instrumented Truck
- Conclusion
  Chapter XI Perturbations are too Severe
  to Measure/Quantify Differences in Performance

HTTCNCLU/TT10A

Track Test

2.3.1 Truck Maintenance Guidelines

Track Test -- Conclusions

- Hunting
  - Best to Worst Case Separated by 15 mph
- Pitch and Bounce
  - No Measurable Difference in Performance
  - More Reserve Travel Remaining for Best Trucks, Off-Resonance Speeds

Modeling Results

Track Test vs Modeling Results

Modeling Results

NUCARS Modeling

- NUCARS (New and Untried Car Analytic Regime Simulation)
- Full multi-body dynamic simulation model
- Car, truck components and wheel sets are modeled as lumped masses with assigned degrees of freedom (DOF)
  - 9 DOF's per body (x,y,z; roll, pitch, yaw; three bending modes)

Modeling Results

NUCARS Modeling (continued)

- Connectors are modeled as perfect springs and dampers
  - Hysteresis
  - Wheel/Rail
  - Parallel or series spring and damper
  - External force
  - Friction elements
- Inputs: Displacements or forces (distance or time-based) any track or measured shape

Modeling Results

NUCARS Modeling Assumptions

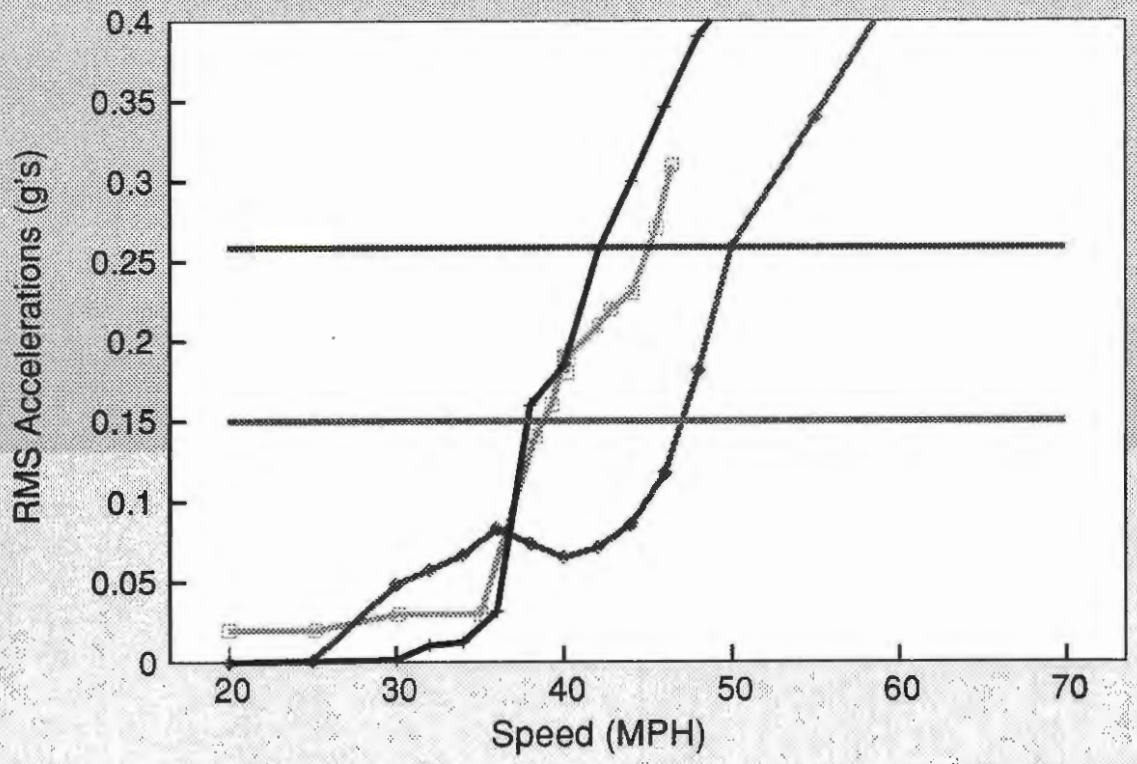
- Characteristics
  - MSU data -- Vertical, lateral and warp stiffness & damping
  - Hand excitation -- Inertias, C.G.
  - Theoretical data
- Wheel/Rail
  - Measured wheels and rails
  - Measured friction levels from track test
- Center plate was lubricated ($\mu = 0.15$)

Note: Car body was modeled as a rigid body (no flexible modes)

Modeling Results

Hunting Test vs Modeling Results Wedge Rise 0.0

Compare Car Body Lateral Acceleration, A-End



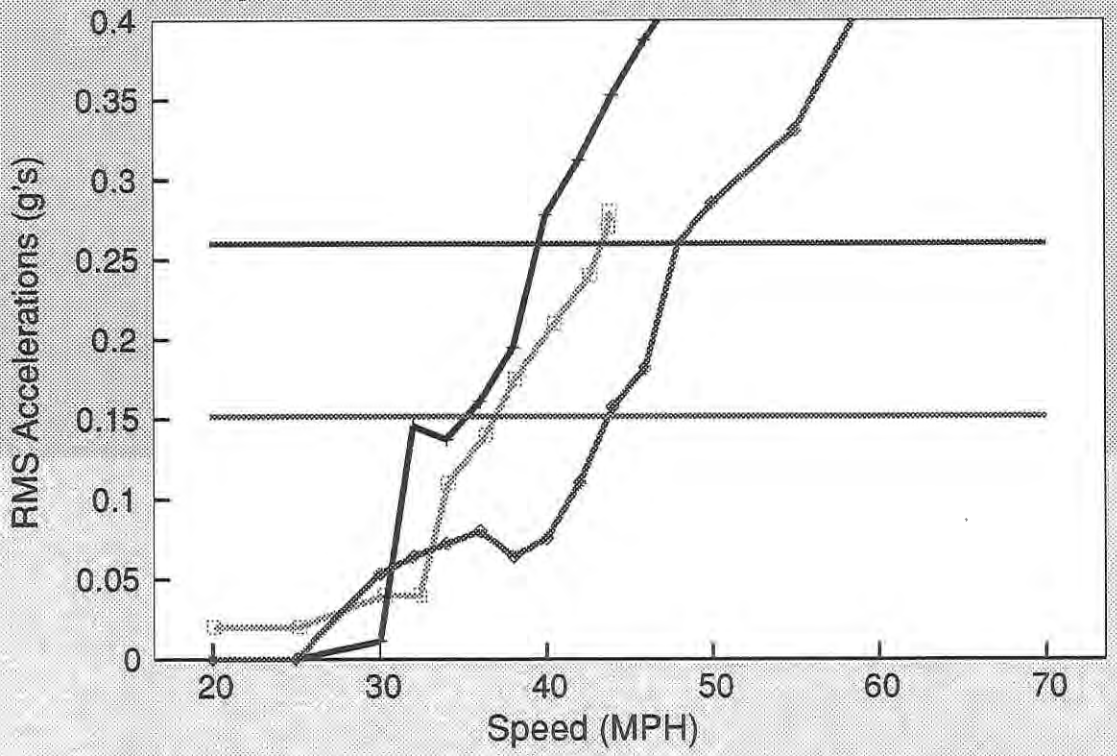
□ Track Test Worn Wheel MU=0.56 + Model Worn Wheel MU=0.5 ◇ Model AAR-1B MU=0.5 — Onset — Chapter XI Limit

hsf1wear/H03.cdr/M02.cdr

Modeling Results

Hunting Test vs Modeling Results Wedge Rise 0.5

Compare Car Body Lateral Acceleration, A-End



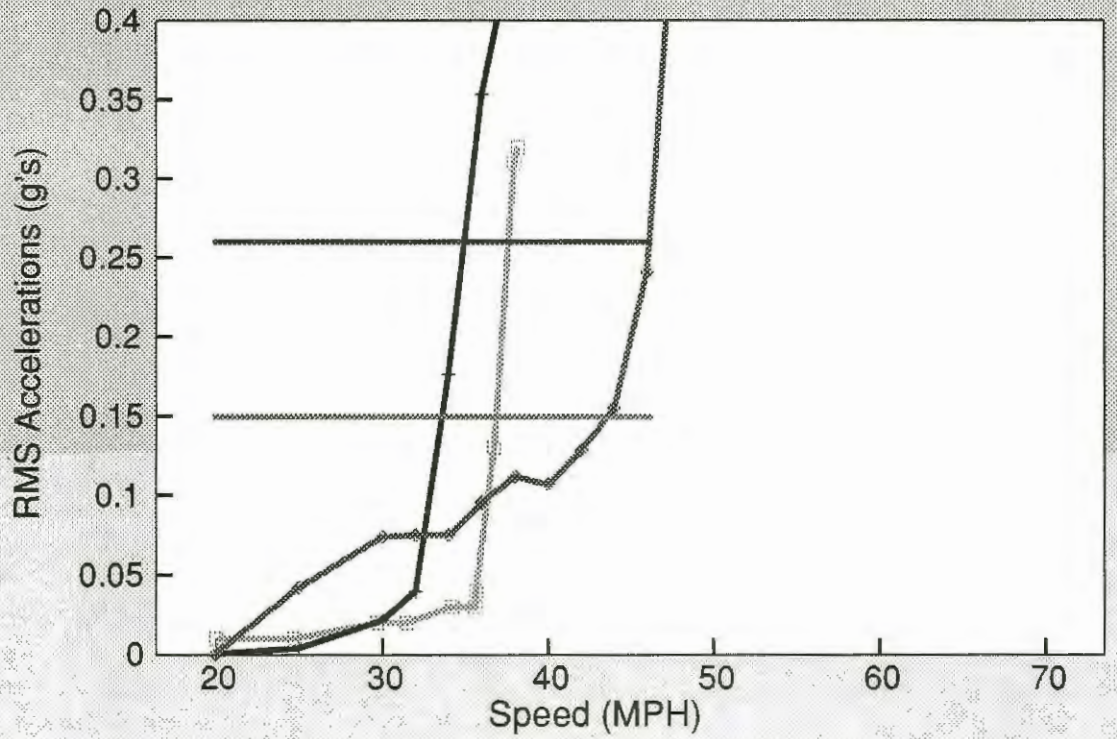
□ Track Test Worn Wheel MU=0.56 + Model Worn Wheel MU=0.5 ◇ Model AAR-1B MU=0.5 —Onset —Chapter XI Limit

hsf1wear/H04.cdr/M03.cdr

Modeling Results

Hunting Test vs Modeling Results Wedge Rise 1.5

Compare Car Body Lateral Acceleration, A-End



□ Track Test Worn Wheel MU=0.43

+ Model Worn Wheel MU=0.5

◇ Model AAR-1B MU=0.5

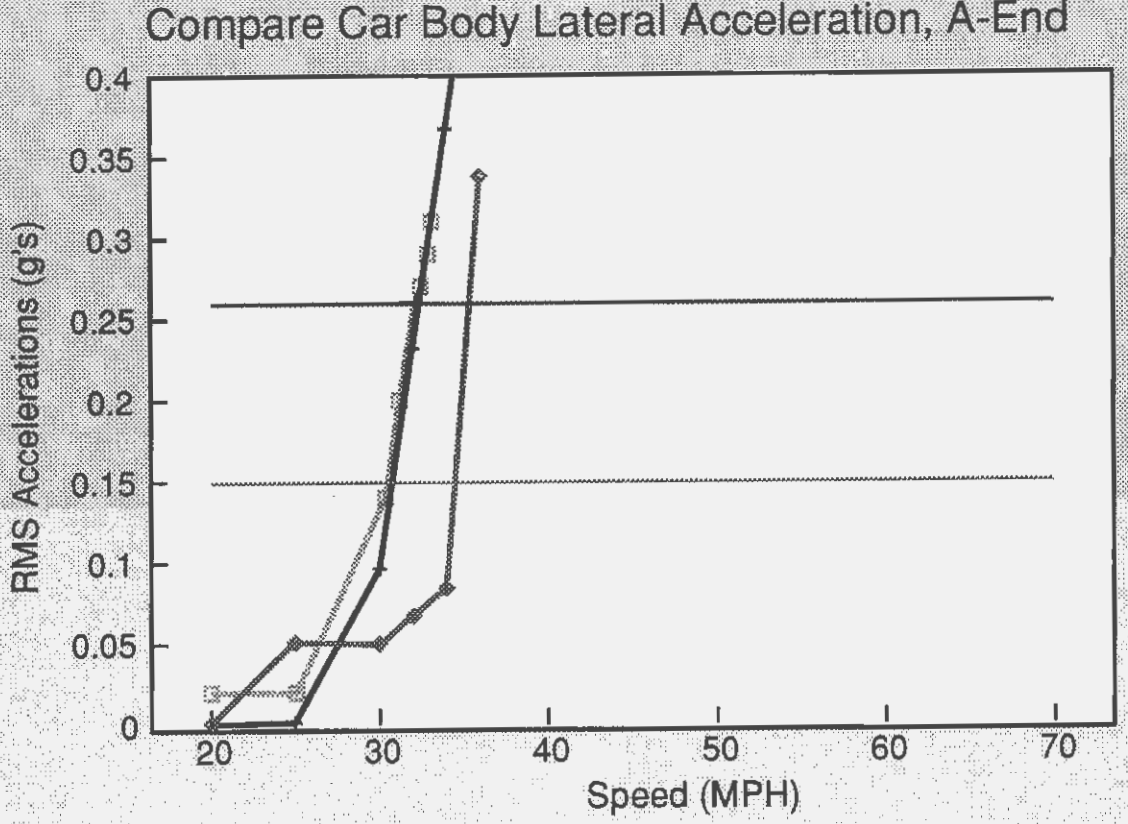
— Onset

— Chapter XI Limit

hsf1wear/H05.cdr/M04.cdr

Modeling Results

Hunting Test vs Modeling Results No Wedges

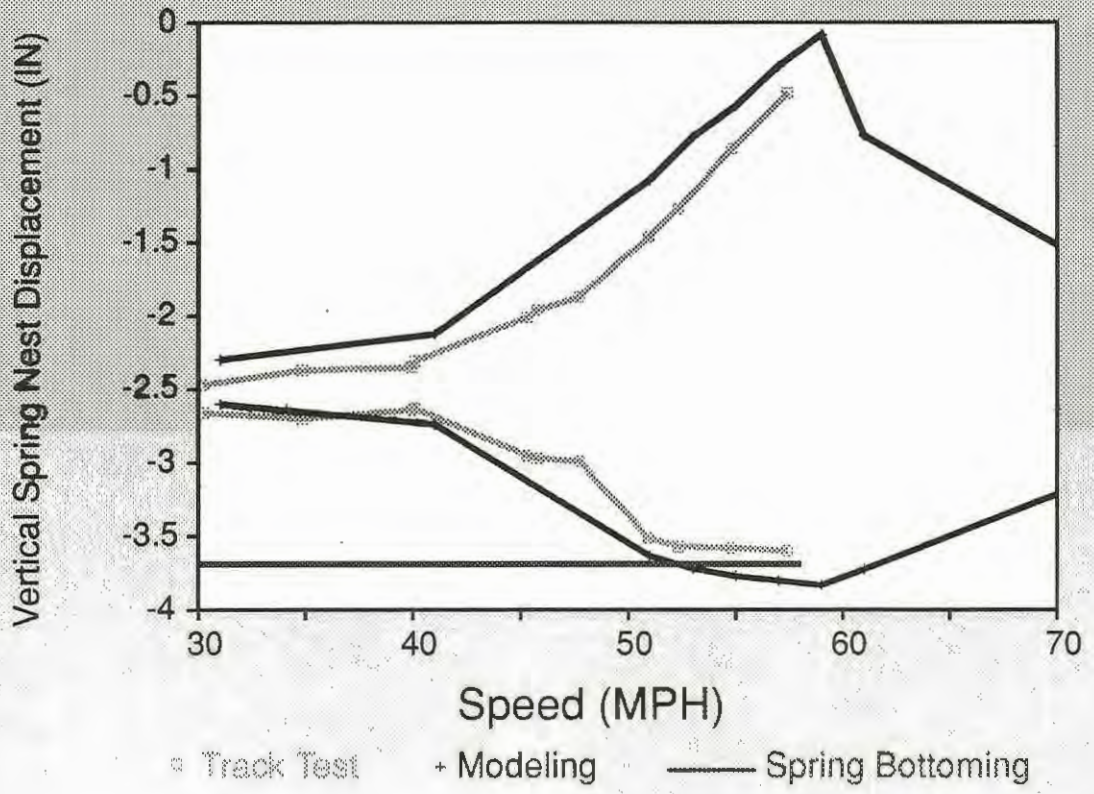


hsf1wear/H02.cdr/M05.cdr

Modeling Results

Pitch and Bounce Test vs Modeling Results Wedge Rise 0.0

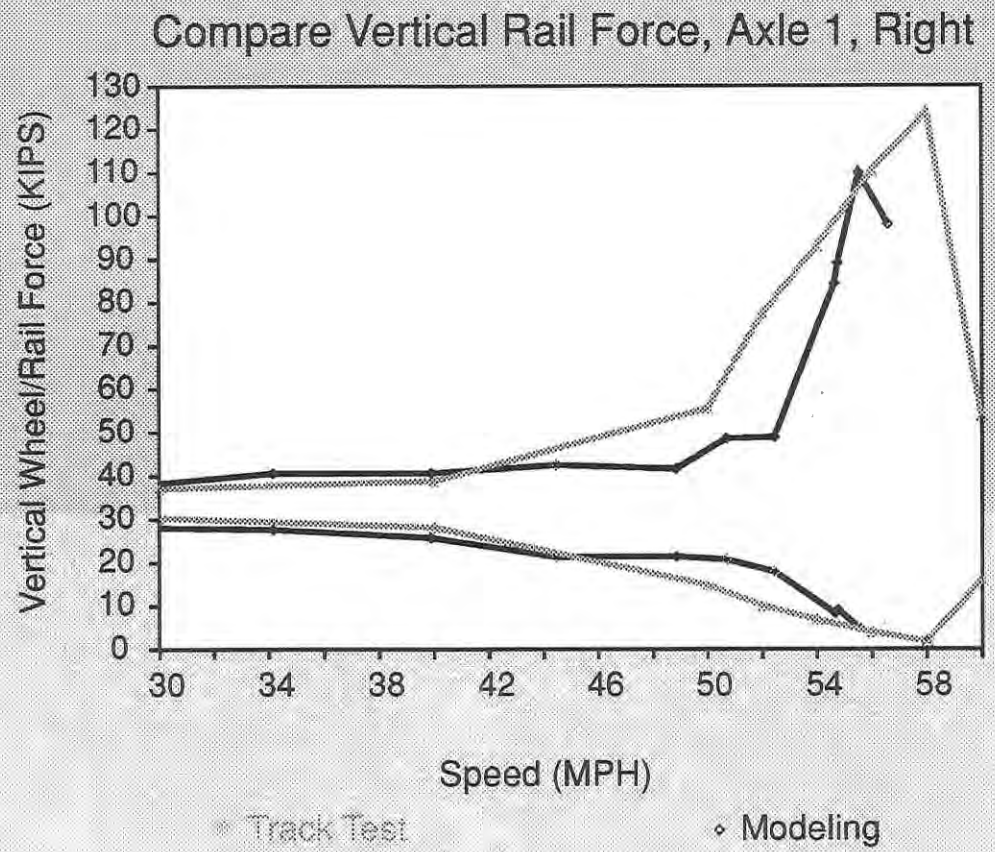
Compare Vertical Spring Displacement, Lead Left



H07.cdr/M06.cdr

Modeling Results

Pitch and Bounce Test vs Modeling Results Wedge Rise 0.5

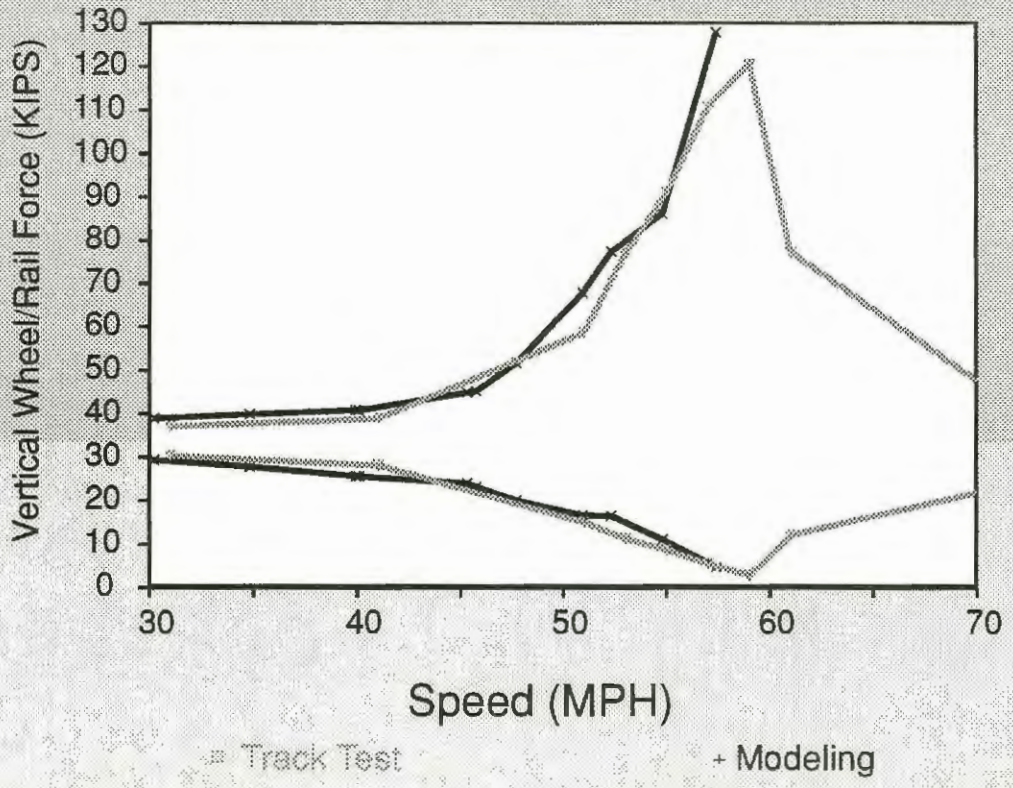


H09.cdr/M07.cdr

Modeling Results

Pitch and Bounce Test vs Modeling Results Wedge Rise 1.5

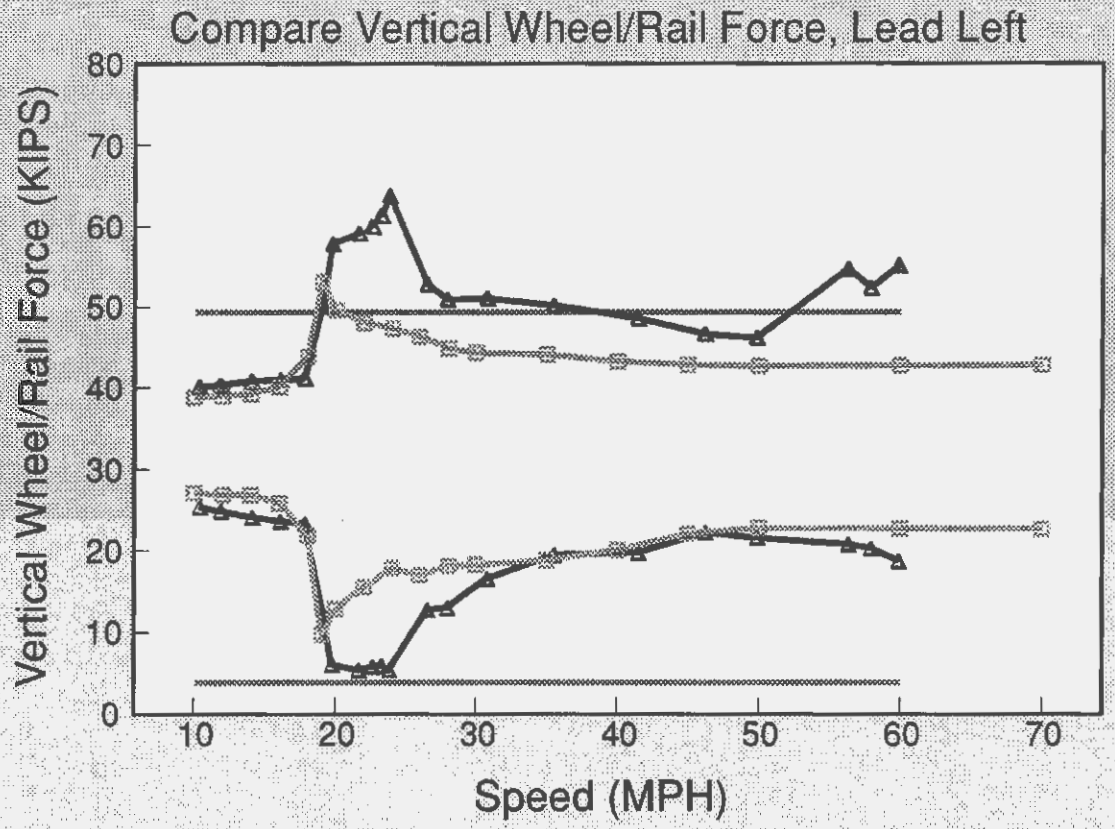
Compare Vertical Spring Force, Axle 1, Right



hsf1wear/H08.cdr/M08.cdr

Modeling Results

Twist and Roll Track Test vs Modeling Results

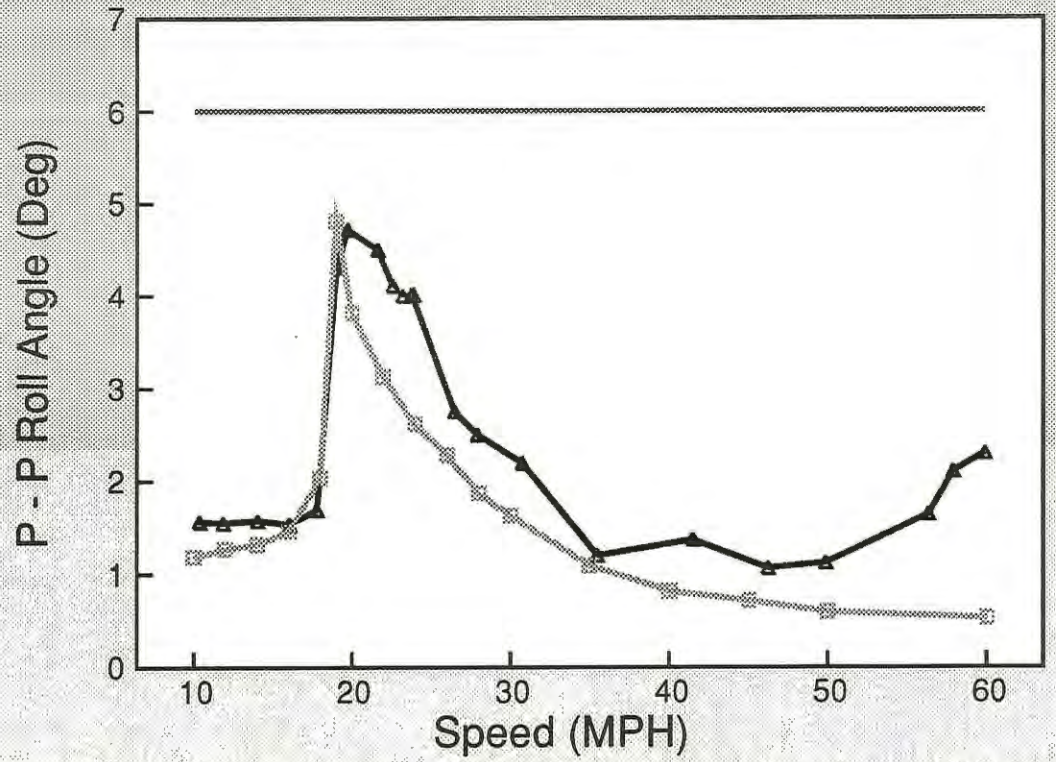


T&R08.cdr/M10.cdr

Modeling Results

Twist and Roll Track Test vs Modeling Results

Compare Peak-to-Peak Roll Angle A - End

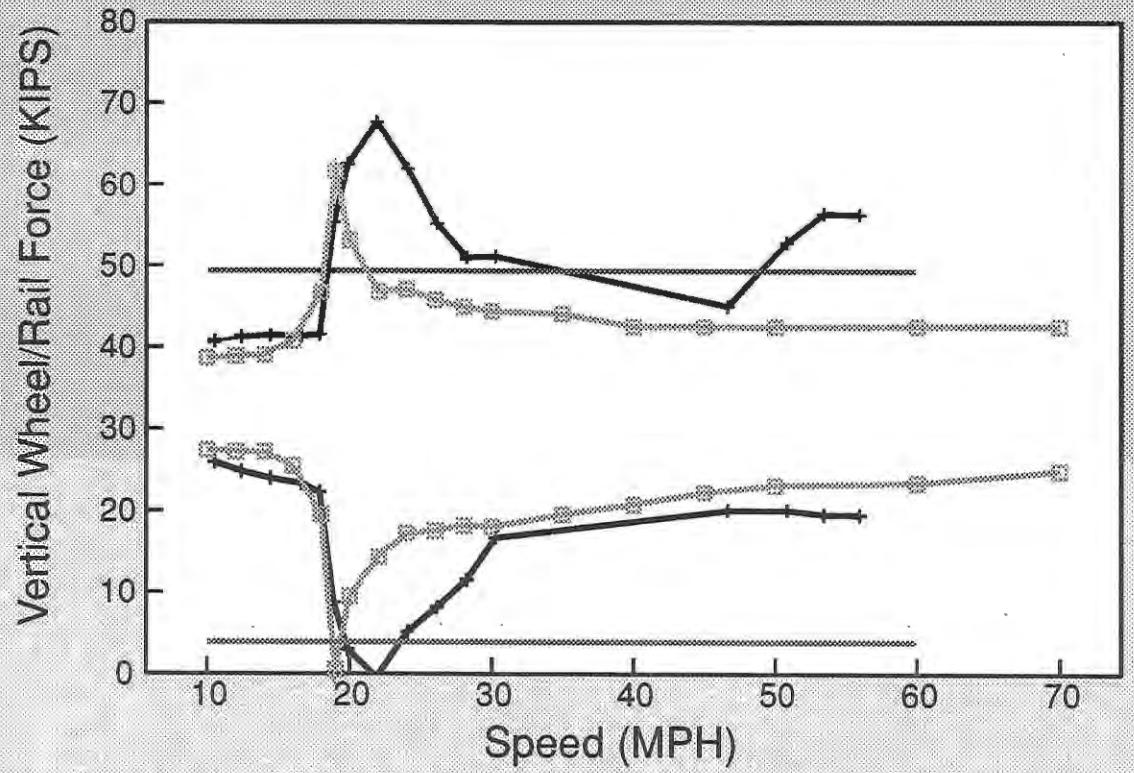


hsf1wear/T&R04.cdr/M11.cdr

Modeling Results

Twist and Roll Track Test vs Modeling Results

Compare Vertical Wheel/Rail Force, Lead Left



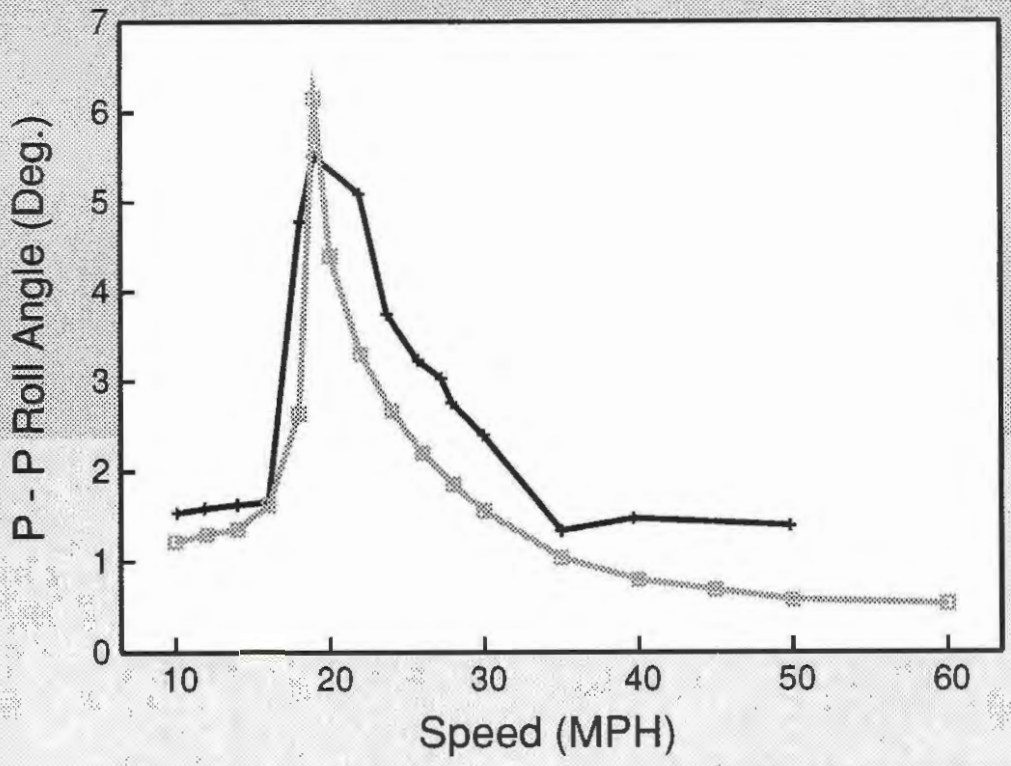
= Model Wedge Rise 0.5 + Track Test Wedge Rise 0.5 —— Chapter XI 10% Limit —— Proposed 1.5 DLF

T&R06.cdr/M12.cdr

Modeling Results

Twist and Roll Track Test vs Modeling Results

Compare Peak-to-Peak Roll Angle A-End

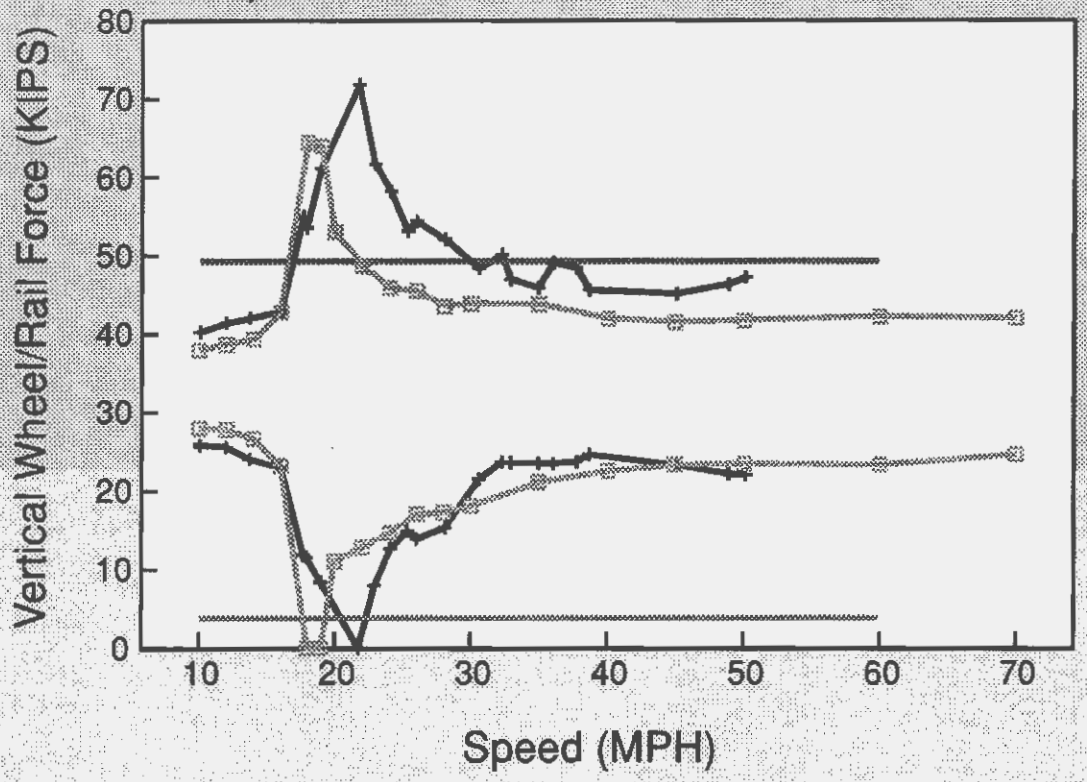


hsf1wear/T&RO2/M13.cdr

Modeling Results

Twist and Roll Track Test vs Modeling Results

Compare Vertical Wheel/Rail Force, Lead Left

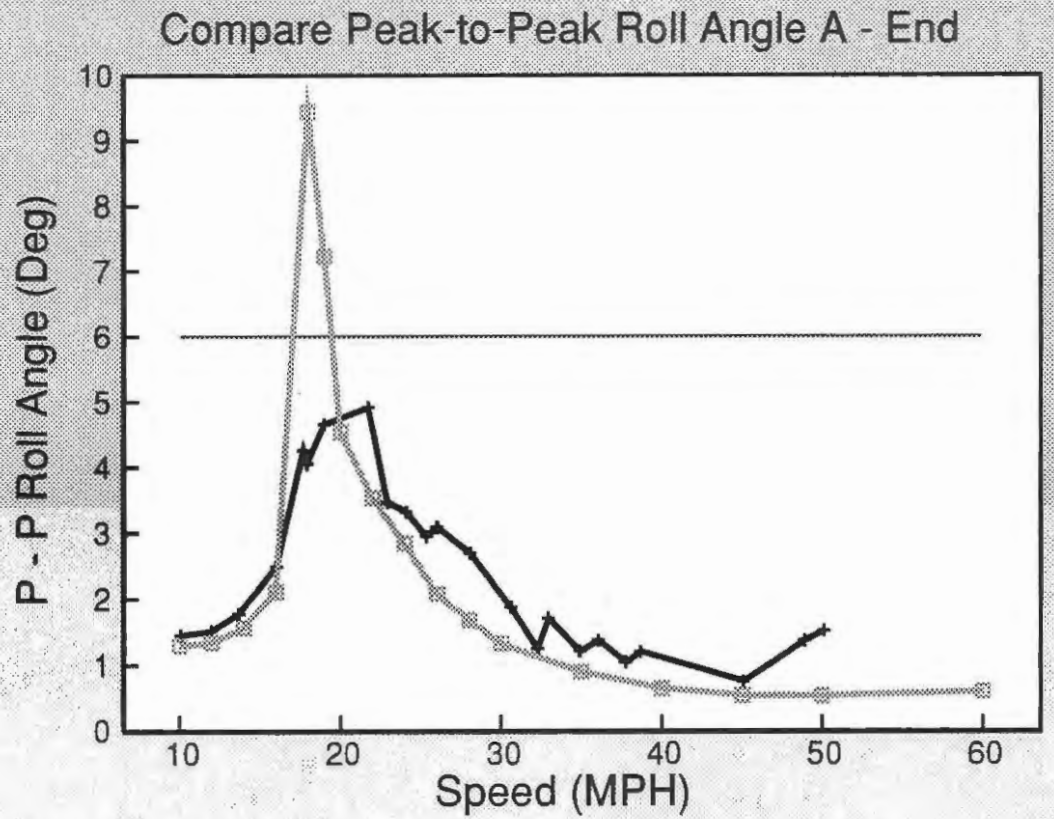


○ Model Wedge Rise 1.5 + Track Test Wedge Rise 1.5 —— Chapter XI 10% Limit —— Proposed 1.5 DLF

T&R07.cdr/M14.cdr

Modeling Results

Twist and Roll Track Test vs Modeling Results

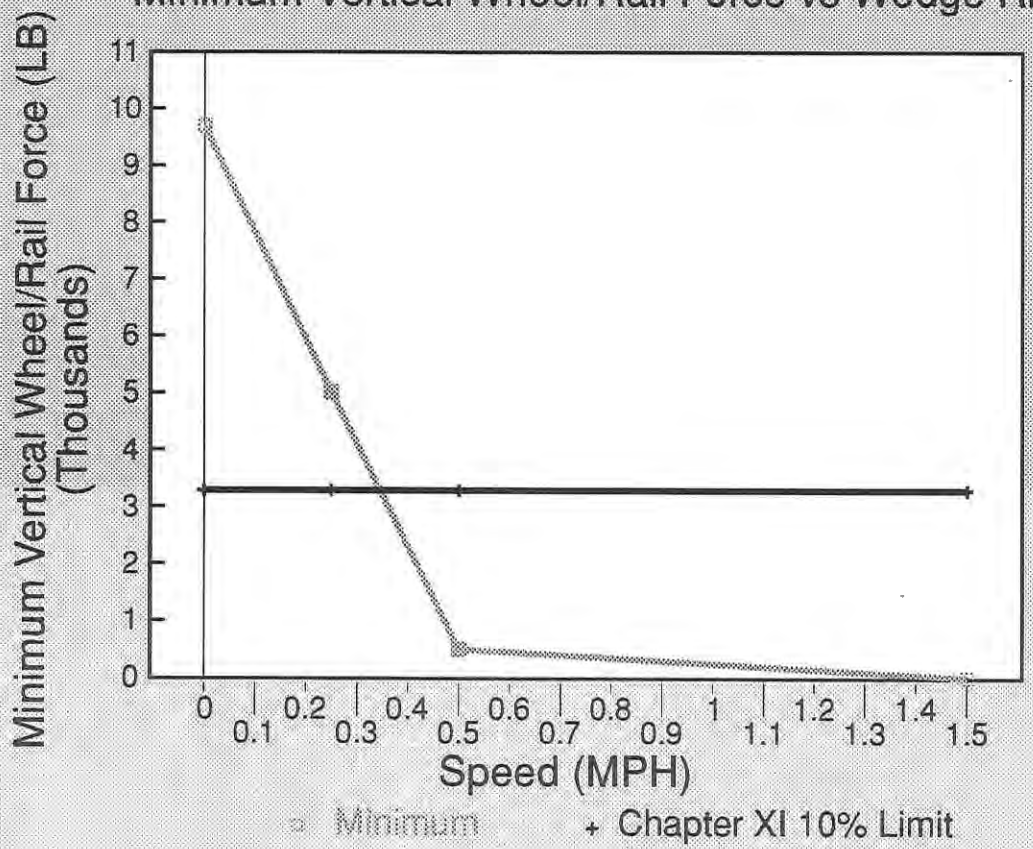


hsf1wear/T&R03.cdr/M15.cdr

Modeling Results

Twist and Roll Modeling Results

Minimum Vertical Wheel/Rail Force vs Wedge Rise

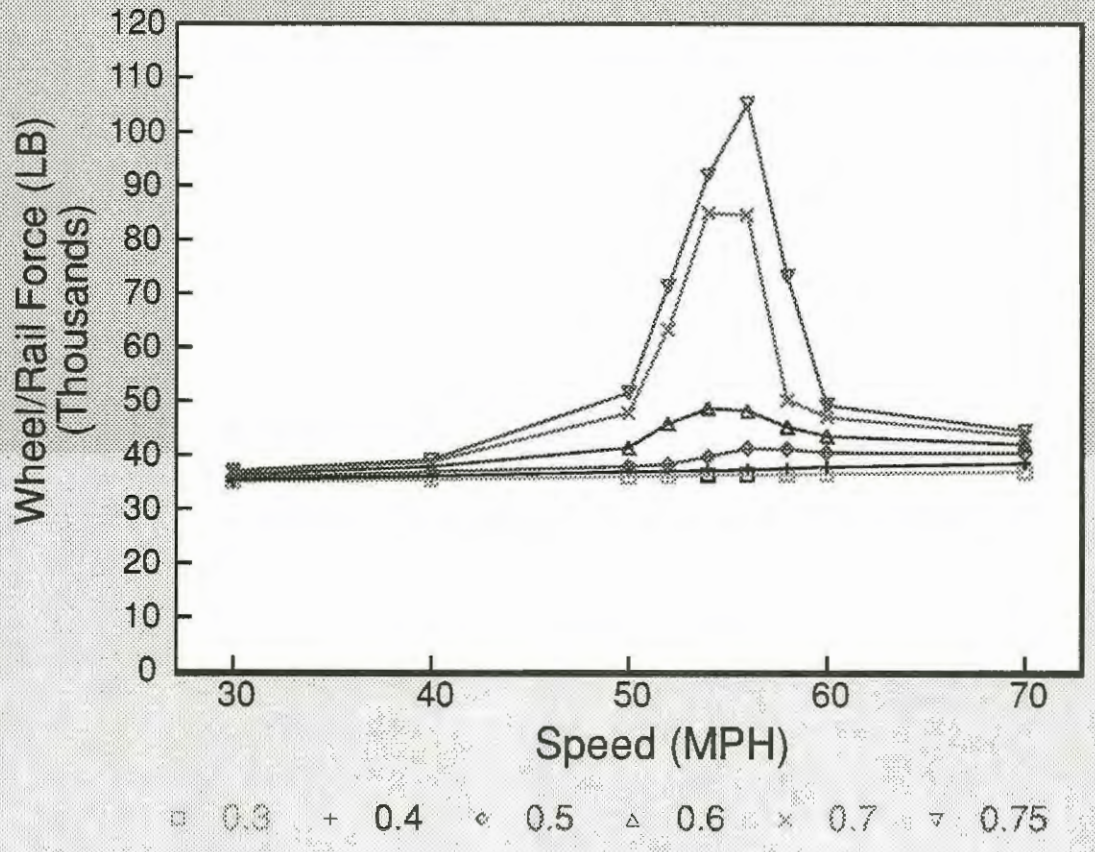


hsf1wear/T&R05.cdr/M16.cdr

Modeling Results

Pitch and Bounce Modeling Results Wedge Rise 0.0

Wheel/Rail Forces vs Speed and Amplitude

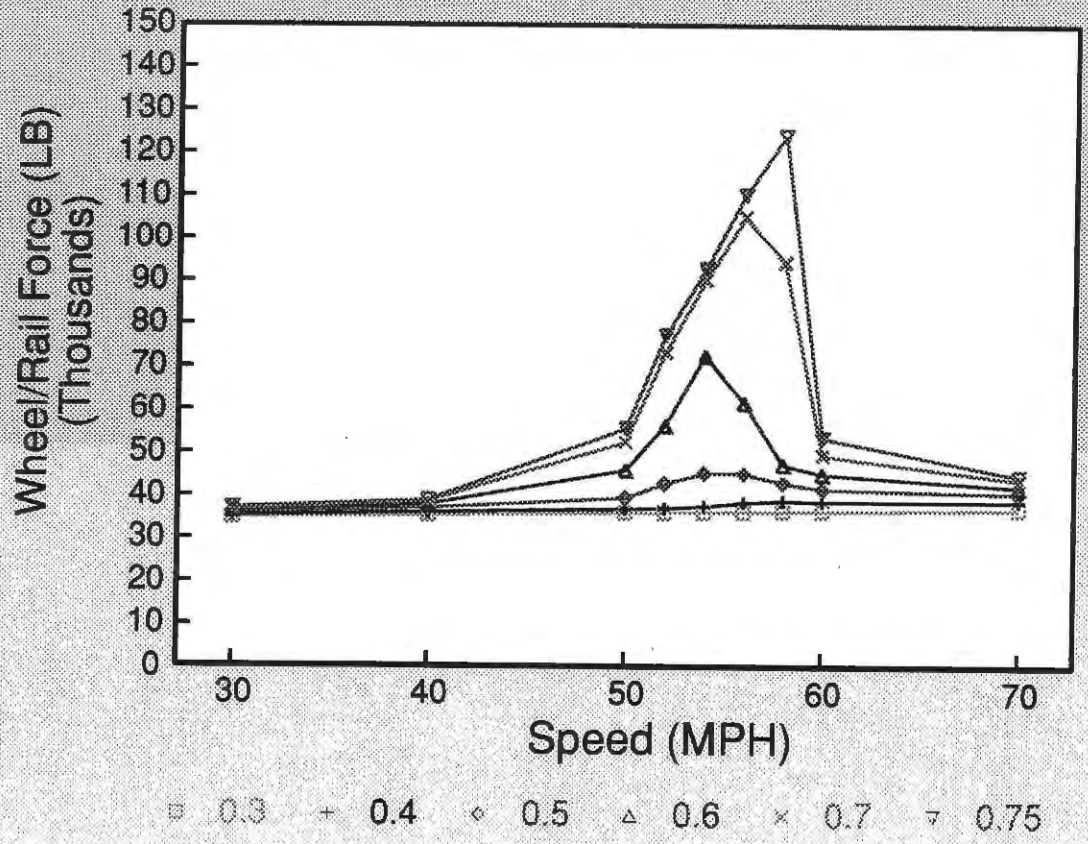


hsf1wear/P&B02.cdr/M17.cdr

Modeling Results

Pitch and Bounce Modeling Results Wedge Rise 0.5

Wheel/Rail Forces vs Speed and Amplitude

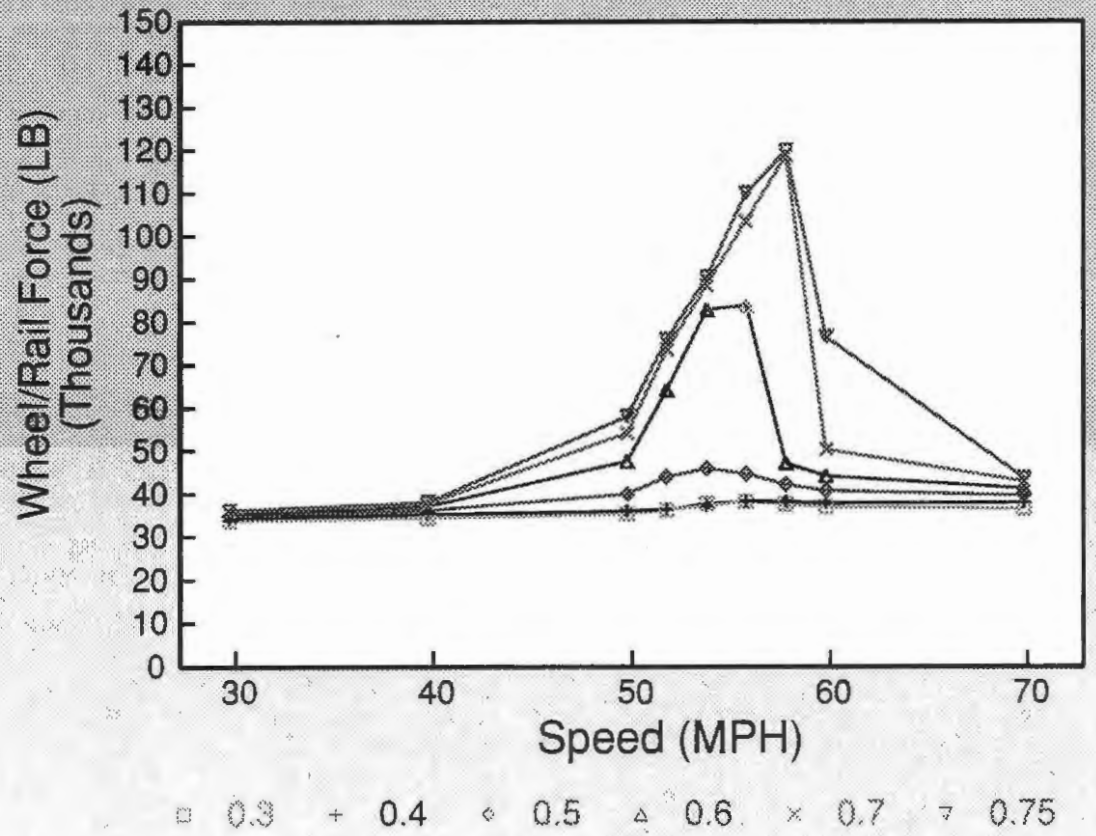


hsf1wear/P&B03.cdr/M18.cdr

Modeling Results

Pitch and Bounce Modeling Results Wedge Rise 1.5

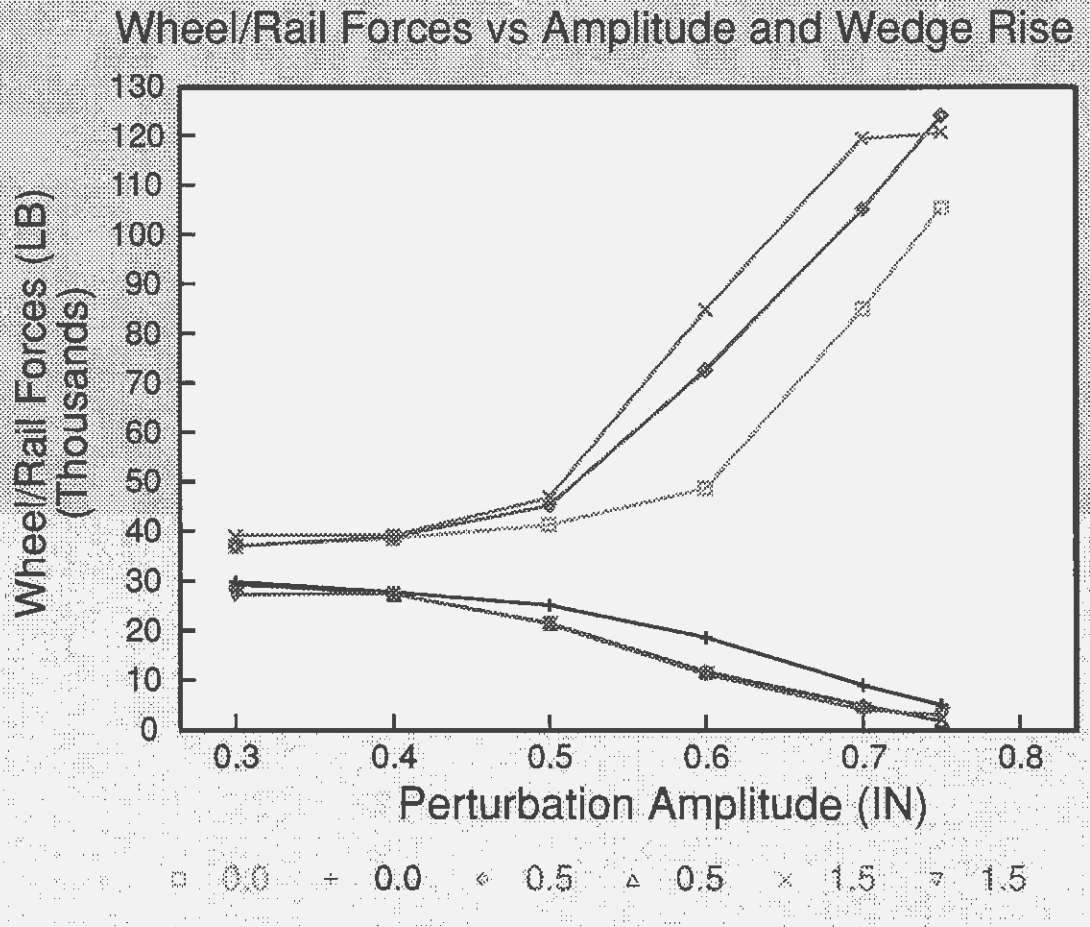
Wheel/Rail Forces vs Speed and Amplitude



hsf1wear/P&B04.cdr/M19.cdr

Modeling Results

Comparison of Pitch and Bounce Modeling Results

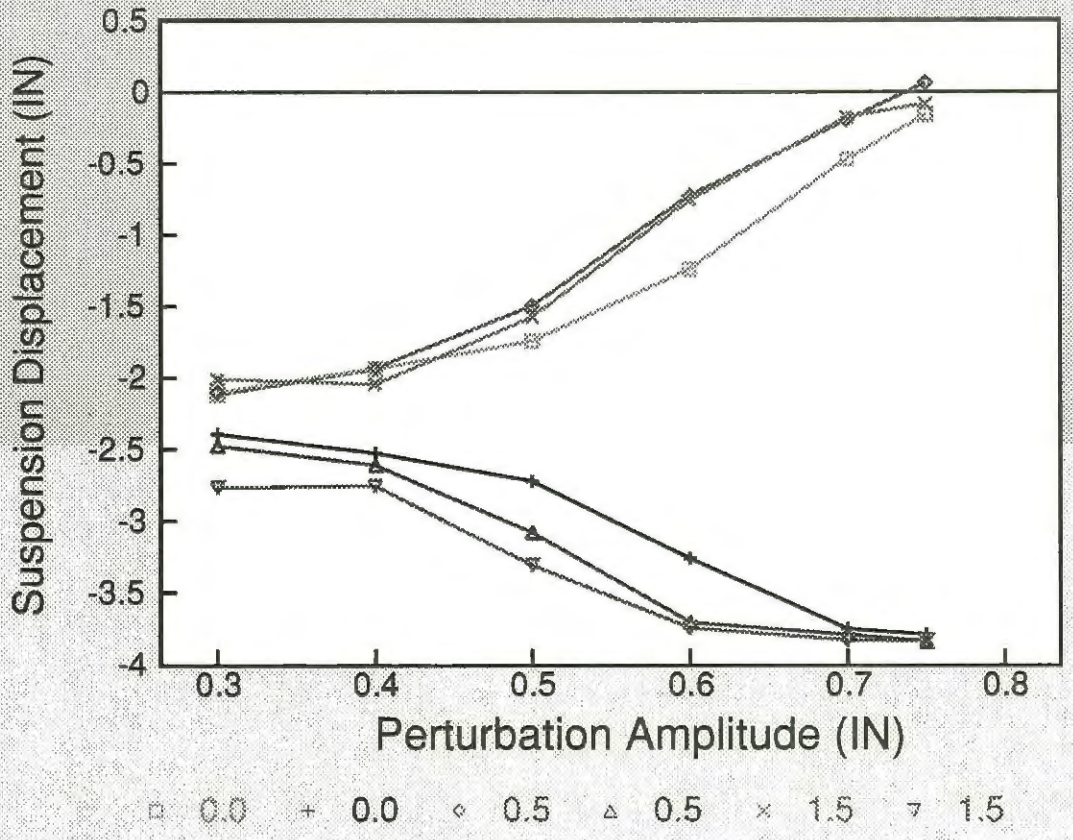


hsf1wear/P&B06.cdr/M20.cdr

Modeling Results

Comparison of Pitch and Bounce Modeling Results

Suspension Displacement vs Amplitude and Wedge Rise

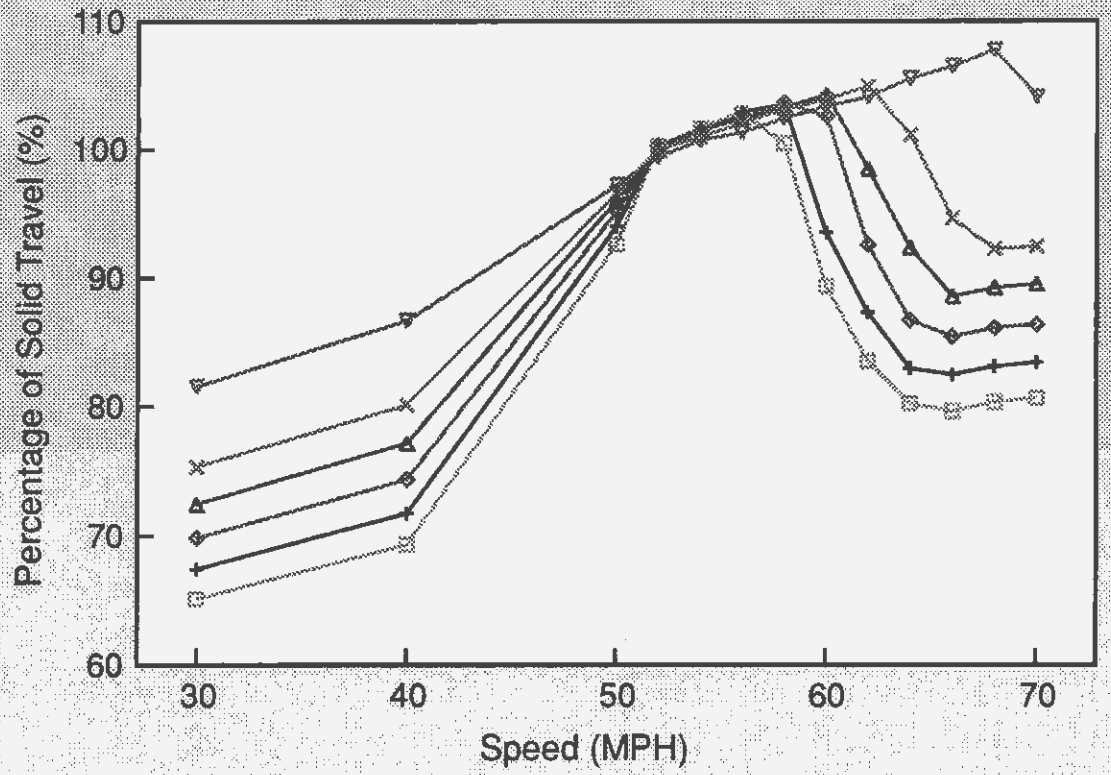


hsf1wear/P&B05.cdr/M21.cdr

Modeling Results

Pitch and Bounce Modeling Results 0.75 Amplitude

Percent Suspension Travel vs Speed, Loss of Travel, and Wedge Rise



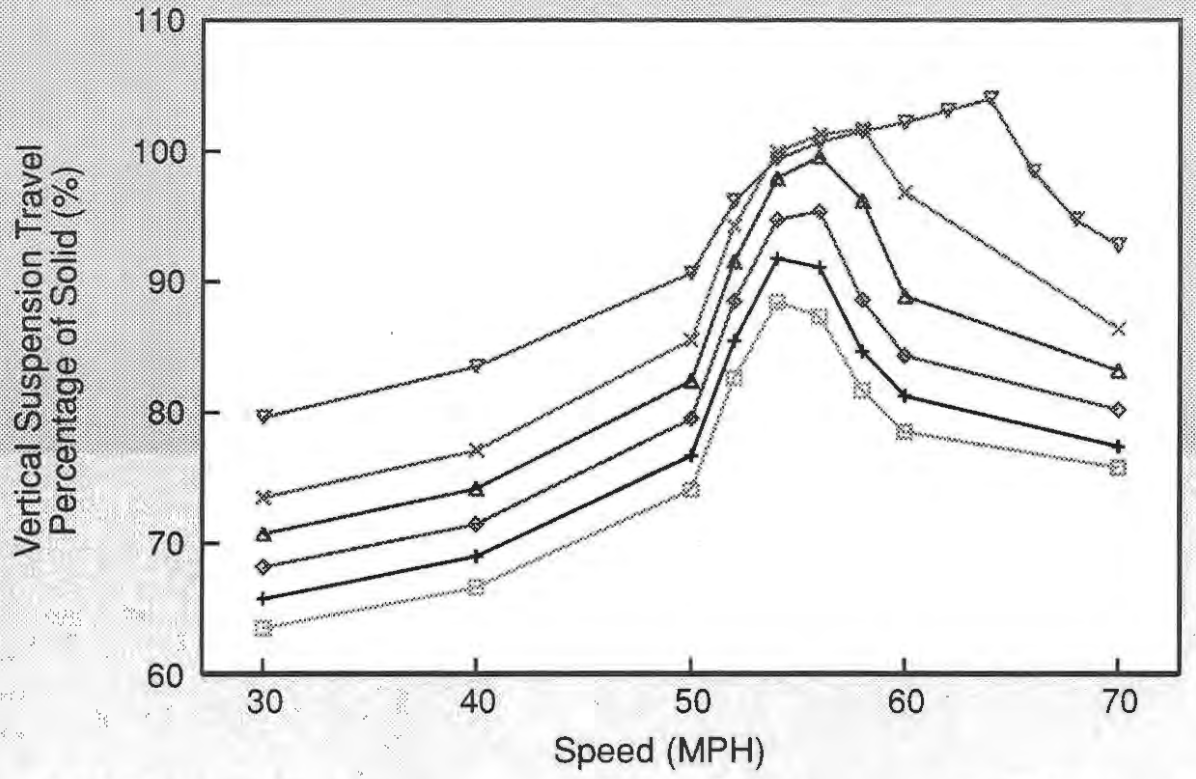
□ 0.0 + 0.25 (3% Loss) ◇ 0.50 (7% Loss) △ 0.75 (10% Loss) × 1.0 (13.5% Loss) ▽ 1.5 (20% Loss)

hsf1wear/P&B09.cdr/M23.cdr

Modeling Results

Pitch and Bounce Modeling Results 0.6 Amplitude

Percent Suspension Travel vs Speed, Loss of Travel, and Wedge Rise

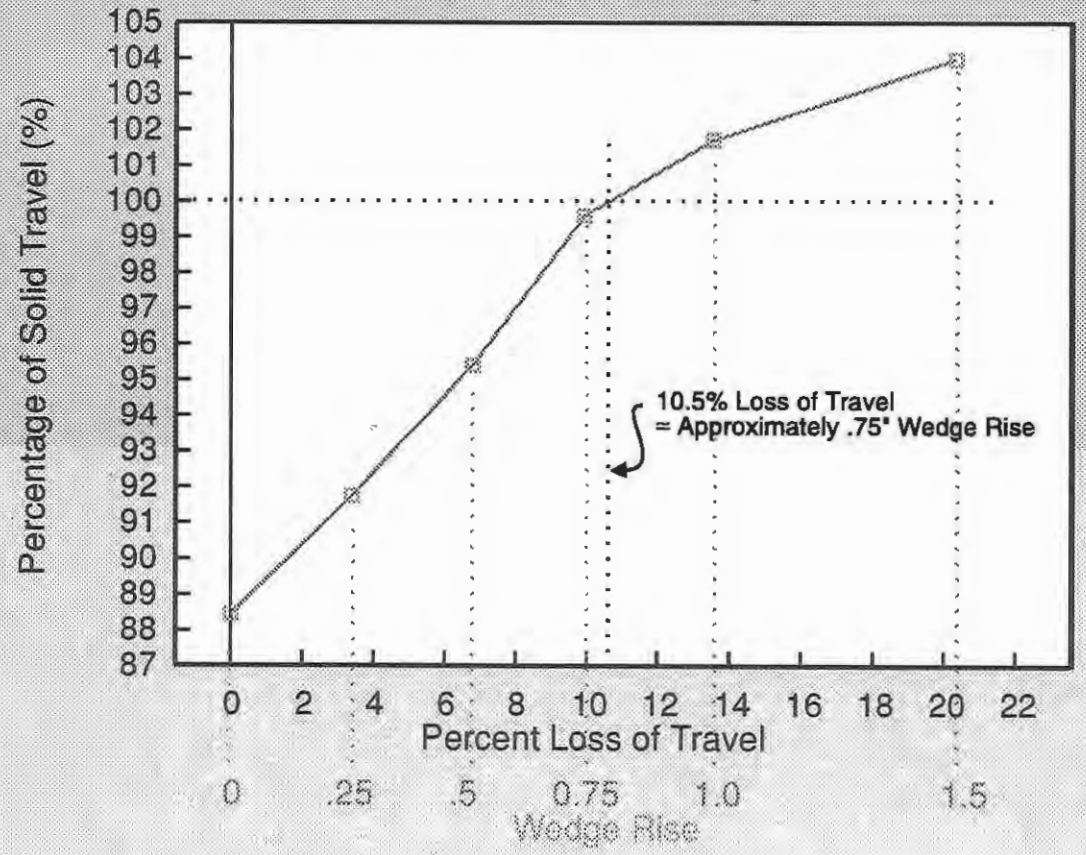


□ 0.0 + 0.25 (3% Loss) ◇ 0.50 (7% Loss) △ 0.75 (10% Loss) × 1.0 (13.5% Loss) ▽ 1.5 (20% Loss)

hsf1wear/P&B08.cdr/M22.cdr

Modeling Results

Pitch and Bounce Modeling Results 0.6 Amplitude

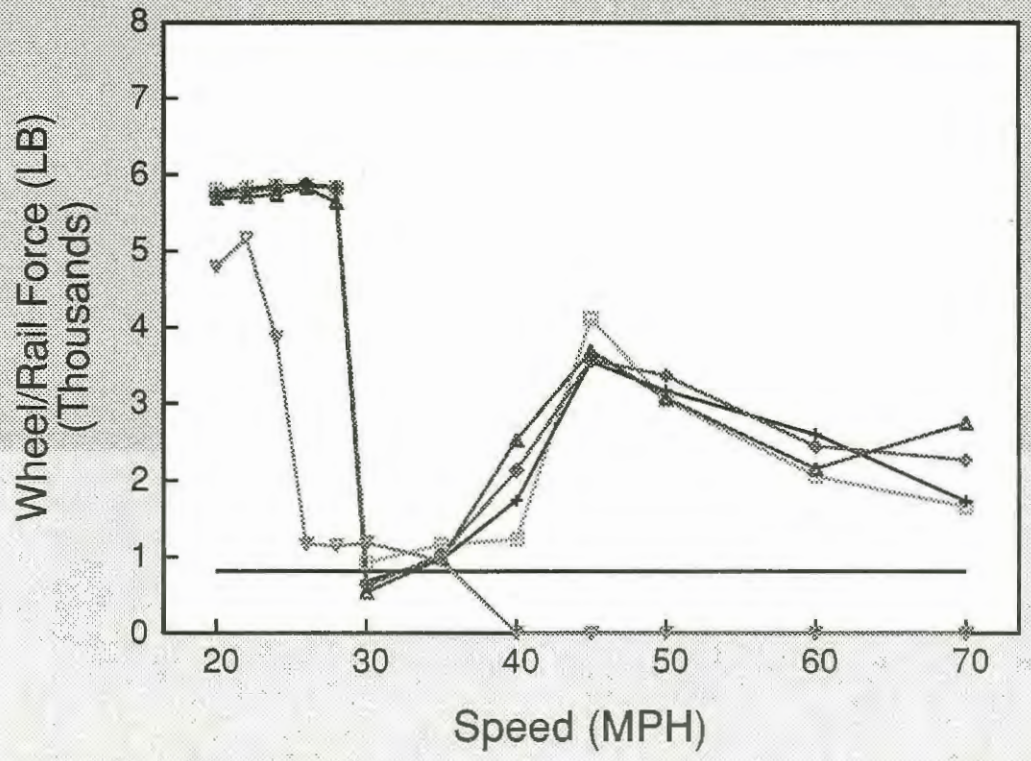


hsf1wear/P&B07.cdr/M24.cdr

Modeling Results

Empty Twist and Roll Modeling Results

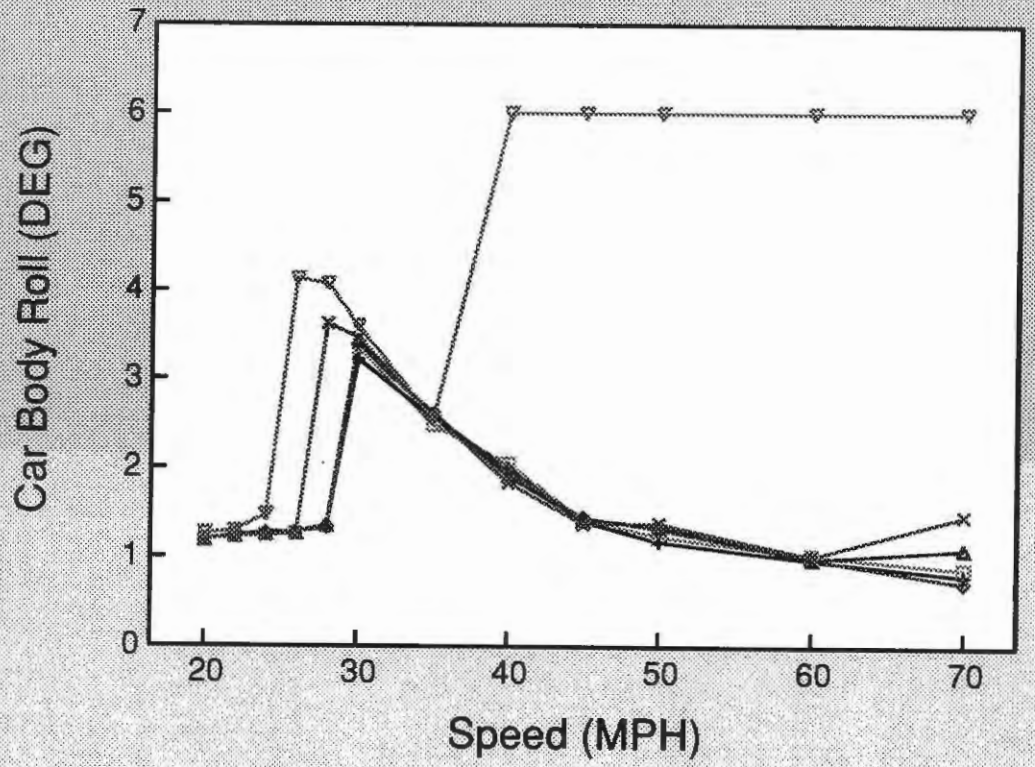
Wheel/Rail Forces vs Speed



T&R12.cdr/M29.cdr

Modeling Results

Empty Twist and Roll Modeling Results

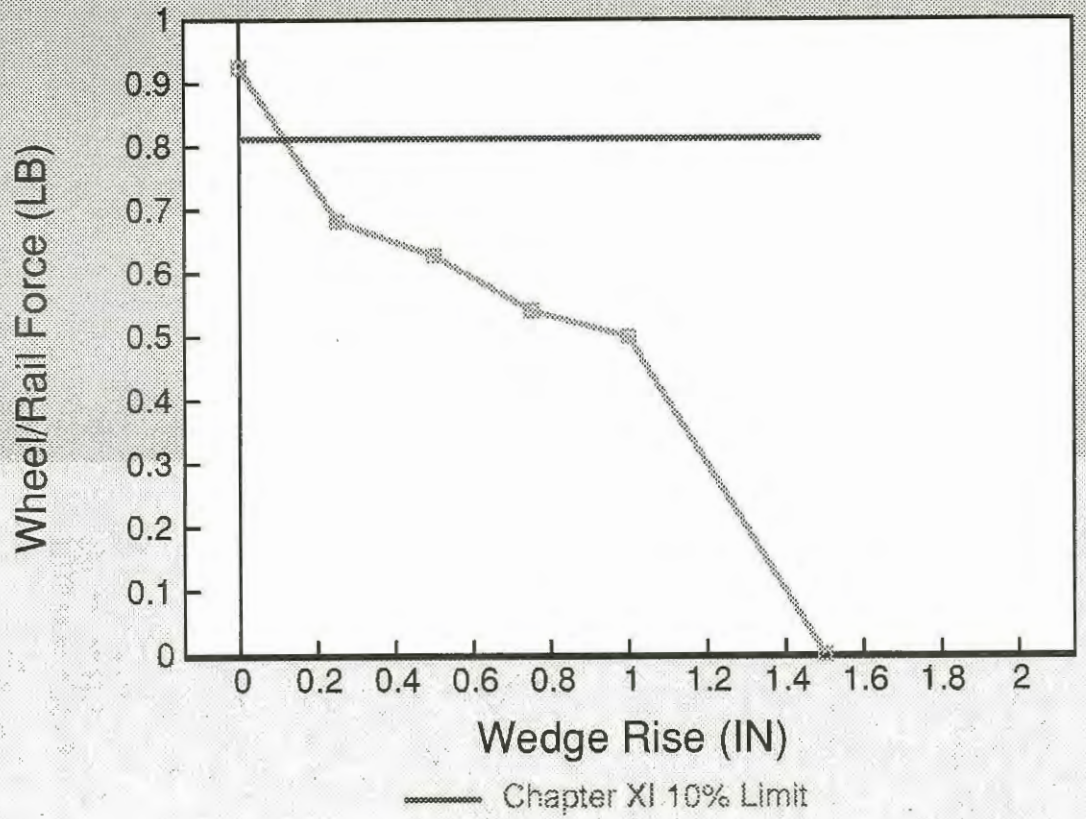


hsf1wear/T&R12/M30.cdr

Modeling Results

Empty Twist and Roll Modeling Results

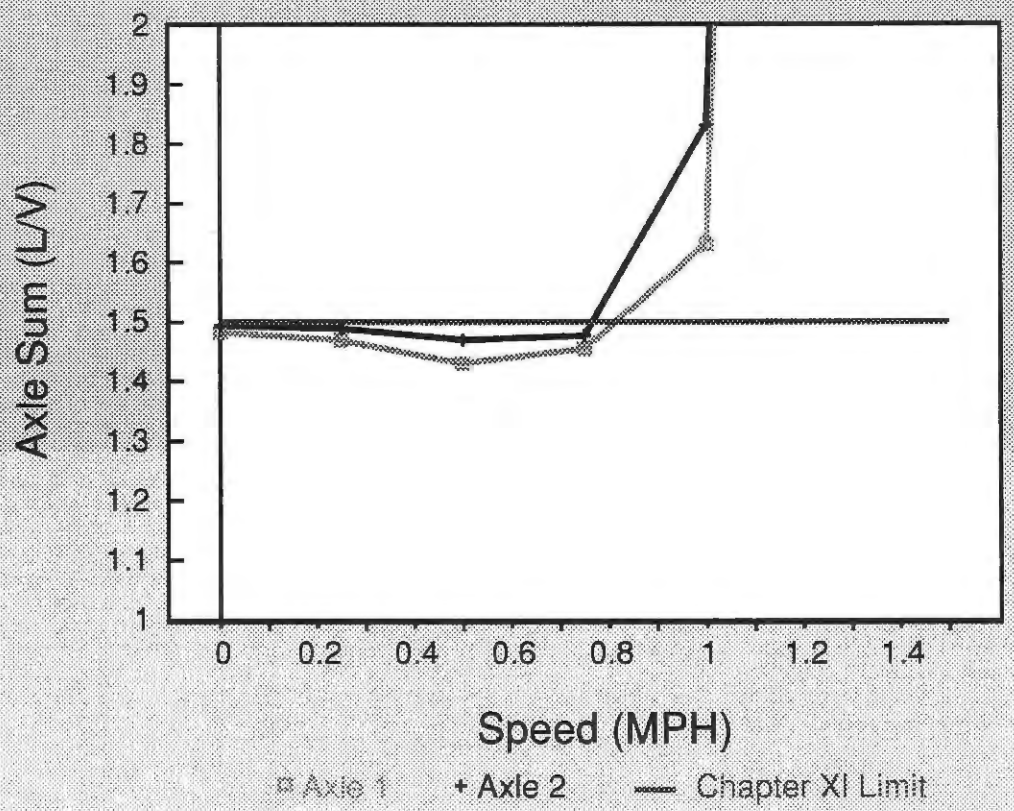
Minimum Wheel/Rail Forces vs Wedge Rise



T&R09.cdr/M31.cdr

Modeling Results

Empty Twist and Roll Modeling Results



T&R11.cdr/M32.cdr

Modeling Results

Modeling Conclusions

- Modeling vs test results compare reasonably well
  - Wedge model in NUCARS requires improvements
- Hunting results show slight improvement using AAR 1B wheel profile
- Pitch and Bounce results obtained with 0.6 amplitude perturbations show:
  - 0 to 0.4 inches of wedge rise pass
  - 0 to 10 percent loss of travel (0 to 0.75 inches of wedge rise) pass

Modeling Results

Modeling Conclusions (continued)

- Twist and Roll (L) results obtained with 0.75 amplitude perturbations show:
  - 0 to 0.35 inches of wedge rise pass
- Twist and Roll (E) results show:
  - 0 to 0.1 inches of wedge rise pass
- Final conclusion:
  - Friction wedge provides insufficient damping for Chapter XI type inputs



MR01.cdr

Inspection Measurements

Inspections/Measurements

Center Plates

A) Diameter
B) Depth

Bolster

A) Gib Spacing
B) Pocket Wear
   1) Slope wear
   2) Slope-to-slope spacing
   3) Lateral clearance
   4) Hardness

IM02.cdr

Inspection Measurements

Inspections/Measurements (continued)

Wedges

A) Vertical contour
B) Lateral contour
C) Hardness

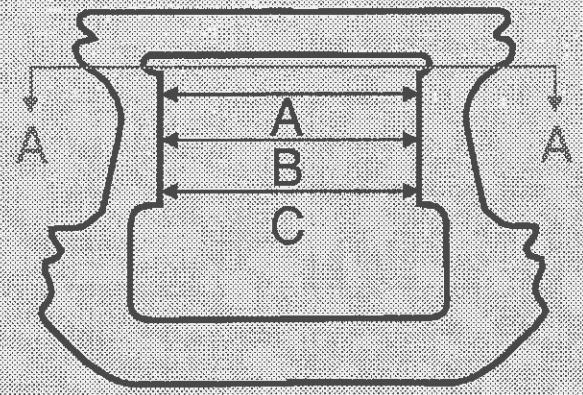
Side Frame

A) Side frame to side frame tolerances
B) Side frame wear plate hardness
C) Pedestal wear

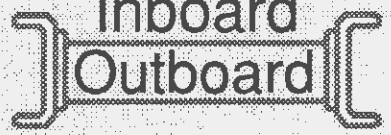
Did not measure Wheel Sets

Inspection Measurements

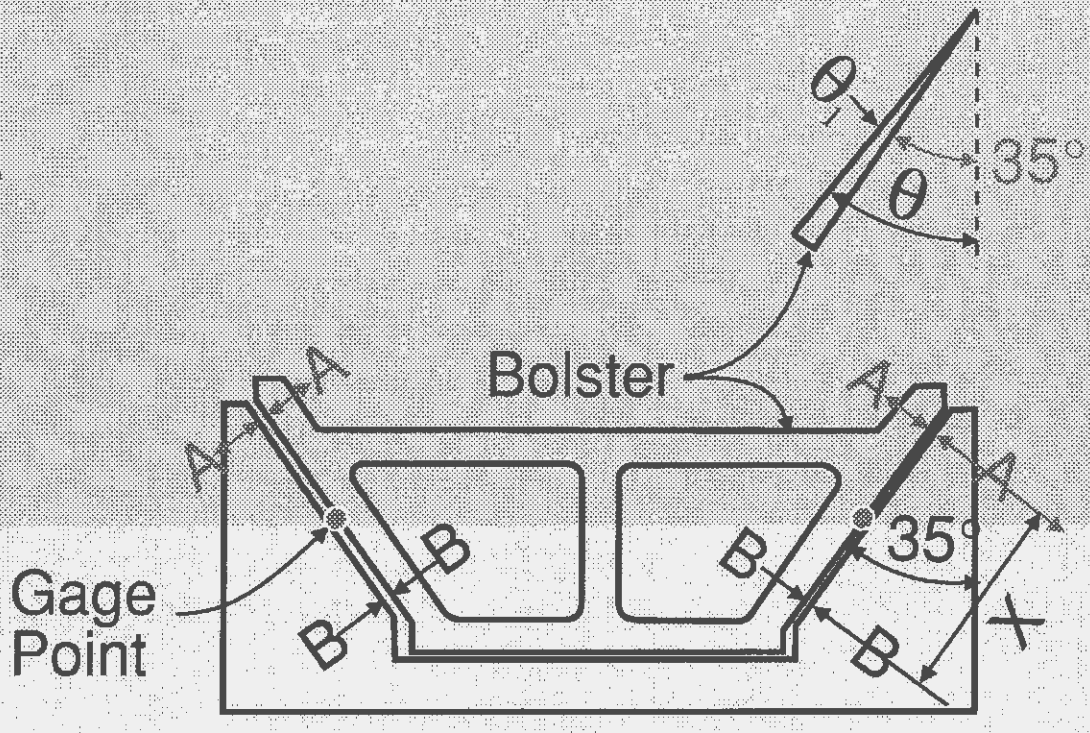
Truck Measurement Locations



Side Frame
Column Spacing
Toe-In = A-C



Section A-A



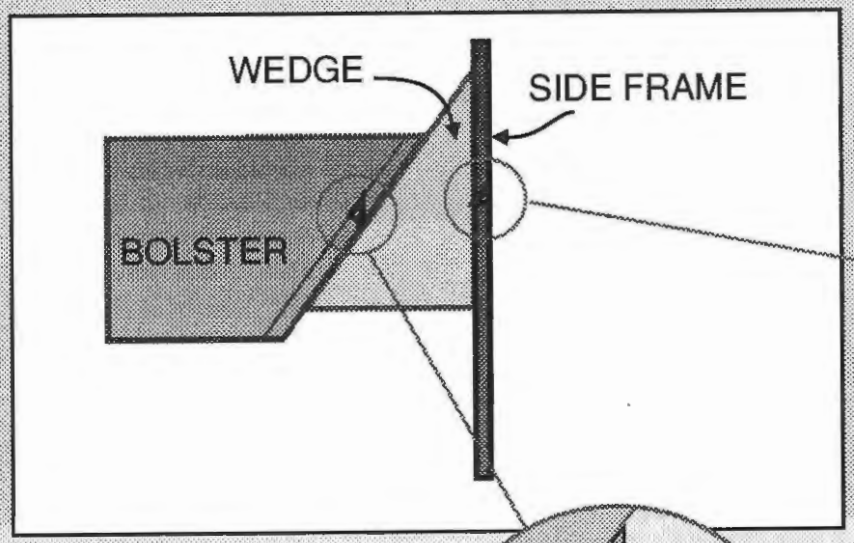
Bolster Wedge Pocket Angle

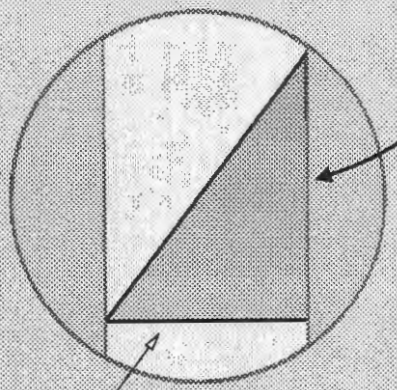
$\theta = 35° + \theta_1$, where $\tan \theta_1 = (B-A)/X$

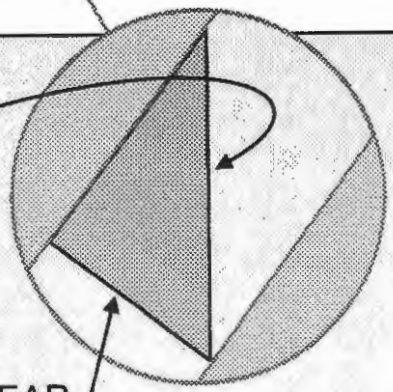
IM18.cdr

Inspection Measurements

Calculation of Wedge Rise







WR-1 = 1.743 * BOLSTER + WEDGE WEAR

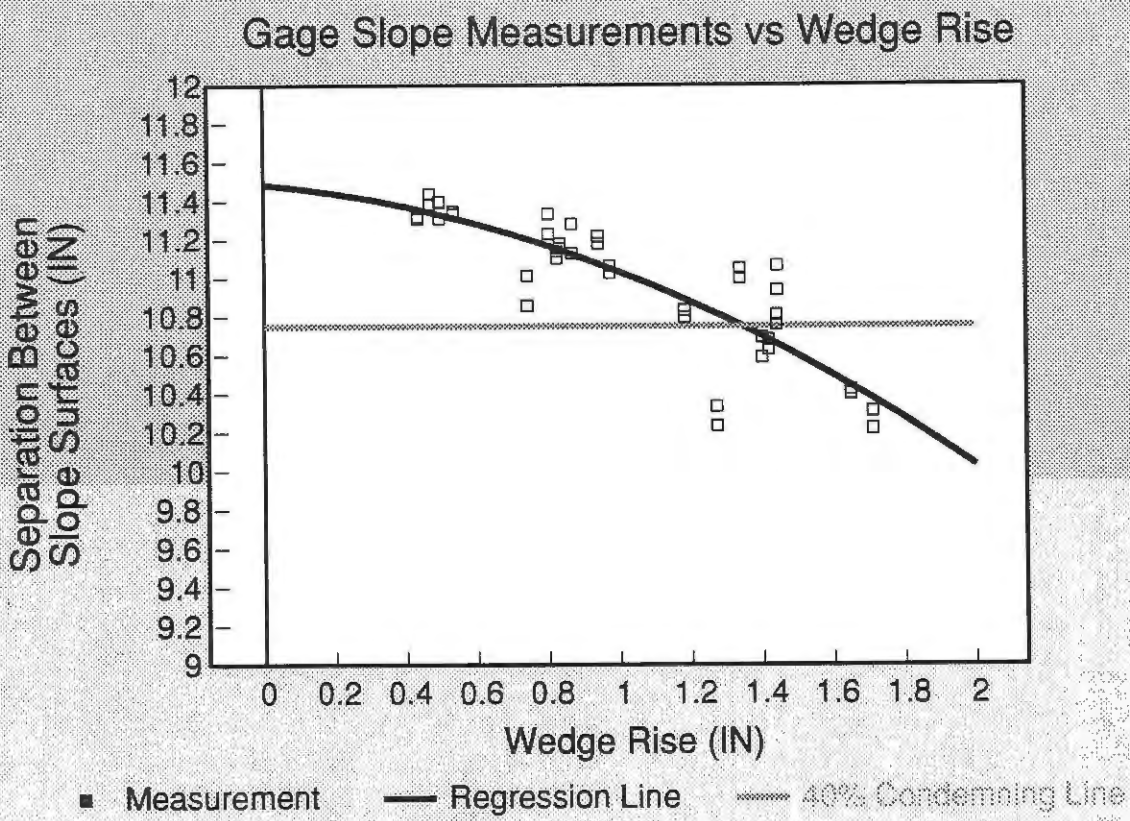
WR-2 = 1.428 * SIDE FRAME + WEDGE WEAR

CSC (WEDGE ANGLE) = 1.743

COT (WEDGE ANGLE) = 1.428

Inspection Measurements

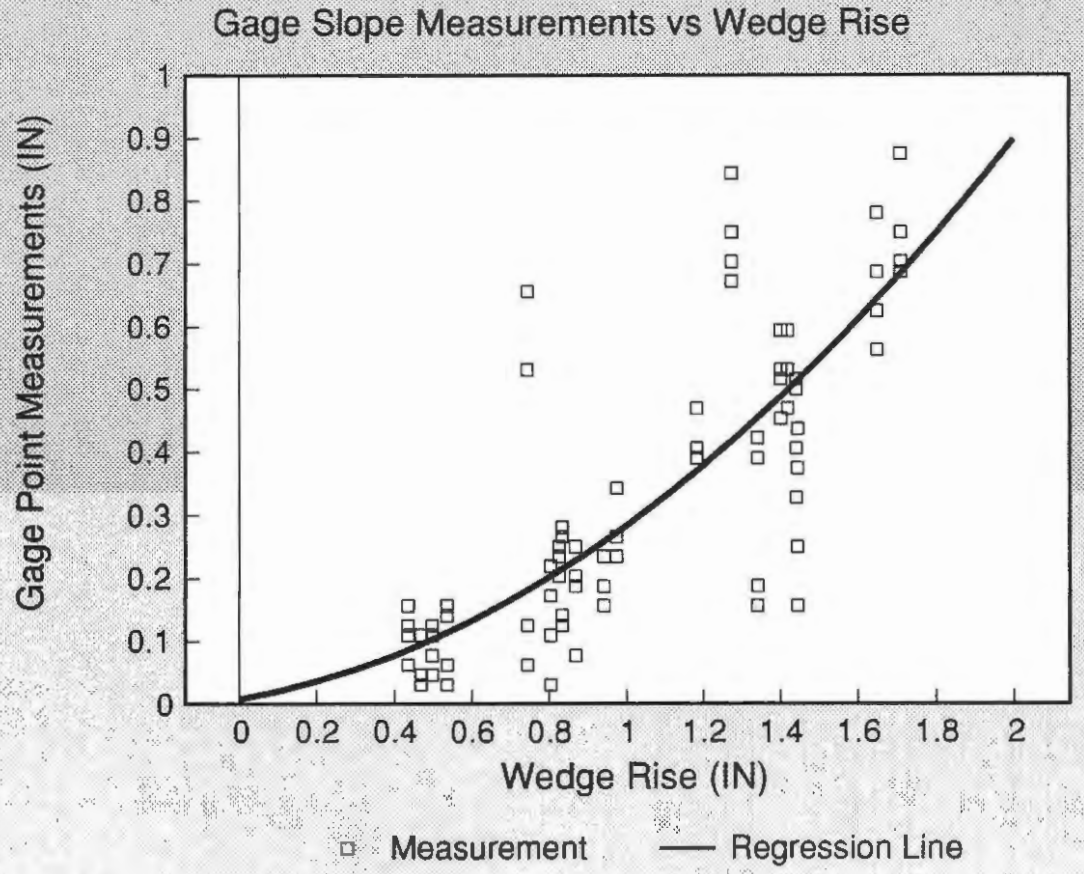
Bolster Slope Gage Measurement



IM03.cdr

Inspection Measurements

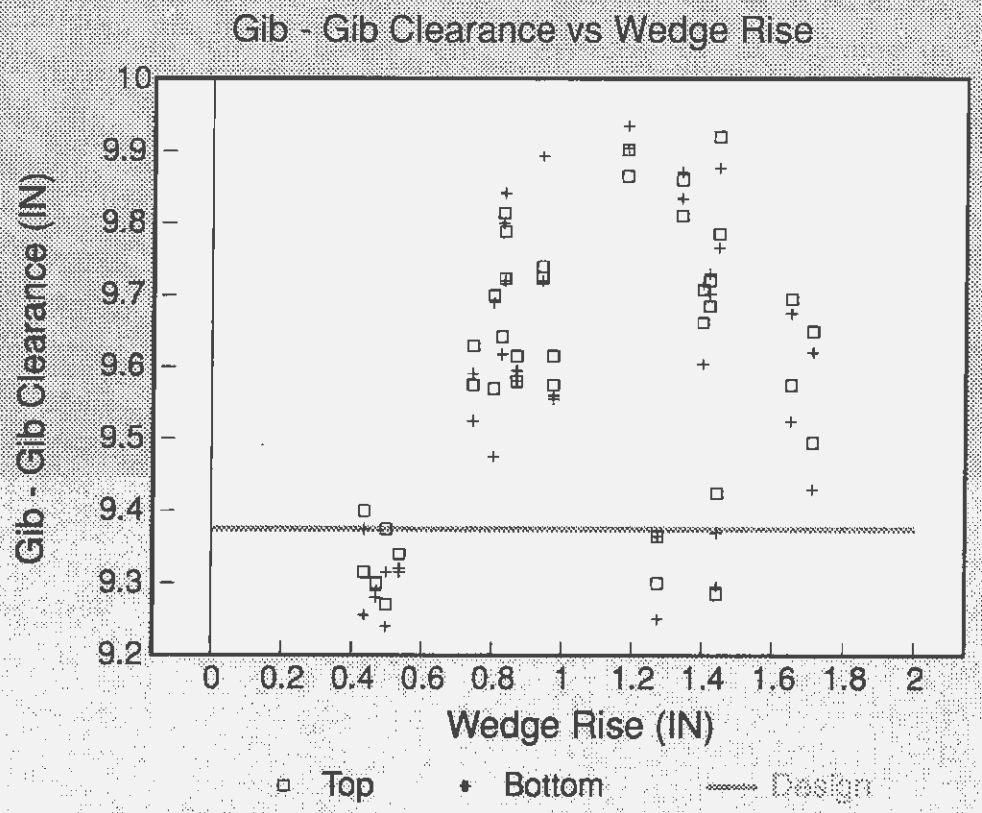
Bolster Slope Gage Measurements



IM04.cdr

Inspection Measurements

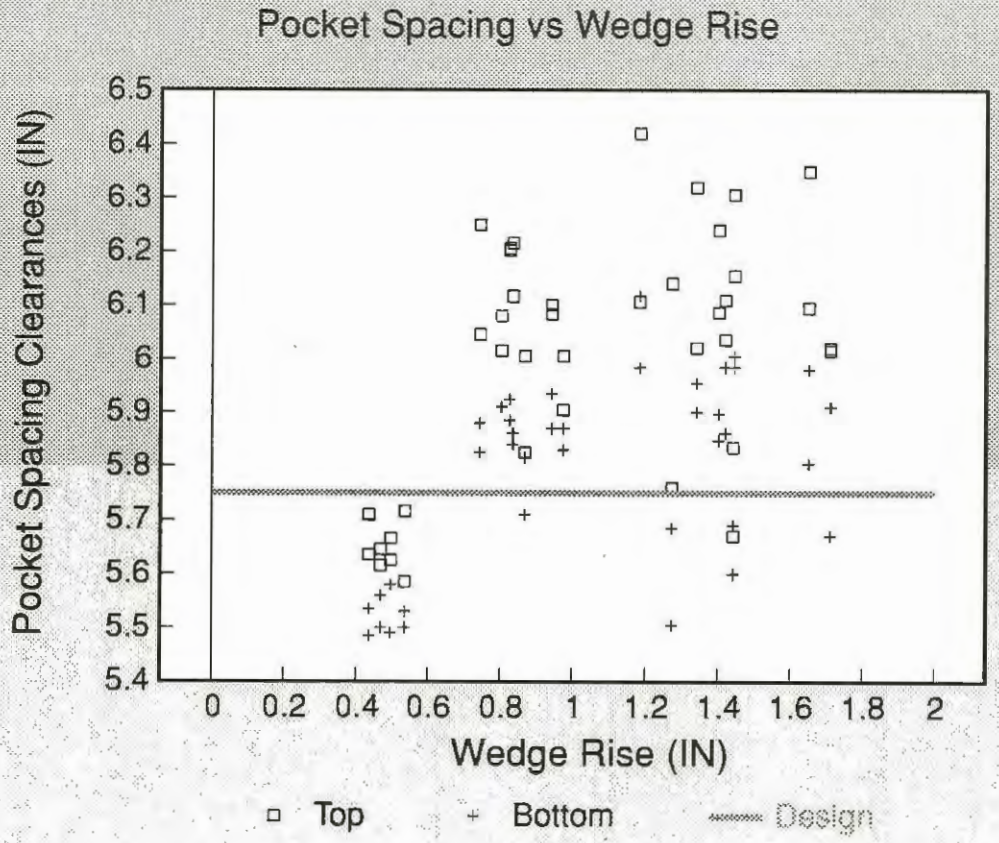
Bolster Gib Wear Measurements



IM05.cdr

Inspection Measurements

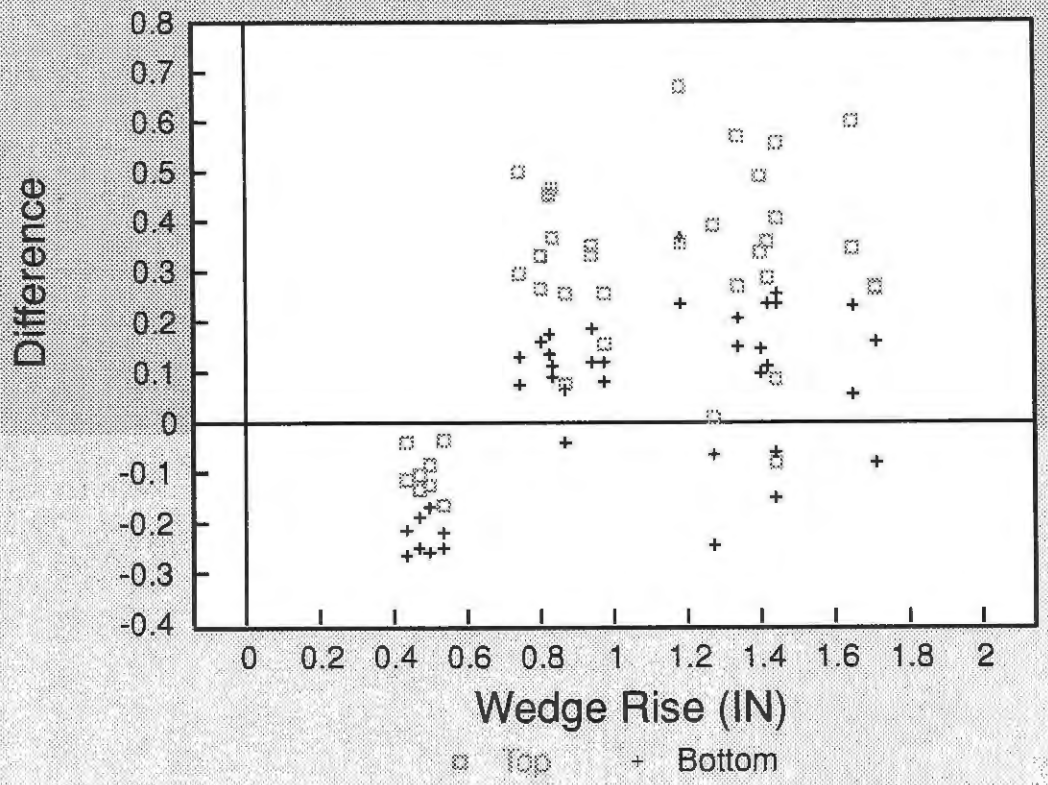
Bolster Wedge Pocket Measurements



IM06.cdr

Inspection Measurements

Bolster Wedge Pocket (Design - Measure)

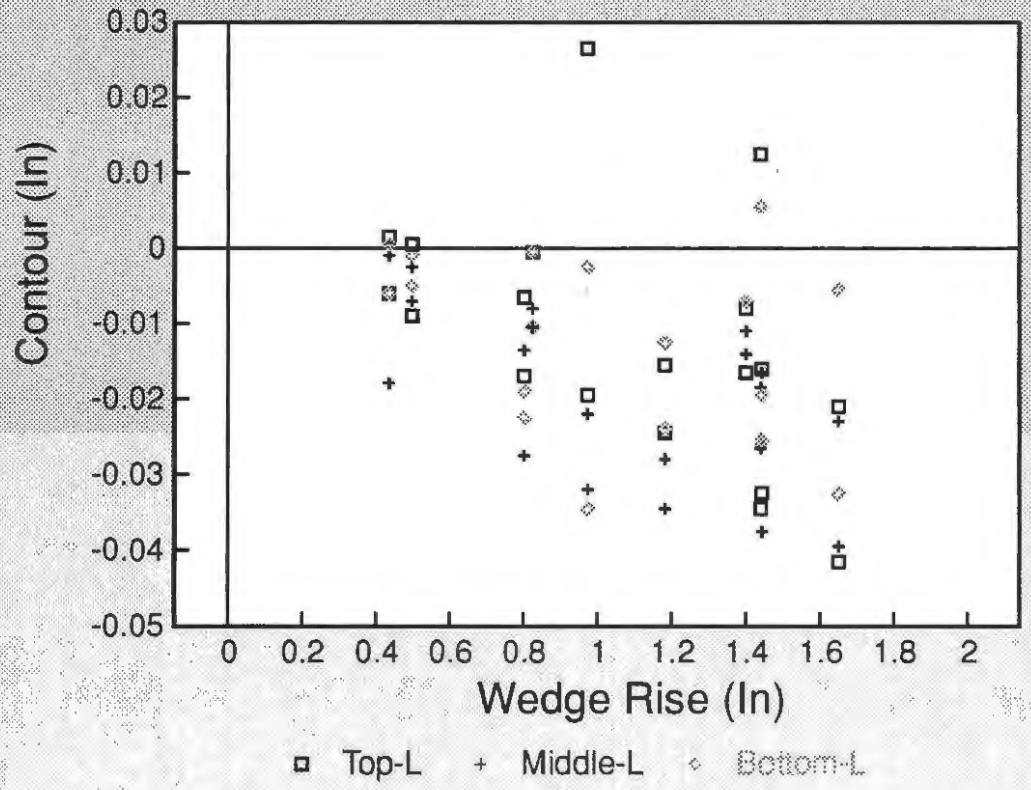


IM15.cdr

Inspection Measurements

Wedge Measurements (Left Side)

Vertical Slope Contour -vs- Wedge Rise

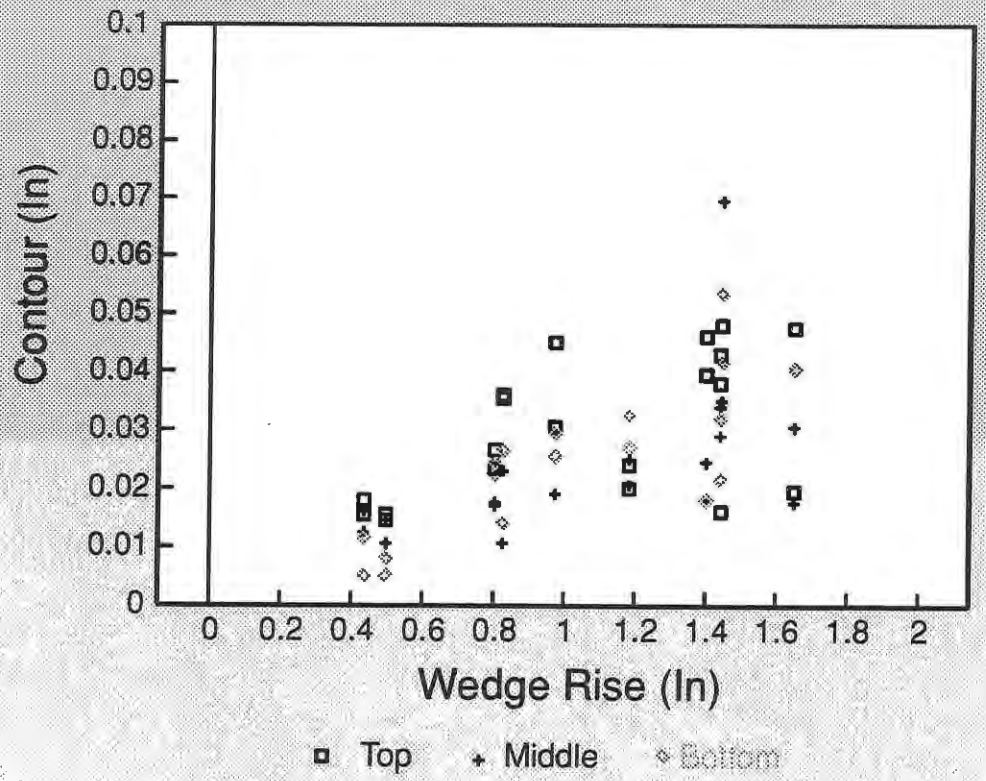


hsf1wear/lm09.cdr

Inspection Measurements

Wedge Measurements (Left Side)

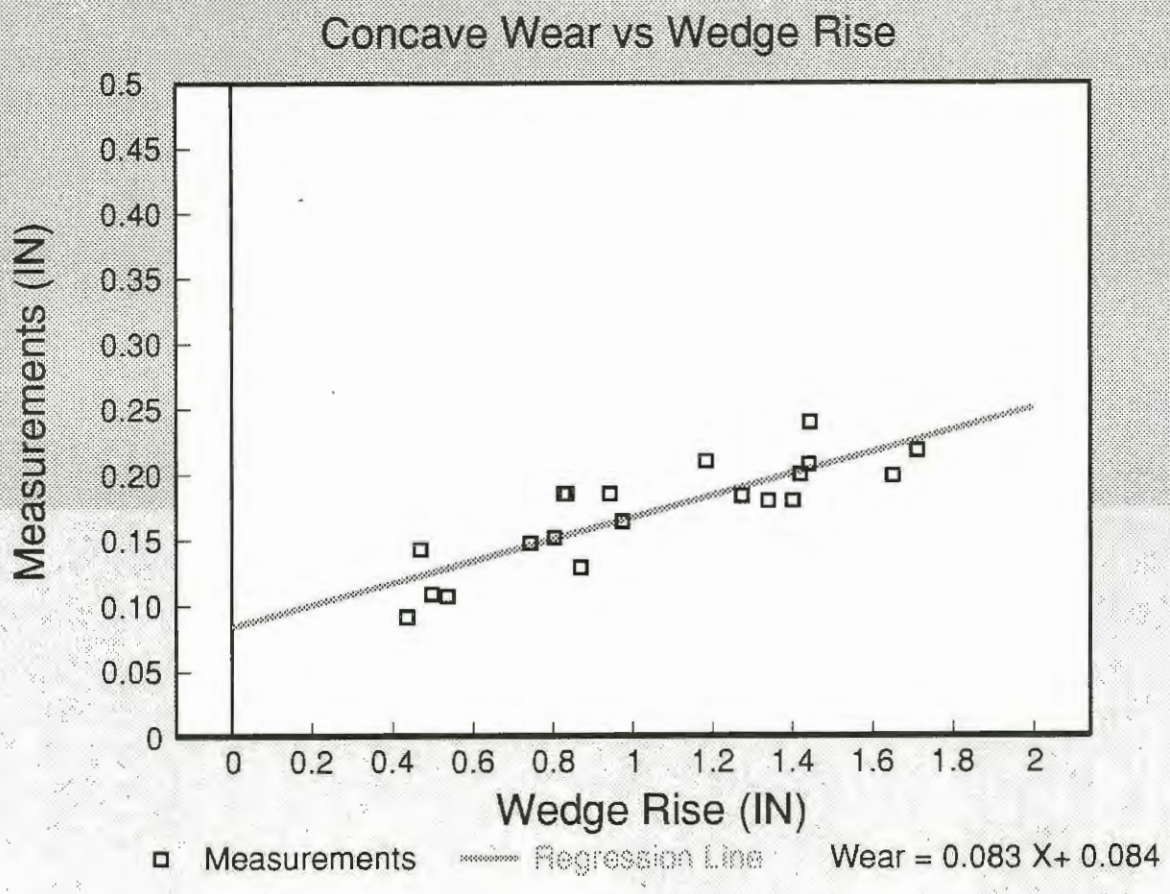
Lateral Face Contour -vs- Wedge Rise



hsf1wear/im10.cdr

Inspection Measurements

Side Frame Wear Plate Measurements

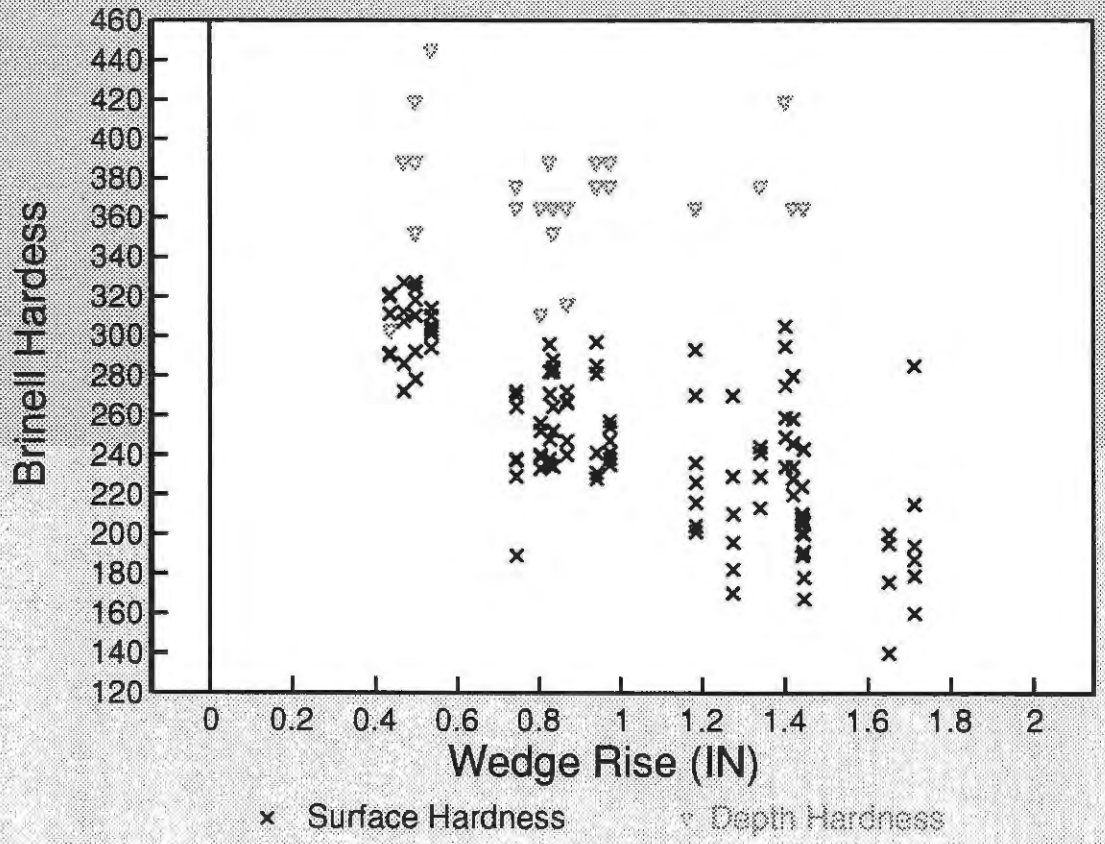


IM08.cdr

Inspection Measurements

Hardness Test

Side Frame Wear Plate vs Wedge Rise

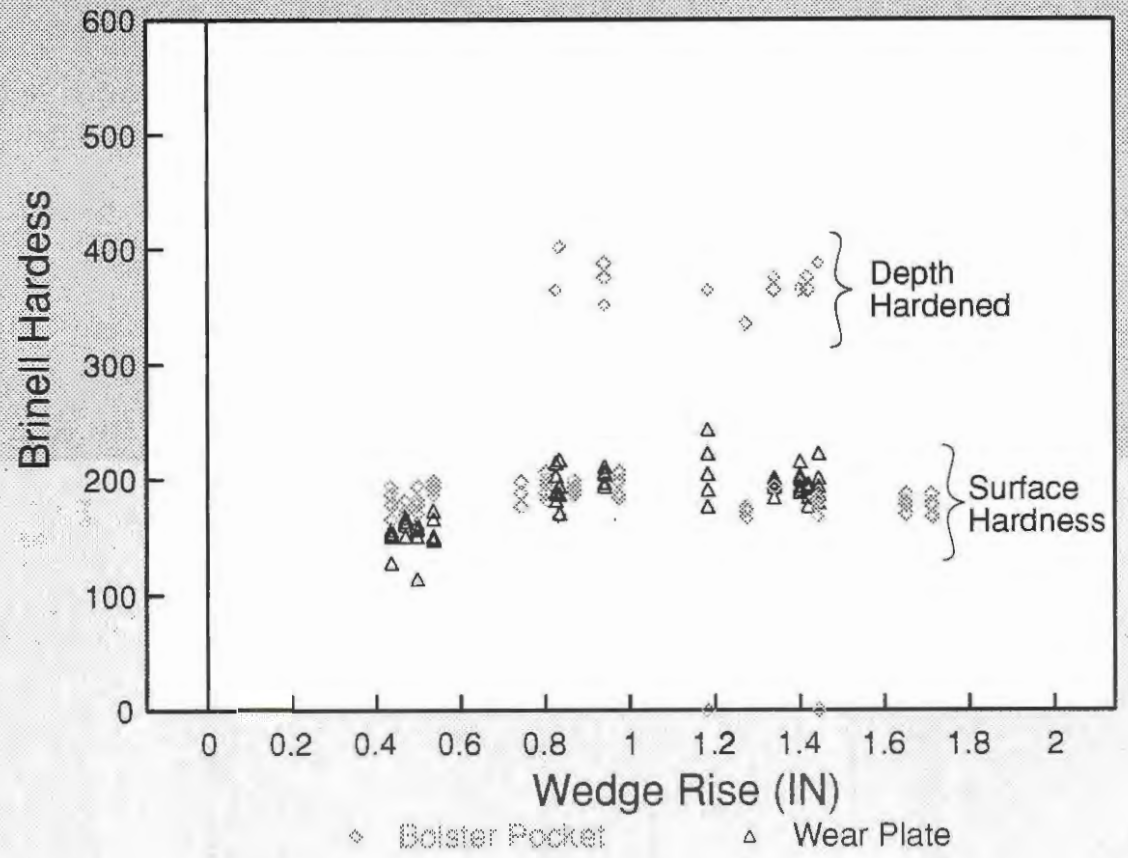


IM11.cdr

Inspection Measurements

Hardness Test

Wear Plates and Casting vs Wedge Rise

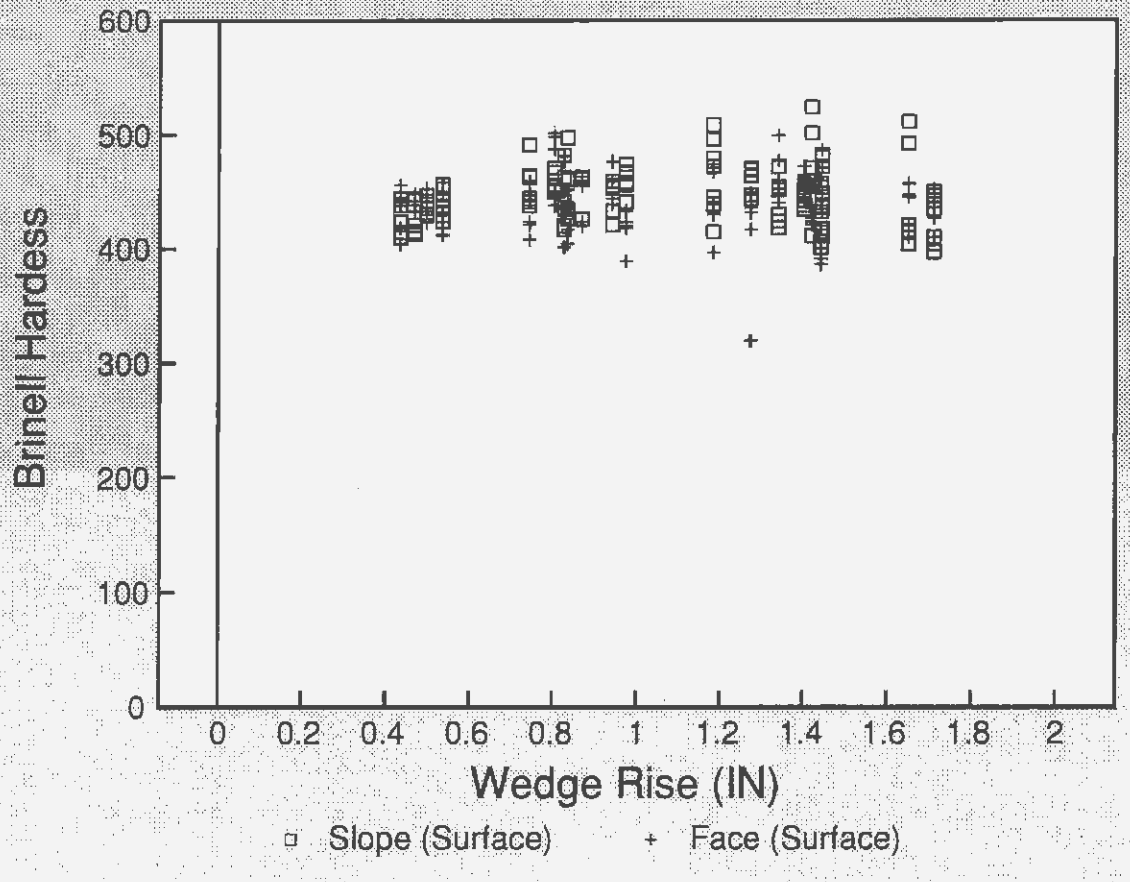


IM12.cdr

Inspection Measurements

Hardness Test

Wedge Hardness vs Wedge Rise



IM13.cdr

Inspection Measurements

Conclusions

- Truck Inspections are not enough to determine which Trucks might perform good or bad
- Hidden wear problems
  - Bolster pocket
  - Side frame wear plates
  - Inner/outer GIB wear problems
- Alignment lugs restricting control coil from properly aligning
- Center bowl wear pattern problems
- Wedges are the hardest component in the friction package

IM14.cdr

ECONOMICS - WHAT THE PROBLEMS COST?

T. S. GUINS & M. B. HARGROVE

ASSOCIATION OF AMERICAN RAILROADS

FREIGHT CAR TRUCKS FOR THE 21ST CENTURY

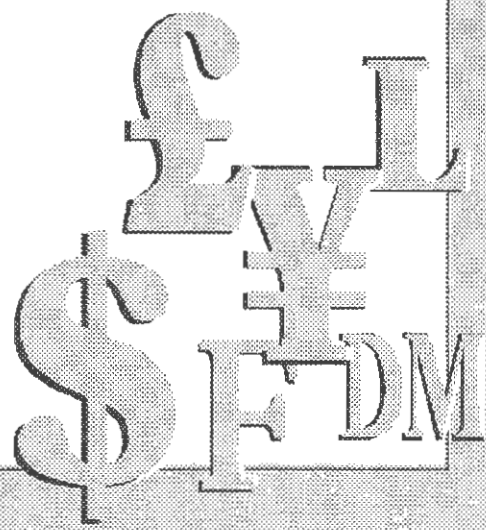
WHAT THE PROBLEMS COST

MIKE HARGROVE
TOM GUINS



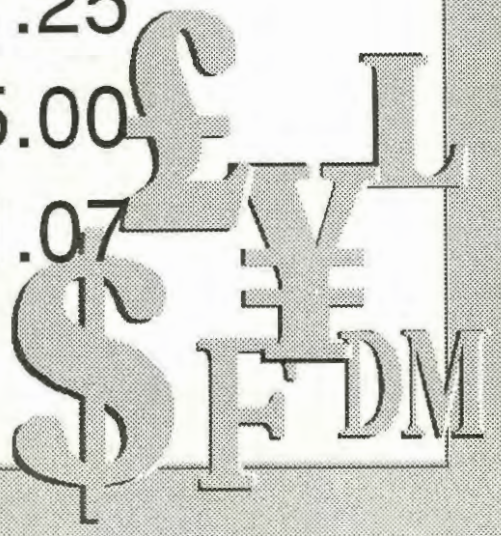
WHAT ARE THE PROBLEMS

- POOR CURVING
- POOR ALIGNMENT
- POOR DAMPING OF VERTICAL FORCES
- INADEQUATE HIGH SPEED STABILITY
- EXCESSIVE WEIGHT
- INADEQUATE BRAKING



WHAT THE PROBLEMS COST POOR CURVING

	$/1000 CAR MILES
WHEEL WEAR	$2.24
RAIL WEAR	$1.25
ENERGY	$5.00
DERAILMENTS	$1.07



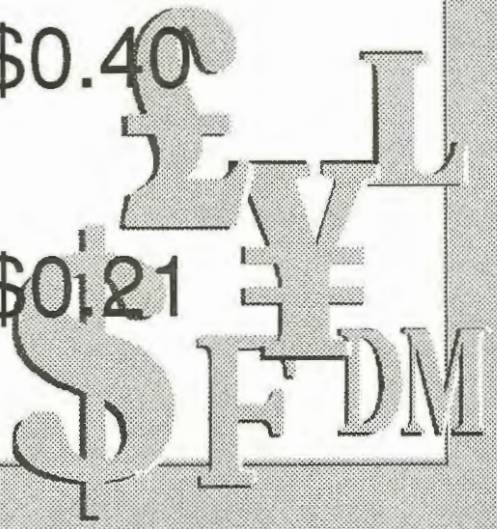
WHAT THE PROBLEMS COST
POOR ALIGNMENT

	$/1000 CAR MILES
ENERGY IN TANGENT TRACK	$4.00



WHAT THE PROBLEMS COST
POOR DAMPING OF VERTICAL FORCES

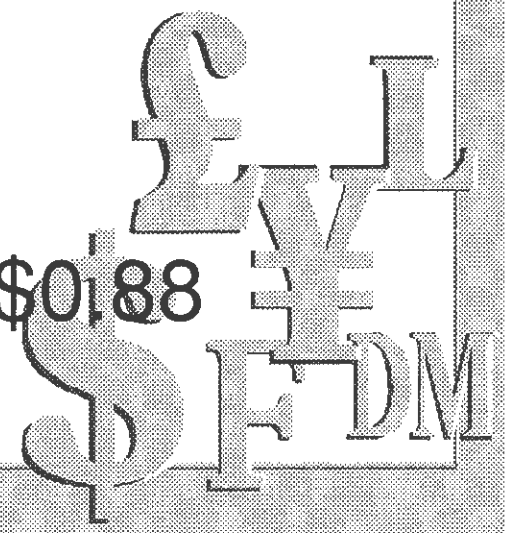
	$/1000 CAR MILES
FATIGUE DAMAGE TO RAIL	$7.00
FATIGUE DAMAGE TO CAR COMPONENTS	$0.40
LADING DAMAGE	
DERAILMENTS	$0.21



WHAT THE PROBLEMS COST

INADEQUATE HIGH SPEED LATERAL STABILITY

	$/1000 CAR MILES
FATIGUE DAMAGE TO CAR COMPONENTS	$1.60
LADING DAMAGE	
LIMITS ON SERVICE QUALITY	
DERAILMENTS	$0.88



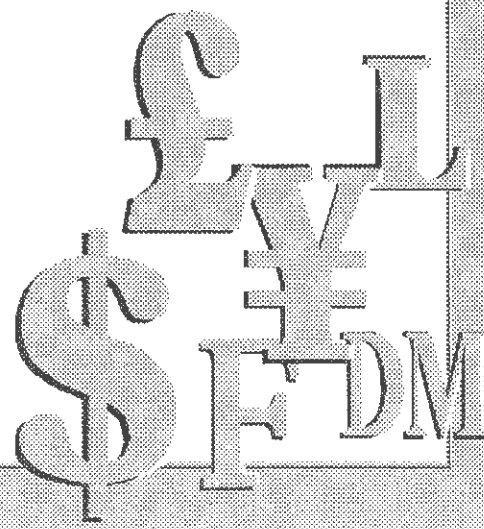
WHAT THE PROBLEMS COST
EXCESSIVE WEIGHT

	$/1000 CAR MILES
OPERATING SAVINGS	$3.50



WHAT THE PROBLEMS COST
INADEQUATE BRAKING

	$/1000 CAR MILES
WHEEL SET DAMAGE	$3.10
LIMITS ON SERVICE QUALITY	



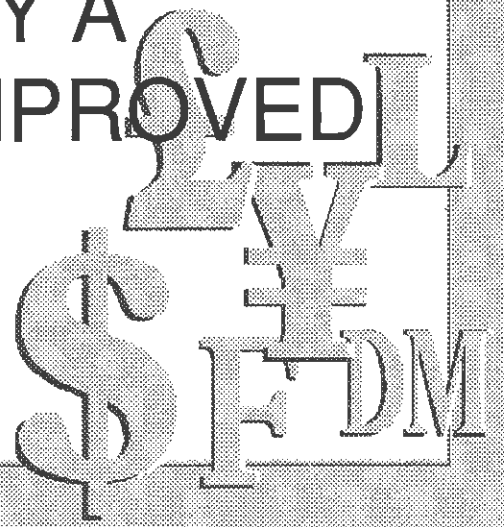
WHAT IS IT WORTH TO SOLVE THESE PROBLEMS

- COSTS ARE PROPORTIONAL TO CAR MILES; THEREFORE, UTILIZATION IS CRITICAL
- LADING DAMAGE AND SERVICE QUALITY ARE SERVICE SPECIFIC



WHAT IS IT WORTH TO SOLVE THESE PROBLEMS

- FOR A GENERAL SERVICE CAR CARRYING DAMAGE RESISTANT, MODERATE VALUED LADING, THE POTENTIAL ANNUAL SAVINGS ARE $750. THIS JUSTIFIES ONLY A MODEST PREMIUM FOR IMPROVED TRUCKS



WHAT IS IT WORTH TO SOLVE THESE PROBLEMS

- FOR A HIGH MILEAGE CAR UTILIZED IN DEDICATED SERVICE, THE ANSWER IS MUCH HIGHER $1800



CP RAIL SYSTEM ANGLE OF ATTACK TRUCK INSPECTION STATION DEVELOPMENT

G. IZBINSKY
W. PAK

CP RAIL SYSTEM

CP RAIL SYSTEM

ANGLE-OF-ATTACK TRUCK* INSPECTION STATION DEVELOPMENT

BY: G. IZBINSKY
W. PAK

---

* PATENT GRANTED

Part I:

1. Definition and significance of angle-of-attack.

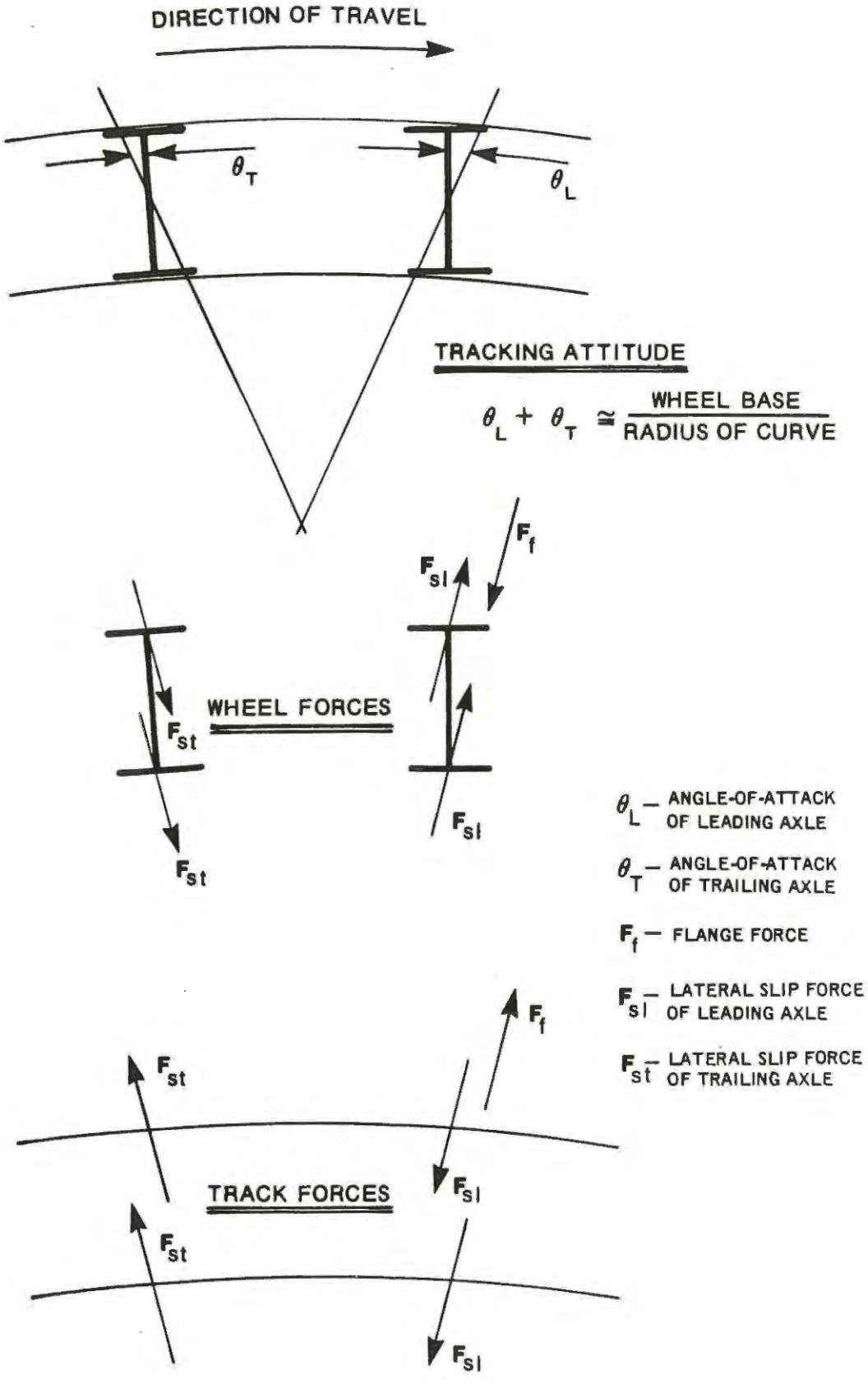
2. Controlled consist test results on curves.

3. Train monitoring results on curves.

Part II:

1. Permanent angle-of-attack hardware development.

2. Train monitoring on tangent track.



TYPICAL TRACKING ATTITUDE AND LATERAL SLIP AND FLANGE FORCES FOR CONVENTIONAL BARBER S–2 TRUCKS

FIG. 21

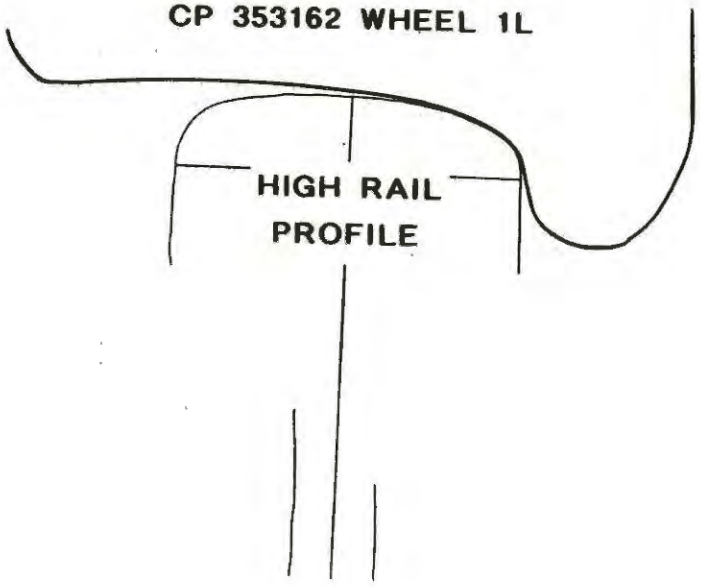
FACTORS AFFECTING ANGLE-OF-ATTACK IN CONVENTIONAL TRUCKS ON CURVES

(1) WHEEL/RAIL PROFILE AND CONTACT CONDITION

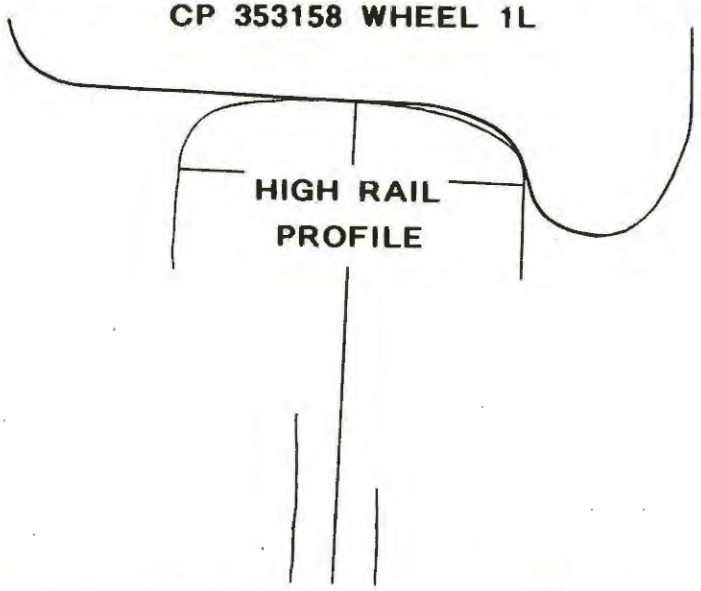
(2) TRUCK SUSPENSION CONDITION

(3) AXLE YAW MISALIGNMENT

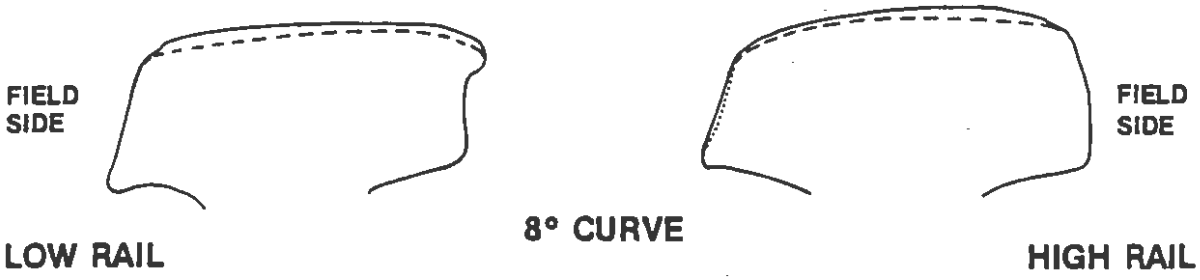
WORN WHEEL PROFILE
CP 353162 WHEEL 1L



NEW WHEEL PROFILE AAR-1B
CP 353158 WHEEL 1L

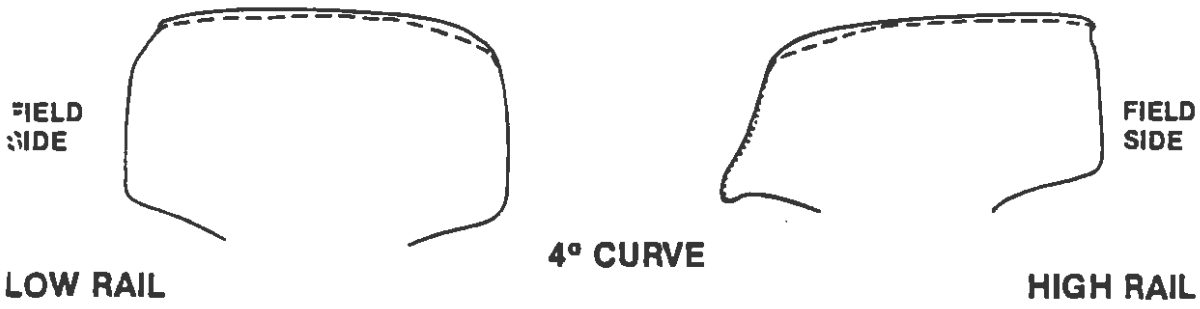


COMPARISON OF NEW & WORN
WHEEL RAIL CONTACT
6 DEGREE CURVE
MP 119.5 WINDERMERE SUBDIVISION



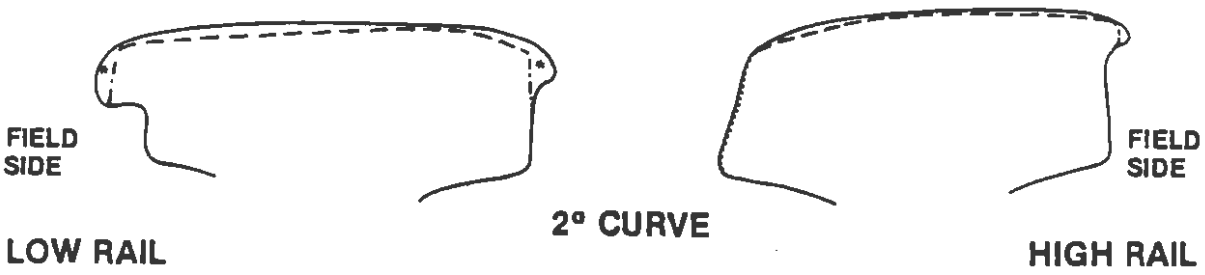
ALGOMA 132 lb. CHROME RAIL (August 1974)

ALGOMA 132 lb. CHROME RAIL (August 1974)



ALGOMA 136 lb. CHROME RAIL (December 1980)

ALGOMA 132 lb. CHROME RAIL (March 1974)



ALGOMA 132 lb. CARBON RAIL (March 1974)

ALGOMA 132 lb. CARBON RAIL (March 1974)

——— RAIL PROFILES BEFORE RAIL GRINDING

– – – – RAIL PROFILES IMMEDIATELY AFTER RAIL GRINDING

····· RAIL WEAR

—·— * hand ground in these locations for mounting profilometer

COMPARISON OF RAIL PROFILES BEFORE AND IMMEDIATELY AFTER RAIL GRINDING

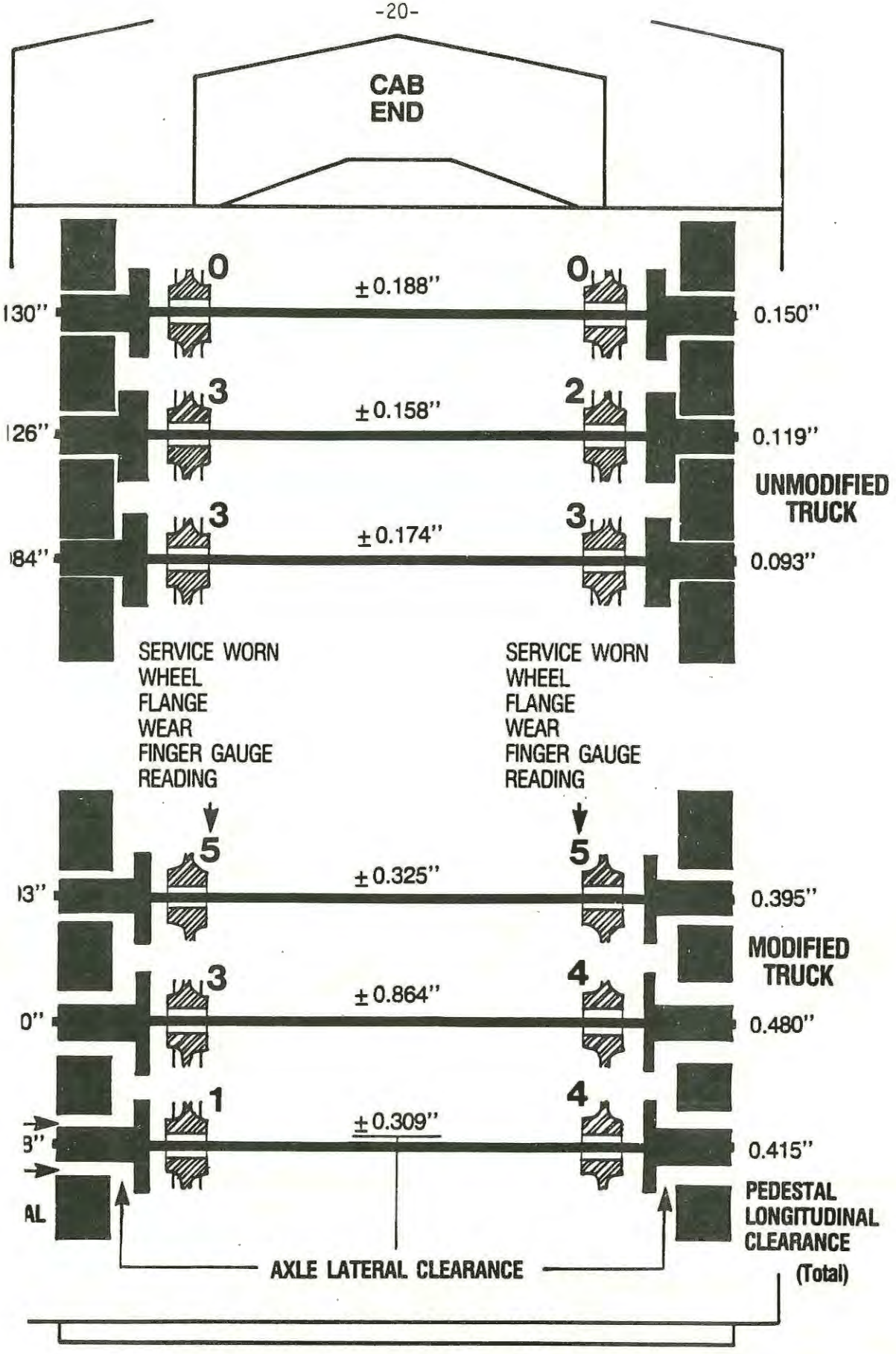
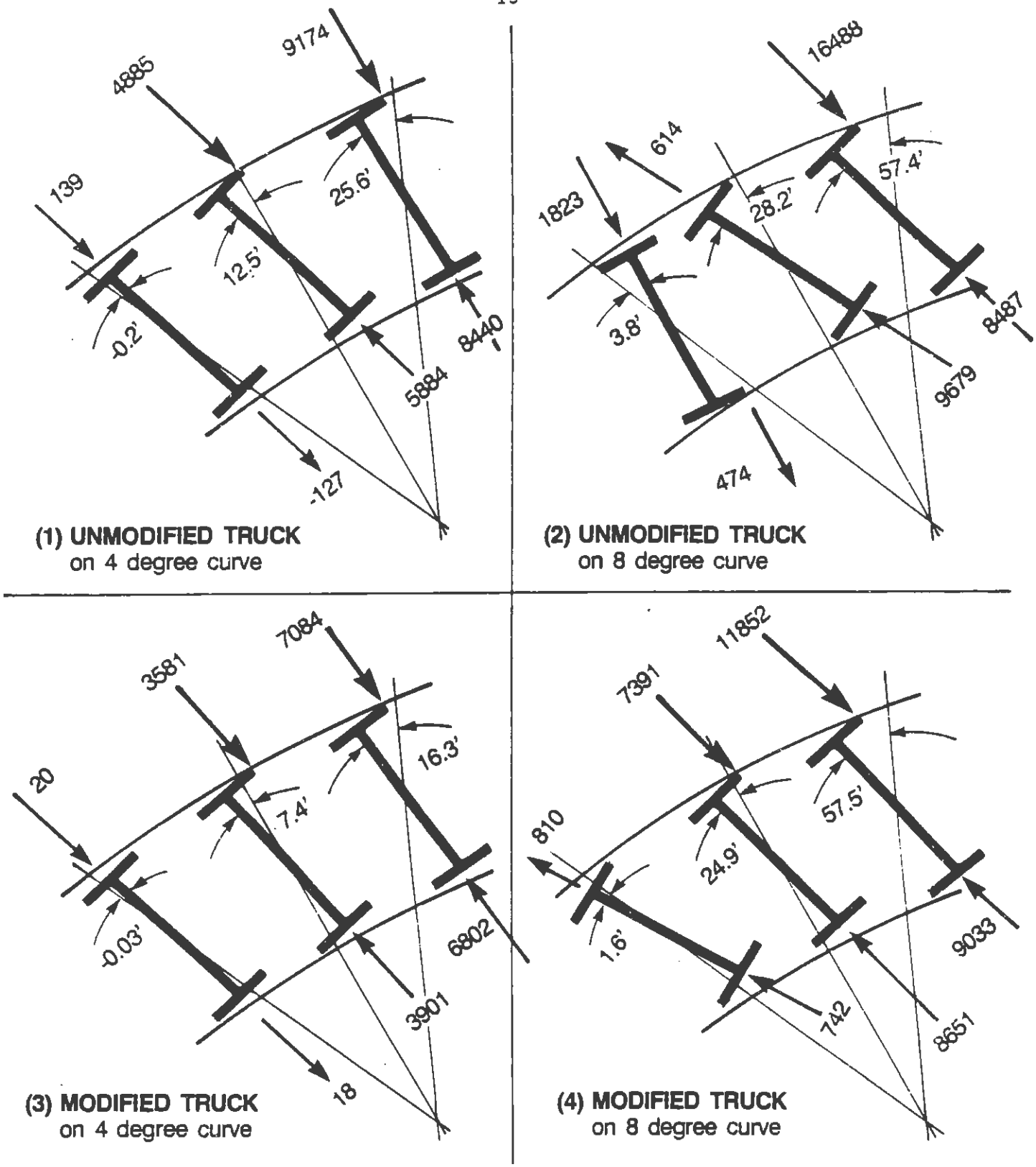
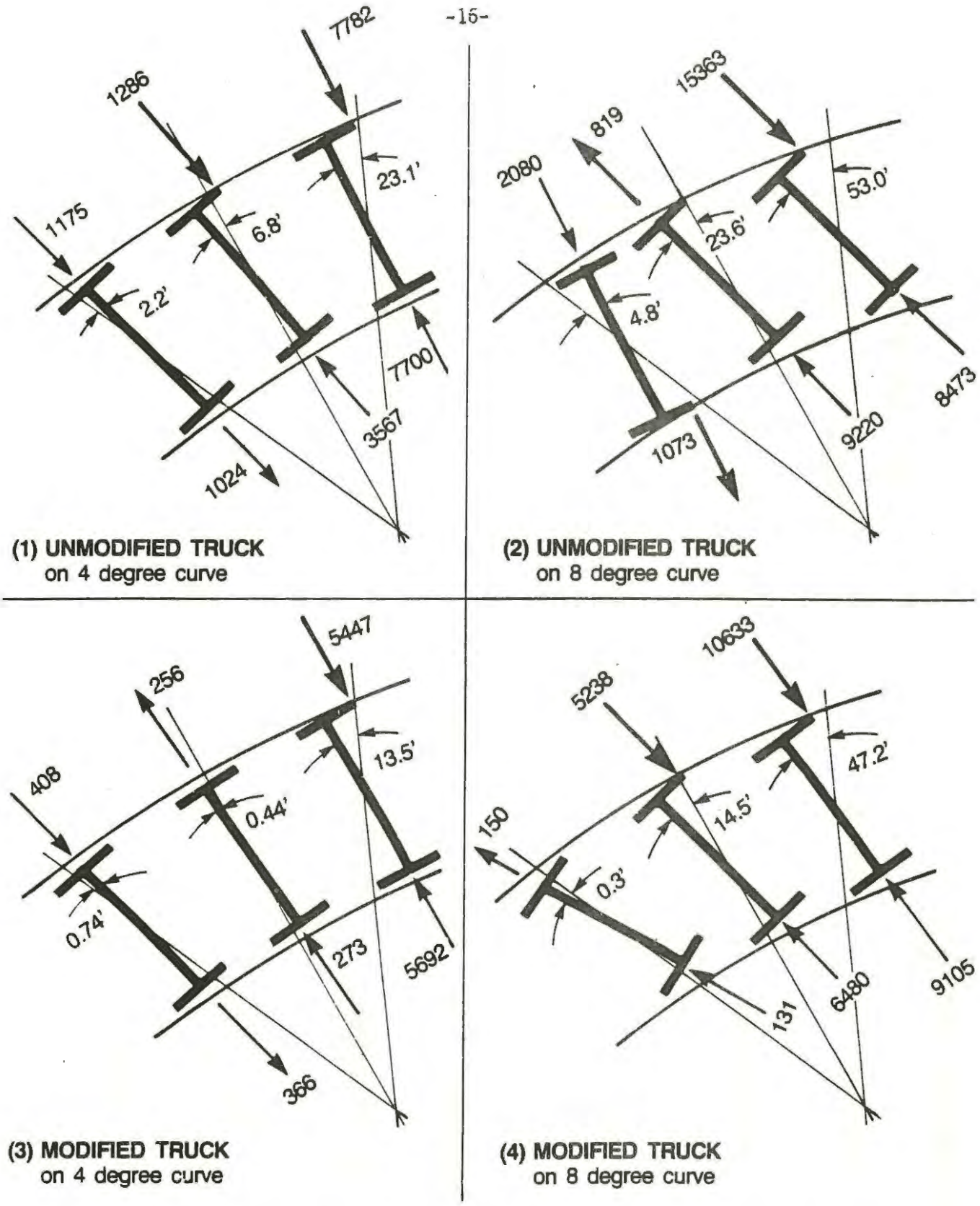


Figure 6 : TEST LOCOMOTIVE 5879 JOURNAL LONGITUDINAL & LATERAL CLEARANCE CONFIGURATION



LATERAL CURVING FORCES (lbf) AND ANGLE-OF-ATTACK (minutes) OF GM SD-40-2 LOCOMOTIVE TRUCK WITH NEW AAR WHEELS AND 3% TRACTION WITH AND WITHOUT PEDESTAL MODIFICATIONS.

Figure 2a



LATERAL CURVING FORCES (lbf) AND ANGLE-OF-ATTACK (minutes) OF GM SD-40-2 LOCOMOTIVE TRUCK WITH NEW HEUMANN PROFILE WHEELS AND 3% TRACTION WITH AND WITHOUT PEDESTAL MODIFICATIONS.

Figure 2b

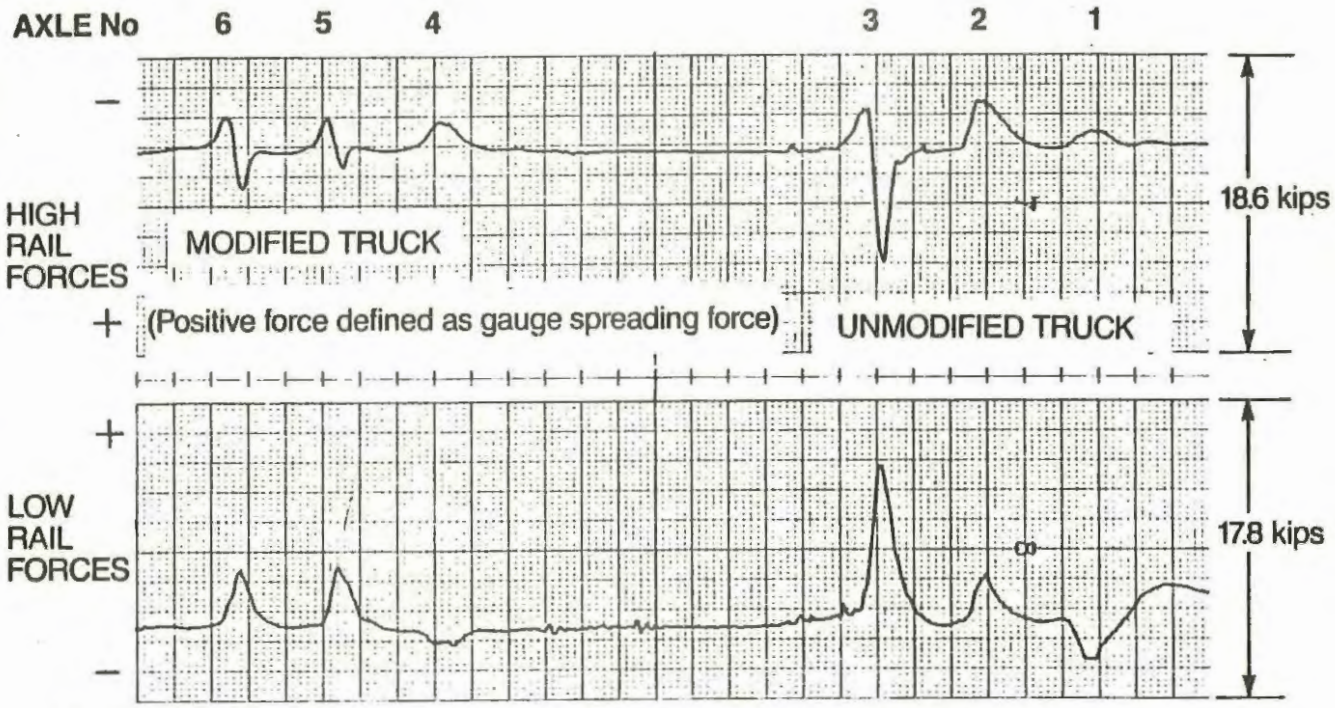
4 DEGREE TEST CURVE, 30 mph

DESCRIPTION	AXLE No.	WHEEL PROFILES		
		AAR	HEUMANN	SERVICE WORN
UNMODIFIED TRUCK	1	26.5'	27.5'	27.2'
	2	16.3'	10.1'	5.0'
	3	-0.3'	-2.0'	-0.4'
MODIFIED TRUCK	4	29.9'	4.1'	-0.3'
	5	16.5'	3.8'	2.9'
	6	5.1'	2.2'	-0.4'

8 DEGREE TEST CURVE, 25 mph

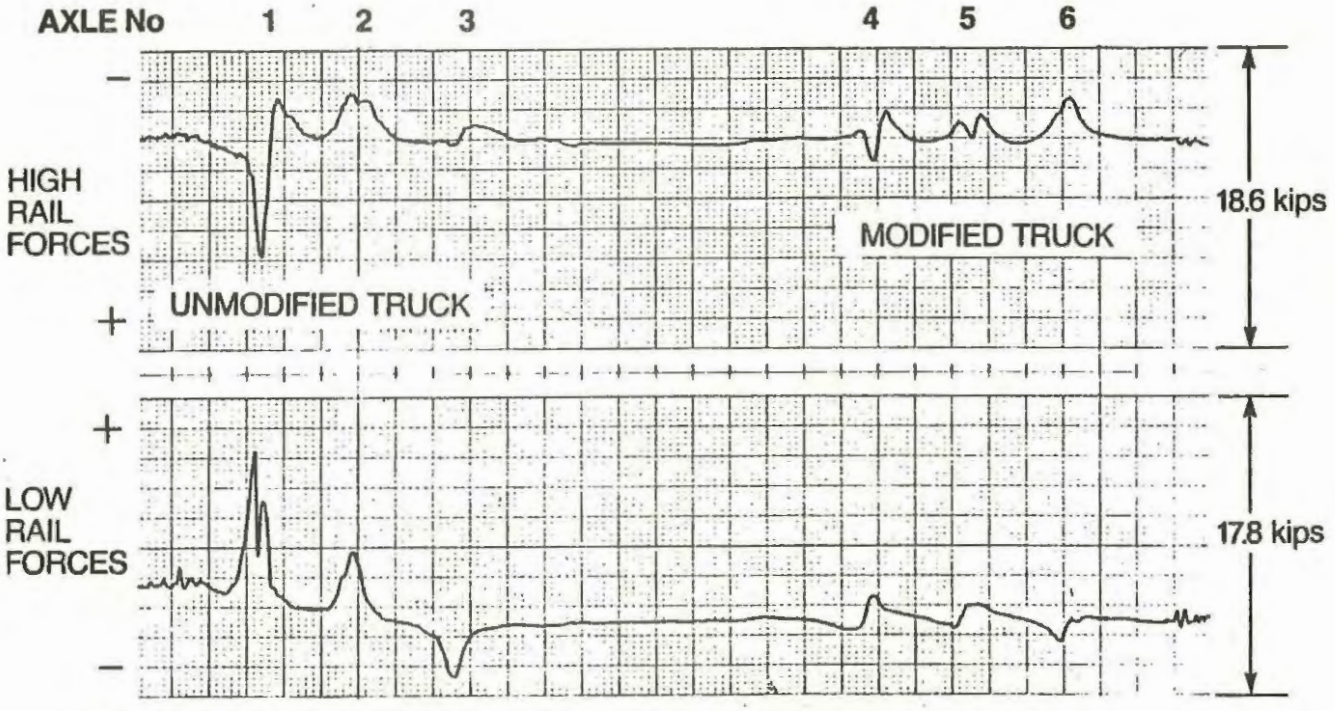
DESCRIPTION	AXLE No.	WHEEL PROFILES		
		AAR	HEUMANN	SERVICE WORN
UNMODIFIED TRUCK	1	62.2'	51.8'	67.5'
	2	30.2'	30.8'	35.9'
	3	-8.2'	-6.5'	-3.4'
MODIFIED TRUCK	4	63.7'	30.3'	44.6'
	5	32.6'	14.4'	20.2'
	6	1.1'	-3.1'	5.9'

SUMMARY OF AXLE ANGLES-OF-A-ATTACK TEST RESULTS



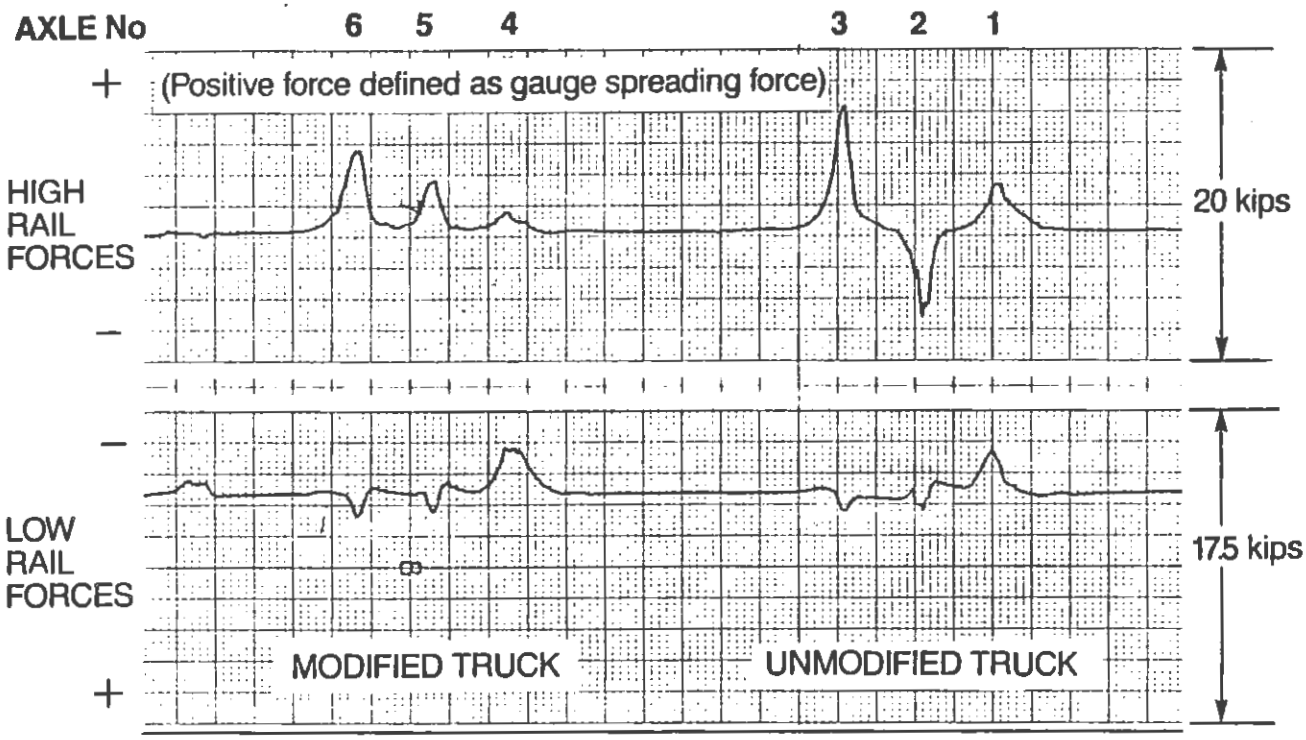
4° CURVE HIGH AND LOW RAIL LATERAL FORCES, 20 mph, WESTBOUND, LOCOMOTIVE UNWYED, NEW HEUMANN WHEEL PROFILE

Figure 18a



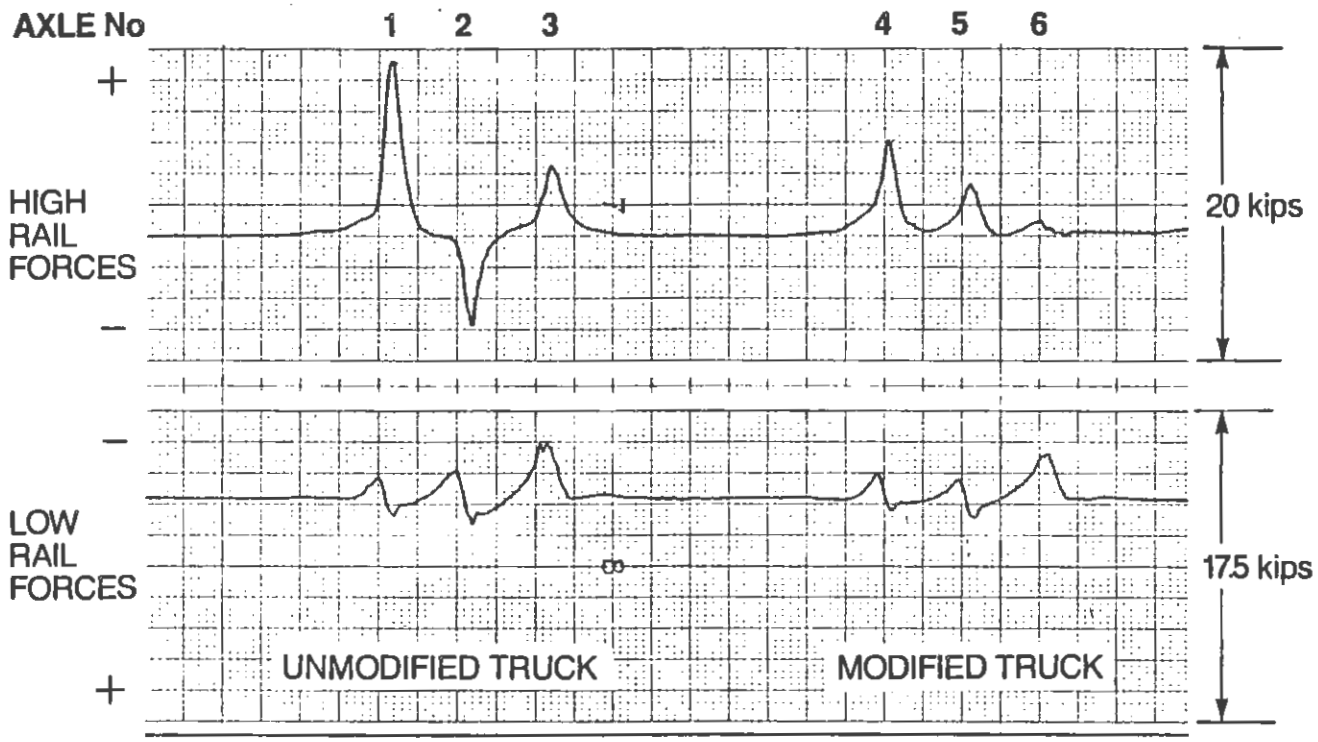
4° CURVE HIGH AND LOW RAIL LATERAL FORCES, 20 mph, EASTBOUND, LOCOMOTIVE UNWYED, NEW HEUMANN WHEEL PROFILE

Figure 18b



8° CURVE HIGH AND LOW RAIL LATERAL FORCES, 25 mph, EASTBOUND, LOCOMOTIVE WYED, NEW AAR WHEEL PROFILE

Figure 19a



8° CURVE HIGH AND LOW RAIL LATERAL FORCES, 25 mph, WESTBOUND, LOCOMOTIVE WYED, NEW AAR WHEEL PROFILE

Figure 19b

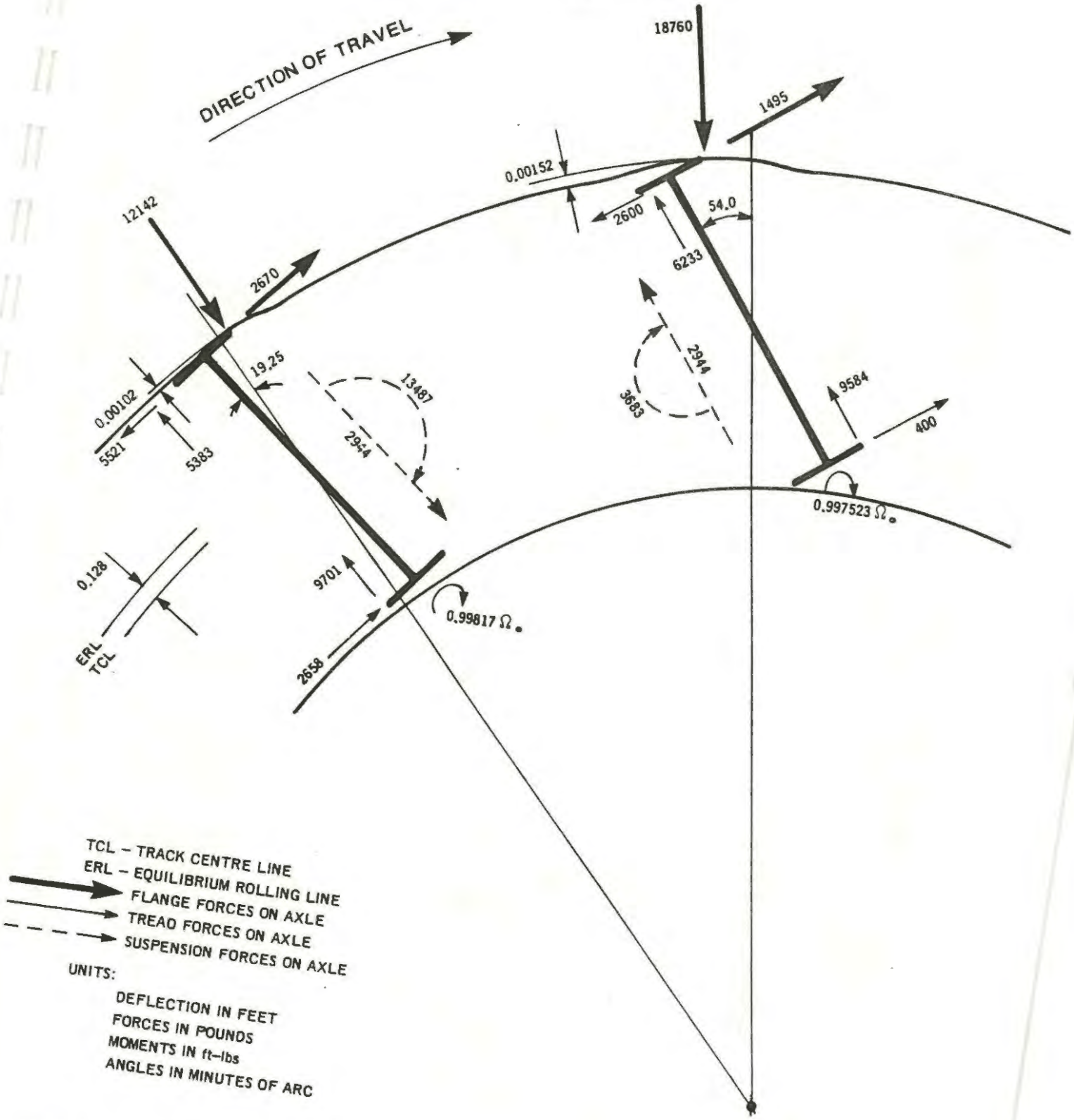
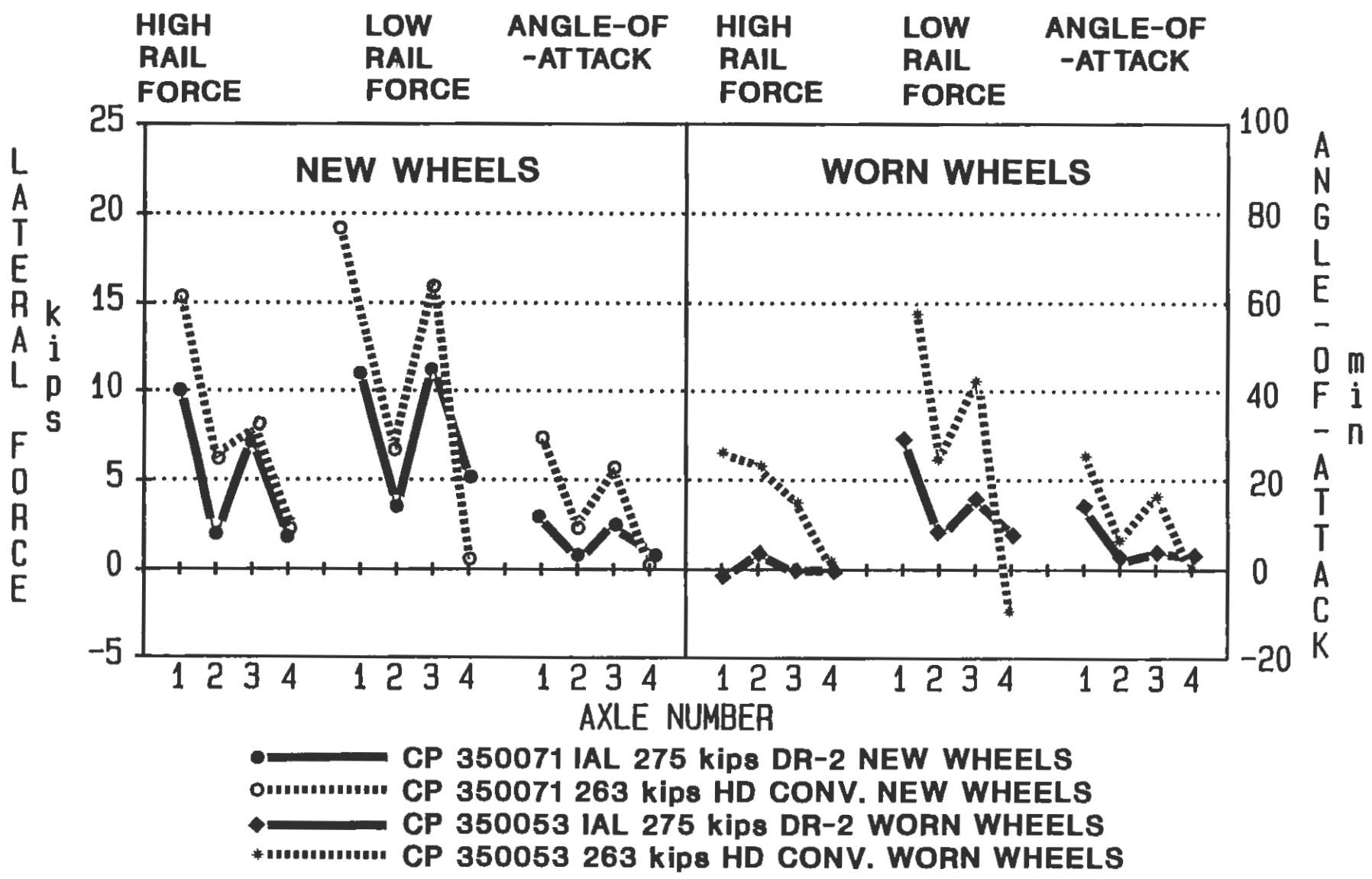
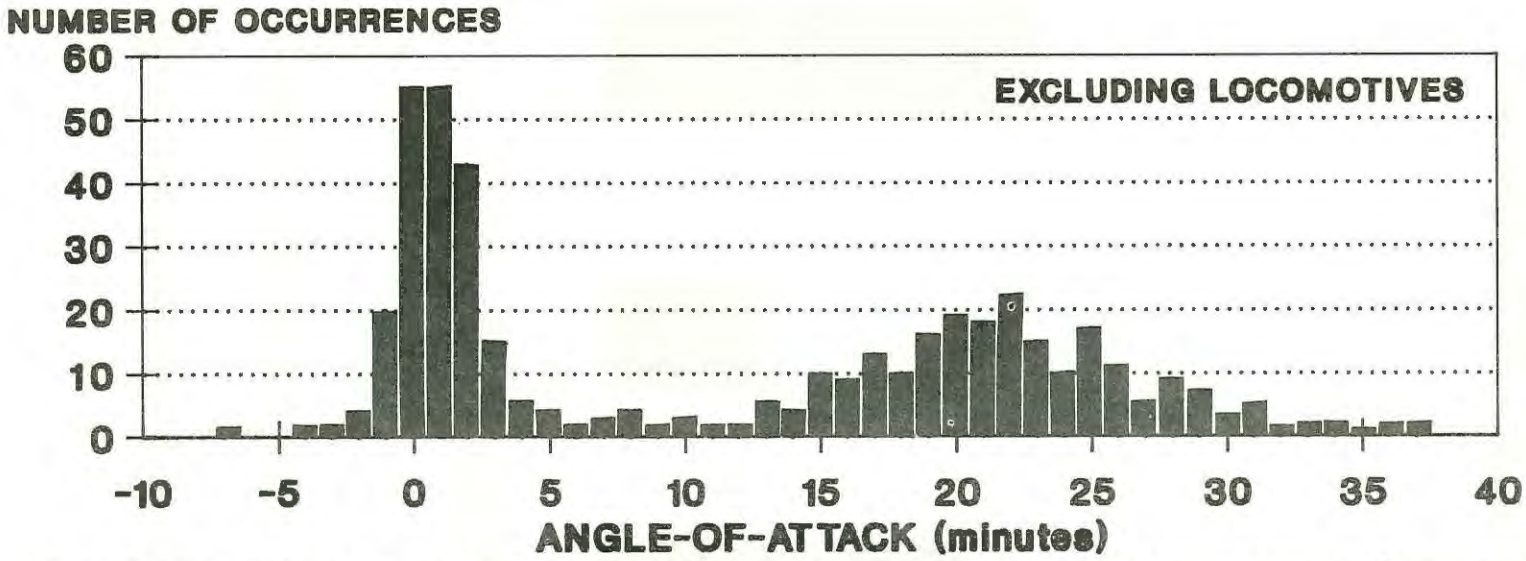


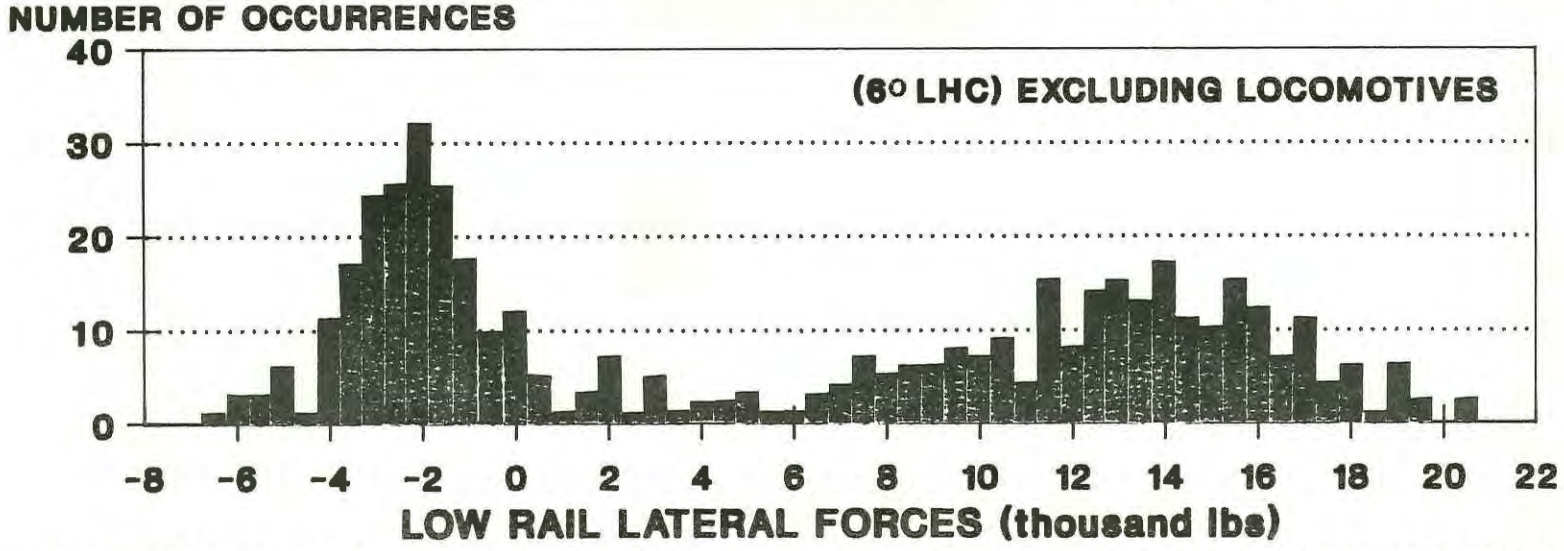
FIG. 1 : STEADY STATE CURVING ATTITUDE OF BS2 TRUCK ON 10 DEGREE CURVE
(NEW WHEEL AND RAIL CONDITION)



RADIAL 275 kips vs CONVENTIONAL 263 kips
CONSTANT SPEED, 6° LHC (EAST SITE) WESTBOUND WYED

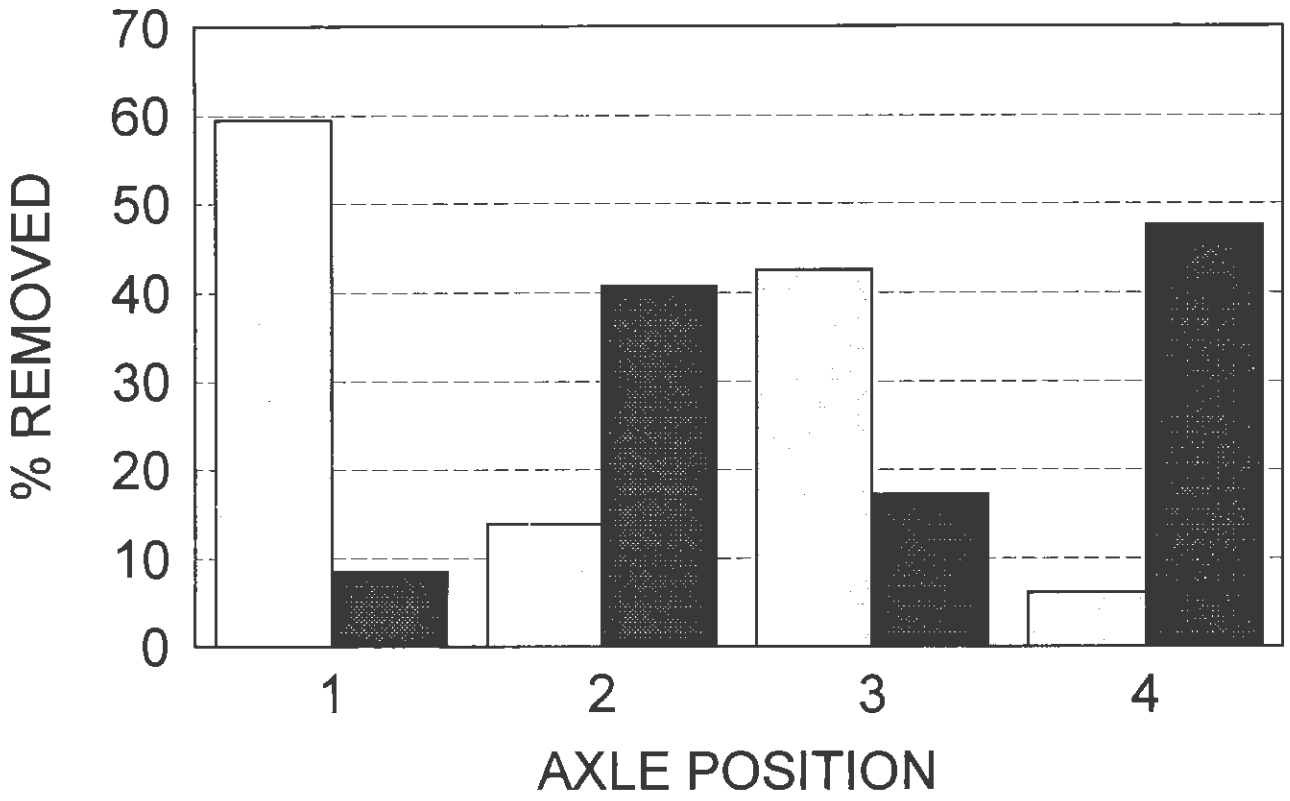


CP RAIL GOLD 1 TRAIN, ANGLE-OF-ATTACK DISTRIBUTION (6° LHC)



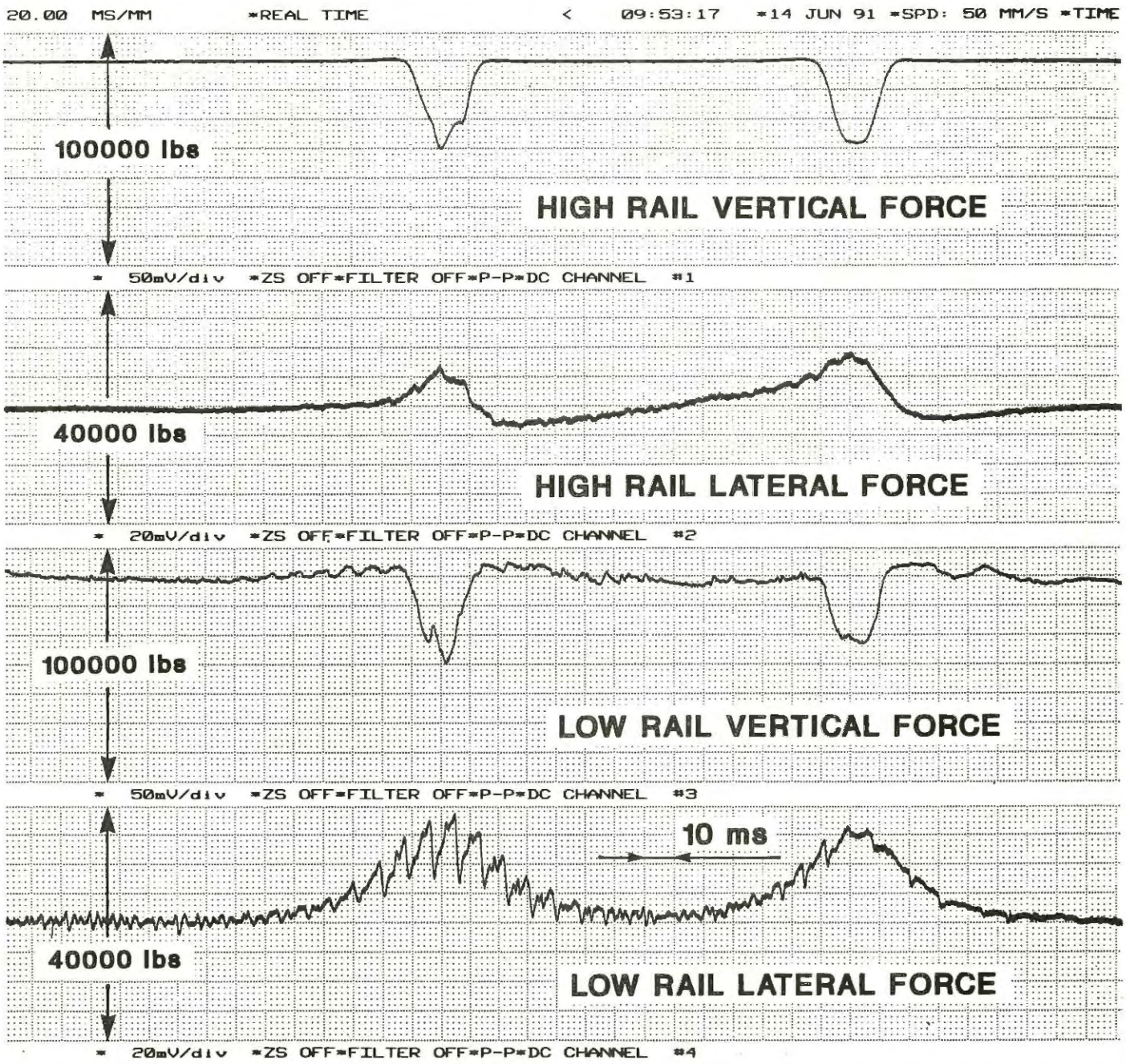
CP RAIL GOLD 1 TRAIN, LOW RAIL LATERAL FORCE DISTRIBUTION

PERCENT OF WHEEL REMOVAL BY AXLE POSITION



B END LEADING - 10 MONTH PERIOD AFTER WHEELS APPLIED NOV-DEC 1989

A END LEADING - 10 MONTH PERIOD AFTER WHEELS APPLIED NOV-DEC 1990

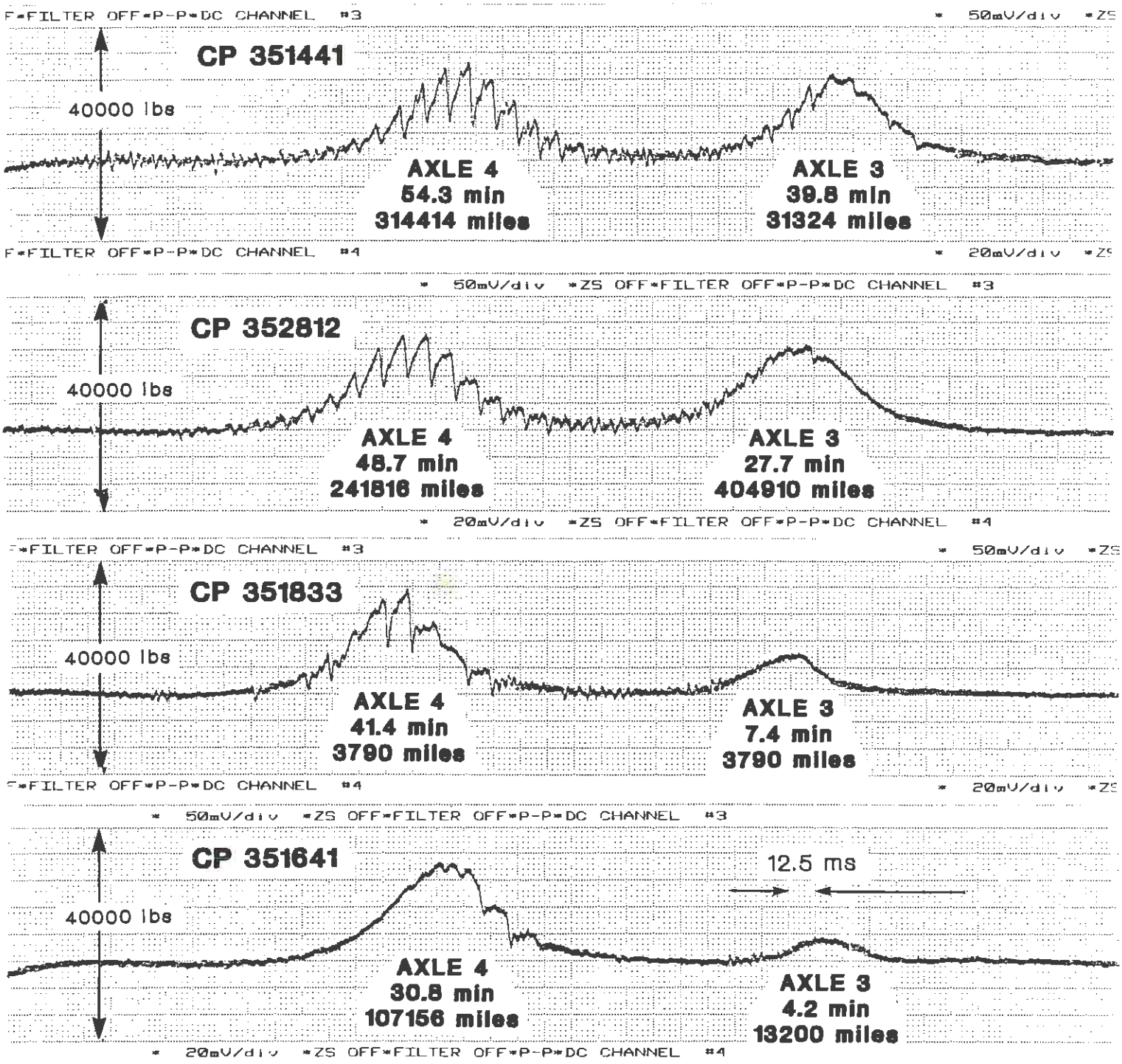


CP 351441

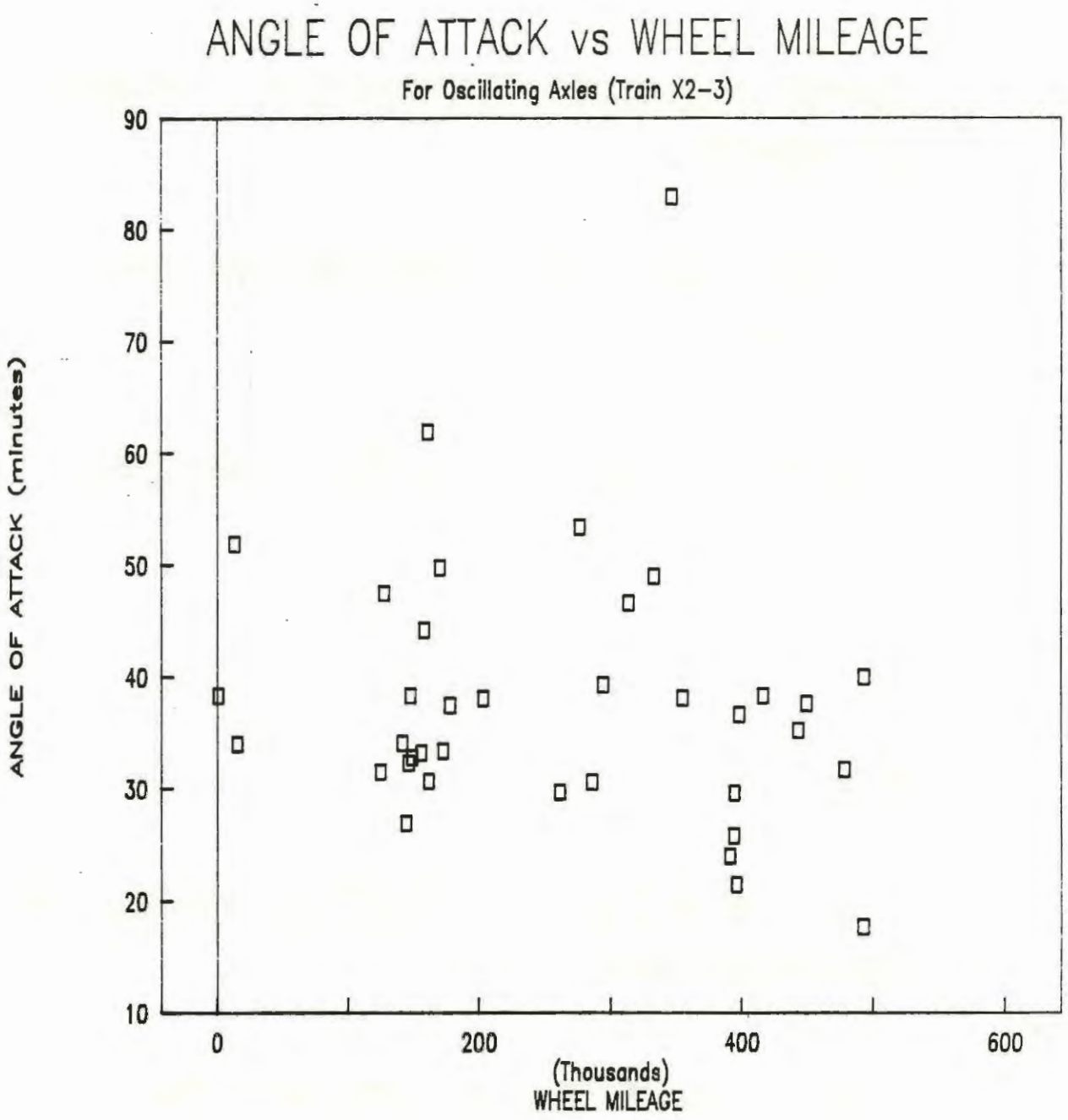
LEAD AXLE, LEAD TRUCK

47.6 MIN. AOA **33.0 MIN. AOA**

LOW RAIL LATERAL FORCE ON AXLE WITH HIGH ANGLE-OF-ATTACK (CONVENTIONAL TRUCK)



TYPICAL LOW RAIL LATERAL FORCE PULSATION
CONVENTIONAL TRUCKS


ANGLE OF ATTACK vs WHEEL MILEAGE
For Oscillating Axles (Train X2-3)
ANGLE OF ATTACK (minutes)
90
80
70
60
50
40
30
20
10
0
200
400
600
(Thousands)
WHEEL MILEAGE

Part I:

1. Definition and significance of angle-of-attack.

2. Controlled consist test results on curves.

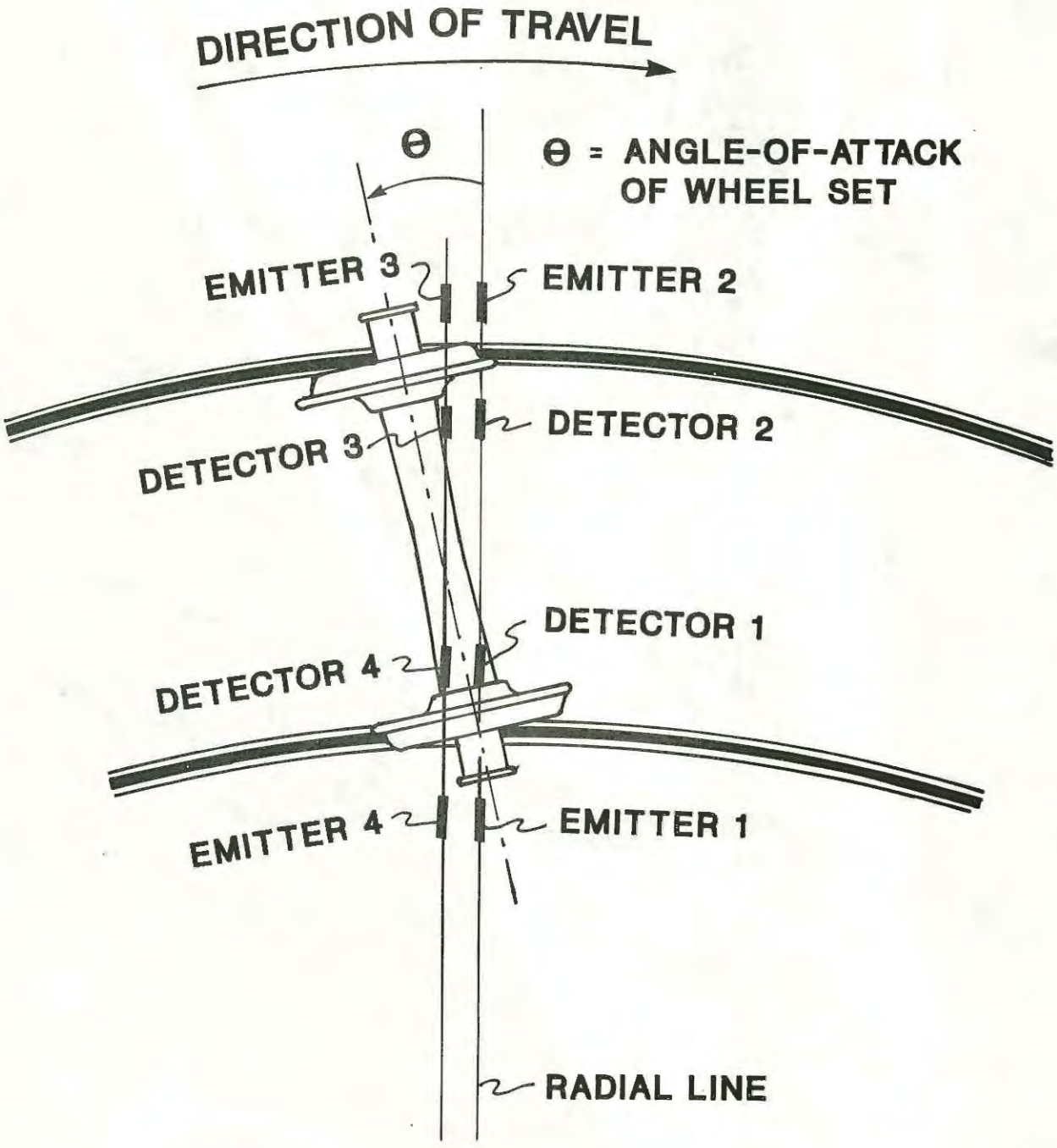
3. Train monitoring results on curves.

Part II:

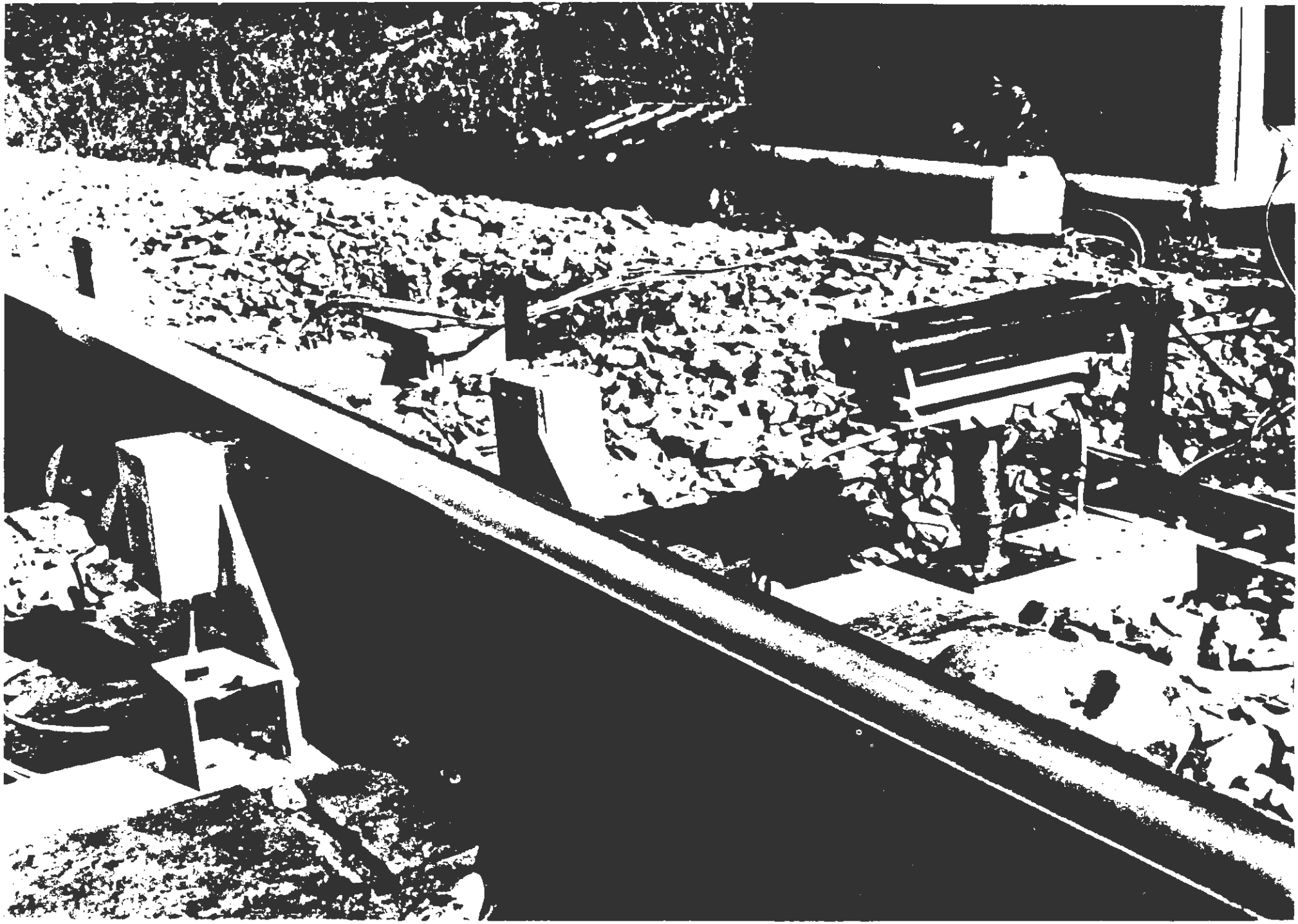
1. Permanent angle-of-attack hardware development.

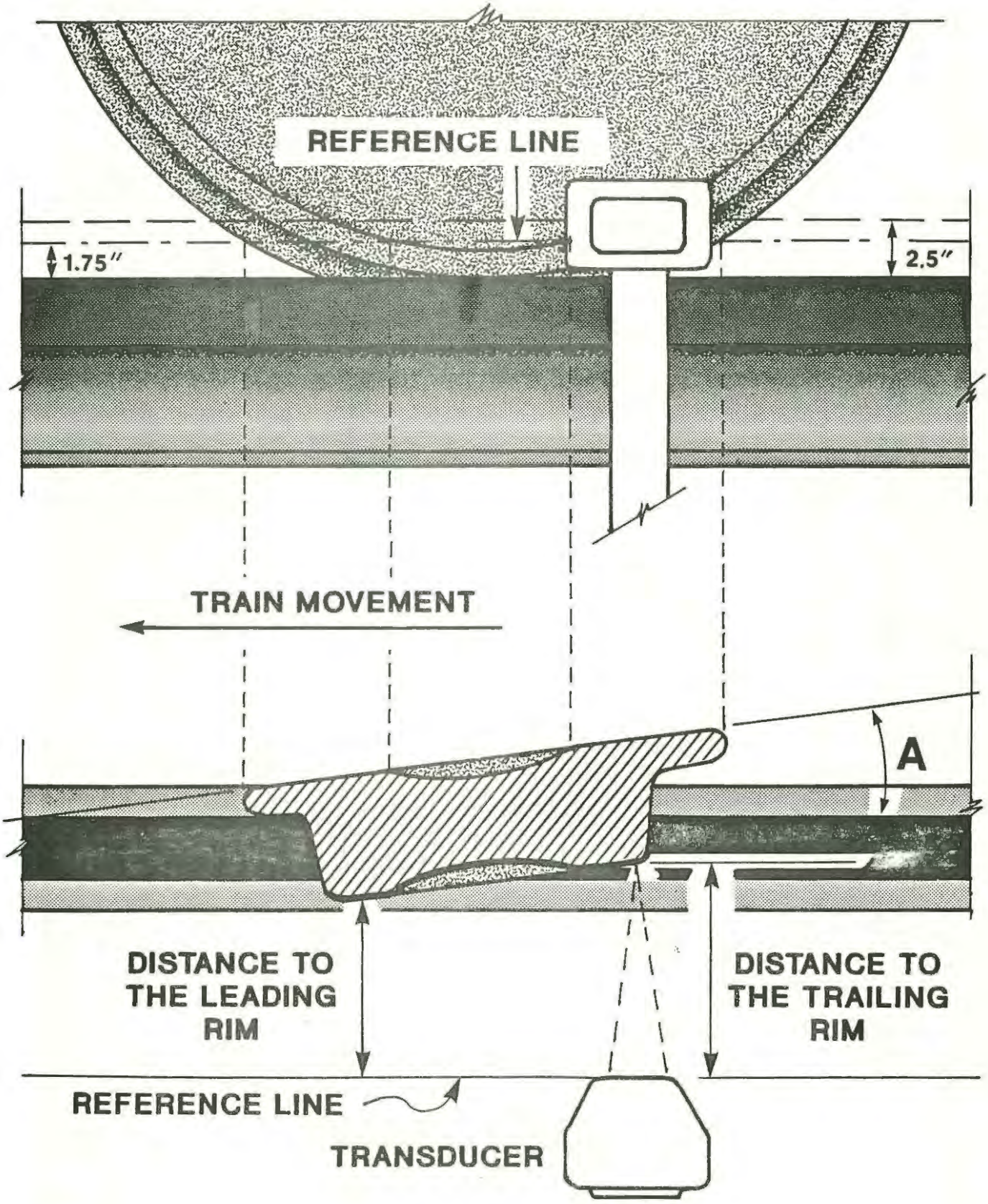
2. Train monitoring on tangent track.

TEST INSTRUMENTATION



ANGLE-OF-ATTACK MEASUREMENT SYSTEM

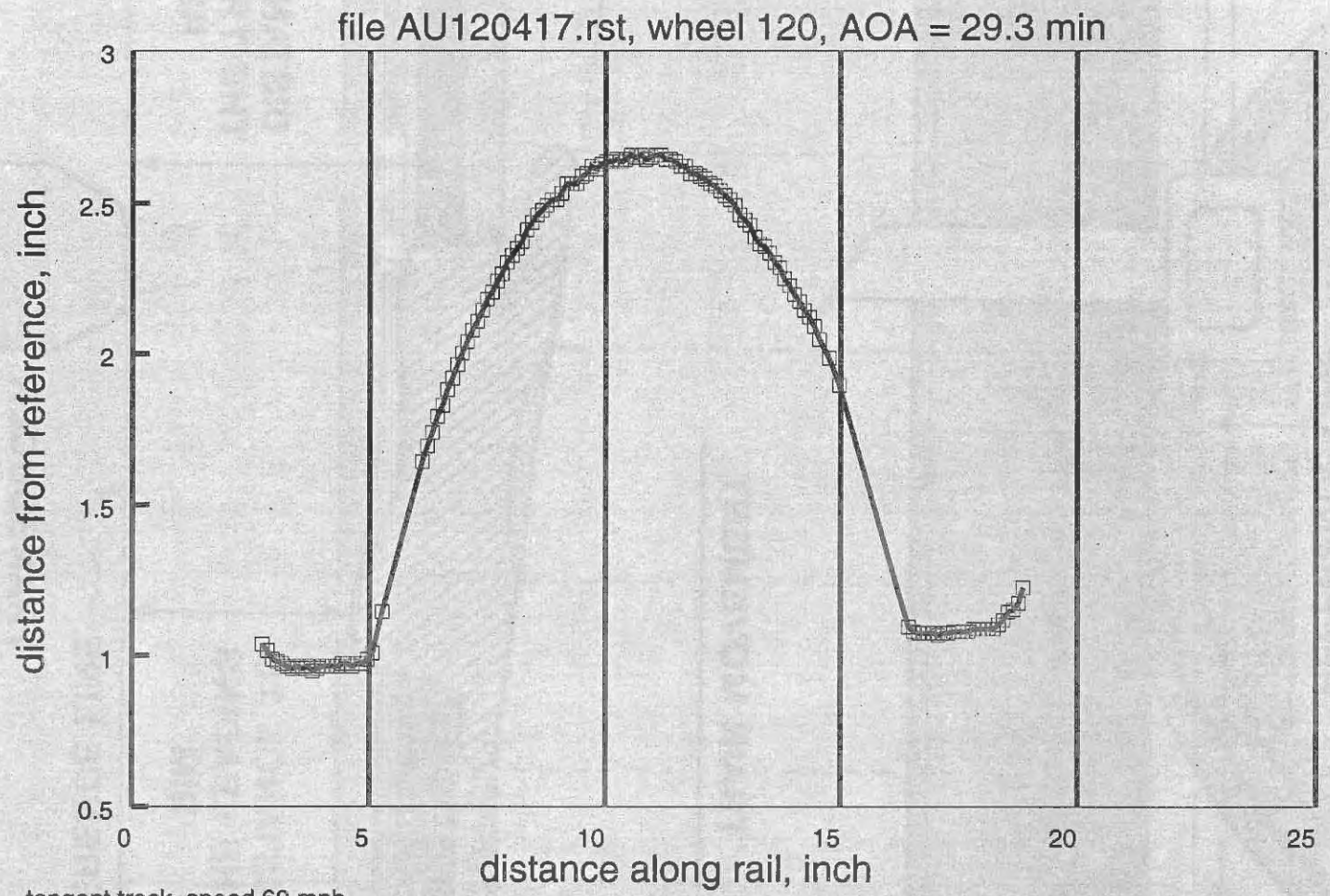




DIRECT MEASUREMENT OF ANGLE-OF-ATTACK

FIG. 5

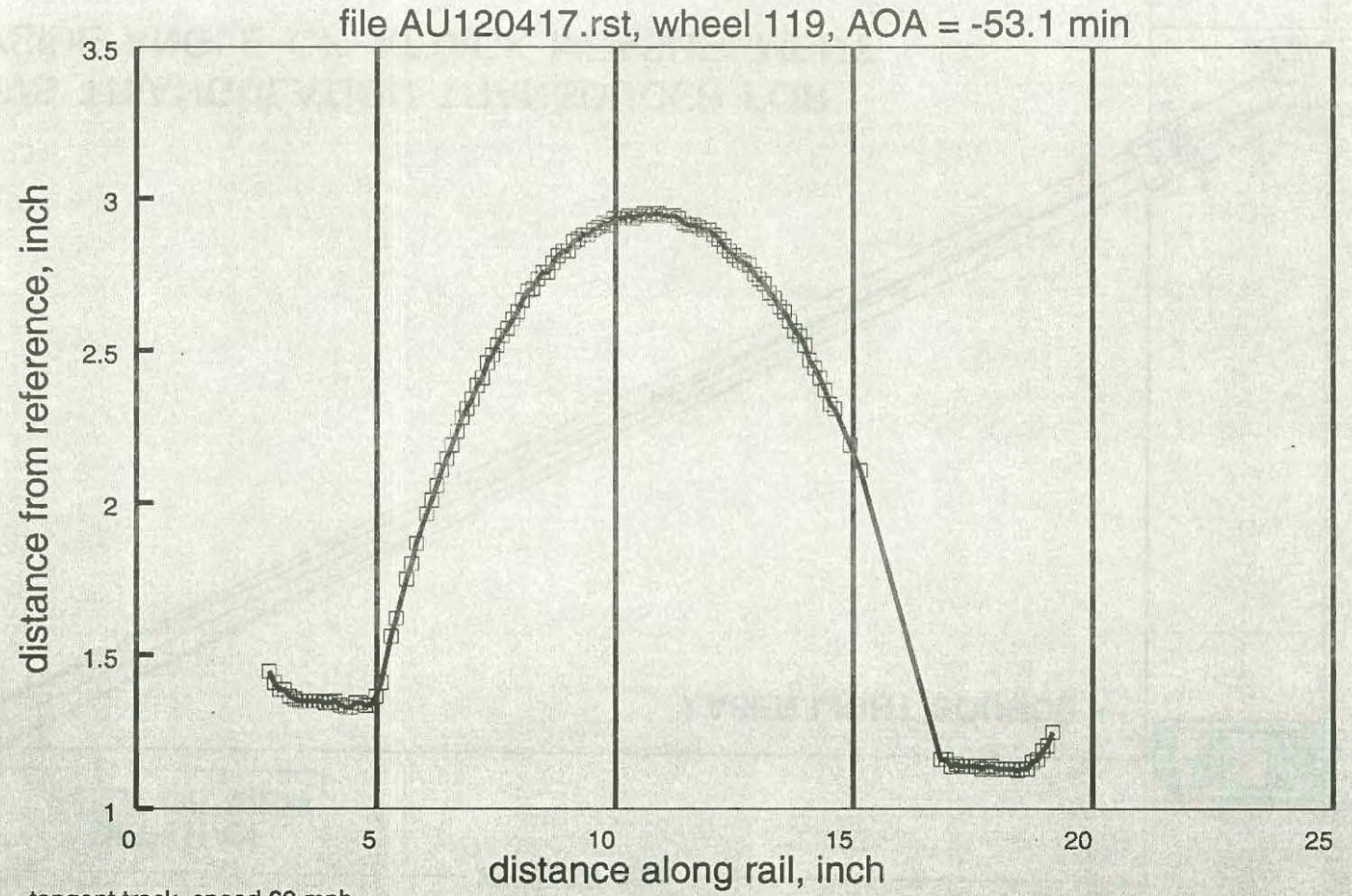
Angle-of-Attack at Ruby Creek, August 12, 1994



tangent track, speed 60 mph

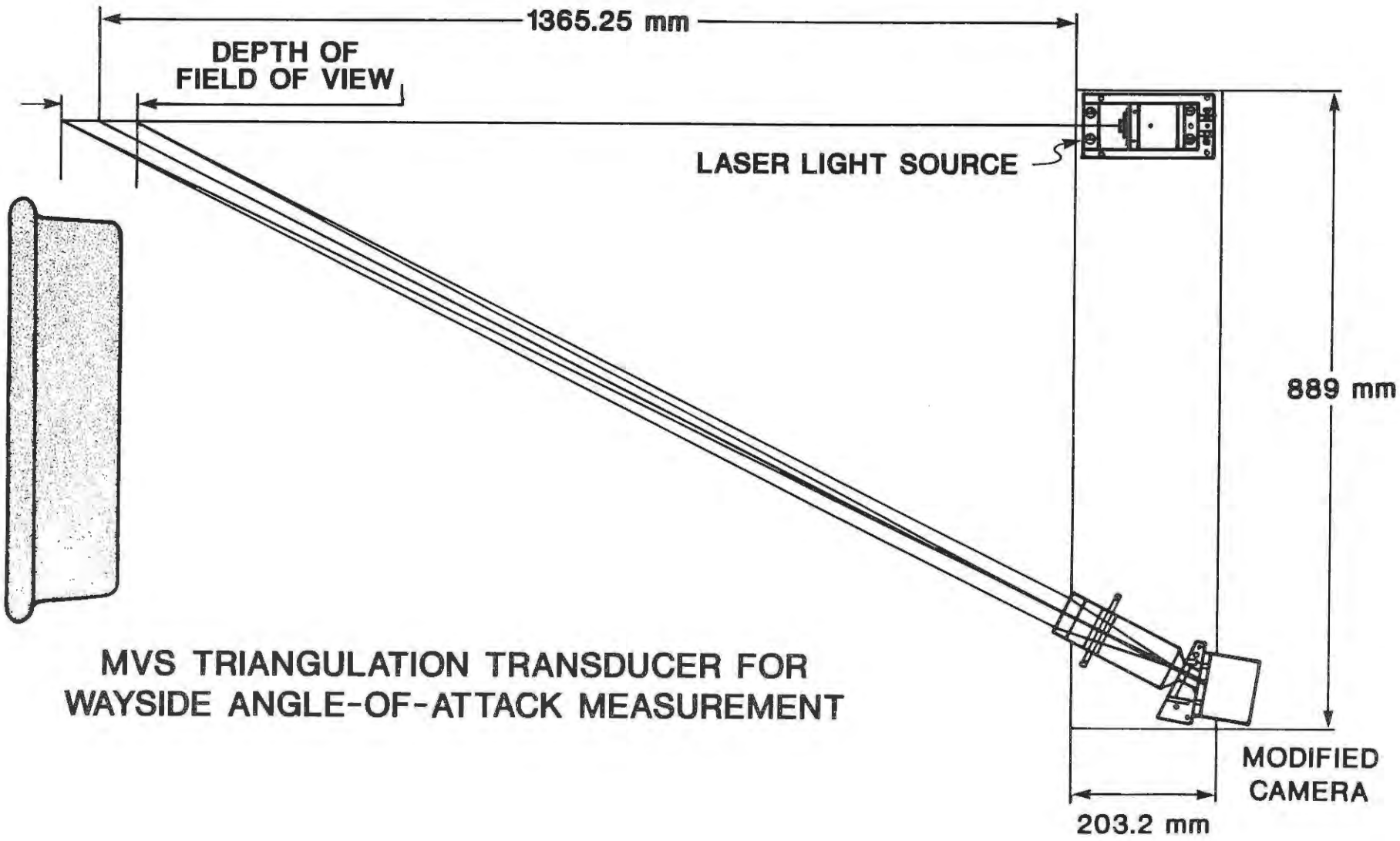
CP Rail System

Angle-of-Attack at Ruby Creek, August 12, 1994

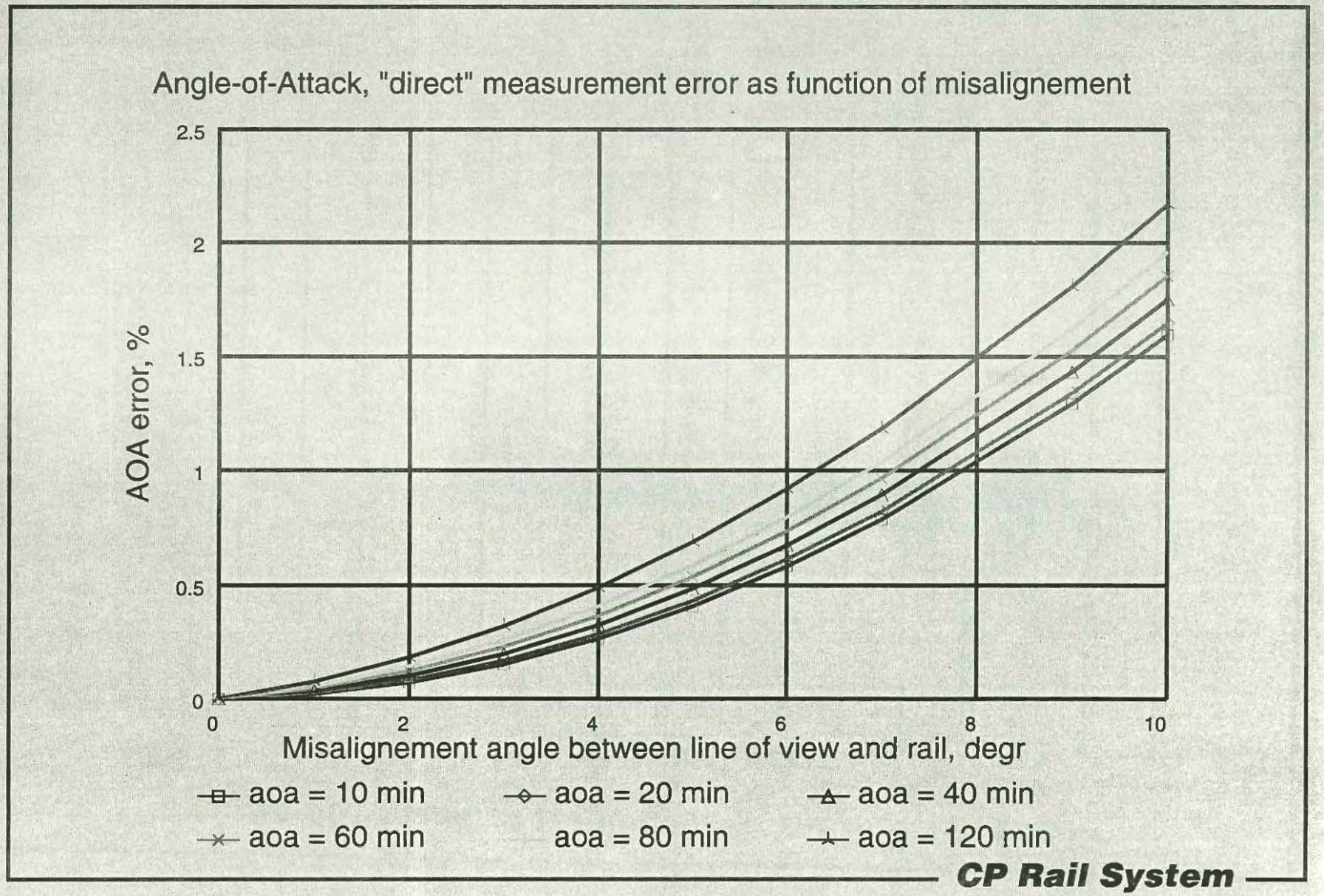


tangent track, speed 60 mph

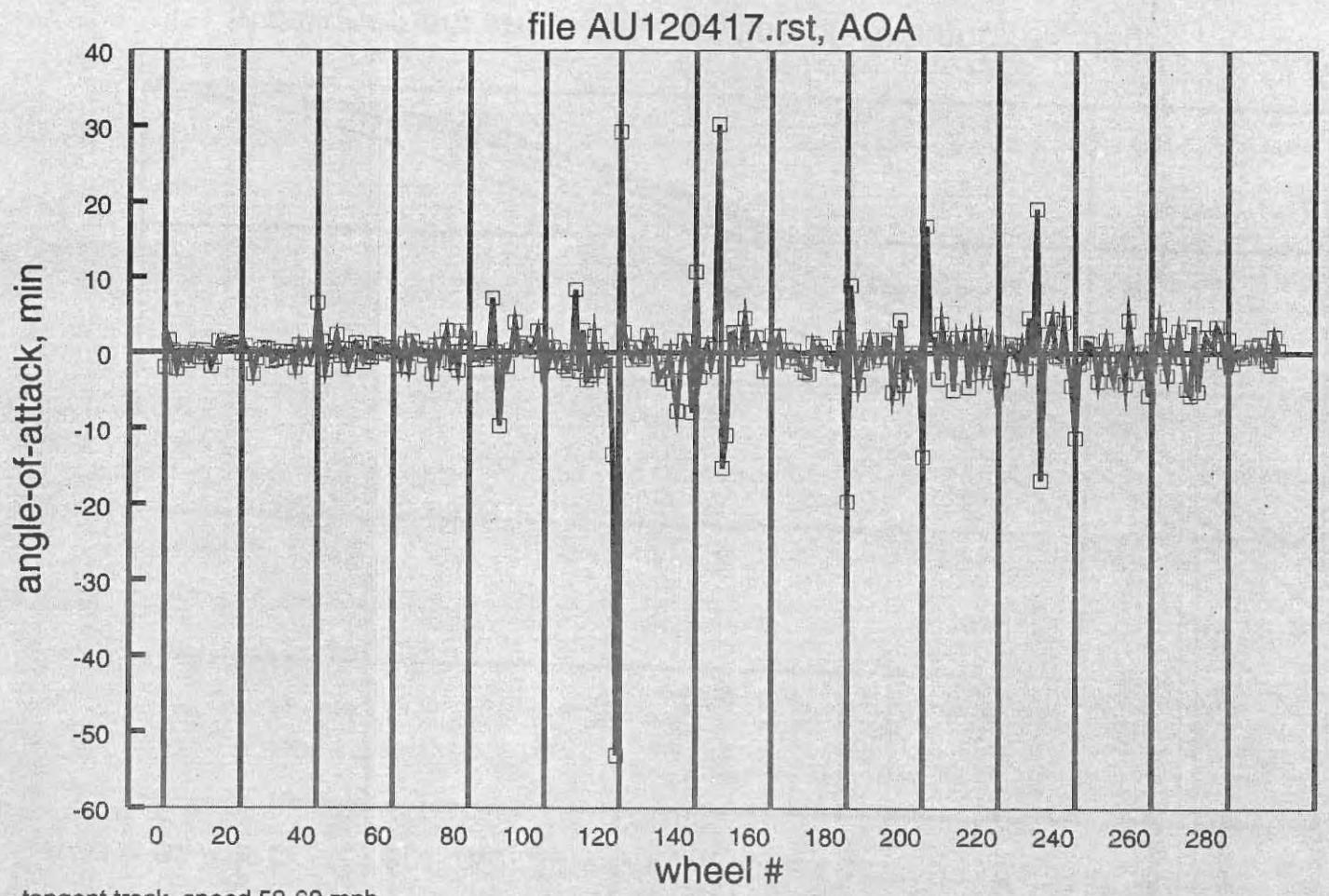
CP Rail System



MVS TRIANGULATION TRANSDUCER FOR
WAYSIDE ANGLE-OF-ATTACK MEASUREMENT


Angle-of-Attack, "direct" measurement error as function of misalignement
AOA error, %
2.5
2
1.5
1
0.5
0
0
2
4
6
8
10
Misalignement angle between line of view and rail, degr
aoa = 10 min
aoa = 20 min
aoa = 40 min
aoa = 60 min
aoa = 80 min
aoa = 120 min
CP Rail System

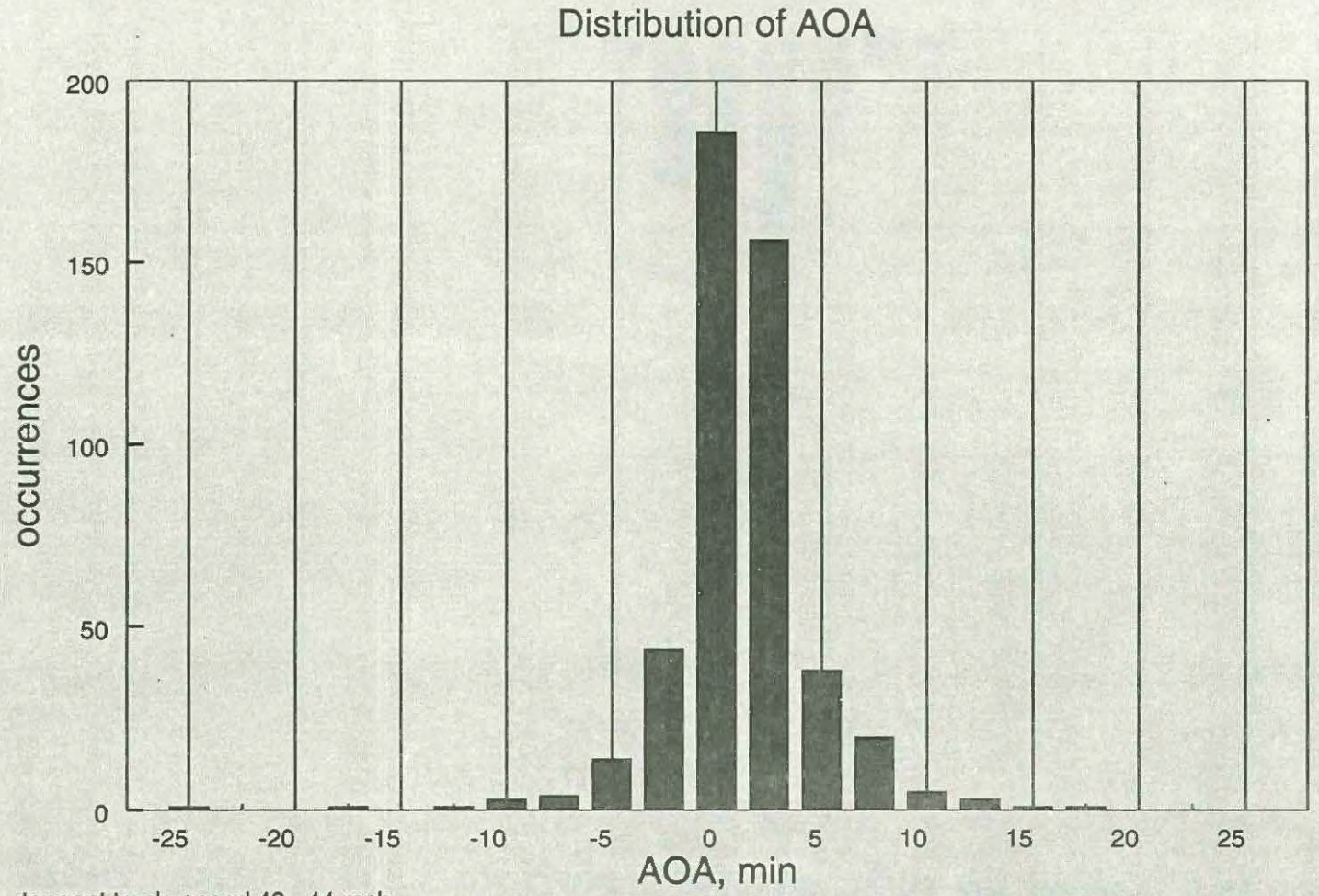
Angle-of-Attack at Ruby Creek, August 12, 1994



tangent track, speed 59-60 mph

CP Rail System

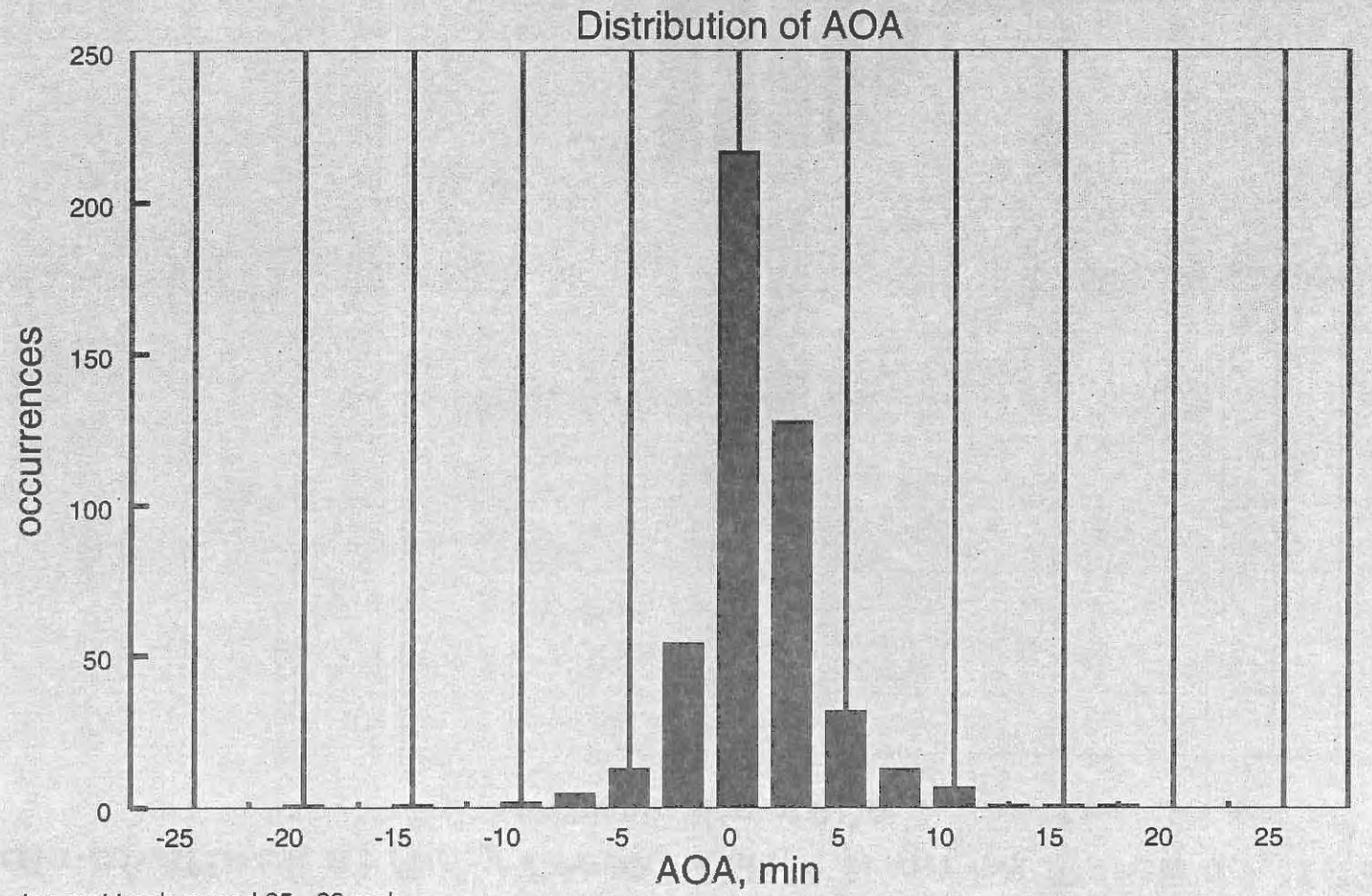
Angle-of-Attack at Ruby Creek, AQUA train on March 11, 1994



tangent track, speed 49 - 44 mph
min: -25.4 min, max: 16.5 min, average: -0.078 min, st.dev: 3.457

CP Rail System

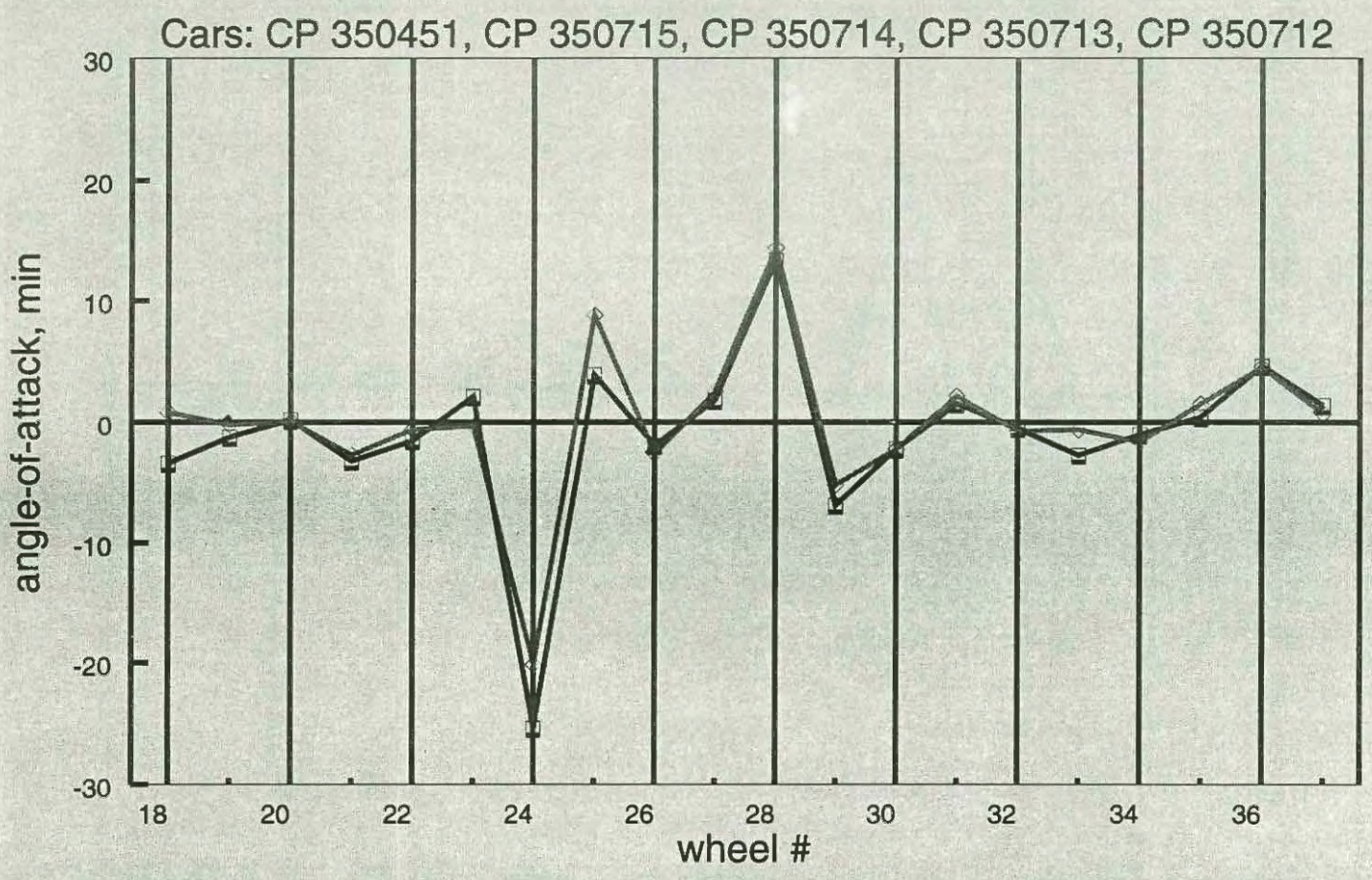
Angle-of-Attack at Ruby Creek, AQUA train on March 14, 1994



tangent track, speed 35 - 38 mph
min: -20.2 min, max: 15.4 min, average: -0.348 min, st.dev: 3.161

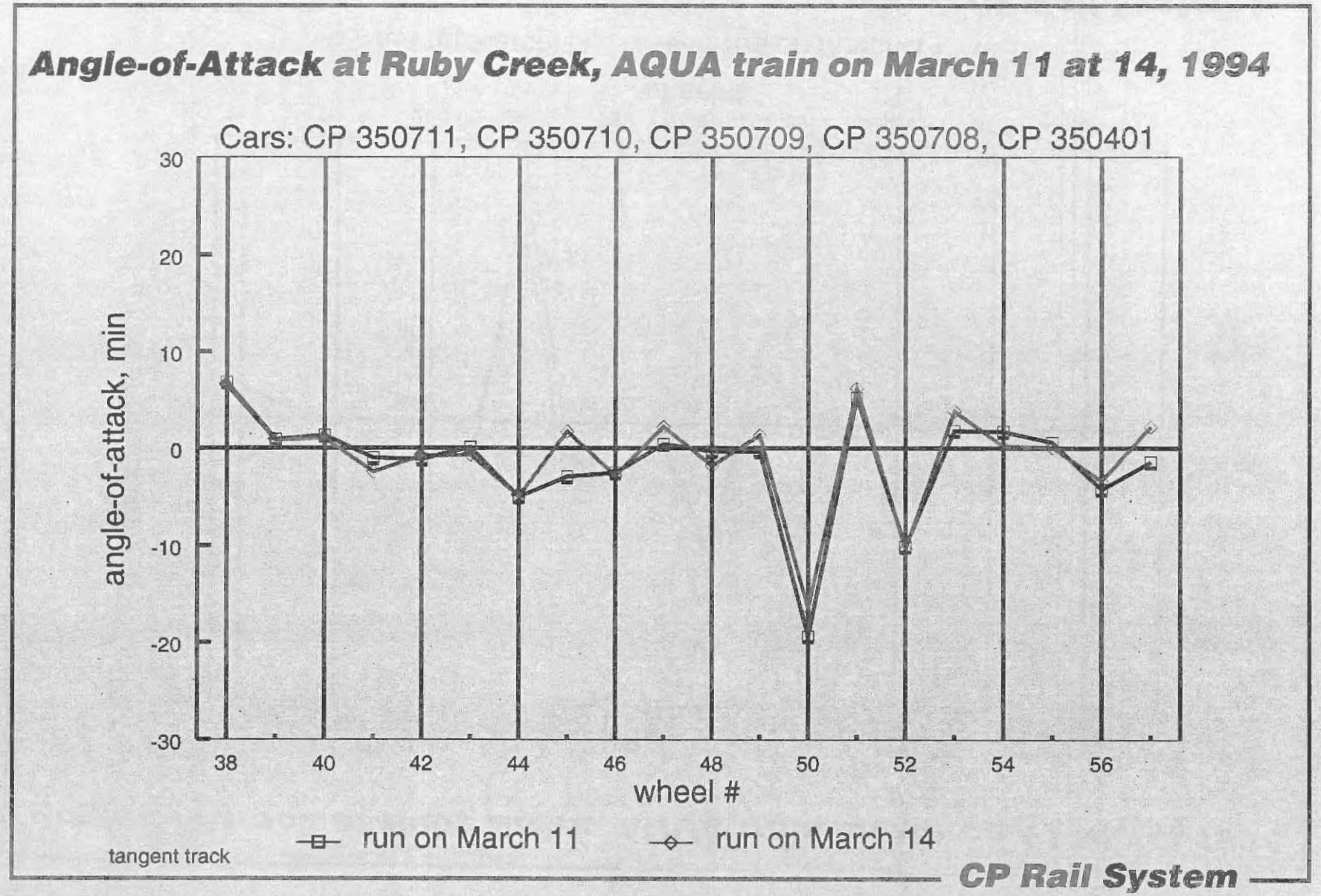
CP Rail System

Angle-of-Attack at Ruby Creek, AQUA train on March 11 at 14, 1994

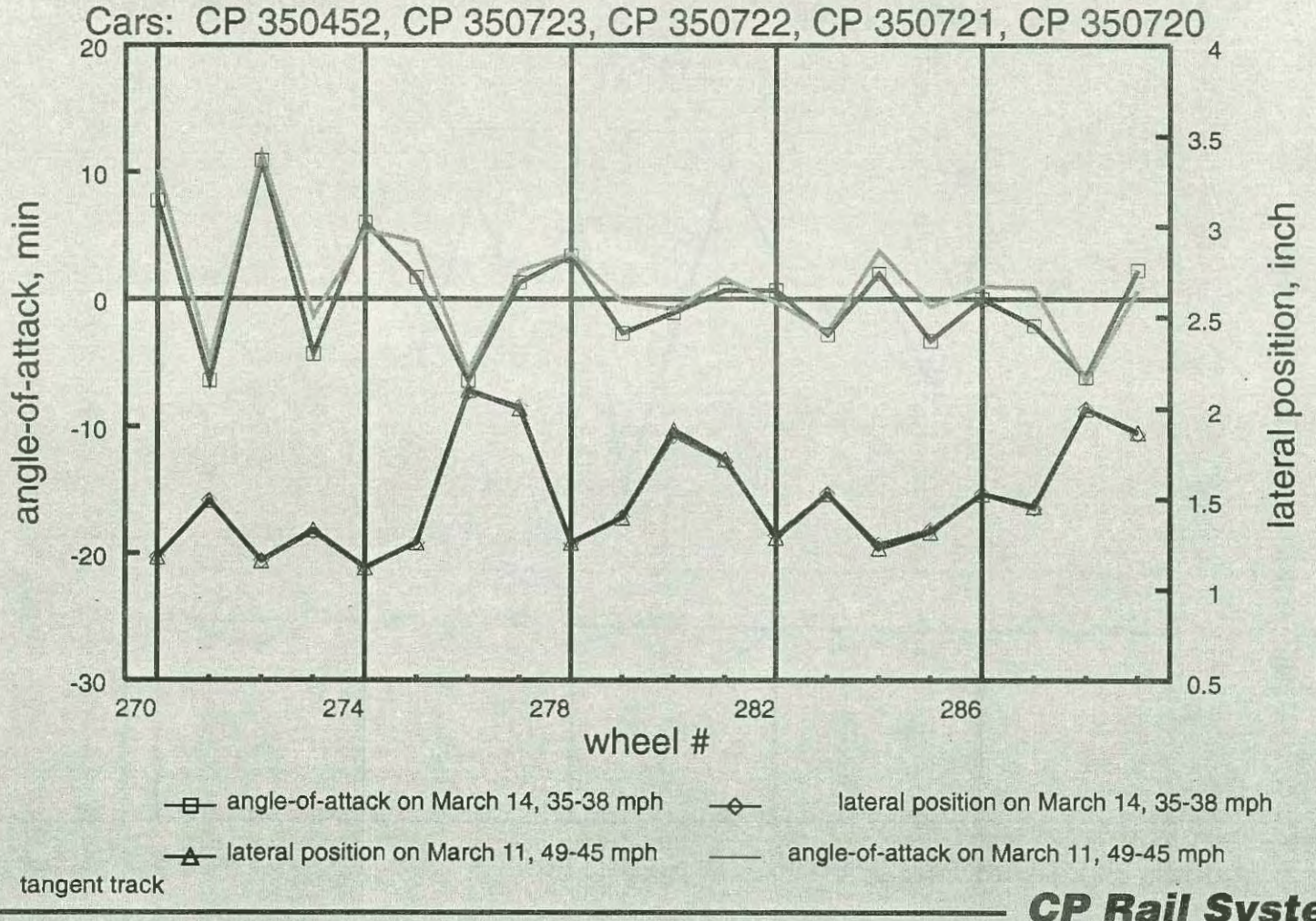


tangent track

CP Rail System


Angle-of-Attack at Ruby Creek, AQUA train on March 11 at 14, 1994
Cars: CP 350711, CP 350710, CP 350709, CP 350708, CP 350401
30
20
10
0
-10
-20
-30
angle-of-attack, min
38
40
42
44
46
48
50
52
54
56
wheel #
run on March 11
run on March 14
tangent track
CP Rail System

Angle-of-Attack at Ruby Creek, AQUA train on March 11 & 14, 1994

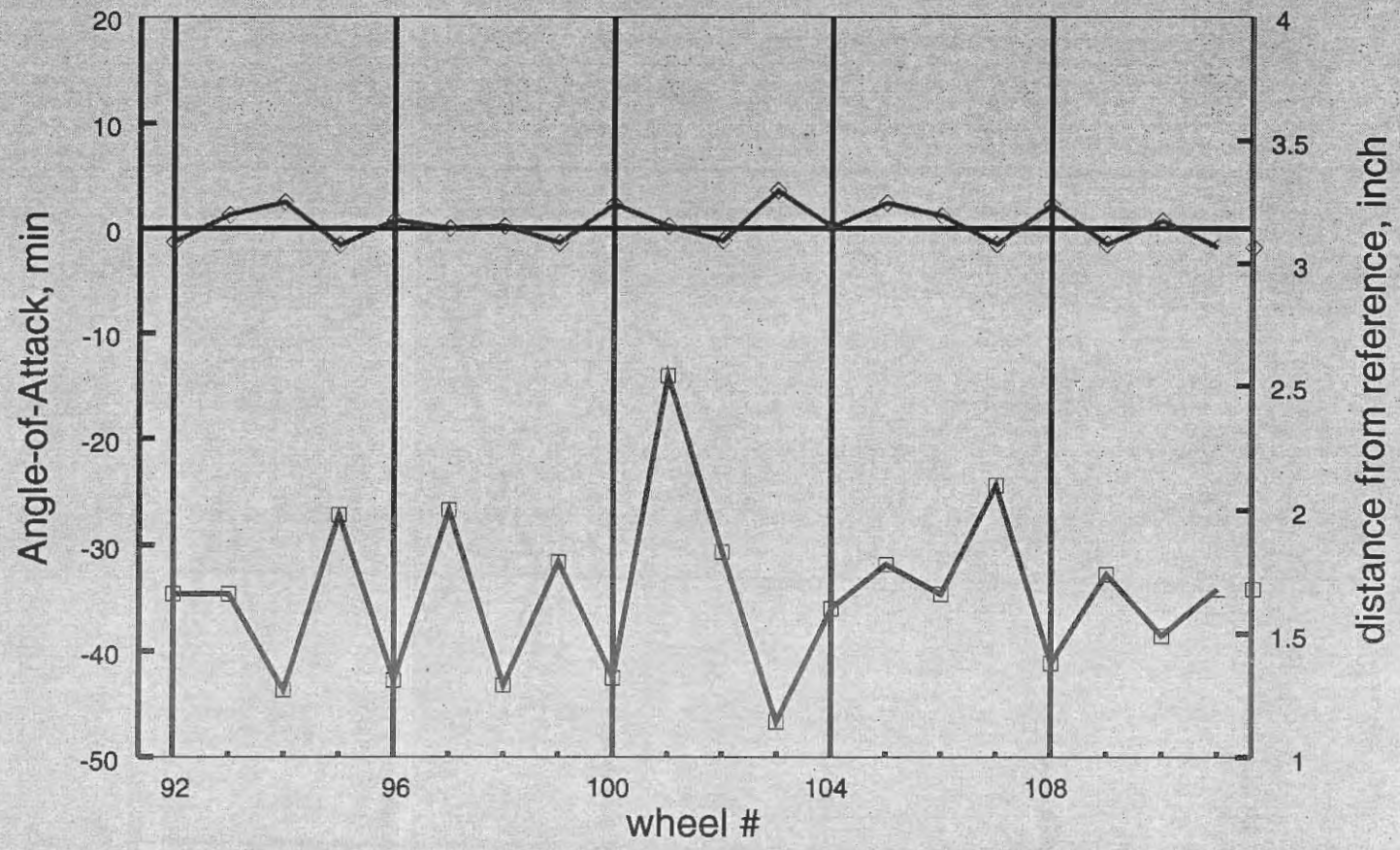


tangent track

CP Rail System

Angle-of-Attack at Ruby Creek, March 13, 1994, 12:06, mix, 48 mph

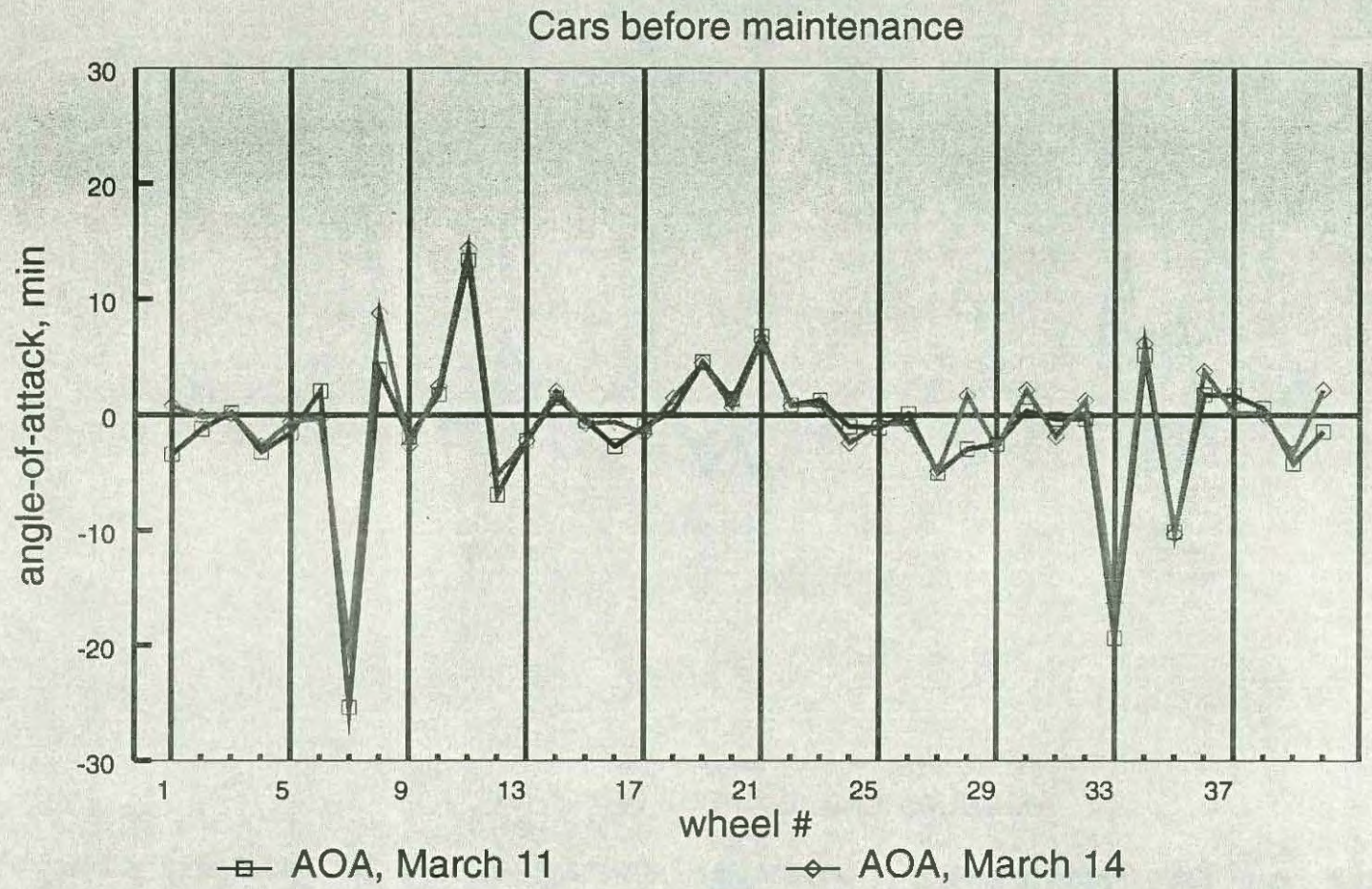
Cars: CPWX 600684, MILW 81243, CPAA 209008, CP 208651, CP 209907



tracking error (ax.100-ax.101) = 1.22 inch

CP Rail System

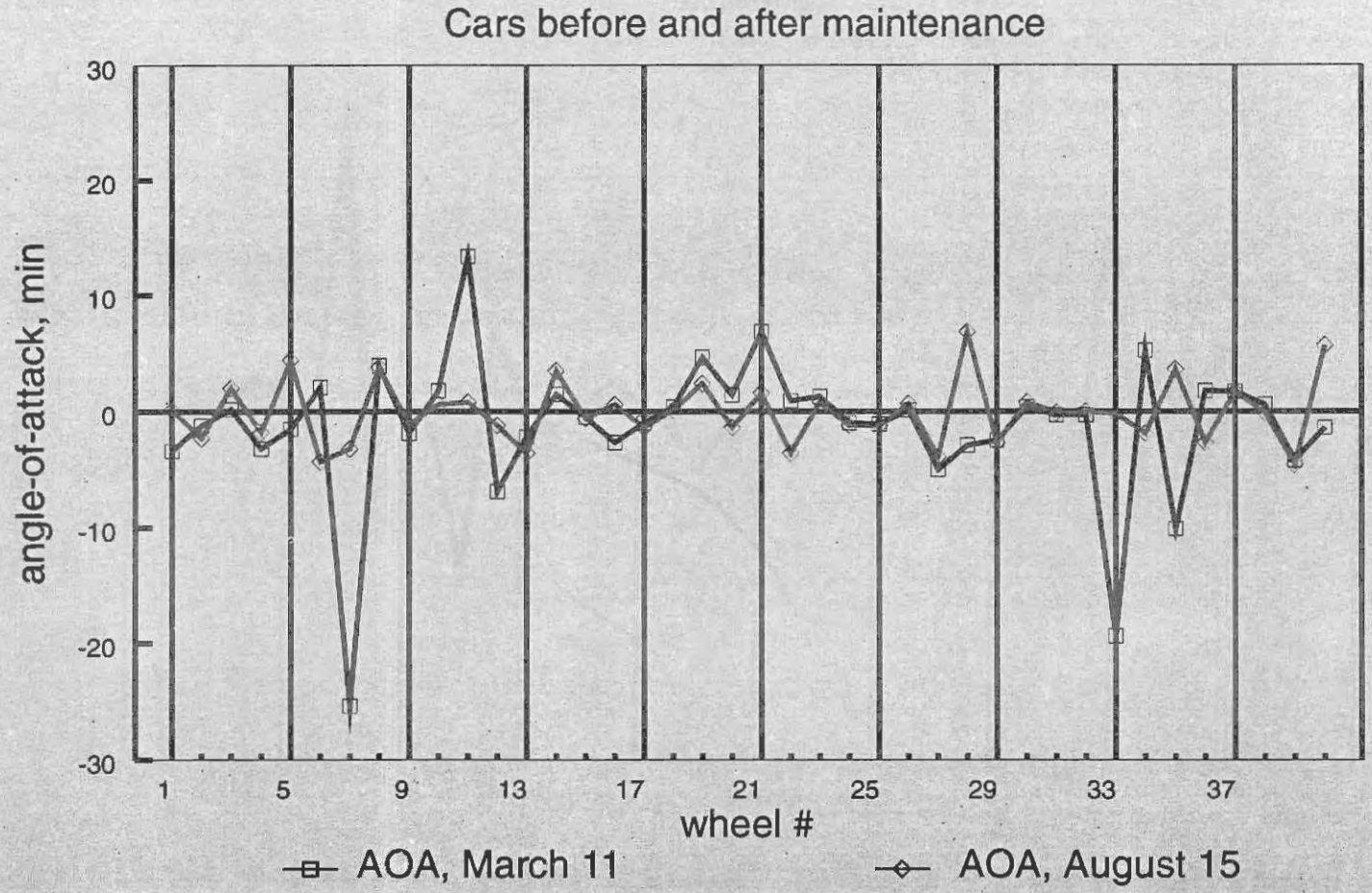
Angle-of-Attack at Ruby Creek, March 11 and 14, 1994



cars CP 350451, cp 350715-cp350708, cp 350401; maintenance in July
tangent track

CP Rail System

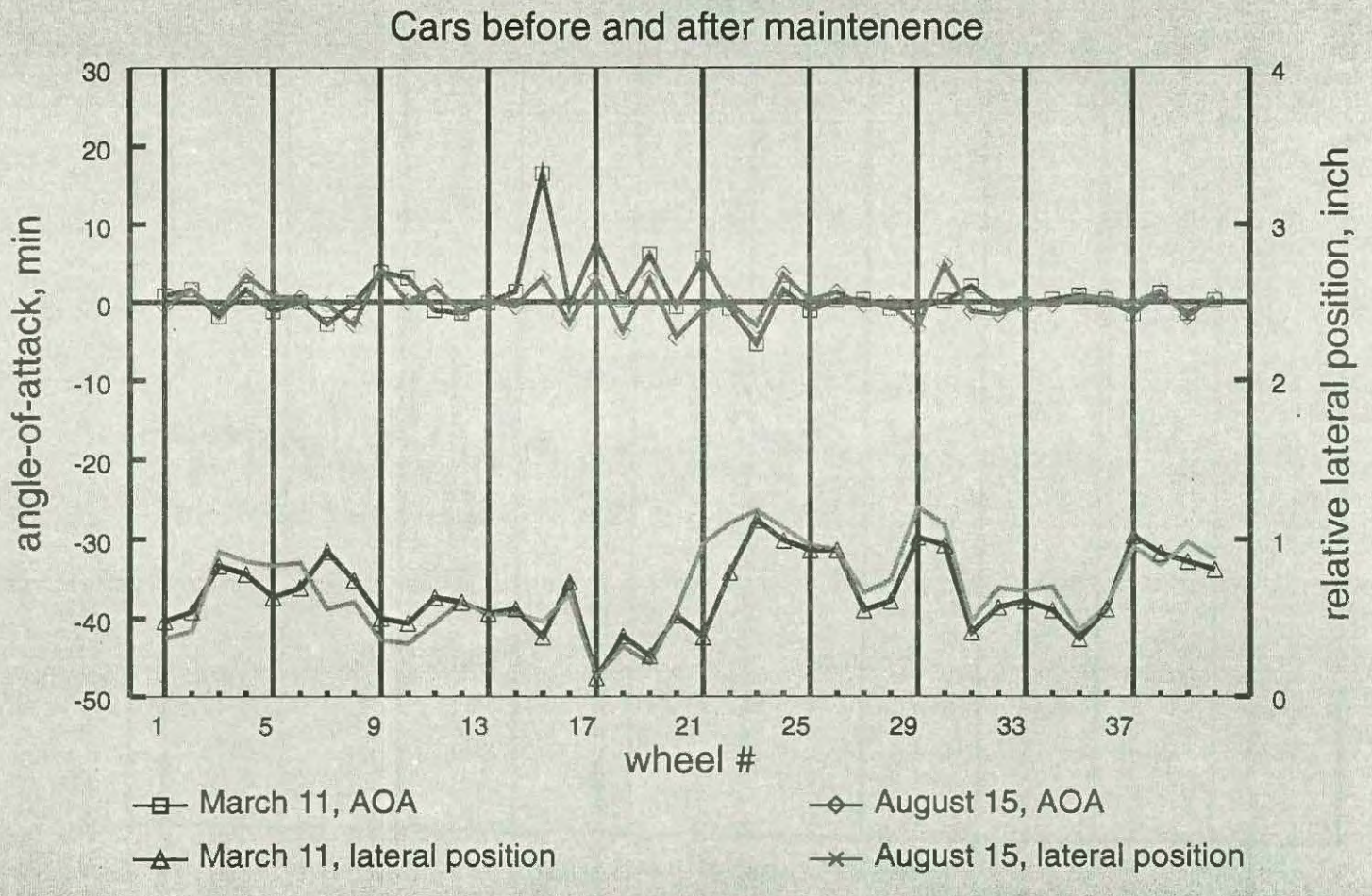
Angle-of-Attack at Ruby Creek, March 11 and August 15, 1994



cars CP 350451, cp 350715-cp350708, cp 350401; maintenance in July
tangent track

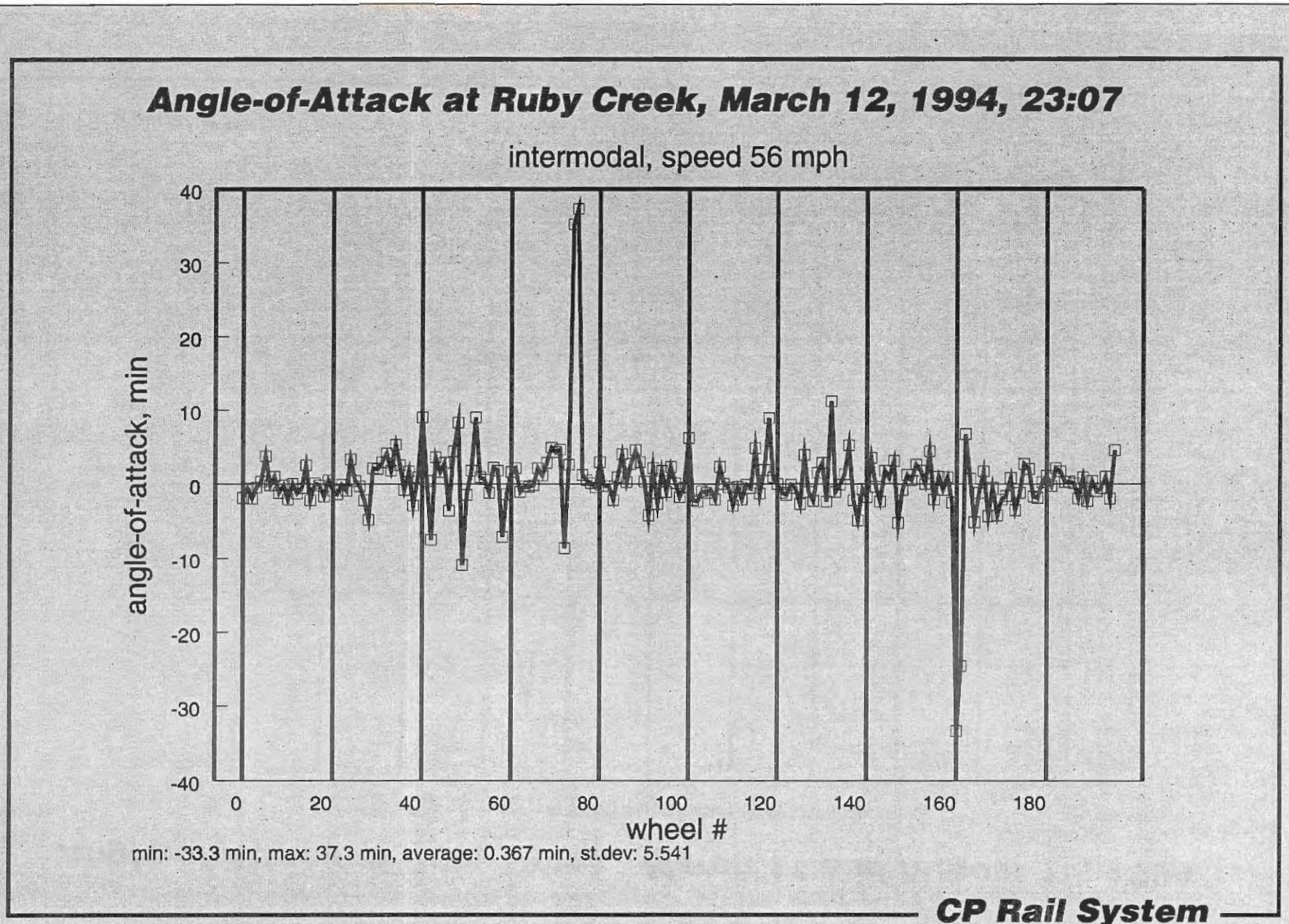
CP Rail System

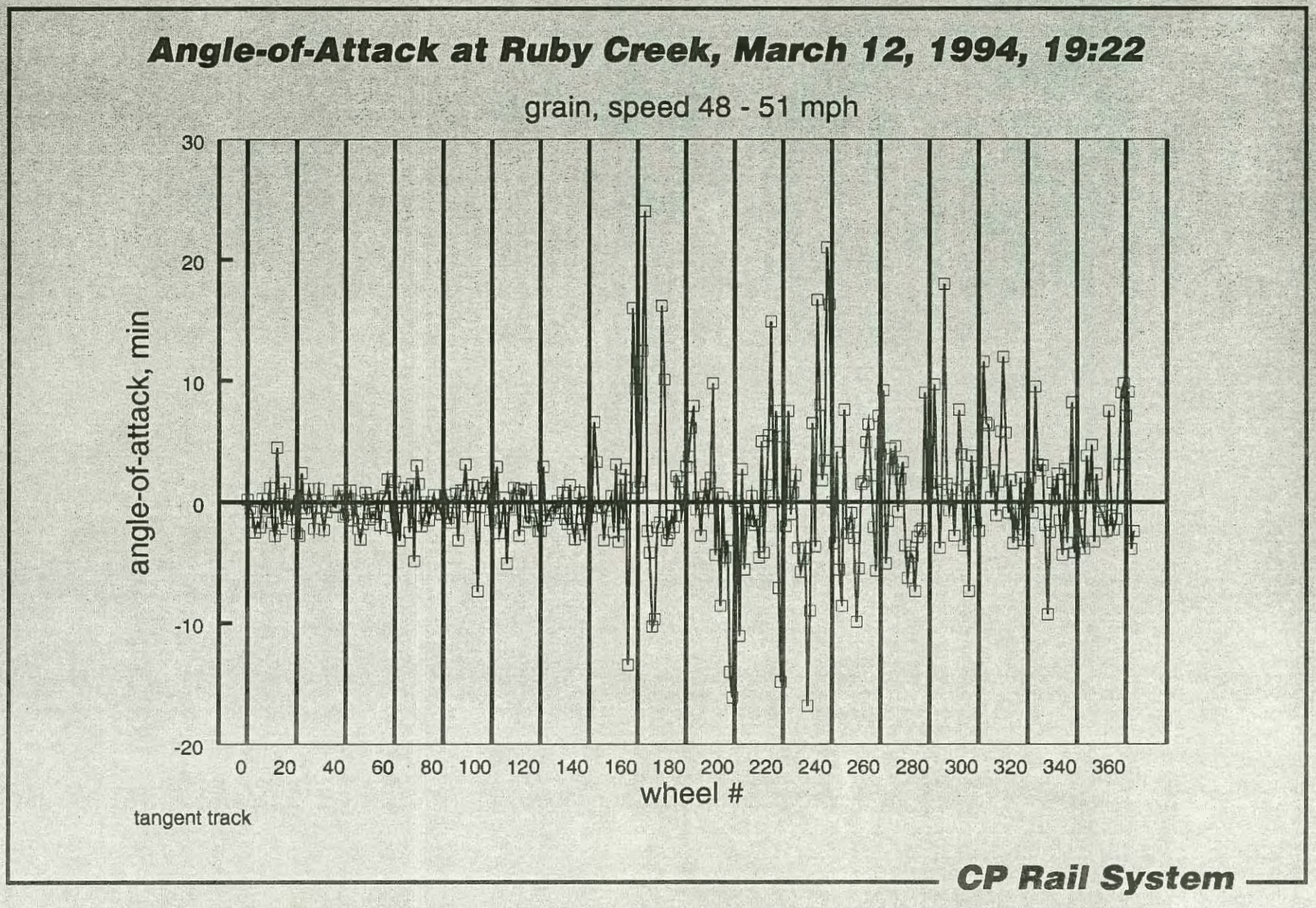
Angle-of-Attack at Ruby Creek, March 11 and August 15, 1994



wheelset #15 found defective and replaced in April
tangent track

CP Rail System


Angle-of-Attack at Ruby Creek, March 12, 1994, 23:07
intermodal, speed 56 mph
angle-of-attack, min
40
30
20
10
0
-10
-20
-30
-40
0
20
40
60
80
100
120
140
160
180
wheel #
min: -33.3 min, max: 37.3 min, average: 0.367 min, st.dev: 5.541
CP Rail System

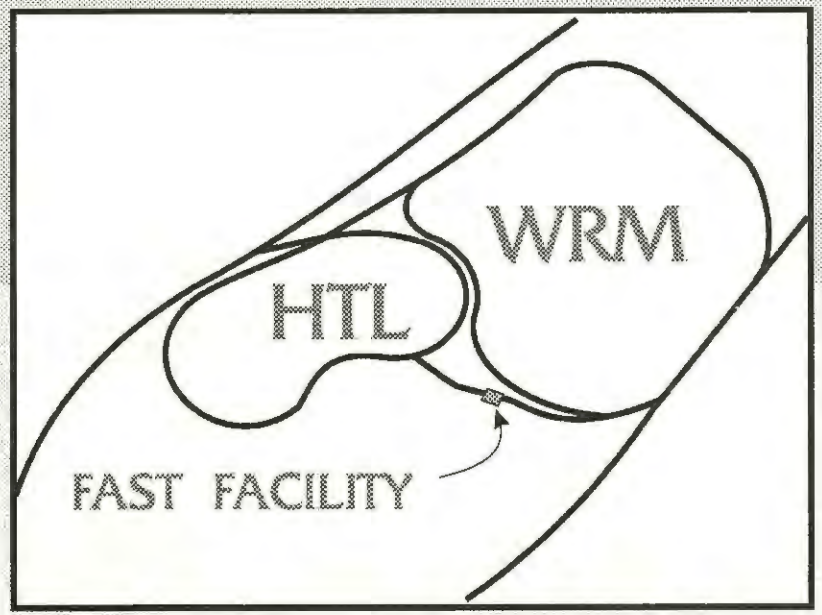

Angle-of-Attack at Ruby Creek, March 12, 1994, 19:22
grain, speed 48 - 51 mph
angle-of-attack, min
30
20
10
0
-10
-20
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360
wheel #
tangent track
CP Rail System

HEAVY AXLE LOAD/ALTERNATIVE TRUCK SUSPENSION MODELING AND TEST

R. L. FLOROM

ASSOCIATION OF AMERICAN RAILROADS

HAL Alternative Suspension Systems



FLOROM00.cdr

HAL ALTERNATIVE SUSPENSION SYSTEMS

▲ PRIMARY OBJECTIVE:

- DETERMINE THE EFFECTS OF ALTERNATIVE SUSPENSION DESIGNS TO REDUCE COSTS OF HEAVY AXLE LOADS ASSOCIATED WITH TRACK DEGRADATION

▲ SECONDARY OBJECTIVE:

- PROVIDE DYNAMIC PERFORMANCE DATA FOR THE ESTABLISHMENT OF 125-TON PERFORMANCE SPECIFICATIONS



HAL ALTERNATIVE SUSPENSION SYSTEMS

GOALS

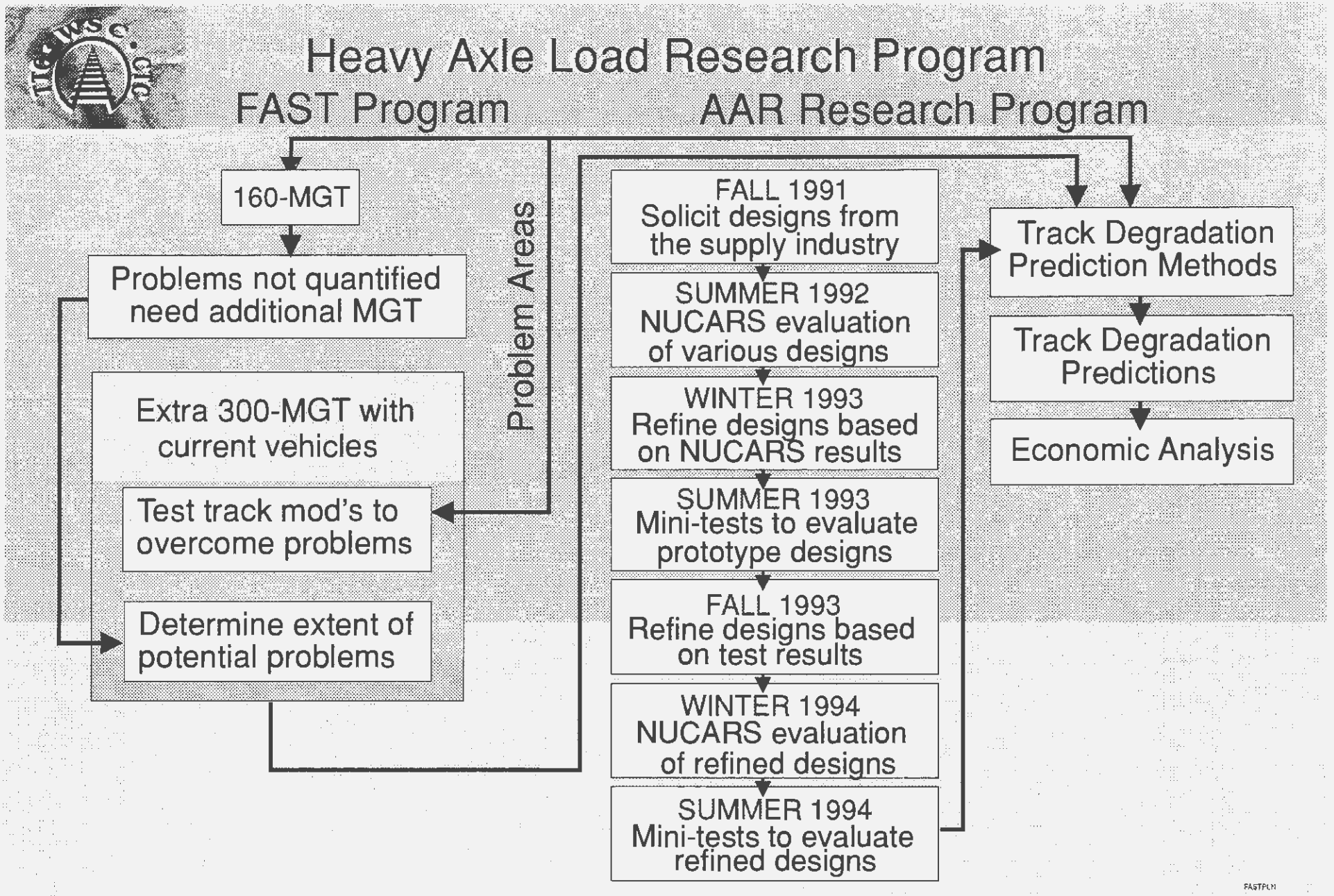
▲ SHORT-TERM GOALS:

- ENCOURAGE DEVELOPMENT OF ALTERNATIVE SUSPENSION DESIGNS
- DEMONSTRATE USE OF ANALYTICAL TOOLS FOR EVALUATION.
- SELECT BEST ALTERNATIVE DESIGN(S)
- CONDUCT FULL-SCALE PERFORMANCE EVALUATION ON TTC TRACKS.
- EQUIP HAL TRAIN WITH ADVANCED TRUCKS

▲ LONG-TERM GOALS:

- REDUCE COSTS OF HEAVY AXLE LOADS.
- REDUCE DYNAMIC LOADS INTO TRACK & CAR BODY STRUCTURE
- PERFORM ECONOMIC STUDIES TO QUANTIFY THE EFFECTS OF SUSPENSION IMPROVEMENTS ON REDUCED TRACK DAMAGE.
- DEVELOP 125-TON TRUCK PERFORMANCE SPECIFICATION.




Heavy Axle Load Research Program
FAST Program
AAR Research Program
160-MGT
Problems not quantified need additional MGT
Extra 300-MGT with current vehicles
Test track mod's to overcome problems
Determine extent of potential problems
Problem Areas
FALL 1991
Solicit designs from the supply industry
SUMMER 1992
NUCARS evaluation of various designs
WINTER 1993
Refine designs based on NUCARS results
SUMMER 1993
Mini-tests to evaluate prototype designs
FALL 1993
Refine designs based on test results
WINTER 1994
NUCARS evaluation of refined designs
SUMMER 1994
Mini-tests to evaluate refined designs
Track Degradation Prediction Methods
Track Degradation Predictions
Economic Analysis
FASTPLN

HAL ALTERNATIVE SUSPENSION SYSTEMS

BENEFITS OF NEW TRUCK DESIGNS

- PREMIUM TRUCKS PROVIDE:
  - IMPROVEMENTS IN STEERING WHICH REDUCE LATERAL LOADS
  - IMPROVEMENTS IN SUSPENSION SYSTEMS REDUCE DYNAMIC VERTICAL LOADS
  - HIGH WARP/SHEAR STIFFNESS REDUCES CURVING FORCES AND THE LIKELIHOOD OF HUNTING
- REDUCED TRACK AND VEHICLE MAINTENANCE
  - TRACK COMPONENT WEAR AND DAMAGE
  - VEHICLE AND TRUCK COMPONENT WEAR AND DAMAGE
- REDUCED FUEL CONSUMPTION
- REDUCED LIKELIHOOD OF DERAILMENT

HAL ALTERNATIVE SUSPENSION SYSTEMS

TRUCK SELECTION METHODOLOGY

- NUCARS MODELING
  - PITCH AND BOUNCE (L)
  - TWIST AND ROLL (L)
  - S.S. CURVING & ENTRY/EXIT (L)
  - HTL/FAST MEASURED TRACK GEOMETRY (L)
  - DYNAMIC CURVING (L)
  - YAW AND SWAY (L)
  - HUNTING (E)
- MINI-TEST ON HTL
- SELECT TEST TRUCKS FOR CHAPTER XI TEST
- CHAPTER XI TEST OF CANDIDATE TRUCKS
- SELECT CANDIDATE TRUCKS
- FULL-SCALE TEST AT FAST WITH PREMIUM TRUCKS



HAL ALTERNATIVE SUSPENSION SYSTEMS

METHODOLOGY FOR ECONOMIC ANALYSIS

- BASELINE COSTS AGAINST IMPROVED TRUCK
- "TYPICAL" U.S. BULK UNIT TRAIN OPERATIONS
- ACCIDENT COSTS (FRA & R-1 REPORTS)
- EQUIPMENT MAINTENANCE COSTS (ERAS DATA BASE)
- TRACK MAINTENANCE COSTS (TRACS)
- ENERGY COSTS (TEM/RECAP)



HAL ALTERNATIVE SUSPENSION SYSTEMS

ESTIMATED SAVING

COST CATEGORY	SAVINGS ($/1000 MILES)	ANNUAL SAVINGS ($) 50,000 MILES	100,000 MILES
ACCIDENTS	1.08	54	108
FUEL	6.66	333	666
CAR MAINTENANCE	3.77	188	377
TRACK	9.70	485	970
WEIGHT REDUCTION	9.62	481	962
TOTAL	34.83	1,541	3,083
NET PRESENT VALUE		14,071	28,143



HAL ALTERNATIVE SUSPENSION SYSTEMS

VEHICLE MODELING ASSUMPTIONS

- EXISTING FAST CAR BODY
  - DIMENSIONS
  - CG
  - INERTIAS
- RIGID BODY (NO FLEXIBLE BODY MODES)
- LOADED 315,000 LB WHEEL ON RAIL
- EMPTY 70,000 LB WHEEL ON RAIL
- TRUCK PARAMETERS
  - IDEALIZED SPRING/DAMPERS
  - MANUFACTURER SUPPLIED CHARACTERISTICS
- FRICTION
  - FRICTION WEDGES = 0.40*
  - CENTER PLATE LUBED = 0.15
  - WHEEL/RAIL AS PER TEST REGIME

* UNLESS OTHERWISE SPECIFIED



HAL ALTERNATIVE SUSPENSION SYSTEMS

TRACK MODELING ASSUMPTIONS

- CHAPTER XI MATHEMATICALLY GENERATED DEFINITIONS
  - PERTURBED SECTIONS
    - TWIST & ROLL - 3/4" 39' SINUSOIDAL INPUT
    - PITCH & BOUNCE - 3/4" 39' SINUSOIDAL INPUT
    - DYNAMIC CURVING - 1/2" 39' SINUSOIDAL & 1" 39' CUSPS
    - YAW & SWAY 1 1/4" SINE WAVES WITH 1" WIDE GAGE
    - WHEEL/RAIL FRICTION = 0.40 HEAD AND FLANGE
    - SIMULATED NEW AAR-1B AND NEW 136 LB RAIL
  - CURVING SECTIONS
    - 3, 5, 7.5, 10 DEGREE CURVES
    - WHEEL/RAIL FRICTION
      - HIGH RAIL = 0.35 HEAD, = 0.15 FLANGE
      - LOW RAIL = 0.4 HEAD AND FLANGE
    - SIMULATED NEW AAR-IB AND NEW 136 LB. RAIL
- FAST/HTL REAL TRACK INPUT
  - MEASURED TRACK GEOMETRY FROM INERTIAL SYSTEM
  - SECTION 7, 25 & 33°(5 °, 6 & TANGENT)
  - WHEEL/RAIL FRICTION
    - OUTSIDE RAIL = 0.35 HEAD, = 0.15 FLANGE
    - INSIDE RAIL = 0.4 HEAD AND FLANGE
    - SIMULATED FAST WORN WHEEL/RAIL PROFILES

HAL ALTERNATIVE SUSPENSION SYSTEMS

TRACK TEST

OBJECTIVES

- IDENTIFY ANY DESIGN DEFICIENCIES
- ASSESS MAINTAINABILITY
- PROVIDE WEAR-IN (MAX. 20,000 MILES) PRIOR TO CHAPTER XI TEST
- VERIFY NUCARS PREDICTIONS

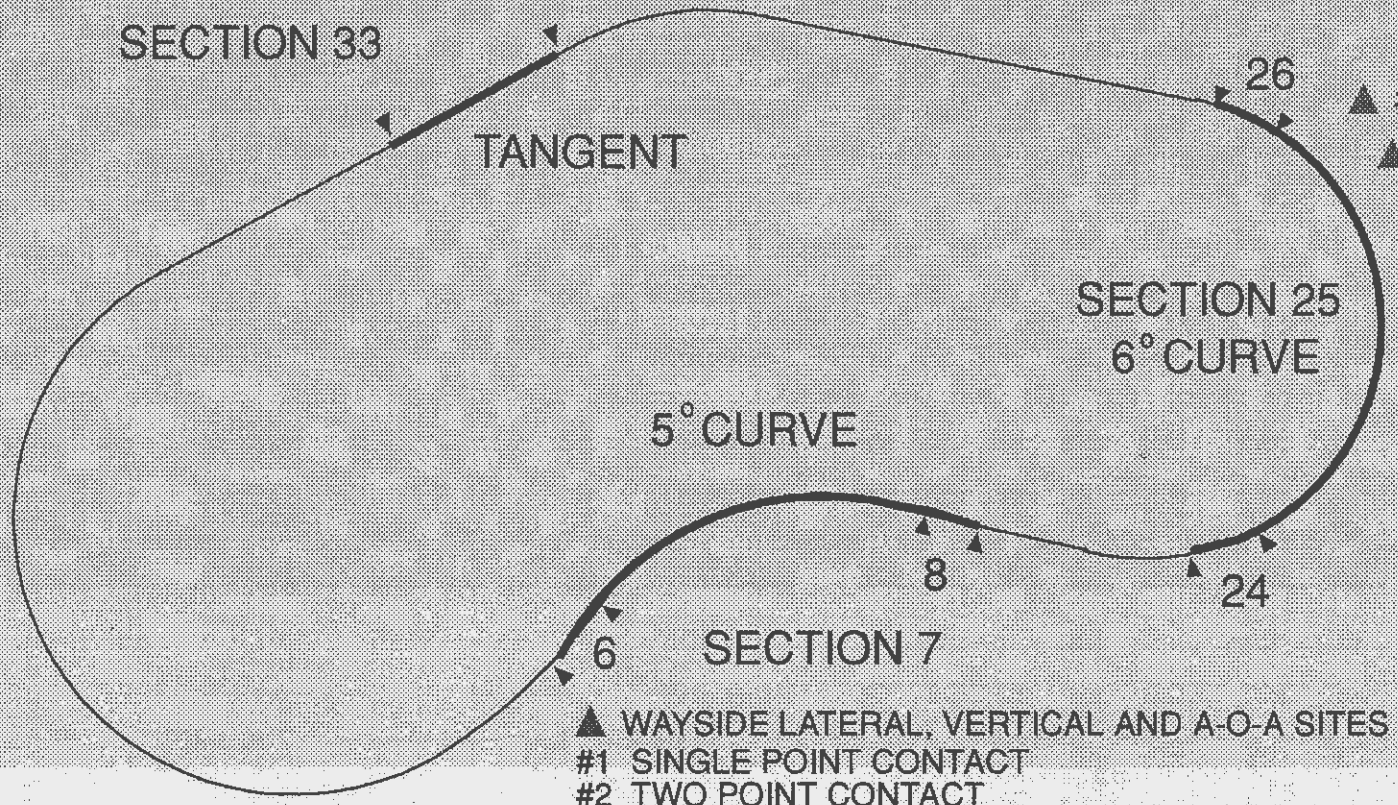
MEASUREMENTS

- LATERAL & VERTICAL WHEEL/RAIL FORCES FROM WAYSIDE
- ANGLE-OF-ATTACK (AOA)
- WHEEL/TRUCK COMPONENT WEAR
- DAILY INSPECTIONS/OBSERVATIONS



HAL ALTERNATIVE SUSPENSION SYSTEMS

FAST/HAL SIMULATIONS & MINI TEST SECTIONS



d:\research\urban\vts\93curve.cdr

HAL ALTERNATIVE SUSPENSION SYSTEMS

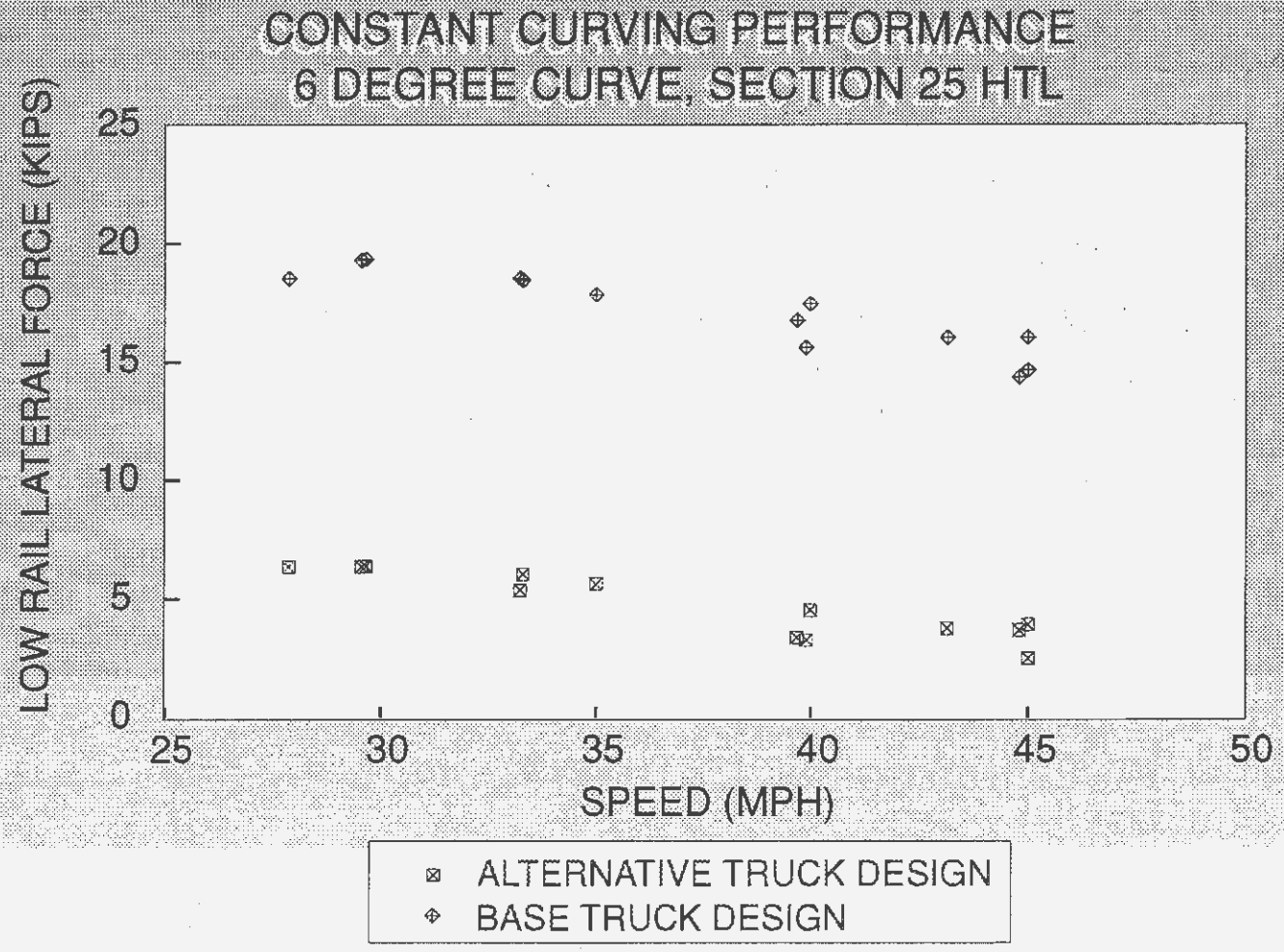
TEN PROPOSED TRUCK DESIGNS

- TRUCK A-1 ..... RADIAL STEERING & HYDRAULIC DAMPERS D7 SPRINGS
- TRUCK A-2 ..... SHEAR PLATE & HYDRAULIC DAMPERS D7 SPRINGS
- TRUCK B ..... STABILIZER FRAME, D5 SPRINGS
- TRUCK C ..... STANDARD 3-PIECE W/WIDER WEDGE & HYDRAULIC DAMPERS D5 SPRINGS
- TRUCK D-1 ..... STANDARD 3-PIECE W/ELASTO WEDGES HYDRAULIC DAMPERS AND CCSB D7 SPRINGS
- TRUCK D-2 ..... CROSS-BRACED & HYDRAULIC DAMPERS D7 SPRINGS
- TRUCK D-3 ..... HYDRAULIC YAW DAMPERS D7 SPRINGS
- TRUCK E ..... RADIAL STEERING & ELASTOMER DAMPING D7 SPRINGS
- TRUCK F ..... SHEAR PLATE & HANGER/ROTATION ARRANGEMENT AT SIDE FRAMES D7 SPRINGS
- TRUCK G ..... FORCED STEERING *

* - DROPPED FROM PROGRAM 10/93

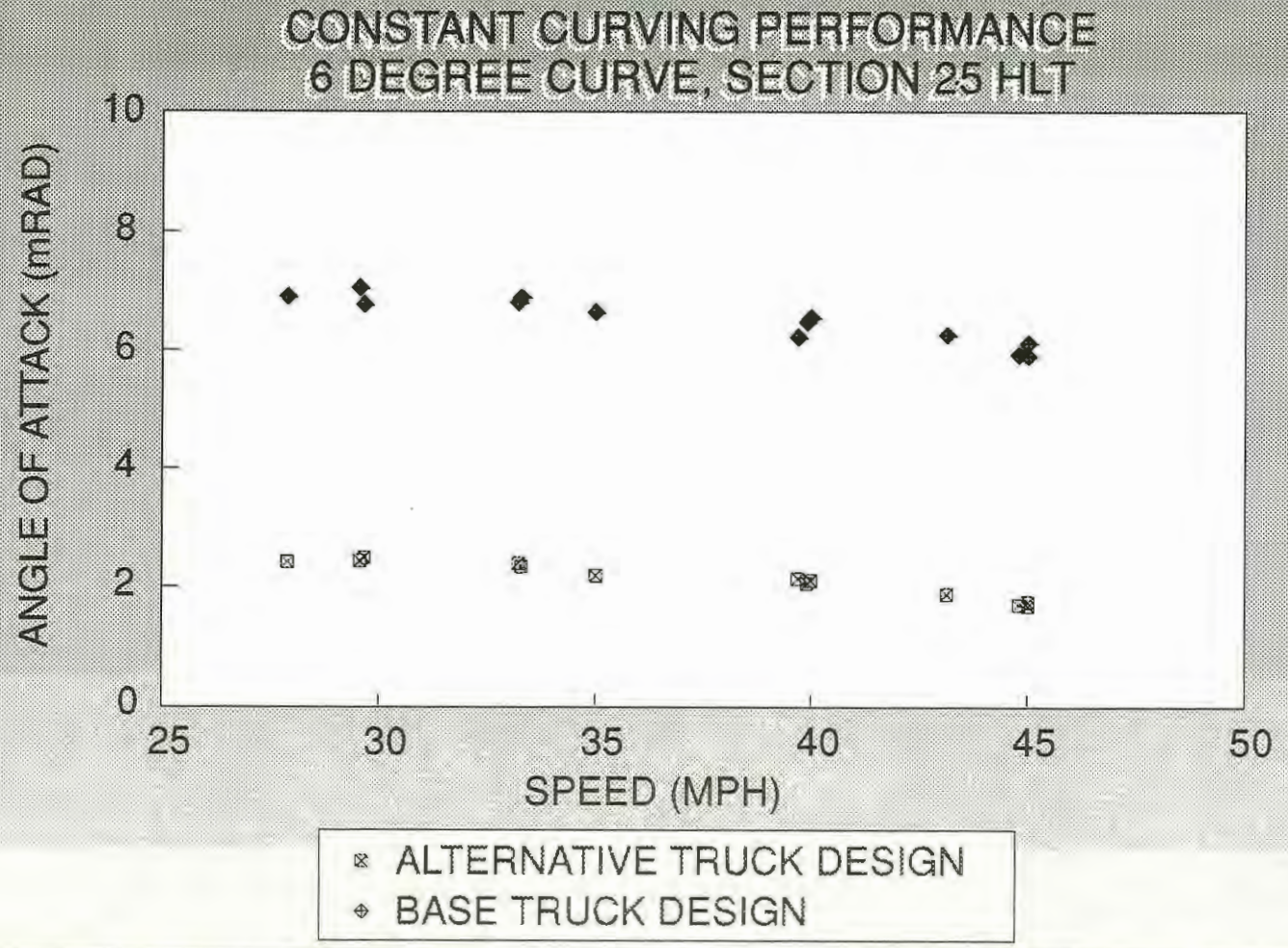


HAL ALTERNATIVE SUSPENSION SYSTEMS



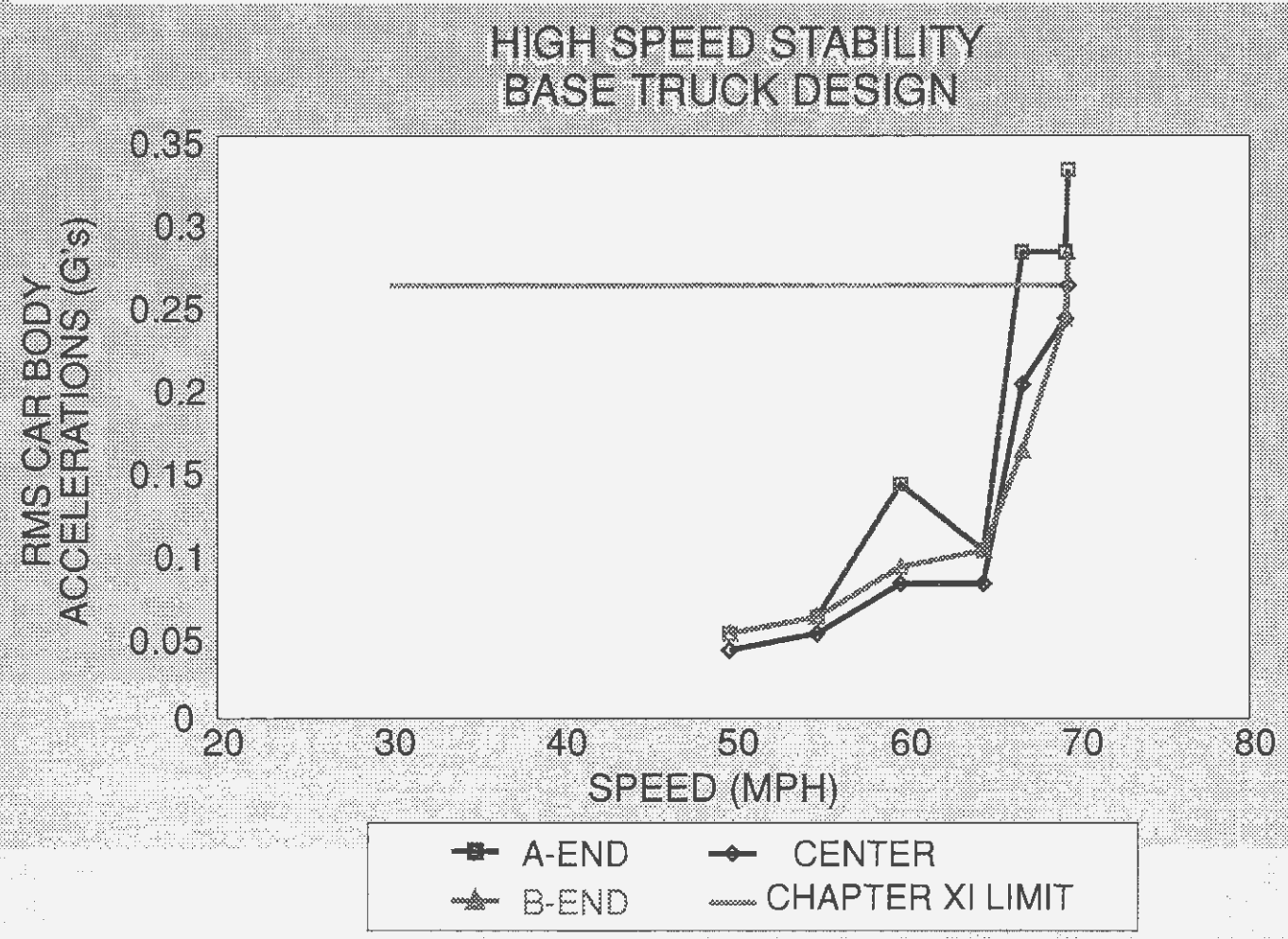
Floromo4.cdr

HAL ALTERNATIVE SUSPENSION SYSTEMS

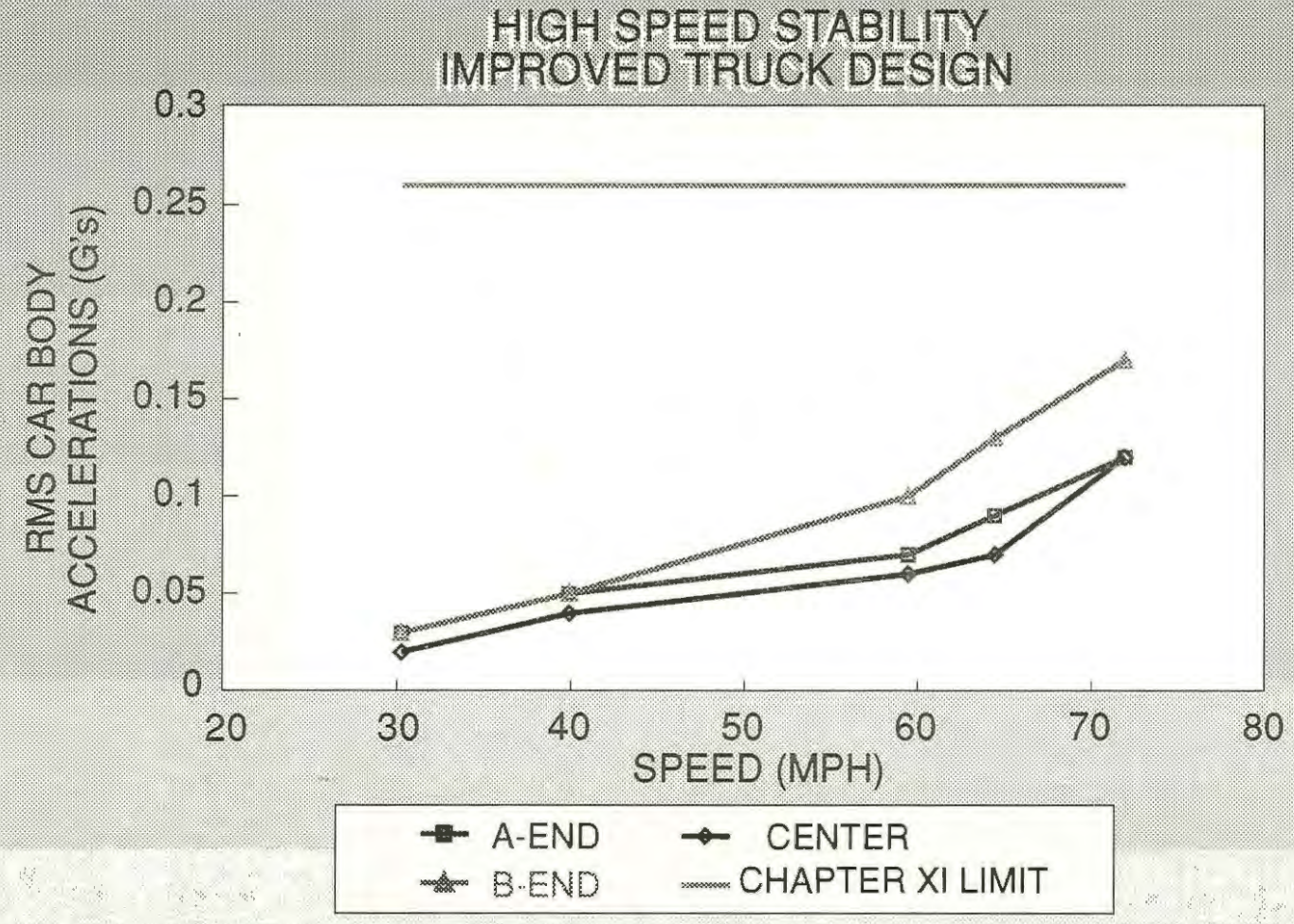


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HAL ALTERNATIVE SUSPENSION SYSTEMS

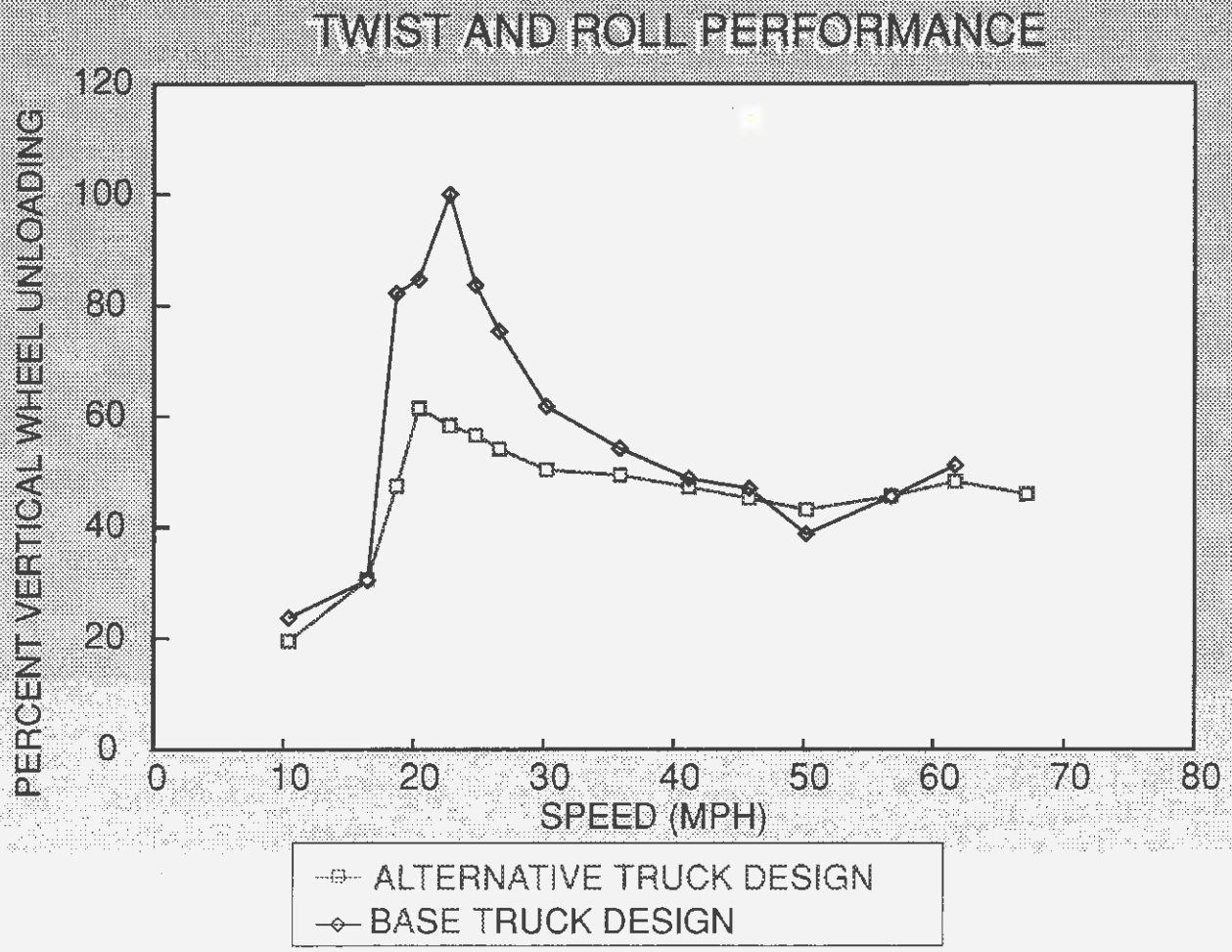


HAL ALTERNATIVE SUSPENSION SYSTEMS



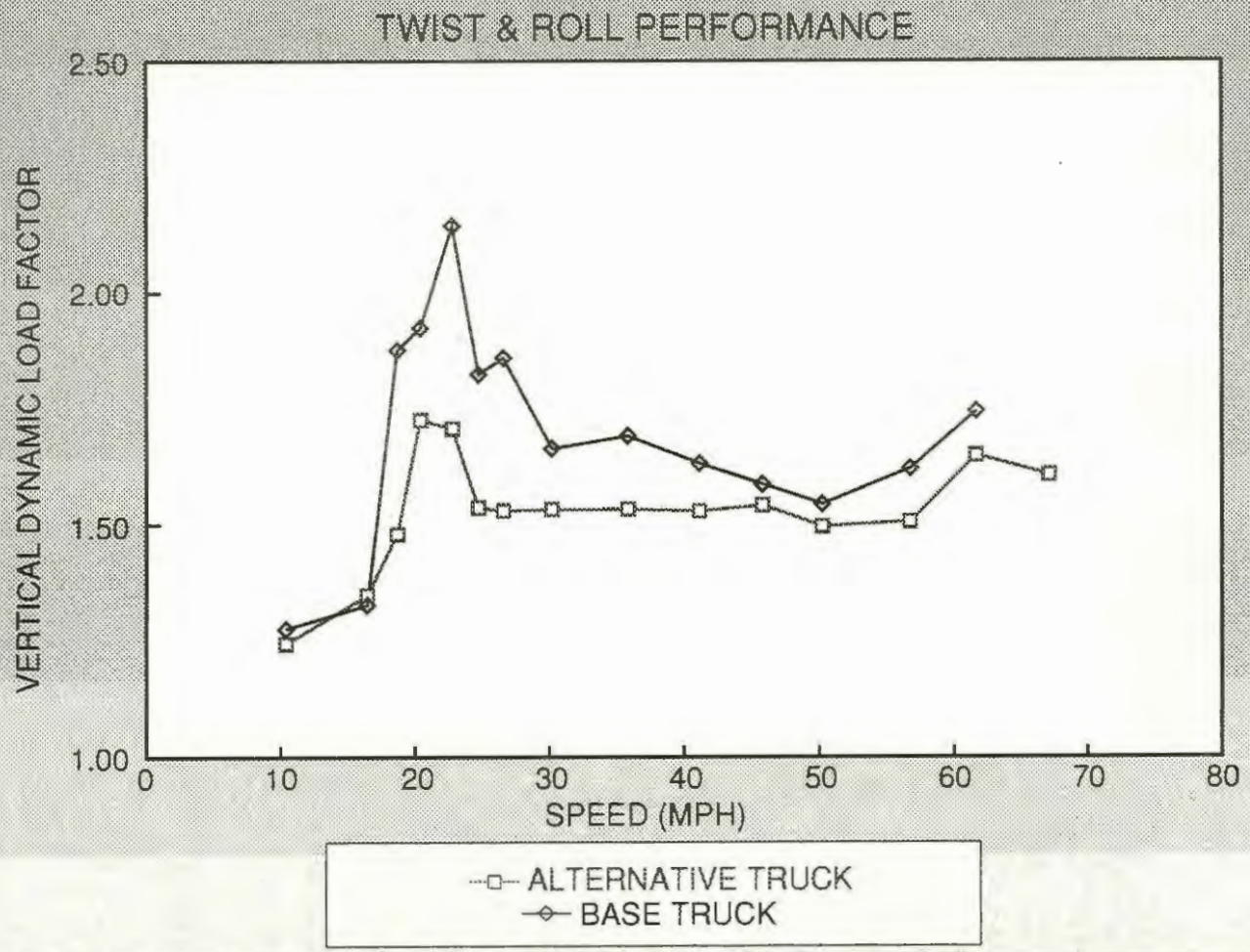
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HAL ALTERNATIVE SUSPENSION SYSTEMS



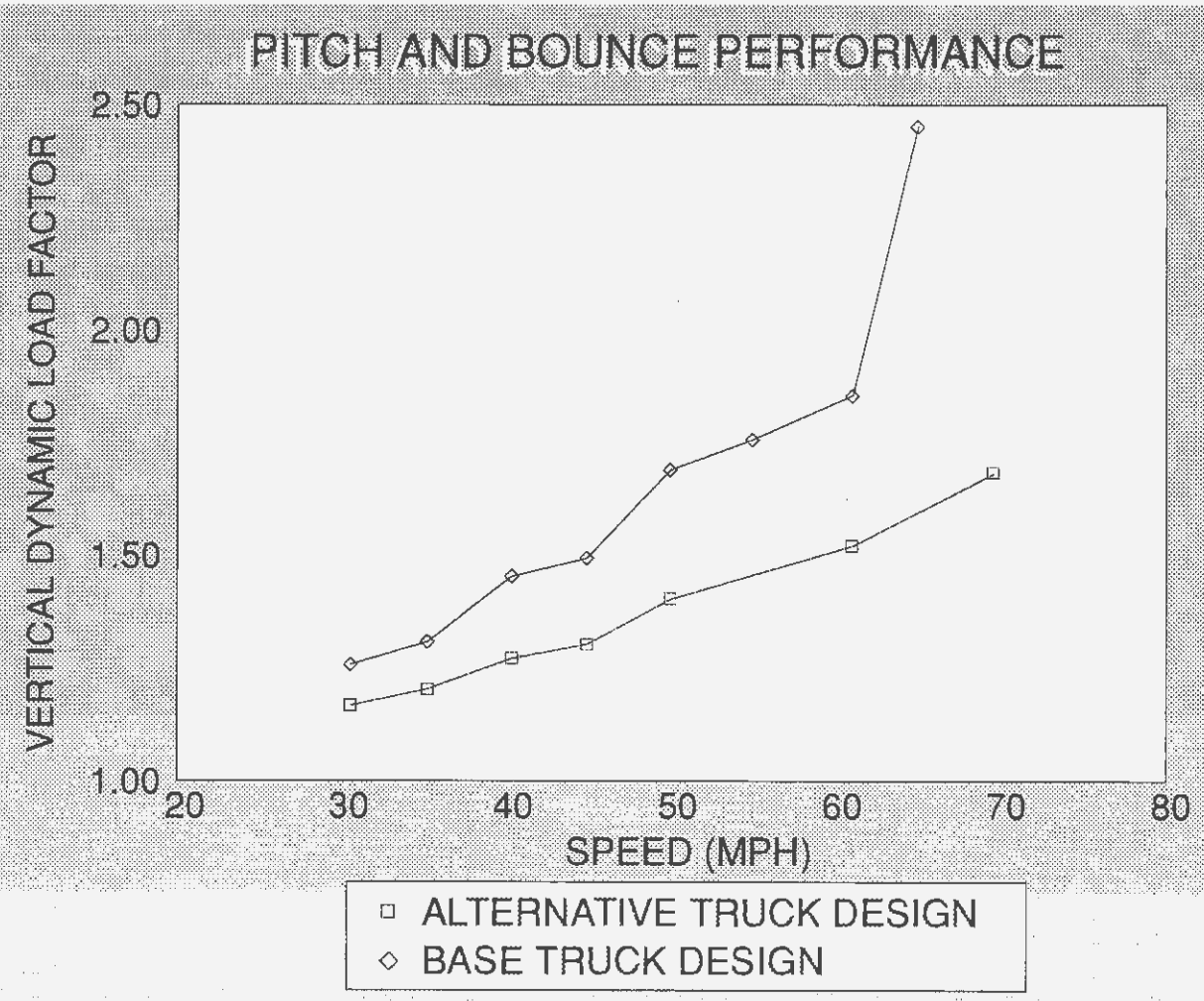
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HAL ALTERNATIVE SUSPENSION SYSTEMS



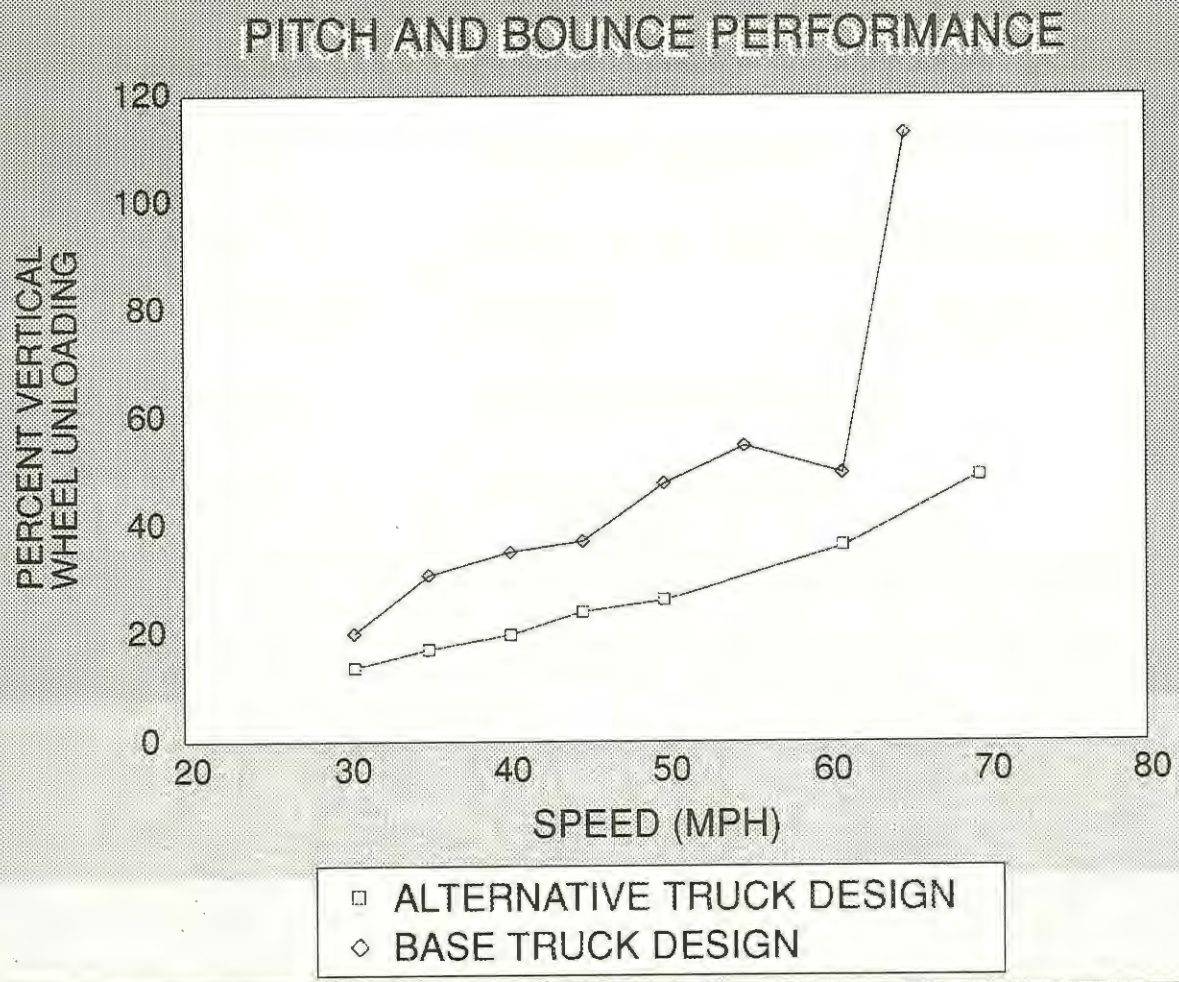
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HAL ALTERNATIVE SUSPENSION SYSTEMS



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HAL ALTERNATIVE SUSPENSION SYSTEMS



FLOROM06.cdr

HAL ALTERNATIVE SUSPENSION SYSTEMS

FUTURE SCHEDULE

- DEC 6, 1993 . . . . . LETTER OF INTENT TO CONTINUE IN PROGRAM
- JAN/FEB '94 . . . . . MANUFACTURERS COMPLETE NUCARS MODELING
- FEB/APR '94 . . . . . NEW HARDWARE
- MAR/AUG '94 . . . . . 10,000 MILE WEAR-IN
- JUL '94/JAN '95. . . . . CHAPTER XI TEST ON "BEST PERFORMERS"
- EARLY '95. . . . . REPORT RESULTS & SELECT TRUCKS
- SPRING '95 . . . . . INSTALL TRUCKS
- SUMMER '95 . . . . . INITIAL MEASUREMENTS
- FALL '95 . . . . . START NEW TRAIN



COMMODITIES BASED PERFORMANCE SPECIFICATION

F. D. IRANI & S. P. SINGH

ASSOCIATION OF AMERICAN RAILROADS

COMMODITIES BASED PERFORMANCE SPECIFICATION

Association of American Railroads



F. D. Irani

Sóm P. Singh

AN ASSESSMENT OF NEEDS AND OPPORTUNITIES-FREIGHT CAR TRUCKS FOR THE 21st CENTURY

A TECHNOLOGY TRANSFER FORUM

September 7-9,1994

Levis Faculty Center

University of Illinois at Urbana-Champaign

B2280. 67

- If you need to haul chickens,
  don't design a Volkswagon Beetle
- If you need to fly over the Rockies,
  don't design a single prop engine plane
- If you need to haul automobiles,
  don't design a coal hopper



GENERAL REQUIREMENTS

- Performance
  - Ride quality
  - Rolling resistance
  - Load spectrum
  - W/R wear
  - Infrastructure
- Performance and wear
  - Limits of wear
  - Limits of performance under stages of wear
- Performance and maintenance
  - Maintenance and wear
  - Maintenance cycle and cost



PAST AND CURRENT DESIGNS

Draft Systems

Buffer Coil Springs → Friction Draft Gears → Cushioned Underframes with Sliding Sills → End of Car Cushion Devices

Suspensions

No Springs Truck (Bogie) → Leaf Spring Trucks (Bogie) → Coil Springs, Spring Plank Trucks → Three Piece Trucks 2½" Travel Coil Springs Friction Damping → Three Piece Trucks 4¼" Travel Coil Springs More Damping → Premium Trucks Radial, Cross Braced

B2280. 70

PROPOSED TRUCK DESIGN PRACTICE(S)

- Vehicle/track interaction consideration
- Vehicle vertical response criterion
- Vehicle lateral response criterion
- Commodity/service considerations



HEAVY AXLE LOAD (125-TON) PROGRAM

Quantify the degree of acceleration in track degradation in major track components under 39-ton axle load vs. 33-ton axle load service

Rail wear	Ballast degradation
Rail fatigue	Subgrade settlement
Rail fastener performance	Turnout performance
Tie degradation	Routine maintenance

HAL TEST OBSERVATIONS (160 MGT)

Rail:

- Increased head-height loss in curves

Ties:

- Only minor increase in damage

Ballast & subgrade:

- More frequent surfacing for ballast
- Slight increase in subgrade maintenance



HAL TEST OBSERVATIONS (160 MGT)

Turnouts:

- Pronounced deterioration

Welds:

- Almost 100% increase in thermite weld failure
- 350% increase in electric flash butt welds on low rail on the 6° curve

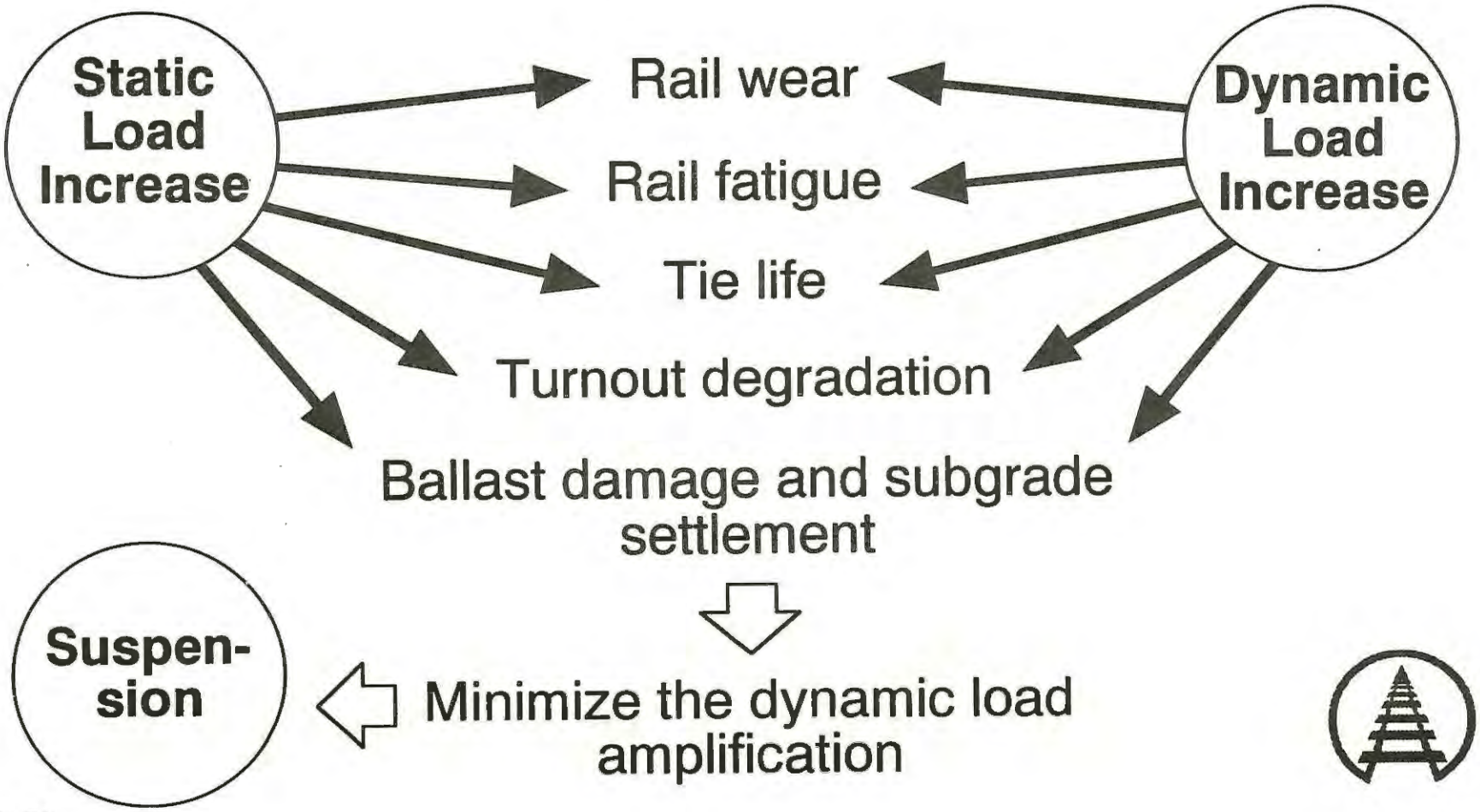
Routine Maintenance:

- A significant increase



HAL ALTERNATE SUSPENSION

Reduce the Increase in Track Maintenance Cost



B2280. 76

PERFORMANCE CRITERIA FOR 39-TON AXLE TRUCK

- Improved curving performance
- Limit dynamic load increment to within the 33-ton axle load environment



PROPOSED PERFORMANCE REGIMES AND ASSOCIATED CRITERIA FOR A 39-TON AXLE LOAD COAL CAR

Regime	*Target*	*Benefits*
Steady-State Curving	Mean $L/V_{WC}<0.5$ Mean $L/V_{AXLE}<0.85$ Curve Res.<0.6	Reduce curving forces Reduce derailment potential Reduce wheel and rail wear
Rock-and-Roll	D.L.F.<1.5	Reduce derailments Reduce rolling stock and track maintenance
Pitch-and-Bounce	D.L.F.<1.5	Reduce derailments Reduce rolling stock and track maintenance

B2280. 78

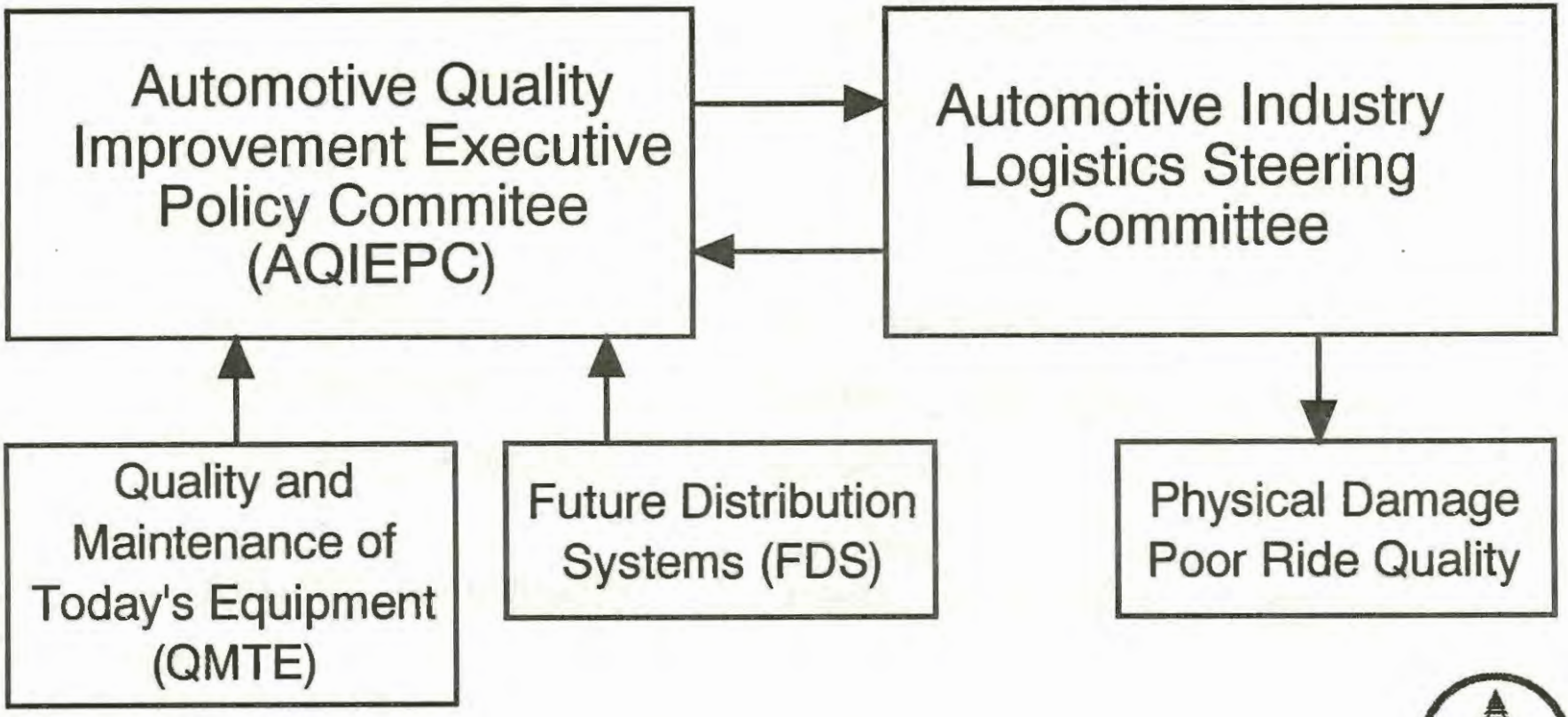
SUMMARY AND CONCLUSIONS

Specific service performance requirements can be used to develop performance specification for new truck designs

The 39-ton axle load truck designs proposed under the HAL program have been evaluated against the proposed performance specification and several of them have substantially met the requirements

Some of the intangible benefits such as improved reliability in performance and reduced derailment risk would add further to the economics of HAL service

NEED FOR A PERFORMANCE BASED DESIGN



B2280. 80

SOURCE OF DAMAGE

- Shock
  - Vertical Transients: Grade Crossings, Switches, Bridges
  - Lateral Transients: As Above
  - Car Defects: Wheel Flats, Solid Springs
  - Operation: Yard Impacts
- Vibration
  - Car Body Rock & Roll: (0.5 to 1.0 Hz; 10 to 25 MPH)
  - Car Body Bounce: (1 to 3 Hz; 40 to 70 MPH)
  - Car Body Modes: - Vertical Bending 10 to 30 Hz
    - Twist 10 to 30 Hz
  - Car Body Hunting: 2.5 to 3.5 Hz



RIDE QUALITY GUIDELINES FOR TRANSPORT OF AUTOMOBILES

- Instrumentation
- Harmonic response performance
- Longitudinal response performance
- Over the road performance
- Allowable limits for each performance regime



B2280. 82

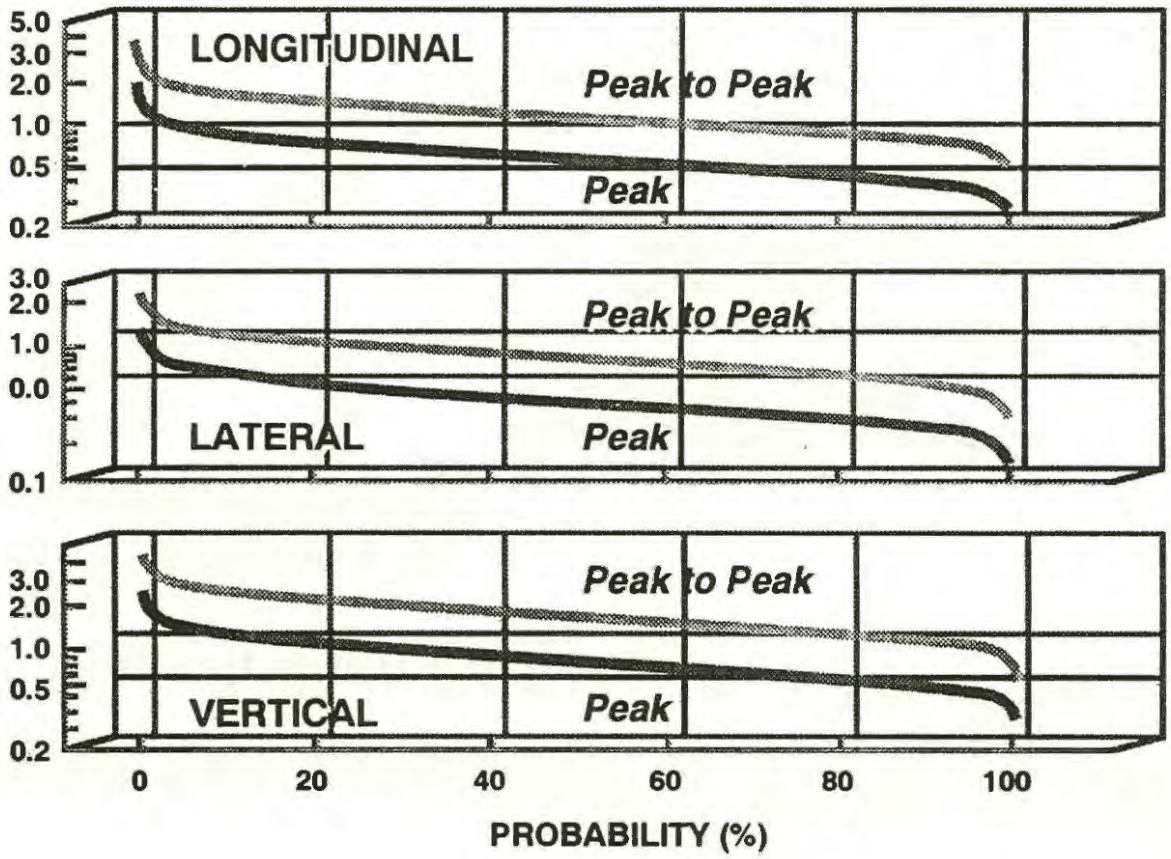
AUTORACK CAR RIDE QUALITY

Objectives

- Quantify vibration and shock environment for today's tri-level
- Study the nature of specific vibration and shock events
- Long term: Recommend a standard methodology for measuring and analyzing ride quality of autoracks



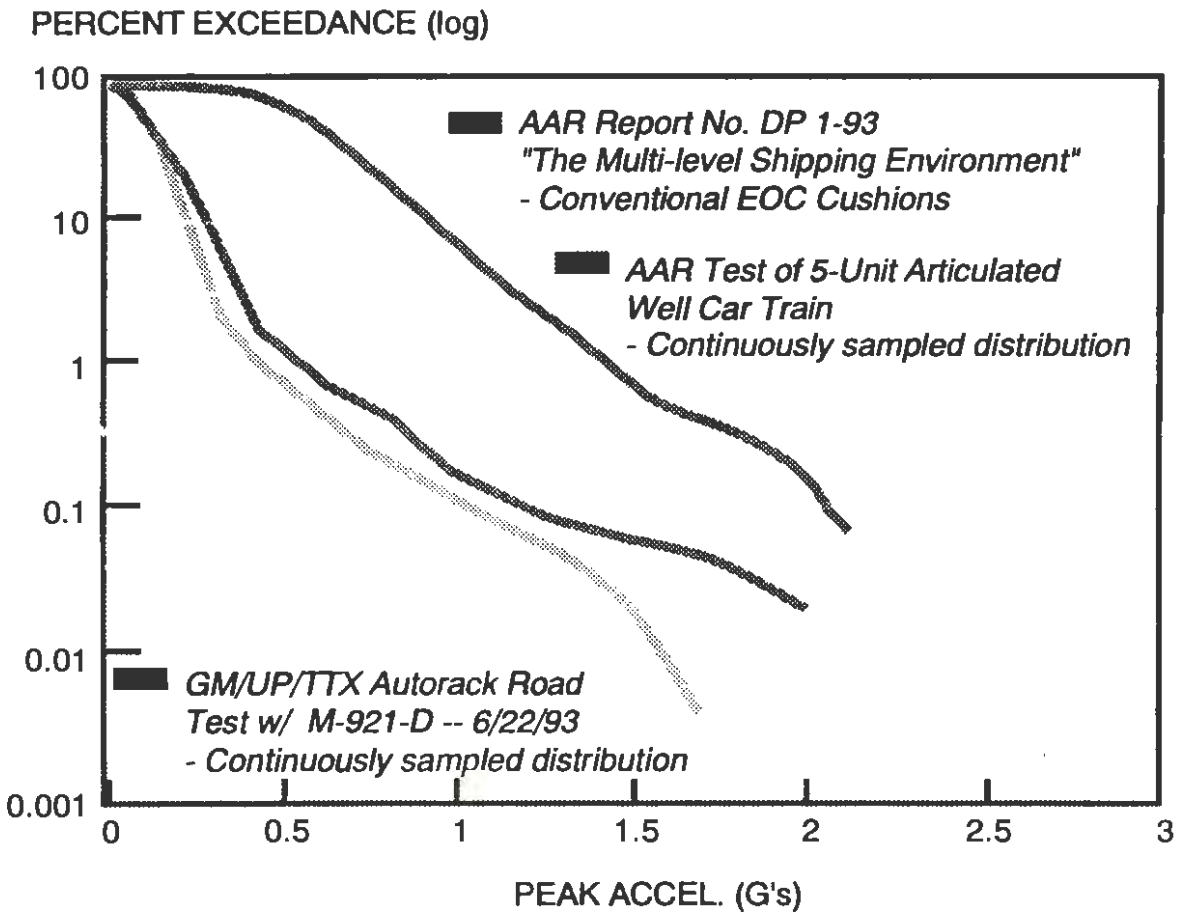
RIDE QUALITY OF EXISTING AUTORACK CARS



B2280. 84

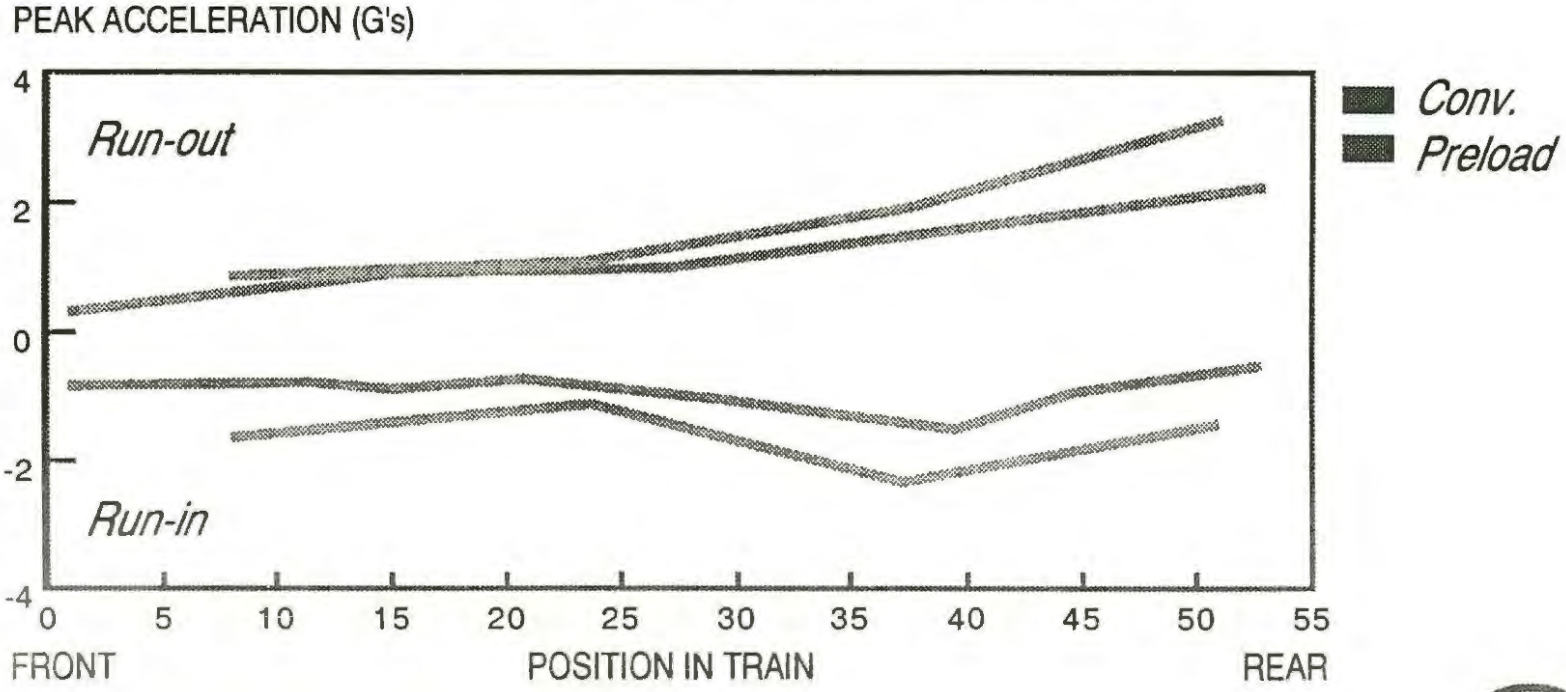
PERCENT EXCEEDANCES vs. PEAK ACCELERATION

Longitudinal Carbody Deck Acceleration



B2280. 85

PEAK LONGITUDINAL ACCELERATION vs. POSITION-IN-TRAIN



Courtesy GM/UP/TTX/Keystone

B2280. 86

ALLOWABLE LIMITS FOR AUTOMOBILE TRANSPORT

- Shock data

- Vertical	Max. ±1.0G Not more than 100 exceedances above ±0.75G for 1,000 miles
- Lateral	Max. ±1.0G Not more than 100 exceedances above ±0.5G for 1,000 miles
- Longitudinal	Max. ±1.0G Not more than 100 exceedances above ±0.8G for 1,000 miles
- Vibration data	Power spectral density Vertical, Lateral RMS Vertical, Lateral (Standard method to be defined)



TRUCK SUSPENSION SPECIFICATION

Summary

- Performance based design necessary to meet the customers expectations
- Performance based design is possible
- Several proposed concepts conform to the FDS specification
- Operational economics will be the ruling factor



CONCLUSION AND FUTURE ACTIVITY

- Today's market environment demands commodity based performance
- FDS ride quality and HAL performance specifications be proposed as car standards
- AAR to start related project: customer-focussed train systems
- If you want to haul nuclear waste design a suspension very, very, very carefully



RECOMMENDED TRUCK MAINTENANCE GUIDELINES

S. K. PUNWANI

ASSOCIATION OF AMERICAN RAILROADS

RECOMMENDED TRUCK MAINTENANCE GUIDELINES

An Assessment Of Needs And Opportunities Freight Car Trucks For The 21st Century

A Technology Transfer Forum

S.K. Punwani
Association of American Railroads

September 7-9, 1994
Levis Faculty Center
University of Illinois at Urbana-Champaign

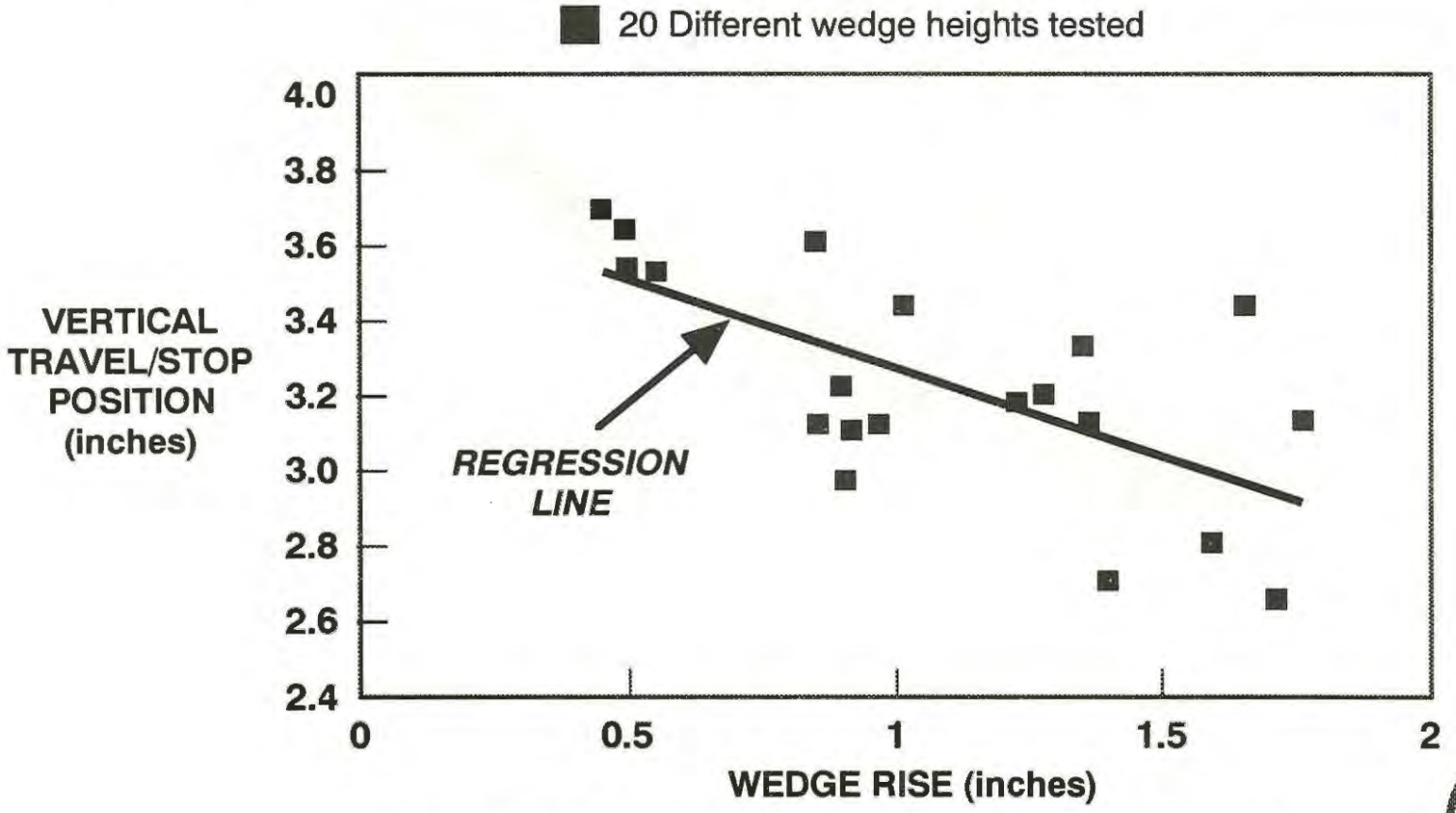
B2219.30

WE HAVE LEARNED THAT --

- Variability in shoe height due to assembly tolerances
- General trend is for shoe rise with service mileage
- Wear patterns cause uneven shoe rise or cause the wedge to lock up showing acceptable shoe rise

LOSS OF TRAVEL DUE TO WEDGE BINDING

Vertical Travel/Stop Position vs. Heights Tested



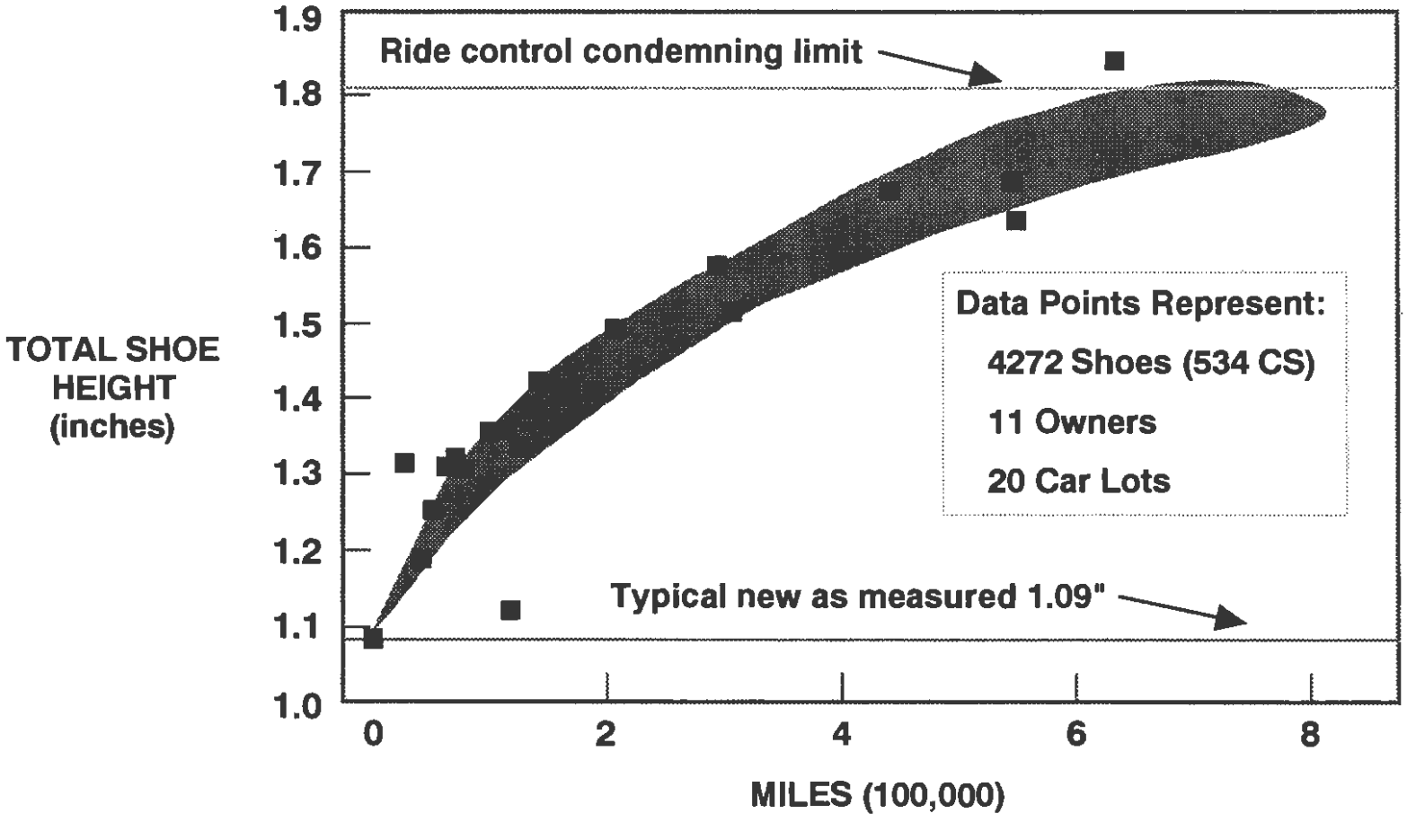
Trucks tested with new coils in all positions

Stop position = -0.5 wedge rise + 3.6875 inches

B2219.31

SHOE RISE VS. MILEAGE

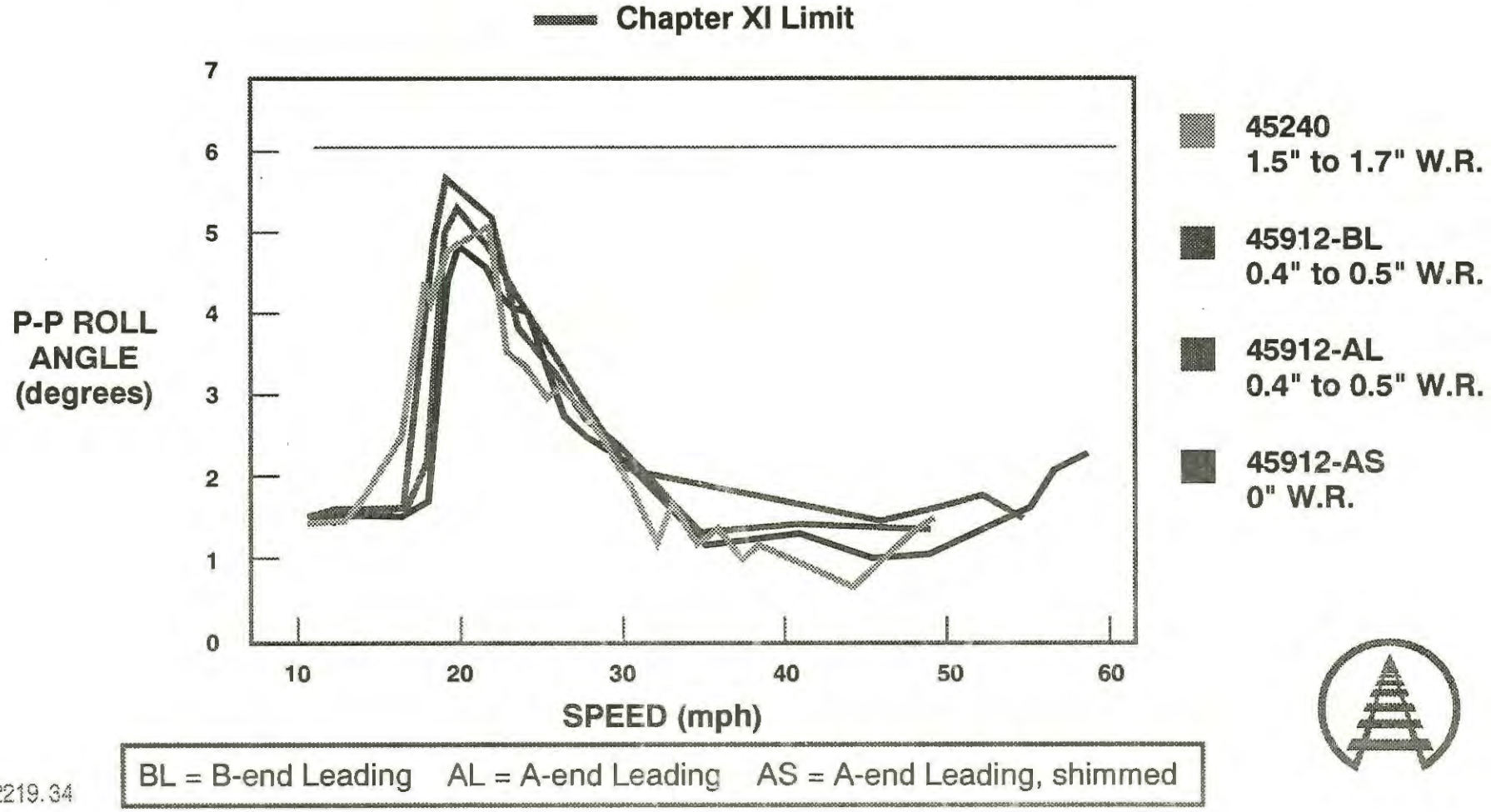
For Coal Unit Trains



B2219.32

TWIST AND ROLL TEST - WORN TRUCK

Comparison Peak-To-Peak Roll Angle, A-End

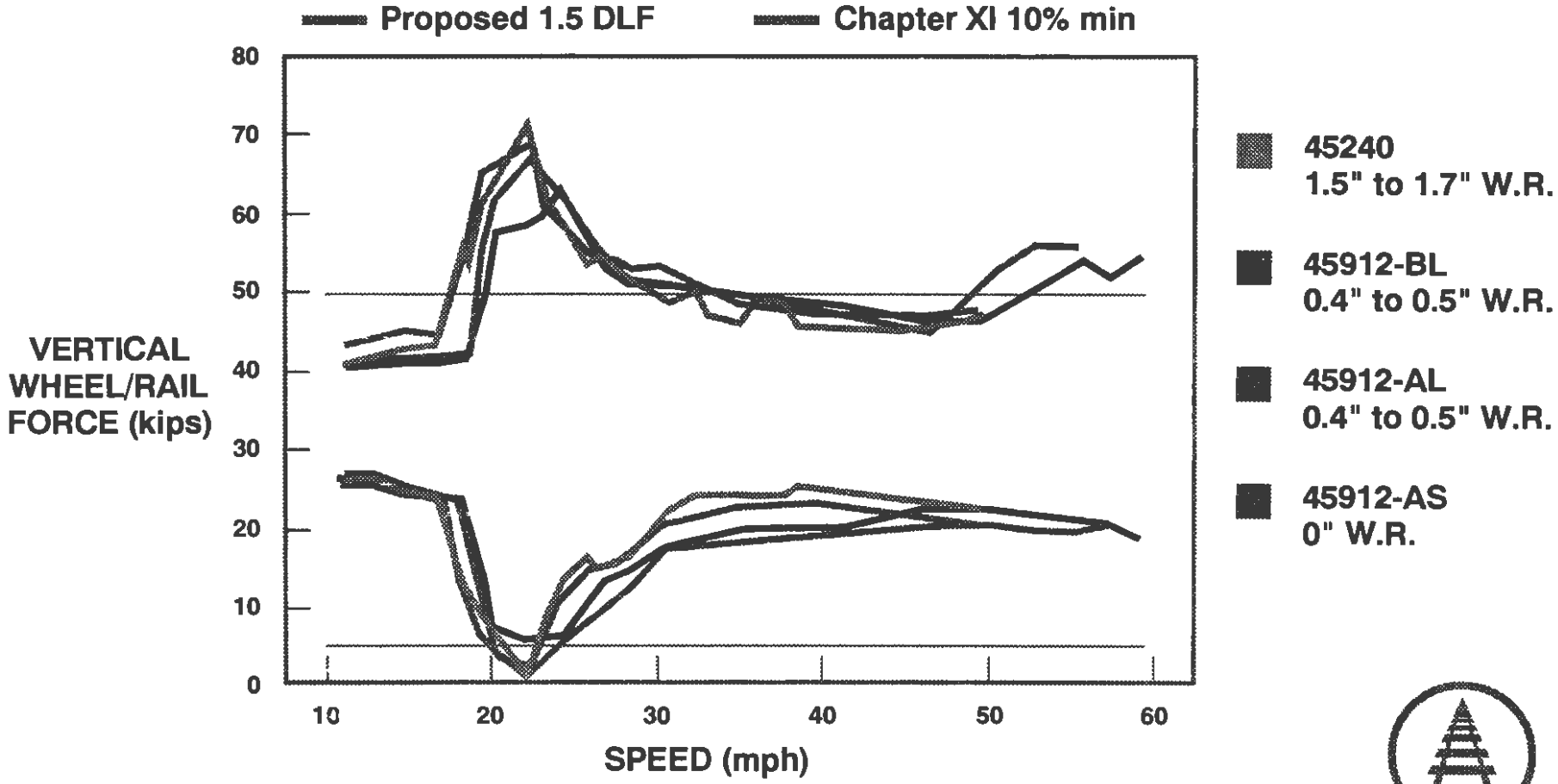


BL = B-end Leading AL = A-end Leading AS = A-end Leading, shimmed

B2219.34

TWIST AND ROLL TEST - WORN TRUCK

Comparison of Vertical Forces, Axle 1 Left

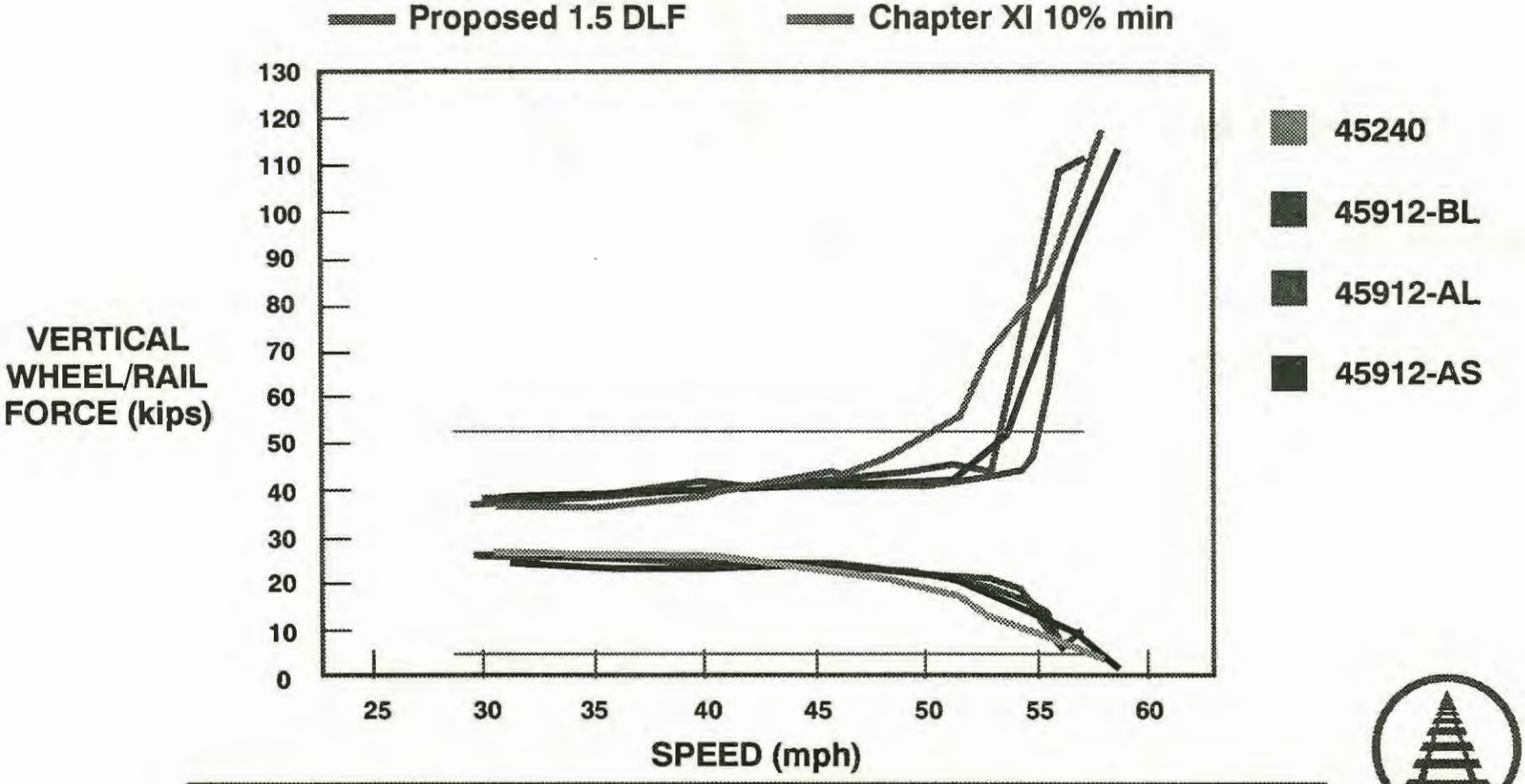


BL = B-end Leading AL = A-end Leading AS = A-end Leading, shimmed

B2210.35

PITCH AND BOUNCE TEST - WORN TRUCK

Comparison of Vertical Rail Force, Axle 1, R

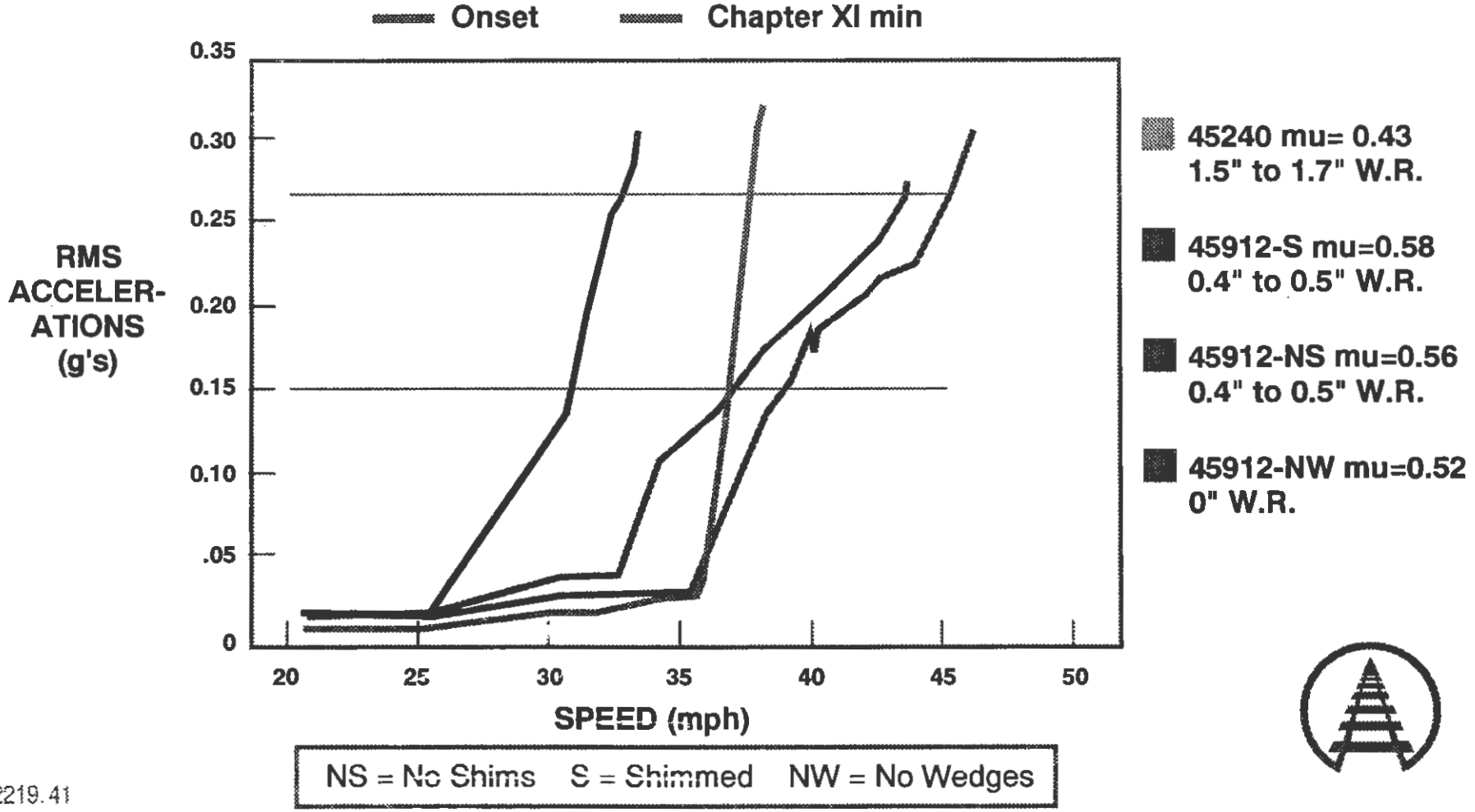


BL = B-end Leading AL = A-end Leading AS = A-end Leading, shimmed

B2219.33

HUNTING TEST - WORN TRUCKS

Comparison Of Carbody Lateral Accelerometer, A-End



B2219.41

FACTORS INFLUENCING TRUCK MAINTENANCE

- Truck wear rates and characteristics
- Loss of dynamic performance with wear
- Car type and utilization
- Related maintenance
- Customer expectations
- Rail network/operating speed implications



ESTIMATED ANNUAL MILEAGE

For Various Cars

CAR TYPE	*ANNUAL MILEAGE*
Coal cars - UT	50,000 - 150,000
Box-type cars	10,000 - 15,000
Intermodal flat	50,000 - 75,000
Autorack	25,000 - 35,000
Double stack	200,000+
Tank cars	10,000



SUGGESTED TRUCK MAINTENANCE CYCLE FOR VARIOUS CAR TYPES

CAR TYPE	SERVICE	MILEAGE
Coal Hopper	Unit Train	500,000
Coal Gon	Unit Train	500,000
Multi-Level	Gen. Serv.	400,000
Box Cars-Equip.	G.S.	175,000#
Covered Hoppers	Grain	300,000
Covered Hoppers	Bulk	300,000
Doublestack	Intermodal	400,000
89-ft. Flat	TOFC/COFC	400,000
Artic. Spin	Intermodal	400,000
Bulkhead Flat	Gen. Serv.	200,000
Tank Cars	Gen. Serv.	175,000#
All Other	Unit Train	400,000
All Other	Gen. Serv.	175,000#

 # Cars not moving this mileage in ten years must have truck inspection and required maintenance/repair done after 15 years.

B2219.38

RECOMMENDED TRUCK MAINTENANCE

Guidelines

- Require truck inspection and repair/maintenance based on mileage.
- Mileage interval should be selected by car type and service. Emphasis on preventative maintenance to avoid out-of-service time and reduce line set out.
- M-214 should be expanded to provide repair practice for each truck design. This includes gages for reuse limits, condemning limits etc. Alternate components should be tested and approved.



POSSIBLE SOLUTIONS TO WEAR PROBLEMS

C. L. URBAN

ASSOCIATION OF AMERICAN RAILROADS

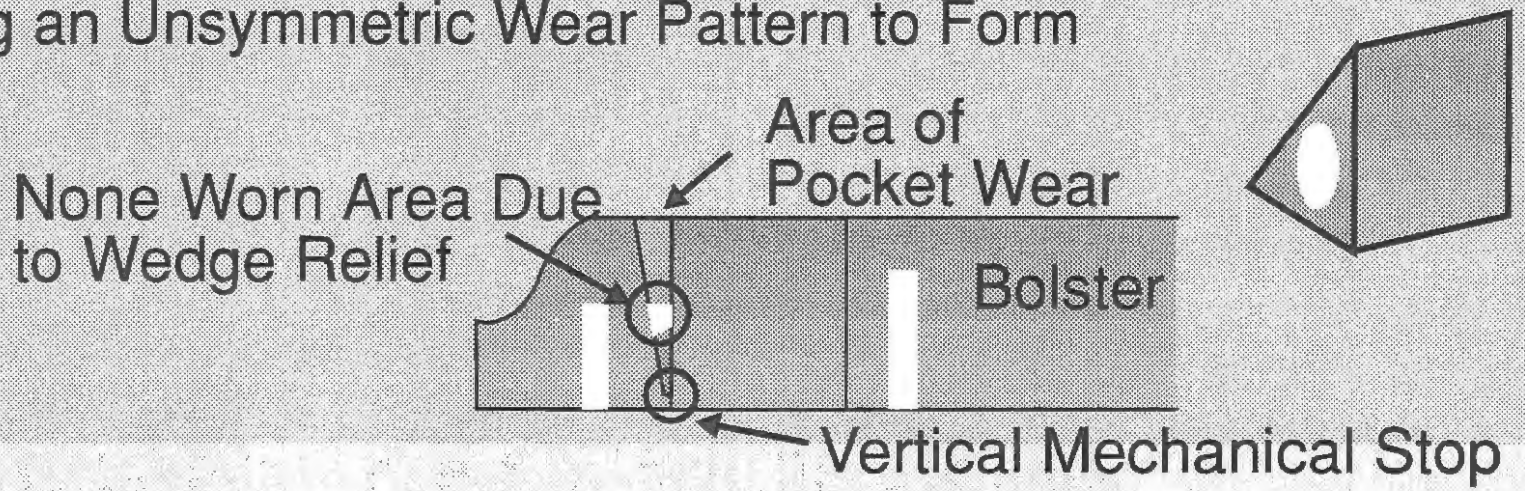


S01.cdr

Solutions

Bolster Pocket Wear Solutions

Outer Bolster Pocket Wall Wear caused by Placement of outer Gib -- Causing an Unsymmetric Wear Pattern to Form

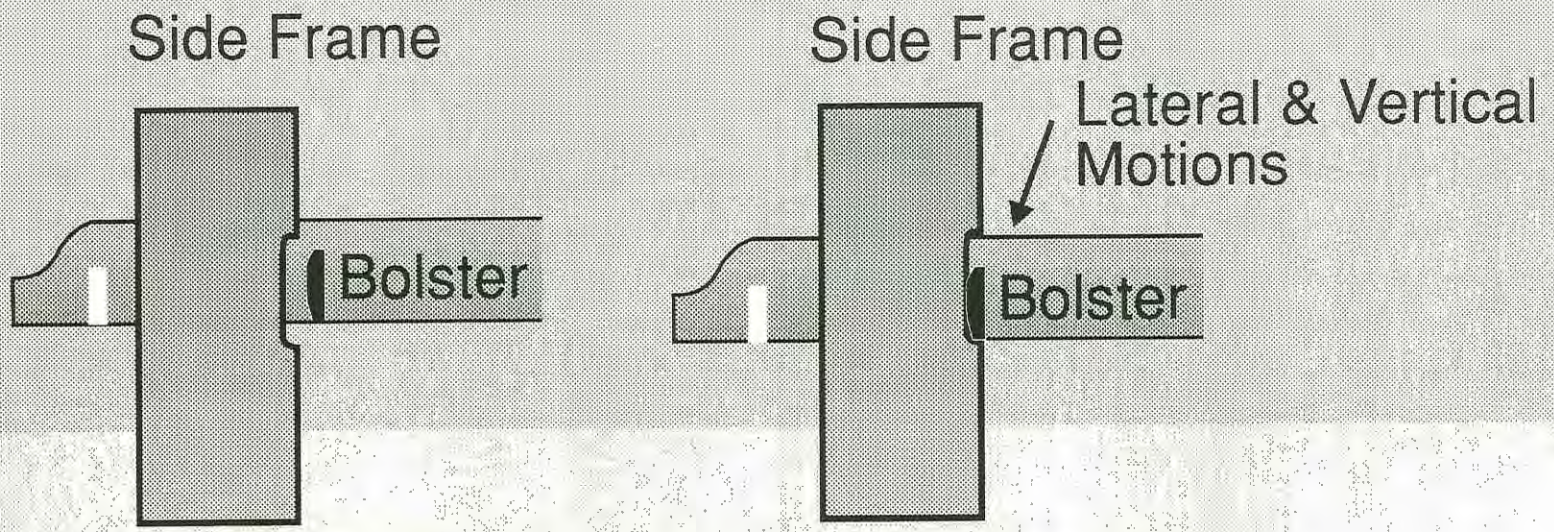


Solutions:
1) Move outer Gib back to Original Position
2) Harden Bolster Pocket and Wear Plates
3) Reduce Hardness of the Wedge

POCKET-S/S02.cdr

Solutions

Inner GIB Wear Creates Vertical Stops--End View

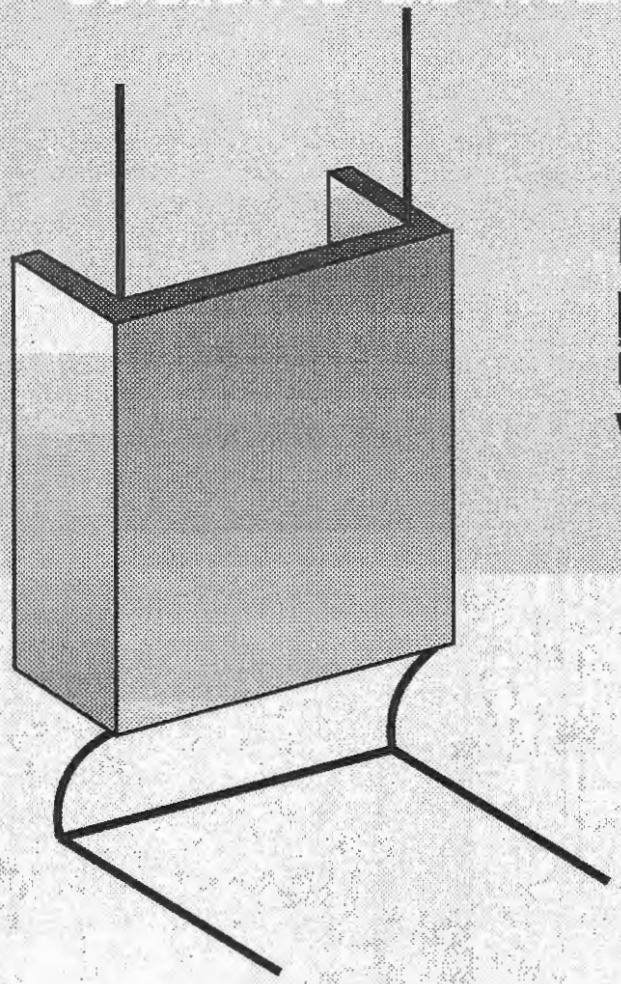


Solution: Harden both Side Frame & Gib
Surfaces to Reduce/Eliminate Wear

INNERGIB/S03.cdr

Solutions

Solution to Gib Wear on Side Frame



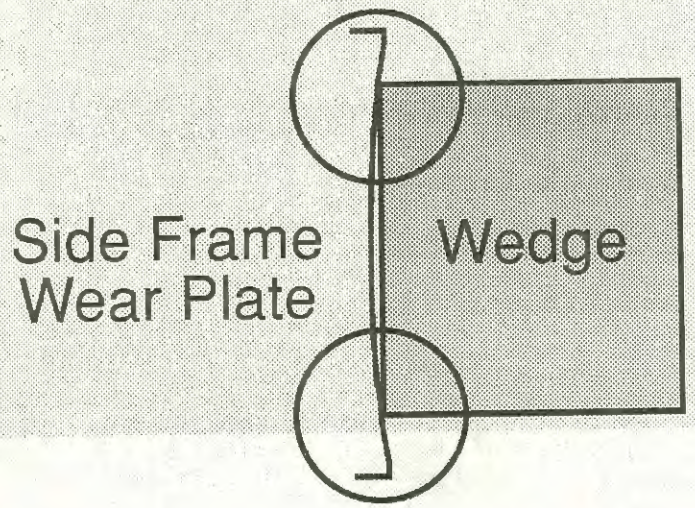
Hardened side frame wear plate reduces gib wear inside and out, increases wear plate strength

GIBPLATE/S05.cdr

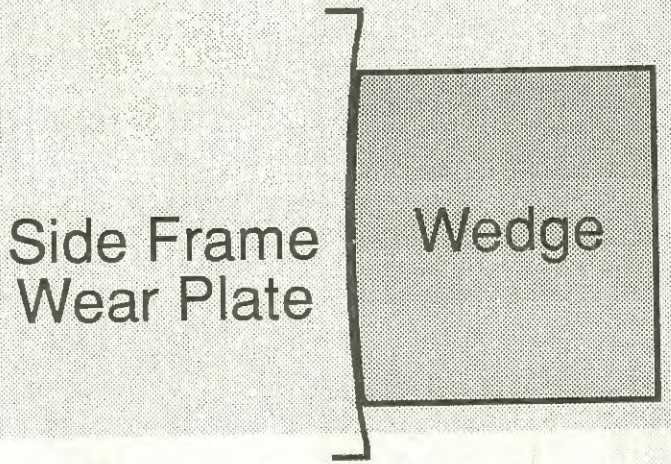
Solutions

Wedge/Wear Plate Solutions--Top View

New Truck--Edge Contact



Worn Truck--Conformal Contact

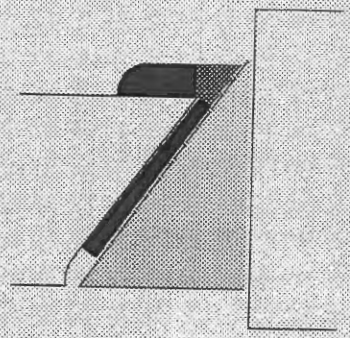


Solution: 1) Follow M-214 on Side Frame Wear Plate Flatness
2) Replace Side Frame Wear Plate and Wedge at Same Time
3) Reduce Wedge Hardness

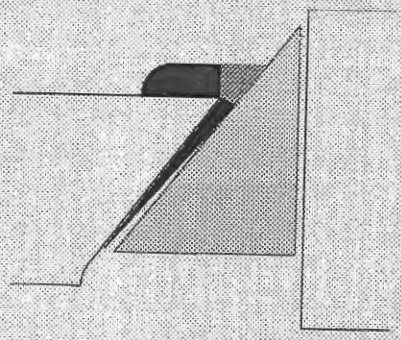
HNEWWORN/S04

Solutions

Bolster Wear Plate



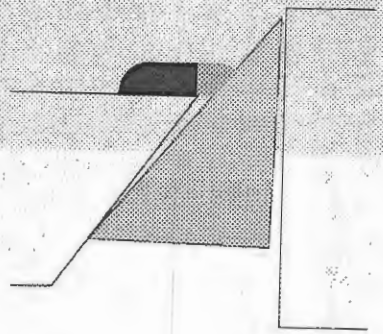
New Condition

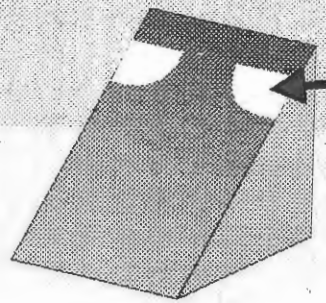


Worn Condition

Wear plate, wears to a wedge shape

Wear plate has fallen or slid out



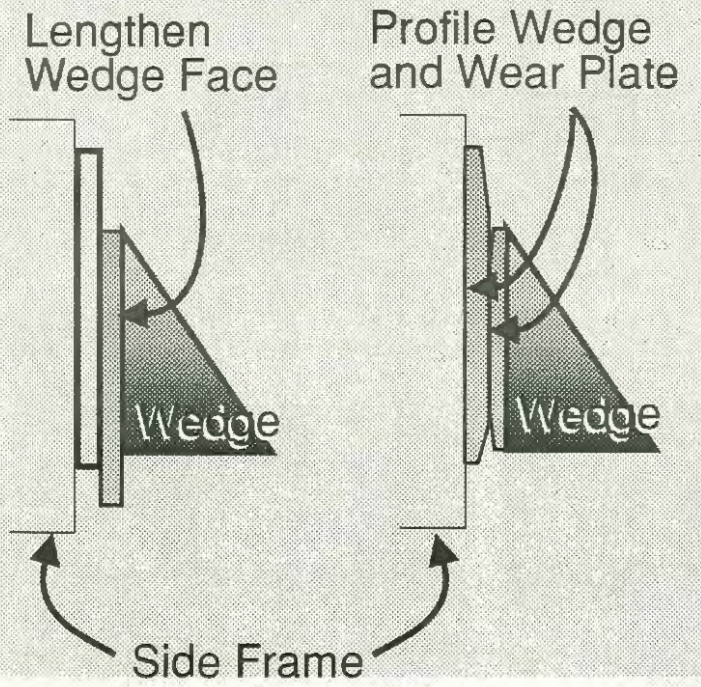


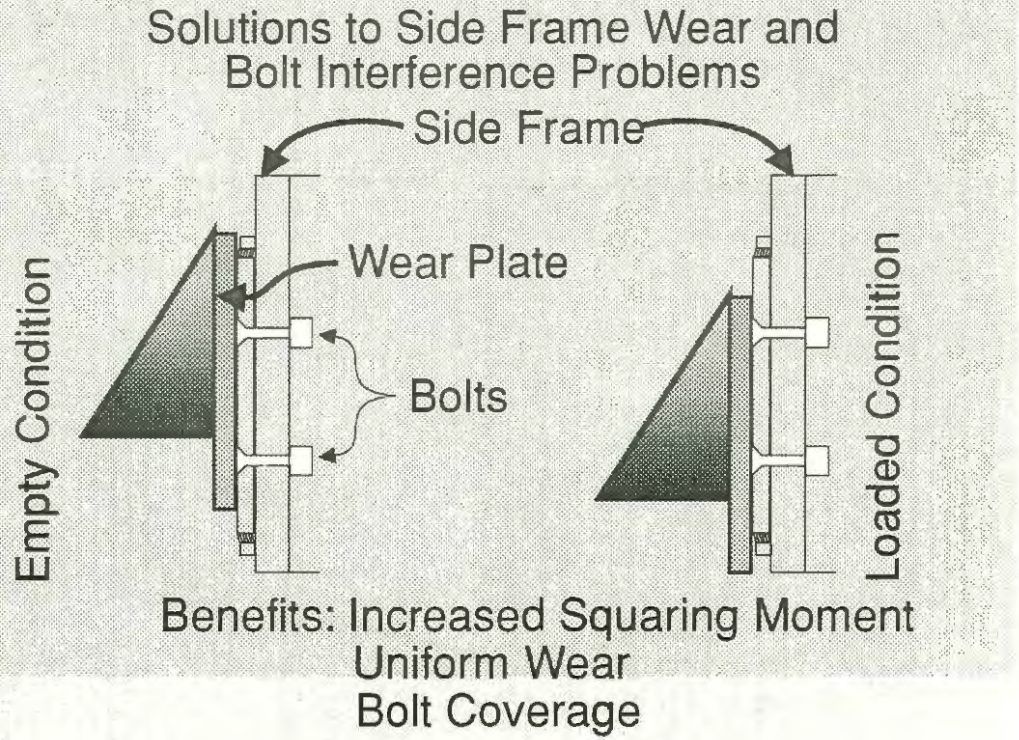
Loss of wear plate increases loads on weld/wedge interface causing mechanical stops to form

Solutions: 1) Softer Welding Rod
2) Reduce Wedge Hardness

hsf1wear/S08.cdr

Possible Modifications To Existing Wedge Designs



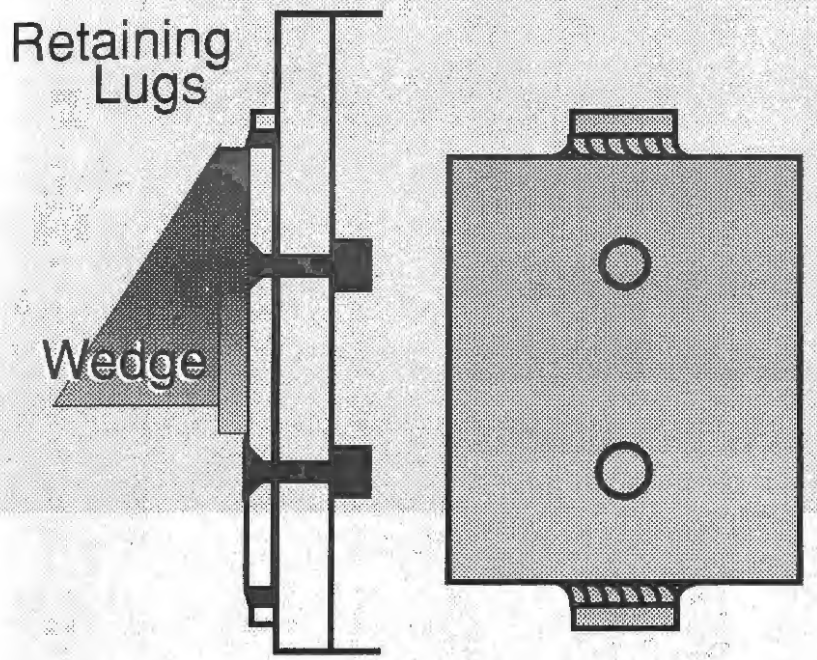


Solutions: 1) Reduce Size of Retaining Lugs
2) Increase/Lengthen Wedge Face Beyond Interference Area
3) Weld Down Sides of Wear Plate
4) Softer Welding Rod
5) Change Profile of Wedge/Wear Plate
6) Reduce Wedge Hardness

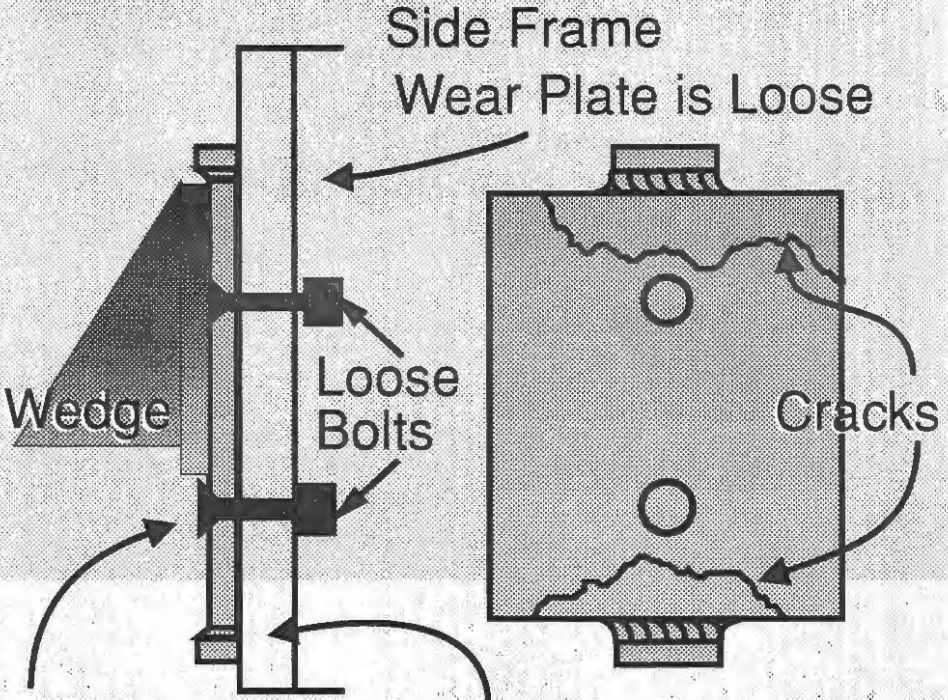
hsf1wear/S07.cdr

Side Frame Wear Plate Solutions

Side Frame Wear Plate-- New Truck



Side Frame Wear Plate-- Worn Truck



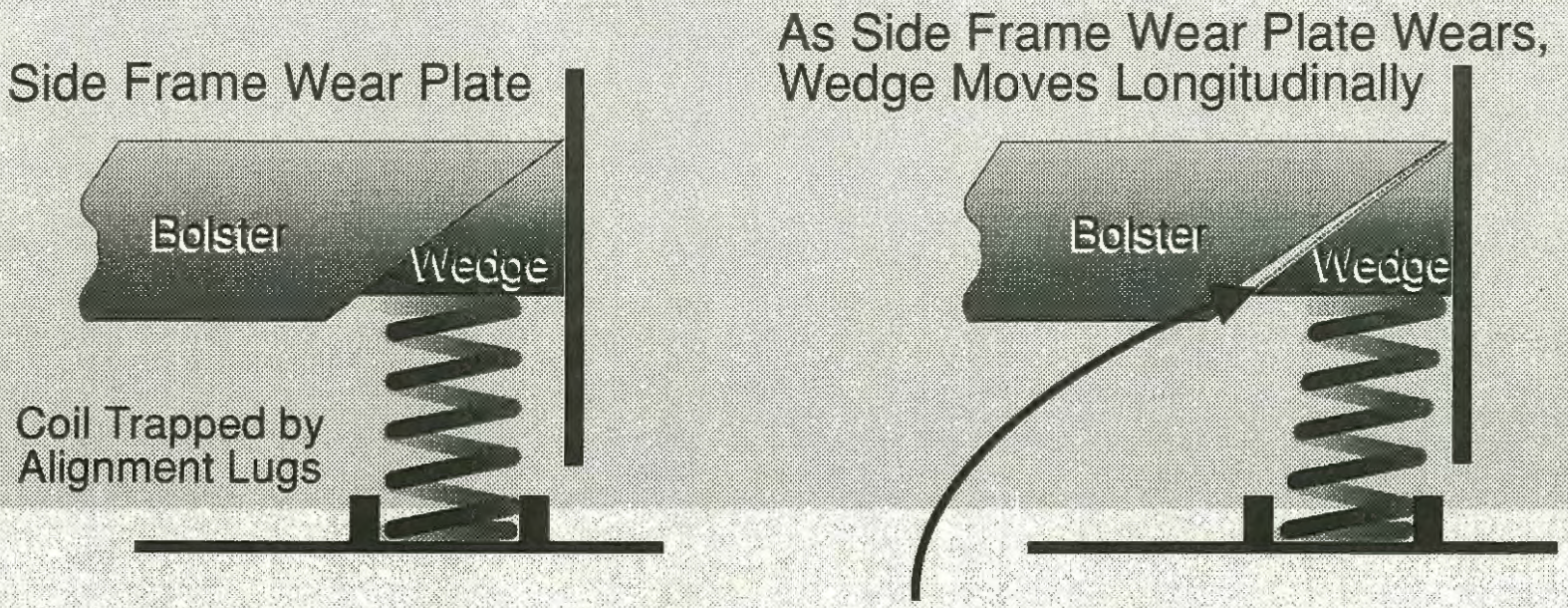
Bolts are Loose Causing a Mechanical Stop

Retaining Lug has not Worn and Welds have Failed

hsf1wear/S06b.cdr

Solutions

Effect of Control Coil Alignment Lugs

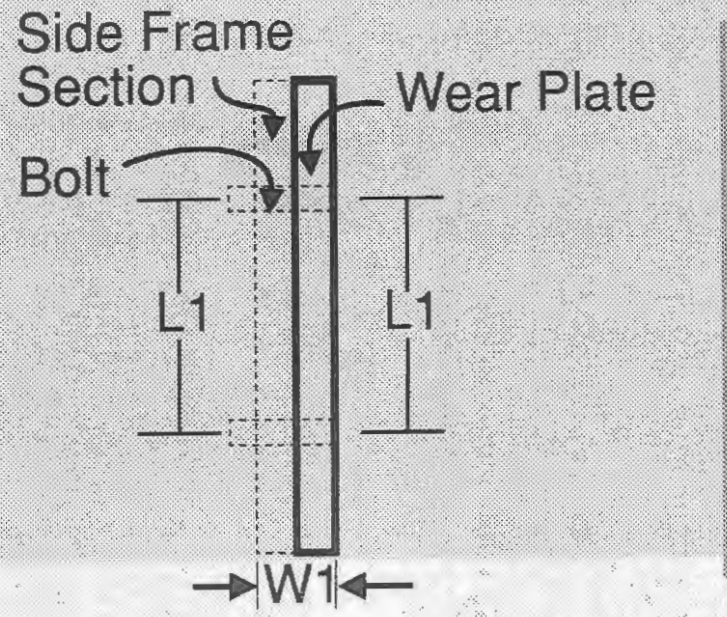


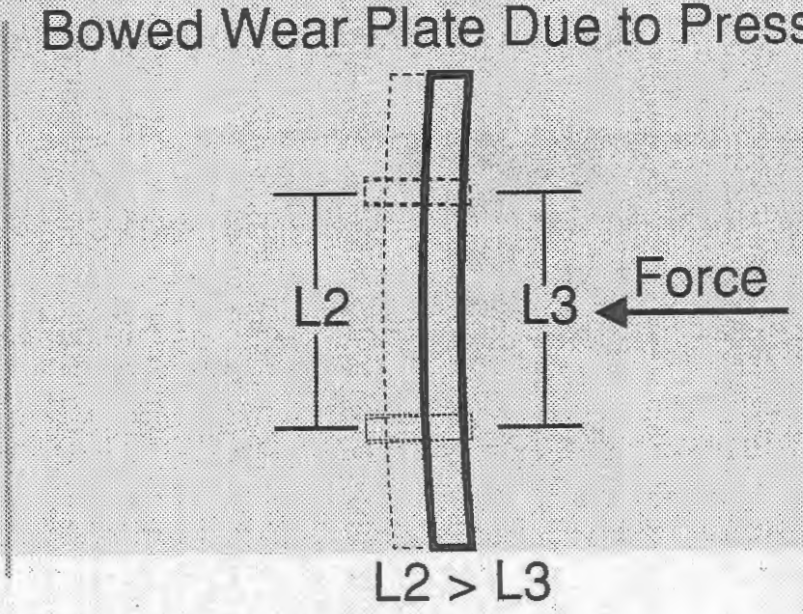
Bolster Slope Wear Due to Wedging Action, Wedge Rocking and Clearance Problems between the Columns

Solutions: Change Alignment Lug Location

Bolt Stretch Due To Cycling Side Frame Forces

Bolts Stretch because of the Cycling of Compressive Forces on Side Frame Wear Plate - Wear Plates get Loose and Damage Occurs





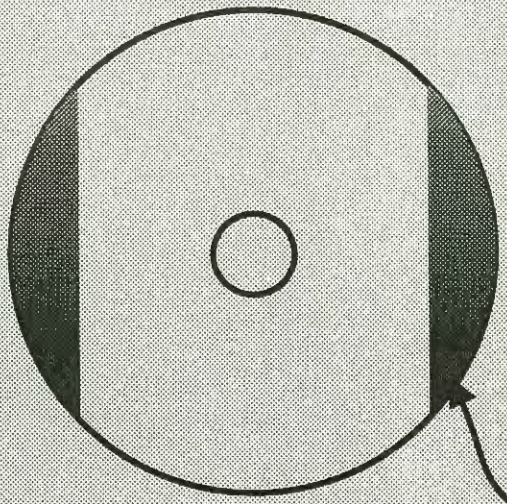
$$W1 - \sqrt{\left(\frac{L2 - L3}{2}\right)^2 + W1^2} = \text{Stretch Of Bolt}$$

Solution: Reduce Bolt Failures by using Thicker Side Frame Section

hsf1wear/S11.cdr

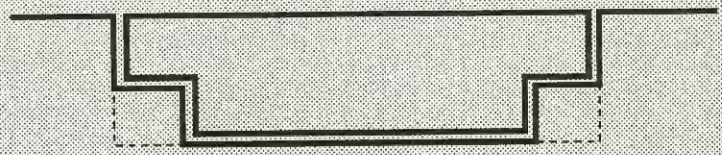
Solutions

Center Plate Locking Problem



Top View

Relief Area



Side View

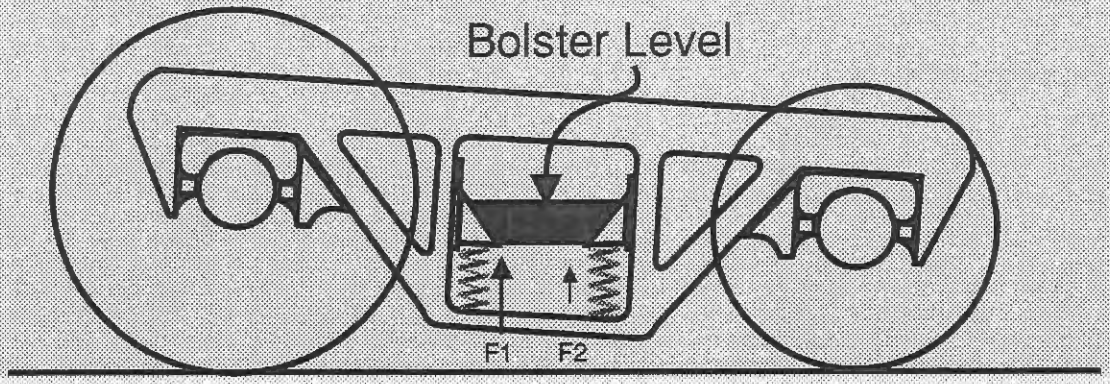
Solution: 1) Use Center Plate Liners
2) Do Not Weld Liners in Place

hsf1wear/S12.cdr

Solutions

Differential Wheel Diameters Create Wear

Differential Wheel Diameter Start Problems with Differential Wedge Heights Causing Abnormal Wear



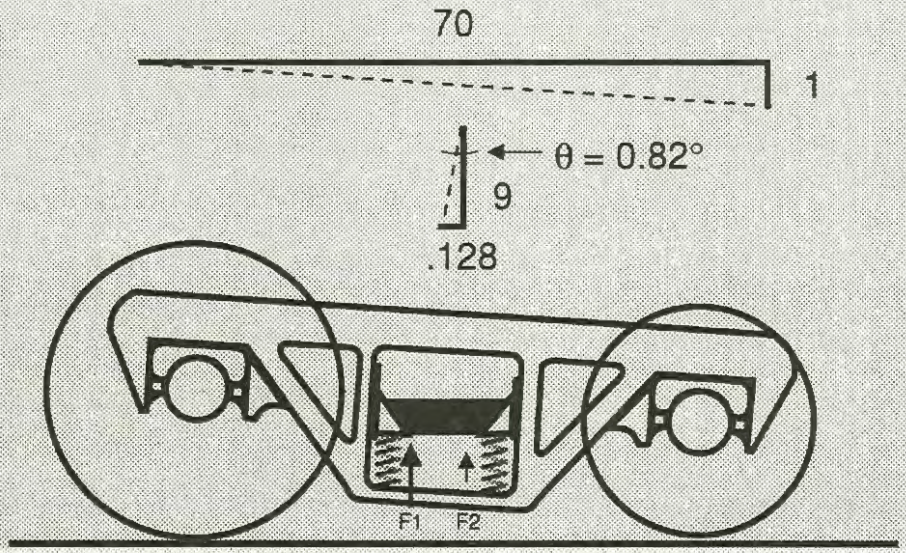
Side View

Tolerances of Side Frame Overshadowed by Side Frame Pitch

If Bolster Remains Level then Force F1> Force F2 Causing Problems with Wear on Bolster Slope Face, Wedges and Side Frames

Solutions

Truck Maintenance Guidelines



Bolster Level or Pitched

Side View

Solutions: 1) Develop Wheel Set to Wheel Set Diameter Tolerance - Tolerance Wheel Diameters must be within 1/2 - 1/4 inch

2) When Lead Axle is Changed out for Wear, both Axles should be Replaced

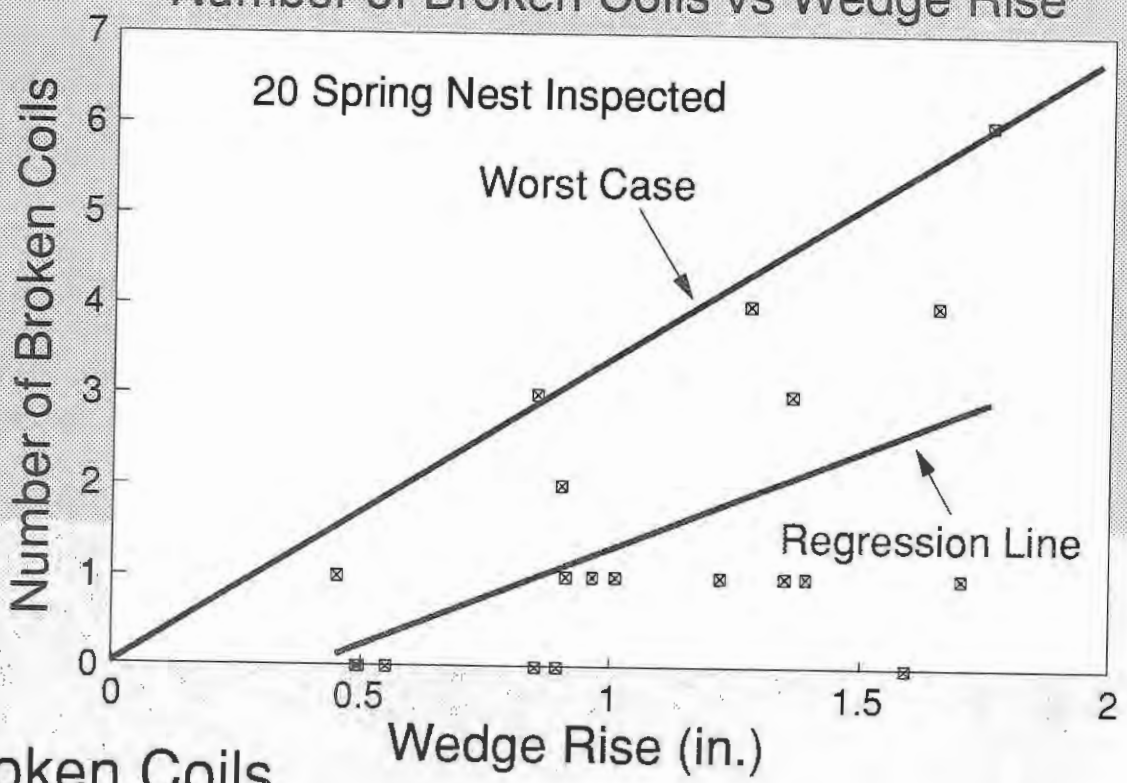
hsf1wear/S14.cdr

Solutions

Truck Maintenance Guidelines

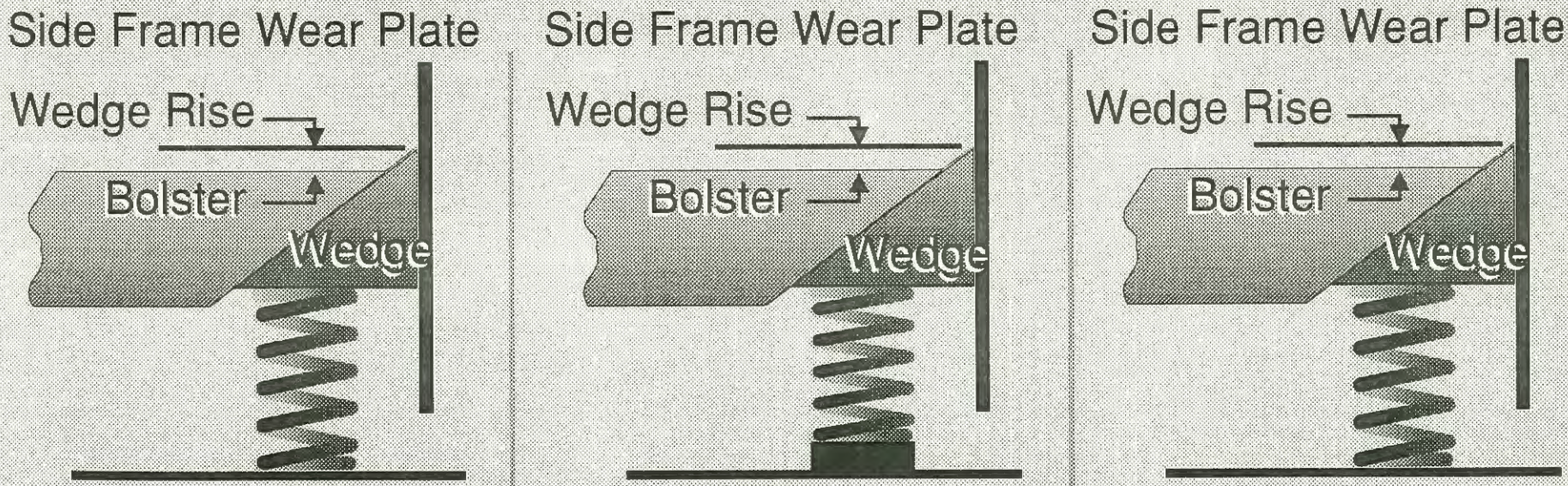
Broken Coils

- Reduce Travel
- Increase Vertical and Lateral Forces on Track and Truck Components
- Increase Wear



Solution: Inspect for Broken Coils

Temporary Fix To Increase Damping



Wedge rise reduces pre-load which reduces available damping

Increase pre-load by placing shims under control coil that is slightly smaller than the wedge rise

Increase pre-load by placing taller control coil under the wedge with approximately same stiffness and solid height

Note: Method works for all damping arrangements

hsf1wear/S10.cdr

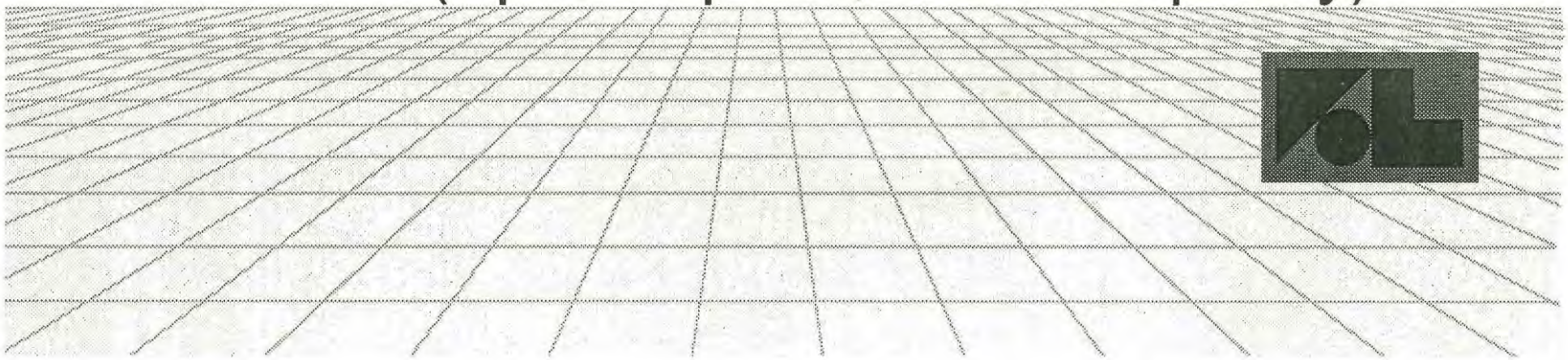
ECONOMICS - PROGRAM VS SPOT MAINTENANCE

P. LITTLE & E. W. SMITH
A & L ASSOCIATES

T. S. GUINS
ASSOCIATION OF AMERICAN RAILROADS

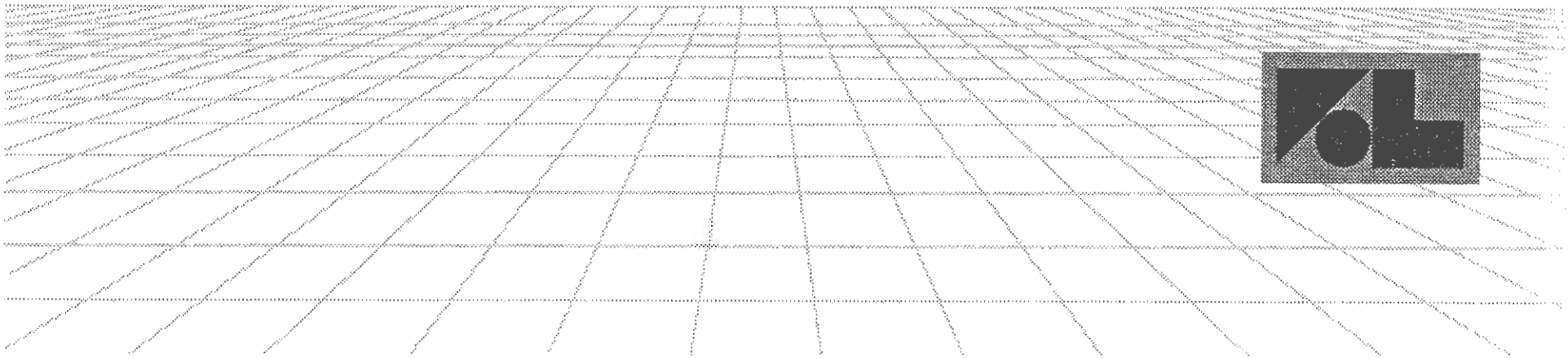
Benefits of "Better" Trucks:

- Derailment Reduction
- Track Maintenance
- Loss and Damage to Lading
- Reduction in Energy Use
- Car Component Wear Reduction
- Other (Spot Repairs, Track Capacity)



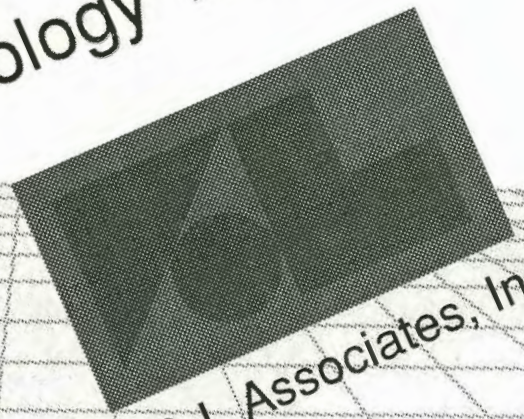
Overview

- Basis for Effective PM Programs:
  - Failure Characteristics of Components
  - Cost of "Failed" vs. "Good"
  - Inspection Policies and Practices
- Model to Analyze Policies
- Preliminary Results of Model



Economics of Truck Maintenance Policies

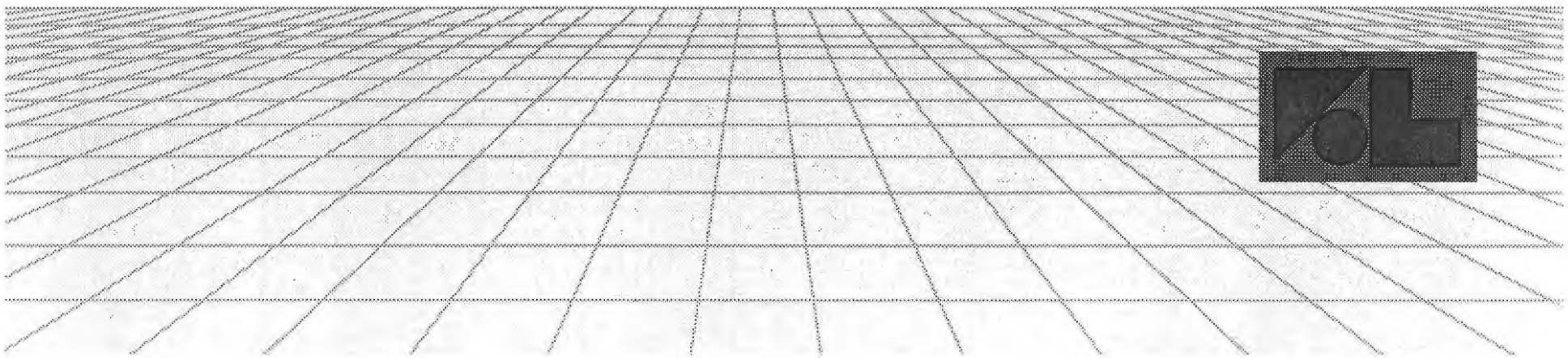
Freight Car Trucks for the 21st Century
AAR Technology Transfer Forum



A & L Associates, Inc.

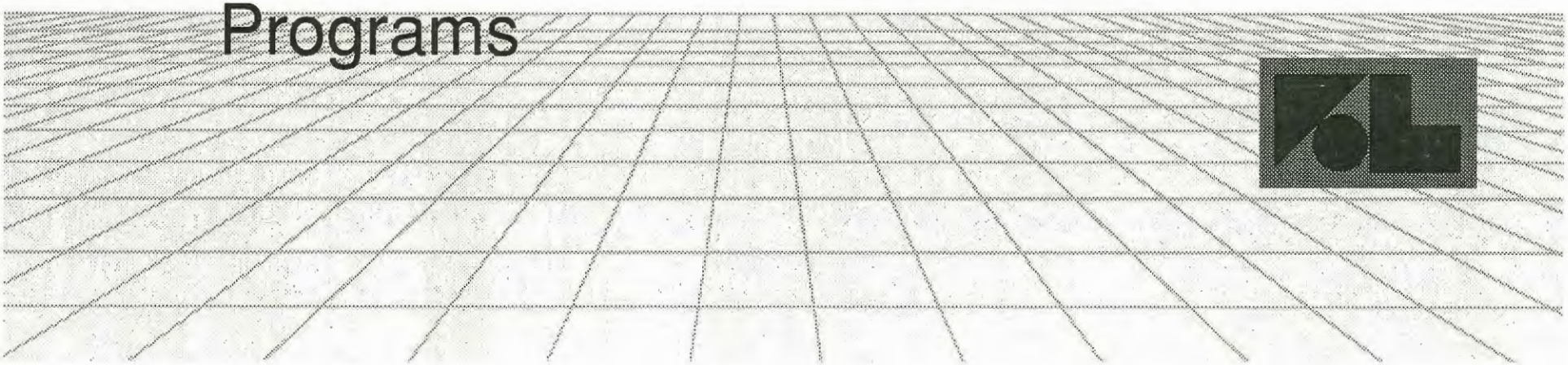
Costs of Truck Maintenance Policies:

- Inspection Costs
- Rebuilding of "Failed" Trucks
- Out-of-Service Costs
- Administrative Costs



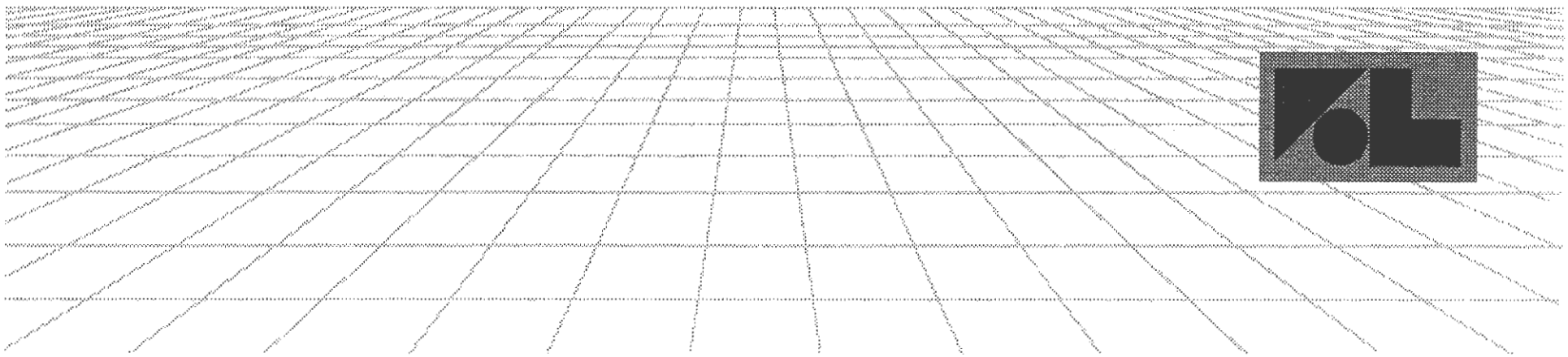
General Description of Economic Model

- Calculates NPV of PM Policy
- Marginal Costs and Marginal Benefits
- Fleet Level Policies Analysis:
  - e.g., Covered Hoppers, Coal Cars, etc.
- No Cost for Effective Maintenance Programs



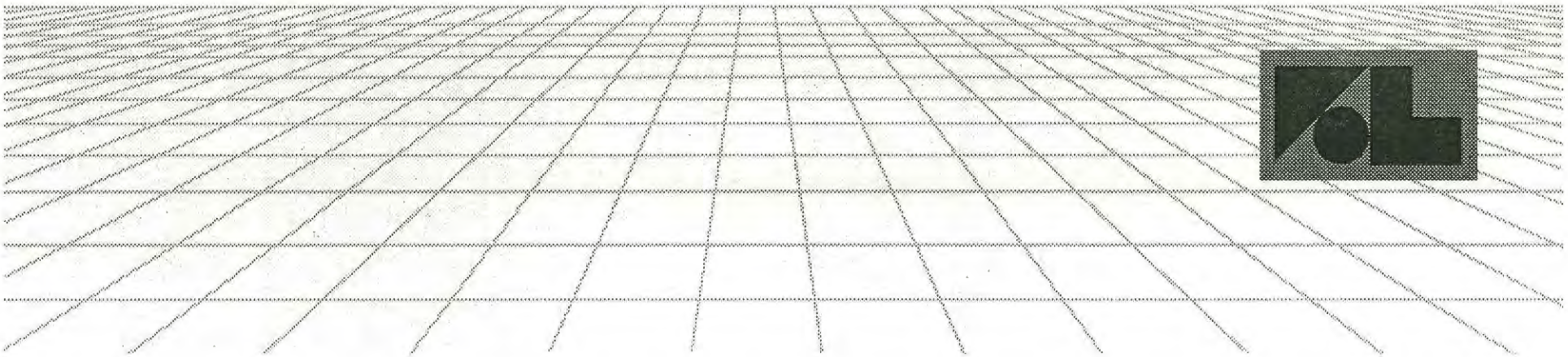
Model Structure

- LOTUS 123 and Javelin combination
- 123 Spreadsheet:
  - Applies Failure Distributions
  - Predicts Steady State Inspection/Rebuild
  - Calculate Annual Mileages



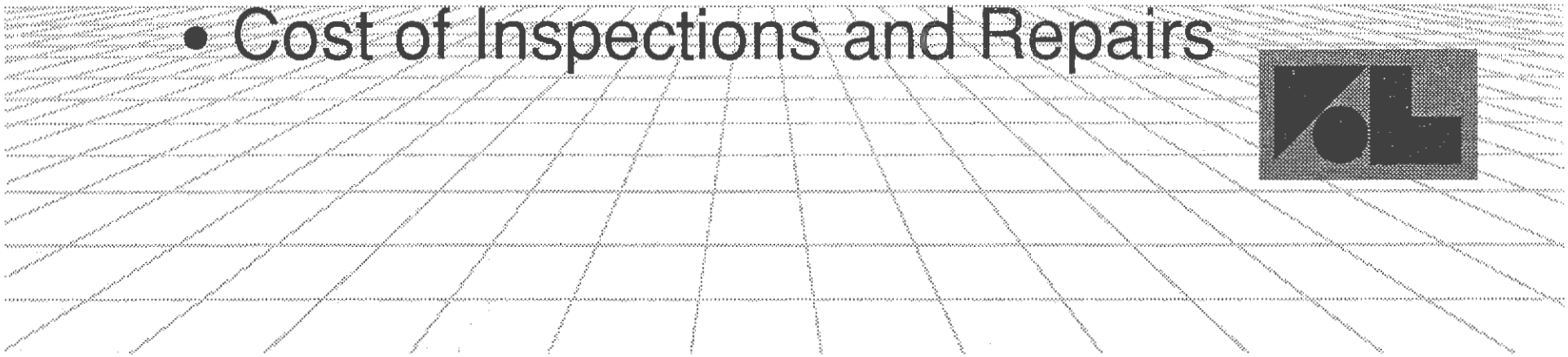
Model Structure II

- Javelin Model:
  - Cost/Benefit Accounting
  - Weights Costs by PM Strategy
  - Applies Deterioration Function
  - Evaluates Economics (discounting, etc.)
  - Output is 25 Year NPV



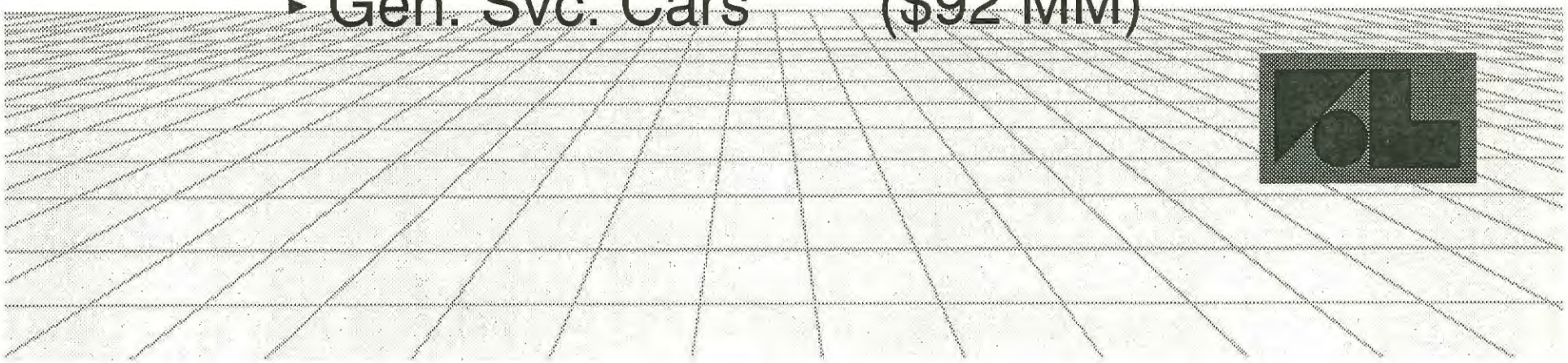
Model Input Requirements:

- Failure Distributions and Deterioration Rates
- Fleet Characteristics
- Current PM Programs of Car Owners
- Benefits of Good Trucks (per Mile)
  - Used NUCARS, TRACS, TEM
- Cost of Inspections and Repairs



Preliminary Results
(Base Case)

• Car Type:	• NPV:
▸ Coal Cars	$15.3 MM
▸ Cov. Hoppers	($30.8 MM)
▸ Double Stack I	$1.7 MM
▸ Double Stack II	$2.8 MM
▸ Tank Cars	($8.1 MM)
▸ Gen. Svc. Cars	($92 MM)



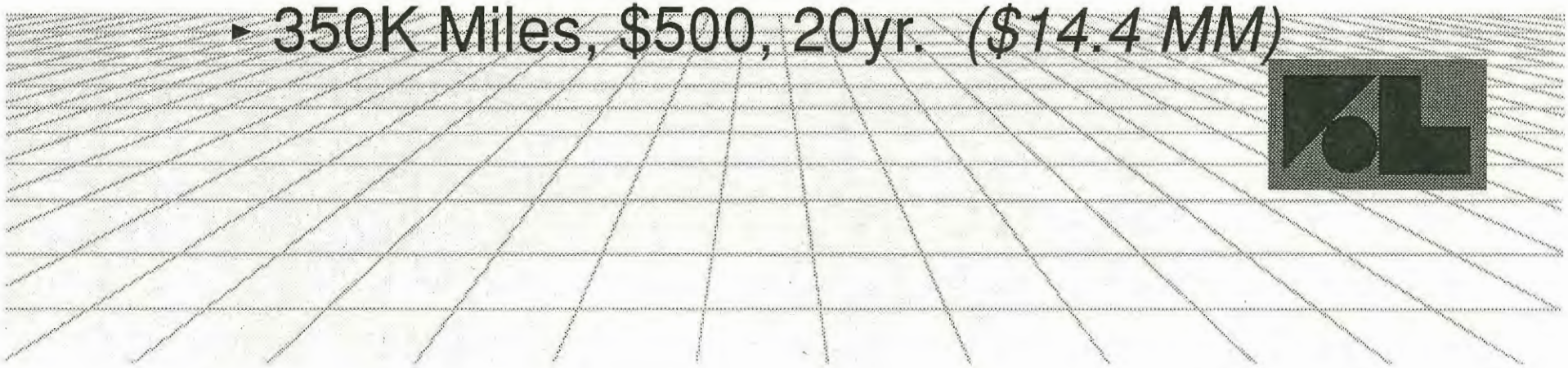
Preliminary Results
(Covered Hoppers)

- Base Case
  - 300K Miles, $1000/Rebuild
  - NPV = ($30.8 MM)
- Alternative 1
  - 400K Miles
  - NPV = ($23.1 MM)
- Alternative 2
  - 300K Miles, $500/Rebuild
  - NPV = +$15.3 MM

Preliminary Results II

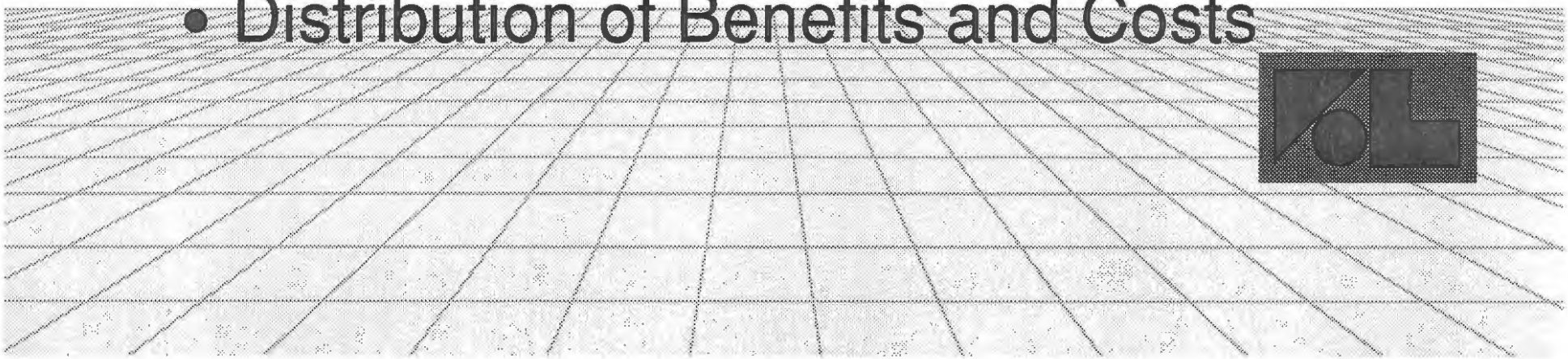
(20+ Alternatives)

- Example I (Covered Hoppers):
  - $500 v. $1000 *$15.3 MM*
- Example II (Tank Cars):
  - Hazmat,175K Miles *($1.2 MM)*
  - Non-Hazmat, 350K Miles $500 *($3 MM)*
- Example III (Gen. Svc. Cars):
  - 350K Miles, $500, 20yr. *($14.4 MM)*



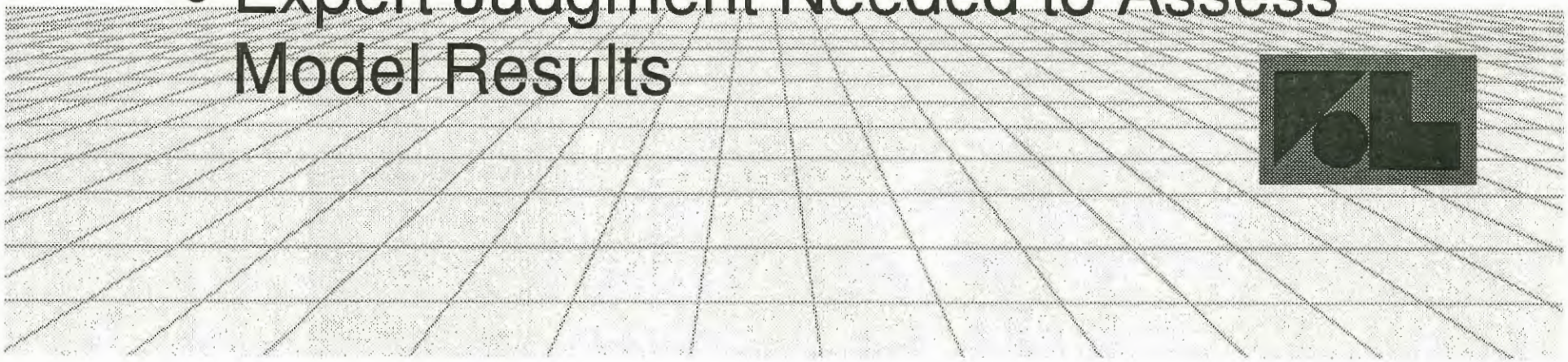
Conclusions

- Economics of Truck Maintenance Policies can be Modelled
- NPV depends heavily on:
  - Cost of Truck Rebuilds/Repairs
  - Annual Mileage
  - Level of PM Participation by Owners
- Distribution of Benefits and Costs



Recommendations

- Information is Needed:
  - Level and Cost of Truck Rebuilding
  - Failure Distributions and Rates of Deterioration
  - Current Maintenance Policies (esp. Low Mileage)
- Expert Judgment Needed to Assess Model Results



RESTORING TRUCKS

D. H. HABEL & M. C. COOK

BURLINGTON NORTHERN RAILROAD

Chicago Truck Forum Presentation
September 8, 1994

Havelock Shops began as a steam locomotive manufacturing facility in 1891. It was later converted to new car construction in 1935. In 1965 it became the heavy / program repair shop it is today. Havelock Shops employs 456 people in it's car shop and another 101 in the wheel plant. It occupies 320 acres with over 500,000 square feet of covered work area. Currently, Havelock produces approximately 22 cars per day.

Burlington Northern, in the past, has not enjoyed the safety performance it is attempting to establish and maintain at this time; as the chart on the next page indicates. We have improved significantly in this area. Our progress at Havelock Shops has mirrored and often times surpassed the railroad as a whole in it's many accomplishments.

One of the many areas of safety we have explored is ergonomics. Our Ergonomic Team has worked with locations and operations throughout the Burlington Northern system by redesigning storage bins, tables, tools, work processes and many other items to make them more ergonomically correct. The team is compiling an Ergonomic Handbook consisting of approximately 350 pages, detailing many of their ideas , innovations and improvements.

Our Truck Shop had one of the highest injury / incident rates of the entire complex due to the heavy components and many awkward positions a carman had to be in to perform repairs. The Truck Shop has almost totally eliminated injuries and now enjoys one of our lowest injury rates of the entire shop complex! The Truck Shop recently achieved **695 days without a lost time injury!** We attribute this to our proactive and new look at safety performance, mandated rest periods, improved ergonomic work positions and tools, and mechanically assisted methods. We will continue to improve!

Burlington Northern has been proactive in employee development and empowerment principles; all of which we are finding necessary to run our businesses economically , efficiently and safely. We have discovered successful ways of involving all our employees in the improvement process. Scheduled employees are currently working closely with management to redesign our Truck Shop from the ground up. A drawing of the new Truck Shop appears on page 3.

We are changing the way we repair trucks for many reasons. Some of these include:

Safety and Ergonomics
Flexibility in performing rule 47/48 repairs
M-214 processes
Utilization of limited working space and storage
A Bearing Test Machine, designed and built by one of our craft employees, that will incorporate and utilize a shock impulse device to determine the condition of the bearing and also locate out-of-round wheels in the process.

The industry continues to move forward with new truck designs, but we realize the process will not happen overnight. We at Burlington Northern and Havelock Shops have enjoyed much success through actively enlisting and utilizing the employee on the floor to help find answers to questions and solve problems. Many of these individuals have worked with trucks for 20 and 30 years. Possibly, the next new truck design is already in the mind of one of our current employees. Only by tapping all possible resources will we be able to eliminate the three piece truck.

Burlington Northern Team

Dave Habel - Steve Johnson - Marty Cook- Rick Smart

INCIDENT STATUS 1994 COMPARED TO 1993 THRU AUGUST

CAR

	1994	1993	1993 YEAR TOTAL
NUMBER OF INCIDENTS	10	28	34
FREQ. RATIO	3.36	9.34	7.76

WHEEL

	1994	1993	1993 YEAR TOTAL
NUMBER OF INCIDENTS	2	4	6
FREQ. RATIO	2.9	6.24	6.54

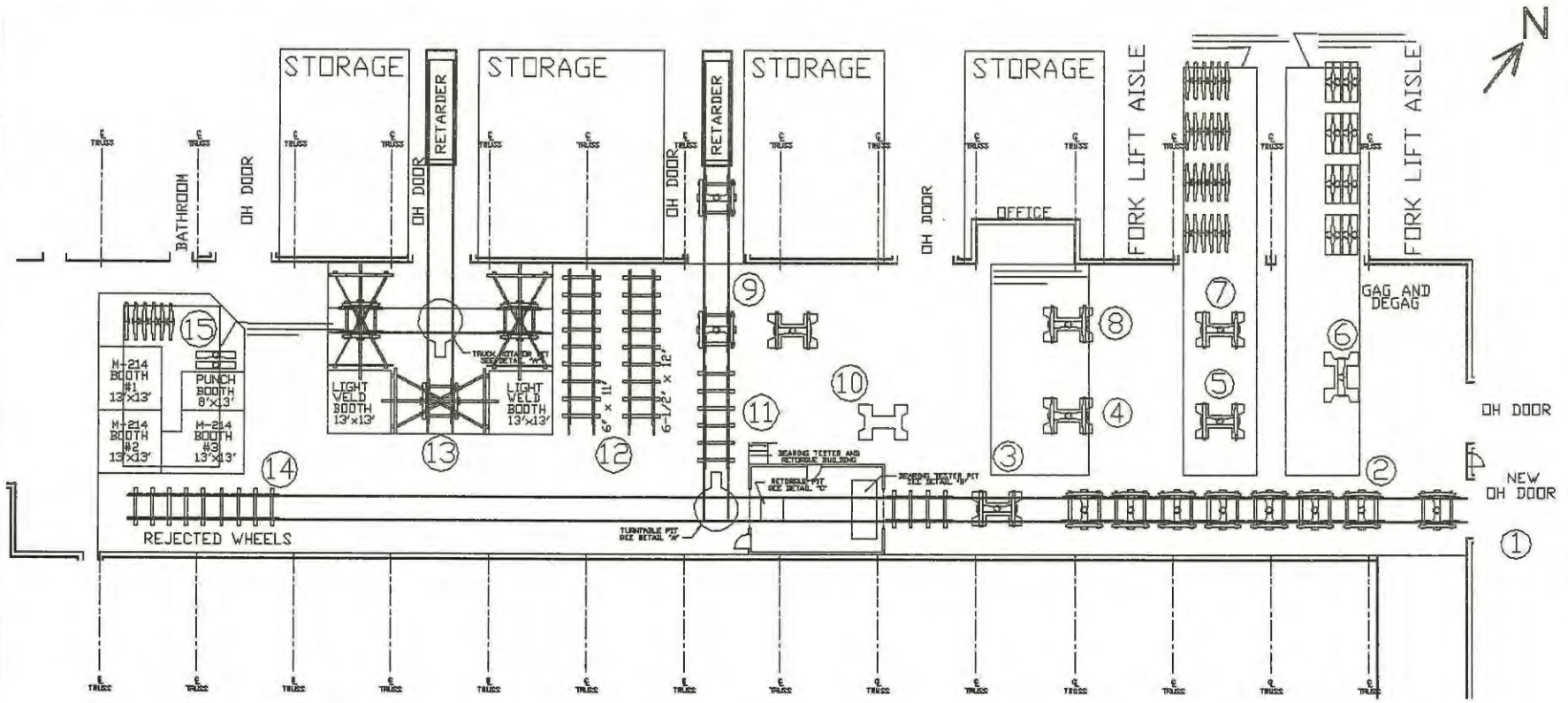
TOTAL CAR AND WHEEL

	1994	1993	1993 YEAR TOTAL
NUMBER OF INCIDENTS	12	32	40
FREQ. RATIO	3.27	8.79	7.55

LAST INCIDENT CAR JULY 21, 1994 49 DAYS

LAST INCIDENT WHEEL MAY 5, 1994 128 DAYS

1994 GOAL OF 50% REDUCTION VERUS 1993
WE PRESENTLY HAVE A 63.12% REDUCTION



STATION DESCRIPTIONS

1. ALL TRUCKS RECIEVING A COMPLETE TEARDOWN WILL ENTER THROUGH THIS DOOR.
2. RIDE CONTROL TRUCKS WILL BE GAGGED BY AUTOMAN AND FRAME KEYS WILL BE REMOVED IN THIS AREA.
3. TRUCKS WILL BE LIFTED OFF OF WHEELS AND PLACED ON SHUTTLE. WHEELS WILL CONTINUE INTO BEARING TESTER/RETORQUE BUILDING.
4. BRAKE RIGGING AND SMALL PARTS WILL BE REMOVED AT THIS LOCATION. BRIDGE CRANE WILL BE USED TO REMOVE BRAKE BEAMS.
5. SPRINGS AND TRUCK SIDES WILL BE REMOVED AT THIS LOCATION USING BRIDGE CRANE. SIDES WILL BE TAKEN OUTSIDE THROUGH DOOR IN THE NORTH WALL.
6. THE BOLSTER WILL BE REMOVED AT THIS LOCATION. IT WILL BE TAKEN TO THE DEGAGGING STATION FOR FRICTION BLOCK REMOVAL, THEN TAKEN OUTSIDE. A NEW BOLSTER WILL BE BROUGHT IN AND WILL HAVE THE FRICTION BLOCKS, SIDE BEARING CAGES, AND DEAD LEVER BRACKETS APPLIED.
7. TRUCK SIDES WILL BE BROUGHT IN FROM OUTSIDE AND INSTALLED DIRECTLY ON THE BOLSTER. SPRINGS WILL ALSO BE INSTALLED AT THIS LOCATION.
8. BRAKE RIGGING AND SMALL PARTS WILL BE APPLIED AT THIS LOCATION. ALSO, BRIDGE CRANE WILL BE USED TO INSTALL BRAKE BEAMS.
9. TRUCK WILL BE LIFTED OFF OF SHUTTLE AND PLACED ONTO WHEELS. ADAPTERS AND FRAME KEYS WILL BE INSTALLED AND AND TRUCK WILL EXIT THROUGH THE NORTH DOOR.
10. SHUTTLES WILL RETURN TO STATION 3 TO RECIEVE NEXT TRUCK.
11. REJECTED WHEELS WILL BE REPLACED WITH NEW WHEELS AND WHEELS WILL BE PAIRED FOR INSTALLATION AT STATION 9.
12. NEW WHEELS WILL BE STORED ON THE TWO TRACKS IN THIS AREA.
13. LIGHT REPAIR AREA. THIS AREA WILL HAVE A TURNTABLE AND THREE A-FRAMES. IT WILL BE USED TO MAKE THOSE REPAIRS NOT REQUIRING A COMPLETE TEAR DOWN OF THE TRUCK.
14. WHEELS HAVING AAR, BEARING, OR RETORQUE DEFECTS WILL BE STORED IN THIS AREA UNTIL THEY ARE MOVED TO THE WHEEL PLANT
15. M-214 RECONDITIONING OF TRUCK SIDES AND BOLSTERS WILL BE DONE IN THIS AREA.

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BURLINGTON NORTHERN

MECHANICAL DIVISION
HAVELOCK SHOPS
LINCOLN, NEBRASKA

TRUCK STATION RENOVATION
GENERAL LAYOUT

RMS	C	8/29/94	1/16" = 1'	RS0109	1-5

RAILROAD BEST PRACTICES

A PANEL DISCUSSION

DAVID K. MEYLER

CP RAIL SYSTEM

TRUCK MAINTENANCE-BEST PRACTICES

David K. Meyler
CP Rail Systems

An Assessment of Needs and Opportunities -
Freight Car Trucks For The 21st Century
A Technology Transfer Forum

September 7-9, 1994
Levis Faculty Center
University of Illinois at Urbana-Champaign

CP RAIL SYSTEM

TRUCK MAINTENANCE

- Based on planned maintenance approach
- Applies to 25,700 revenue cars and 1,100 OCS cars
- Inspections based on 200,000 mile intervals
- All condemnable components returned to FMS refurbishing facility at Weston
- 70% of cars have been PM'ed
- 2,800 cars planned for 1995
- Policy presently being reviewed for low mileage cars

CP RAIL SYSTEM

FLEXIBLE MANUFACTURING SYSTEM

- Reconditions to M214 standards
  - All side frames restored to 3 button dimensions
  - All column wear face surfaces machined
  - Column openings restored to nominal dimensions
  - All side frames receive stainless steel liners
  - All bolster pockets receive stainless steel liner
- Since 1990, 25,000 side frames and 6,000 bolsters have been reconditioned by FMS

RICHARD W. DAWSON

TTX COMPANY

TRUCK MAINTENANCE-BEST PRACTICES

Andrew Nibouar
TTX Company

AN ASSESSMENT OF NEEDS AND OPPORTUNITIES -
FREIGHT CAR TRUCKS FOR THE 21ST CENTURY
A TECHNOLOGY TRANSFER FORUM

September 7-9, 1994
Levis Faculty Center
University of Illinois at Urbana-Champaign

ECONOMIC BENEFIT

- **Planned reconditioning programs result in reduced out-of-service time, higher shop efficiencies, and lower replacement part costs**
- **Helps reduce railroad set outs**
- **Planned reconditioned results in better control of work being completed and a more uniform product**
- **Less damage to customer product**

TTX MILEAGE BASED RECONDITIONING CRITERIA

- **Tear down inspections based on actual measurements**
- **Trucks**
  - **Friction shoe rise, bolster gib spacing, thrust lug wear, wear plate condition side frames and bolsters, friction shoe condition**
- **Hitches, articulated connectors, brakes, side bearings, adapters, wheels and axles, end-of-car cushion devices, couplers, etc.**

TTX MILEAGE BASED RECONDITIONING CRITERIA

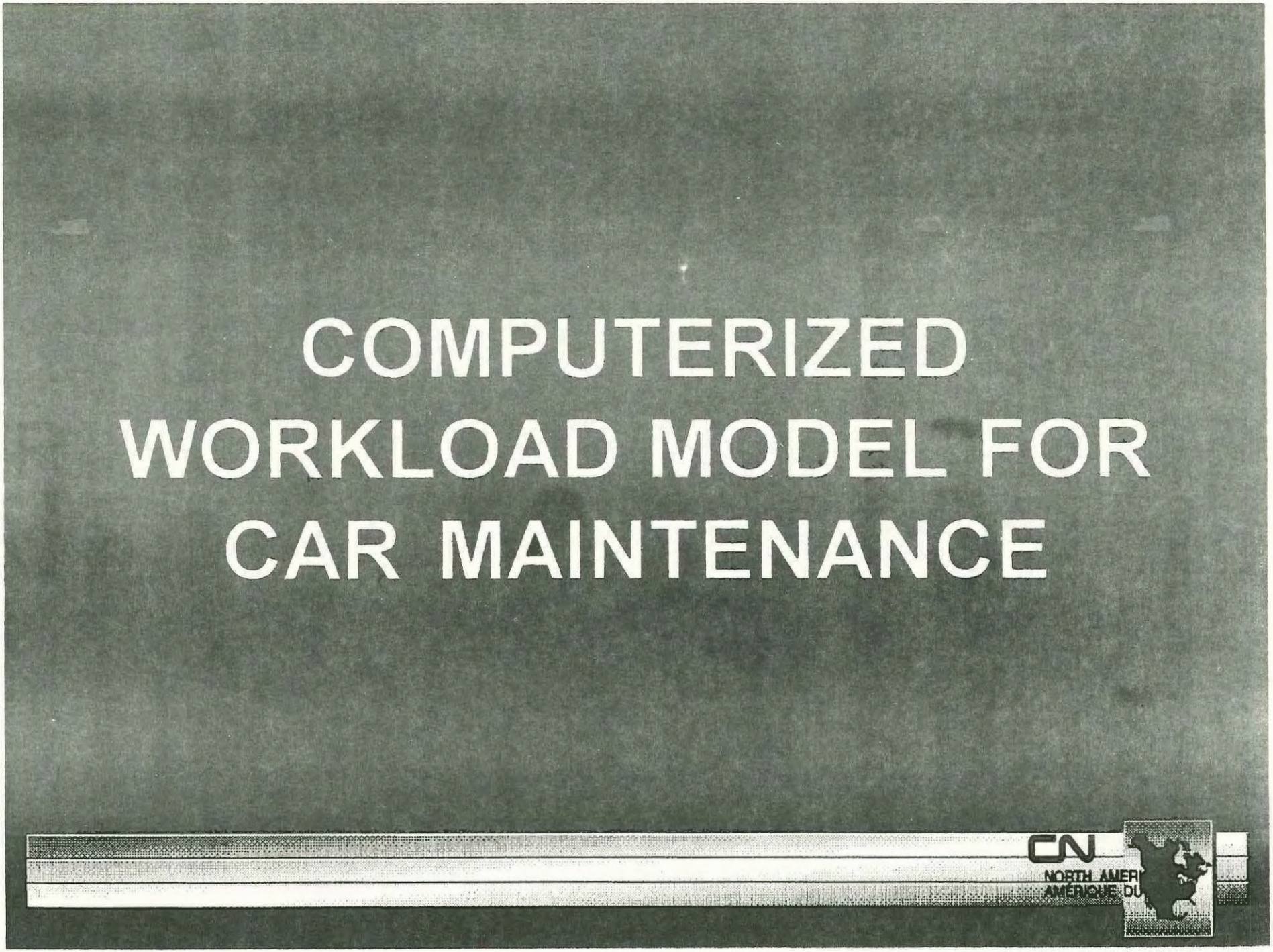
Continued

- **Average miles per year and type of service**
- **Market and economic considerations**
- **Conversion and upgrade programs**
- **TTX/TTC - Pueblo testing**
- **Over-the-road unattended revenue service performance testing**

WILLIAM J. LANDERS

CN NORTH AMERICA

COMPUTERIZED WORKLOAD MODEL FOR CAR MAINTENANCE



COMPUTERIZED WORKLOAD MODEL

- IDENTIFIES CARS DUE FOR "PIM" BASED ON MILEAGE.
- MILEAGE CRITERIA :
  - VALIDATED BY FIELD INSPECTIONS AND CUSTOMER FEEDBACK.
  - BASED ON CAR TYPE AND COMPONENTS.
  - DEPENDS ON CAR TYPE, MILES / YEAR AND TYPE OF SERVICE.
- DETERMINES CURRENT AND FUTURE FLEET CONDITION AND WORKLOAD REQUIREMENTS.



TYPICAL CRITERIA BY CAR TYPE

TYPE	MILEAGE	MILES/YR	YEARS
BOX	350,000	25,000	14
OPEN HOPPER	300,000	25,000	12
COVERED HOPPER	300,000	30,000	10
GONDOLA	200,000	15,000	13
5-PAKS	1,000,000	100,000	10
AUTO RACK	400,000	45,000	9
BULKHEAD FLAT	240,000	30,000	8
FLAT CAR	300,000	25,000	12
ROTARY GONDOLA	650,000	70,000	9

PREVENTIVE INSPECTION & MAINTENANCE " PIM "

- GENERAL REPAIRS PERFORMED ON ALL CARS.
- ALL TRUCKS REBUILT TO NEW CONDITION.
- CARS RECEIVING " PIM " ATTENTION PRIOR TO BEING REMOVED FROM SERVICE FOR MECHANICAL REASONS.
- COMPONENTS ARE REBUILT / REPAIRED INSTEAD OF RENEWED.
- REDUCED OUT OF SERVICE TIME WITH CONTROLLED MOVEMENT OF CARS.

COMPUTERIZED WORKLOAD MODEL FOR CAR MAINTENANCE



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SUPPLIER RECOMMENDED PRACTICES

A PANEL DISCUSSION

CHARLES MOEHLING

AMERICAN STEEL FOUNDRIES

TRUCK MANUFACTURERS' ENGINEERS' COMMITTEE

REPRESENTS

AAR Approved Manufacturers of Side Frames and Bolsters
(Members)

Manufacturers of Truck Snubbing Components
(Ex-Officio Members)

PURPOSE

Work with the AAR to:

- ✓ Establish Interchangeability Between Side Frames & Bolsters of a Given Type
- ✓ Establish Dimensional Standards to Assure Interchangeability
- ✓ Establish Specifications and Standards for the Design, Testing and Manufacture of Freight Car Trucks and their Components.

WORKS WITH

AAR Car Performance Subcommittee

AAR Staff

Technical Committees Representing:

- ✓ Car Builders
- ✓ Suppliers of Related Equipment

TRUCK MANUFACTURERS' ENGINEERS' COMMITTEE

CURRENT MAKE UP

MEMBERS:

- ✓ American Steel Foundries
- ✓ Buckeye Steel Castings
- ✓ National Castings Incorporated

EX-OFFICIO MEMBERS:

- ✓ Standard Car Truck
- ✓ A. Stucki Company

DESIGNERS - MANUAL SECTION D-II

- ✓ American Steel Foundries
- ✓ Buckeye Steel Castings
- ✓ National Castings Incorporated
- ✓ Standard Car Truck

AAR REQUIREMENTS
SIDE FRAMES AND BOLSTERS

AAR MANUAL SECTION D
TRUCKS AND TRUCK DETAILS

KEY SPECIFICATIONS:

- ✓ M-201, Steel Castings (Manual Section A-II)
- ✓ S-300, Basic Freight Car Truck Design Data
- ✓ M-202, Specification for Truck Bolsters
- ✓ M-203, Specification for Truck Side Frames, Cast Steel
- ✓ M-210, Specification for Purchase and Acceptance of AAR Approved Side Frames and Bolsters Applicable to Interchange Freight Cars
- ✓ M-1003, Specification for Quality Assurance (Manual Section J)

AAR REQUIREMENTS
SIDE FRAMES AND BOLSTERS

AAR MANUAL SECTION D-II, AAR CODE FOR DESIGNATING DESIGN FEATURES FOR SIDE FRAMES AND TRUCK BOLSTERS

✓ Interchange Code:
- Side Frames
- Bolsters

✓ Code Sheets

✓ Code Drawings

✓ Code Update
- All Designs in Production
- Latest Standard

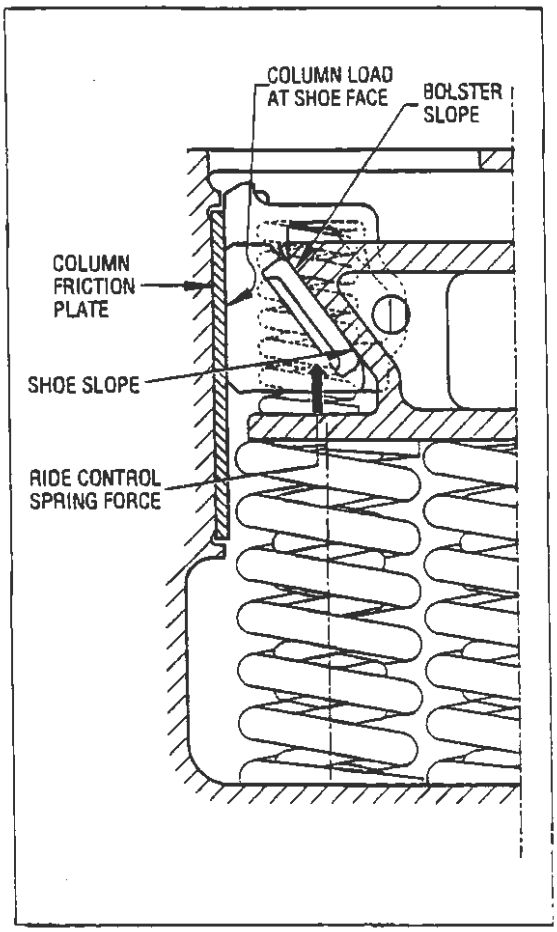
✓ Designer Responsibility
- Code Sheets
- Code Drawings

Super Service Ride Control and Ride Control Trucks

70, 100 & 125 Ton Capacity

Proper functioning of the Super Service Ride Control and Ride Control Trucks is dependent on the satisfactory condition of the control elements. Although the life expectancy of these cannot be predicted in terms of years due to the many service variables involved, they function as friction snubbers so that some wear is inherent in their action.

Wear in such a system can take place on: 1) the column friction plate, 2) the shoe face, 3) the shoe slope and 4) the bolster shoe pocket slopes. These are shown in the figure at the right. In all cases the wear is such as to permit the shoe to rise in the pocket with a consequent relaxation of the spring and a reduction in the "column load". Hence, A MEASUREMENT OF SHOE HEIGHT IS A COMPREHENSIVE MEASURE OF TOTAL SNUBBING ELEMENT WEAR.[1]



The most accurate method of measuring shoe height is with a yoke type gauge, page 8, or shoe height gauge, page 10. Both drawings are fully dimensioned for its manufacture and contain instructions for its use. Note that this measurement can be made with the car light, loaded, or partially loaded.

With Ride Control Trucks manufactured after mid 1976, built-in visual indicators are provided, and are also capable of indicating total wear and are also independent of the load in the car. See page 7, for proper use of these visual indicators. All Super Service Ride Control Trucks have these visual indicators.

In addition to both of these, a limit of wear groove is cast into the shoe face. This, when it is reached, indicates a need for shoe replacement based on shoe face wear. (See page 18)

[1] Reduction in "Column Load" can also result from a decrease in Ride Control spring capacity due to springs taking a permanent set. Control spring free heights should be checked as shown in instructions on page 44.

Gauge For Checking Combined Wear Of Assembled Control Elements

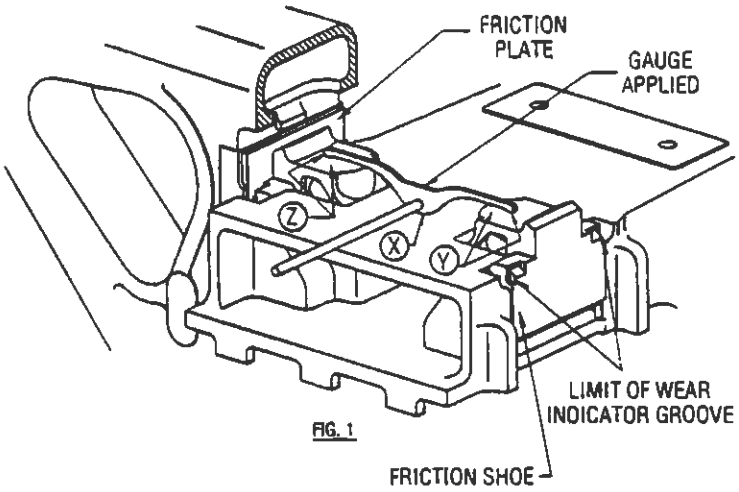


FIG. 1

NOTES:

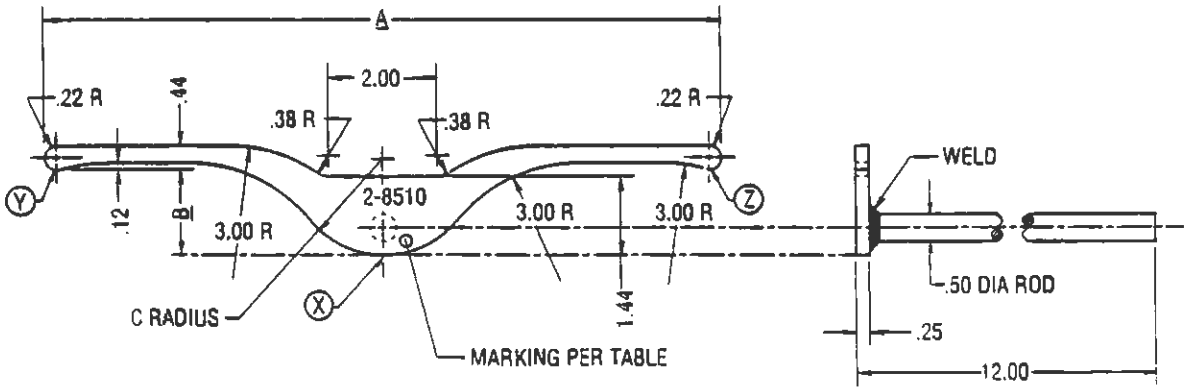
RIDE CONTROL ELEMENTS ARE IN WORKING ORDER WHEN GAUGE CONTACTS BOLSTER AT "X" AS ILLUSTRATED IN FIG. 1.

CONTROL REMAINS BUT REPAIR IS INDICATED AT THE TIME GAUGE CONTACTS BOTH FRICTION SHOES AT "Y" AND "Z" AND BOLSTER AT "X".

MATERIAL: STEEL
ALTERNATE - ALUMINUM 7075-T6 OR 7079-T6 (HARD COAT ANODIZED).

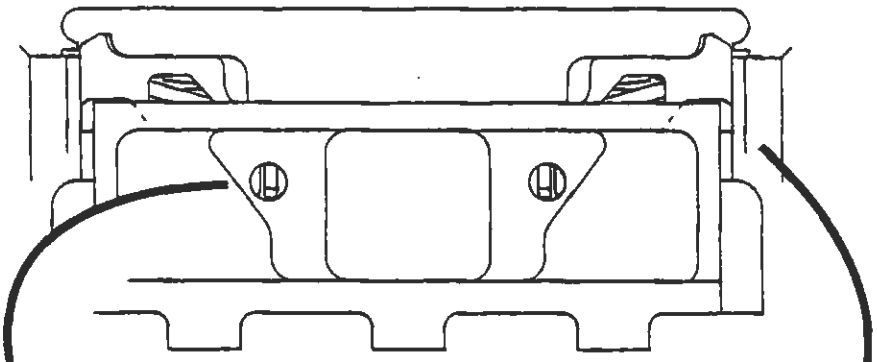
UNDERLINED DIMENSIONS ±.005 .

MARKING: LETTERS & NUMBERS .19 STAMPED.

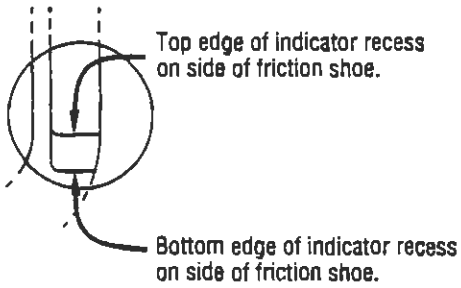


TYPE OF TRUCK	A	B	C	MARKING	LINE NO.
70 & 100 TON RIDE CONTROL TRUCK 125 TON RIDE CONTROL TRUCK (GRADE "C")	12.00	1.812	1.84	70 & 100 TON RIDE CONTROL TRUCK 125 TON RIDE CONTROL TRUCK (GRADE "C")	1
SUPER SERVICE RIDE CONTROL TRUCK	12.50	1.562	1.75	SUPER SERVICE RIDE CONTROL TRUCK	2
70 TON LOW LEVEL RIDE CONTROL TRUCK 5'-3" WHEELBASE	12.00	2.000	1.75	70 TON LOW LEVEL RIDE CONTROL TRUCK 5'-3" WHEELBASE	3
70 TON LOW LEVEL RIDE CONTROL TRUCK 5'-1" WHEELBASE	9.75	1.625	1.75	70 TON LOW LEVEL RIDE CONTROL TRUCK 5'-1" WHEELBASE	4

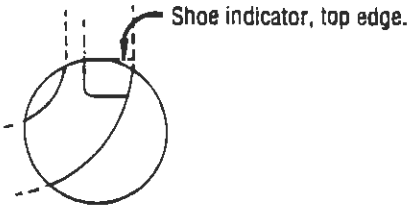
Shoe Height Visual Indicators



End elevation of bolster illustrating control indicators.



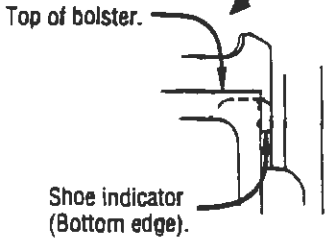
When shoe indicator is between top and bottom of holes in bolster, No Maintenance Is Required.



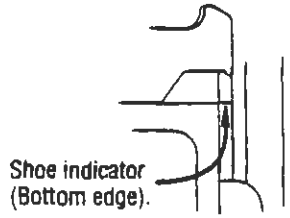
Control Remains But Repair Is Indicated when shoe indicator is in alignment with top of hole in bolster.

NOTE:
It may be necessary to average out the position of both shoe indicators if the bolster is off center.

NOTE:
This feature on all Super Service Ride Control trucks and all Ride Control trucks produced after mid 1976.



When bottom edge of shoe indicator is below the top of bolster, No Maintenance Is Required.

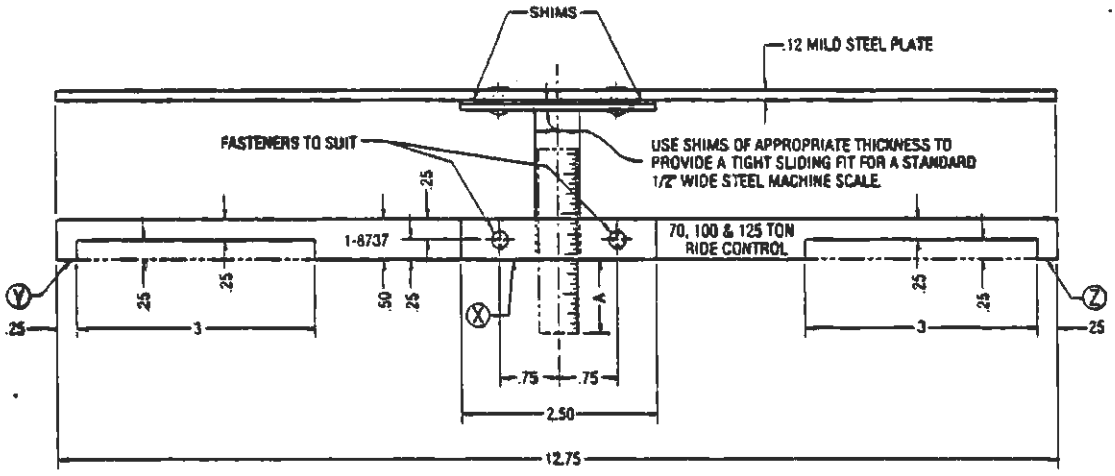


Control Remains But Repair Is Indicated when bottom edge of shoe indicator is in alignment with top of bolster.

NOTE:
It may be necessary to average out the position of both shoe indicators if the bolster is off center.

NOTE:
This feature on all Ride Control and all Super Service Ride Control trucks produced starting in 1980.

Shoe Height Gauge



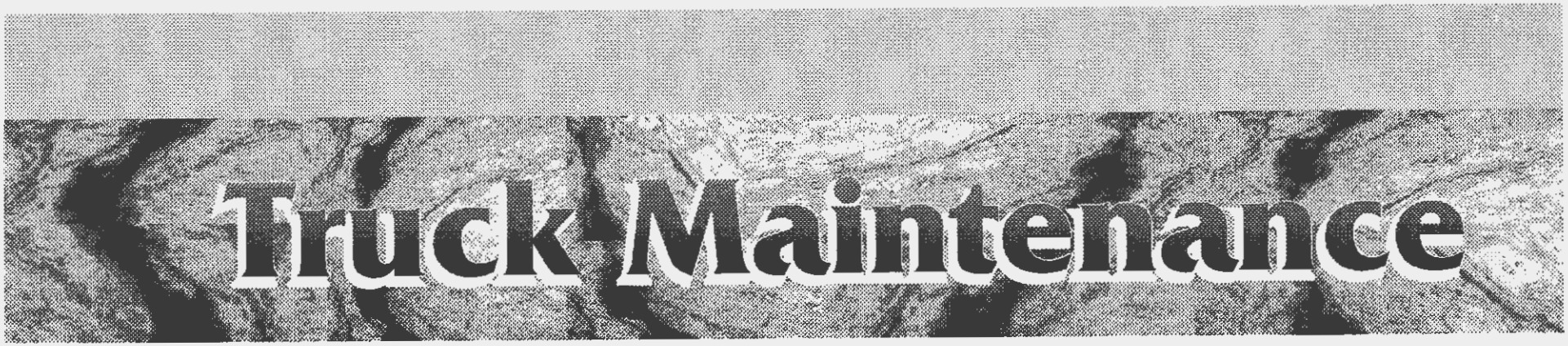
NOTES:
SURFACES "X", "Y" AND "Z" MUST BE IN THE SAME PLANE.

MARKING LETTERS & FIGURES .19 STAMPED.

	DIMENSION A		LINE NO.
	NOMINAL NEW DIMENSION	RECOMMENDED LIMIT OF WEAR	
70 & 100 TON RIDE CONTROL TRUCK (ALL TRAVELS)	.938	1.812	1
125 TON RIDE CONTROL TRUCK (GRADE "C" ONLY)	.938	1.812	2
70 TON LOW LEVEL RIDE CONTROL TRUCK 5'-3" WHEELBASE	1.125	2.000	3
70 TON LOW LEVEL RIDE CONTROL TRUCK 5'-1" WHEELBASE	.750	1.625	4
100 TON SUPER SERVICE RIDE CONTROL TRUCK (ALL TRAVELS)	.688	1.562	5

JACK R. DOWNES

BUCKEYE STEEL CASTINGS



Jack R. Downes
Director Engineering
& Mass Transit

Mark R. Carifa
Product Engineer

Buckeye Steel Castings

Subsidiary of Worthington Industries, Inc.

buc01.cdr

Why Freight Trucks Should be Maintained

- Worn friction components produce wedge rise.
- Wedge rise reduces the vertical, lateral and warp damping levels.
- The loss in damping force results in less control from twist and roll, and pitch and bounce.
- The loss in warp restraint increases the propensity of the truck to hunt.

When Buckeye XC-R Trucks Should Be Maintained

- We recommend trucks be rebuilt after one inch of wedge rise.
- The 100-ton XC-R has a column force of 6,550 pounds and a squaring moment of 56,500 in • lbs in the new condition. (4,940 lbs & 42,590 in • lbs for 70-ton XC-R)
- After one inch of wedge rise the column force is reduced to 3,630 pounds and the squaring moment is reduced to 31, 300 in • lbs. (2,740 lbs & 23,660 in • lbs for 70-ton XC-R)
- Our test data indicates that an increase in squaring moment (warp stiffness) provides better control of truck hunting.

buc03

How to Inspect Buckeye XC-R Trucks for Wear

- The Buckeye XC-R side frame features a visual wear indicator consisting of a large slot with two cast arrows.
- The two arrows are one inch apart.
- One inch of wedge rise is indicated when the top of the wedge aligns with the top arrow. Truck maintenance is generally recommended after one inch of wedge rise.

JAMES F. WRIGHT

A. STUCKI COMPANY

INTRODUCTORY REMARKS
PANEL DISCUSSION ON TRUCK MAINTENANCE,
AAR'S TRUCK CONFERENCE - CHAMPAIN, ILLINOIS

JAMES F. WRIGHT **Sept 8, 1994**

One hundred years ago, there were doubtless discussions in the Railroading fraternity about the needs for trucks in the upcoming Twentieth Century. This is a sketch of a truck which was common at that time.

SLIDE OF DIAMOND ARCH-BAR TRUCK OF ABOUT 1890

The three piece truck has been evolving for over 100 years. A continuing desire to improve three-piece truck performance has caused many individuals and companies, including A. Stucki, to develop improvements to enhance its utility, and we've heard about some of those efforts during this conference. The contemporary product of this evolution is a design which does a number of things rather well.

SLIDE WITH BULLETS FOR POINTS BELOW

The three-piece truck is inexpensive to produce and to maintain. It is simple and easily assembled with few tools. No adjustments of the basic mechanism are required, other than the relatively simple matching of side frame axle centers. It does a satisfactory job of equalization of wheel loads on the same rail.

A revolutionary system-designed truck as a replacement for the three piece truck has been the dream of many people for a long time. Most of these designs did not gain wide acceptance.

You may remember such designs as the Chrysler truck,

SLIDE OF THE CHRYSLER TRUCK

the Scissors truck,

SLIDE OF THE BARBER SCISSORS TRUCK

the Rockwell truck,

SLIDE OF THE ROCKWELL TRUCK

and others.

I don't mean to imply that the lack of success of many past efforts means that we should ignore all new ideas. Many service conditions require performance beyond the abilities of the standard three-piece truck. We must EAGERLY evaluate all proposals which offer a chance for the sort of quantum advance desired, but these

innovations must be measured against the formidable combination of performance and economy of the three-piece truck enhanced by modern technology.

It is possible to run a three piece truck for more than a million miles with no maintenance other than replacement of brake shoes and worn wheels.

SLIDE OF MILLION MILE TRUCK

The truck shown in this slide was subjected to just over a million miles in unit coal train service, and at the time it was removed from service could have continued "as is". It was removed because we wanted to display it at the 1992 RSA show, since it utilized several Stucki products.

The gibs were unworn,

SLIDE OF THE COLUMN GIBS

the friction pockets were unworn,

SLIDE OF THE FRICTION POCKETS

there were no broken springs, and the centerbowl vertical liner looked almost new. The wheels had lasted an average of about 700,000 miles; and best of all, the truck's condition was not untypical of the 50 cars similarly equipped. It looked so good

that experts in truck design doubted our truthfulness when we told them the mileage.

What must be done to make this performance routine rather than extraordinary? The design choices for the truck components are important, the truck should be suited to the intended service, and it should be maintained intelligently. This brings us to the subject of today's panel.

SLIDE OF BULLETS FOR POINTS BELOW

A truck which is providing substandard performance will exhibit tell-tale evidence of its problems. A proper reading of this evidence is necessary if we are to maintain the truck appropriately. Such things as wedge rise, evidence of spring rotation, worn gibs, damaged elastomer components, and unusual wheel wear can suggest the nature of a needed repair.

One cost effective measure is training which will change the attitudes of maintenance personnel from performing the "most expedient" repair to performing the "most effective" repair. For example, it is possible (and legal) to install our RFE Elastowedge friction elements into bolster pockets which are somewhat worn. If the pockets are accurately built up to something like the original unworn contour and slope, and ground smooth, they probably will never need to be repaired again, and the Elastowedges will last indefinitely. However, there is a temptation to try to

get by with as little work in renovation of the pockets as possible, resulting in poorer performance and shorter life.

Component suppliers spend a lot of time developing service literature. We make videotapes and send people all over the country to help our customers to train maintenance personnel. Use the supplier's services. They are one of the best resources available.

The common thread here is training. TRAINING PAYS. To review: read your truck, make the best repair possible, and use the resource provided by your supplier's field service team. Thank you

WILBUR G. STREAMS

NATIONAL CASTINGS, INC.

NATIONAL TRUCK DESIGNS

- C - 1
- SUPER C -1 WEDGELOCK
- SWING MOTION

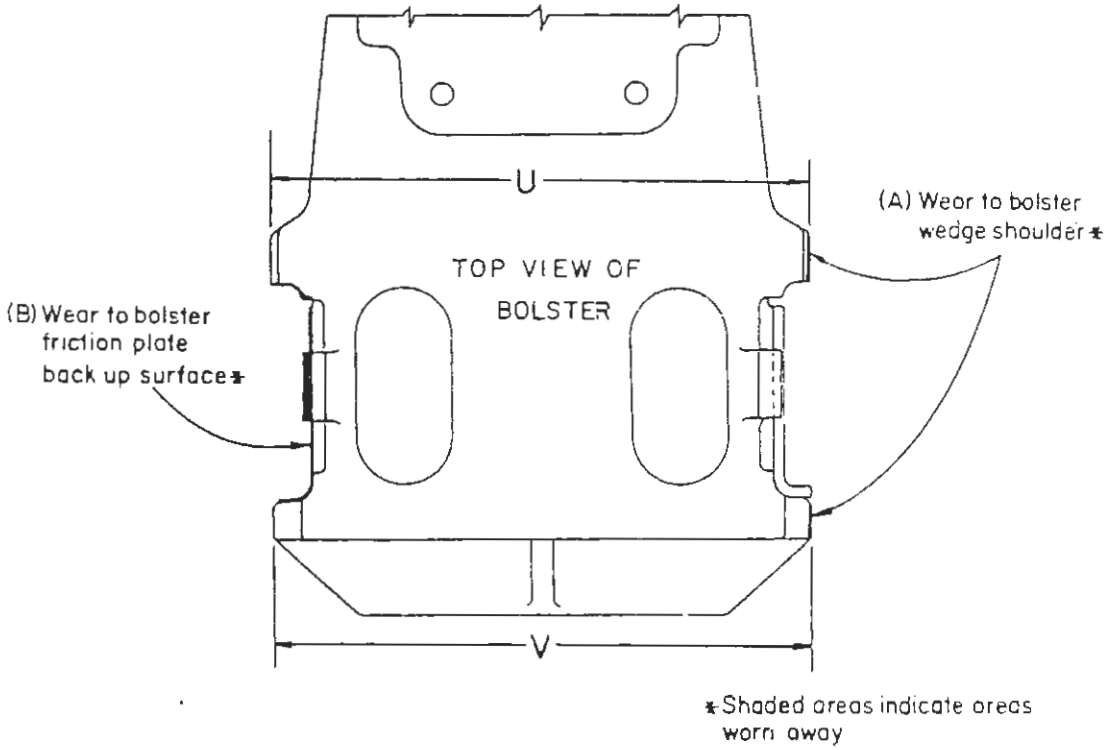


Figure 14

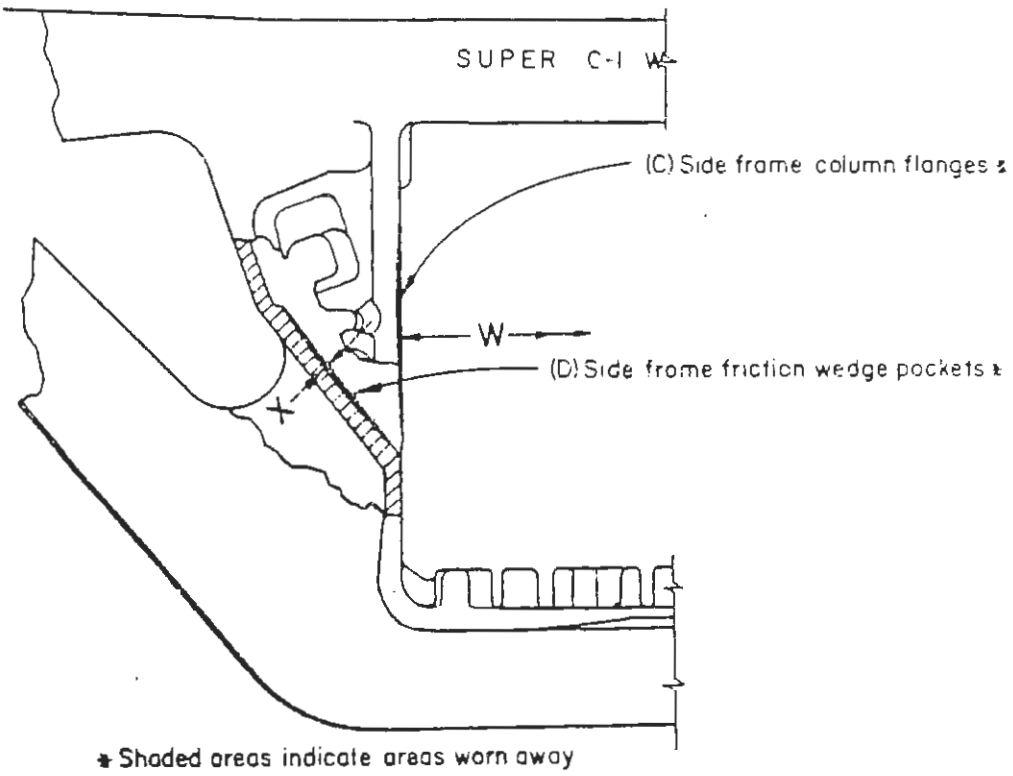
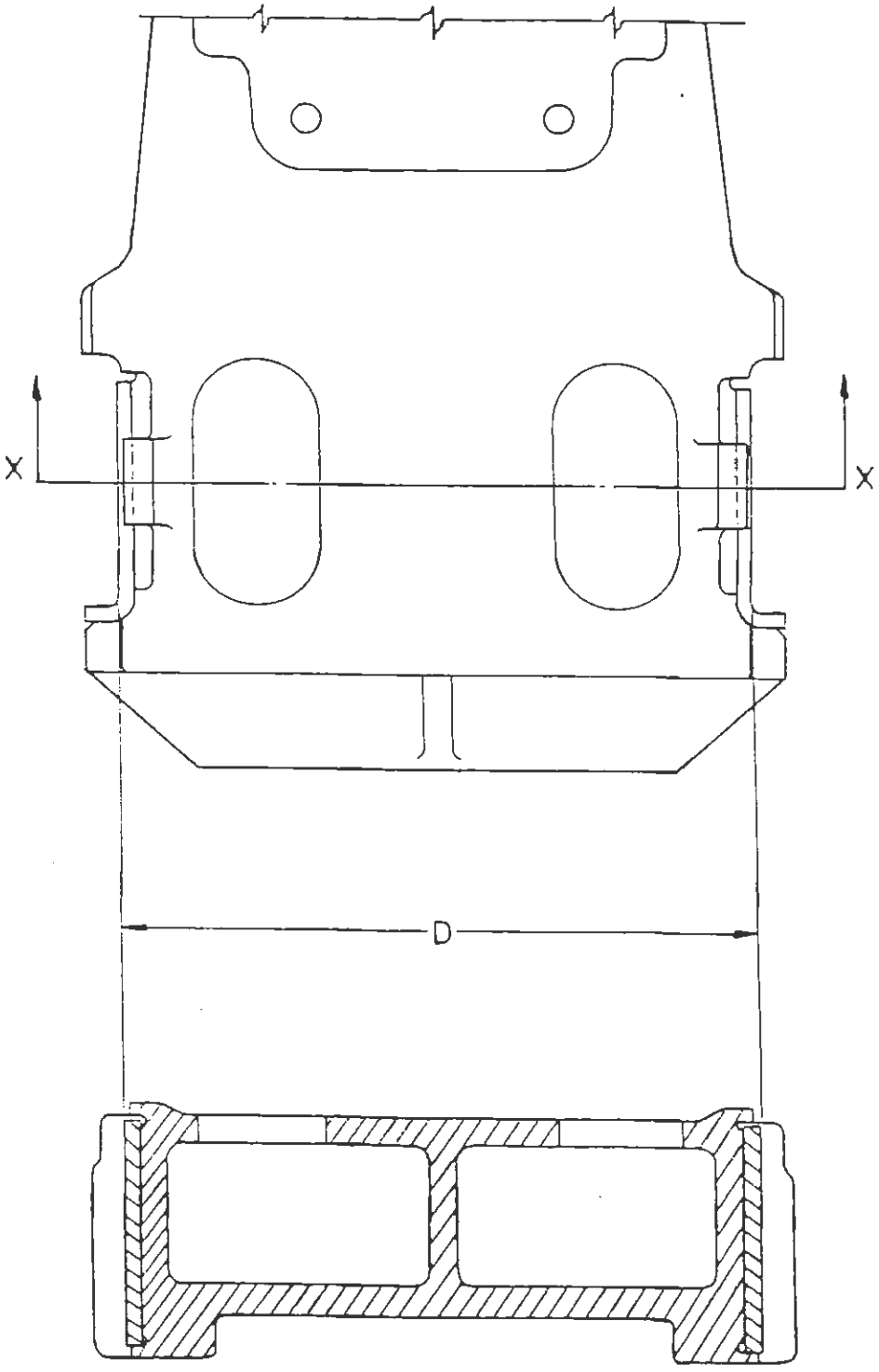


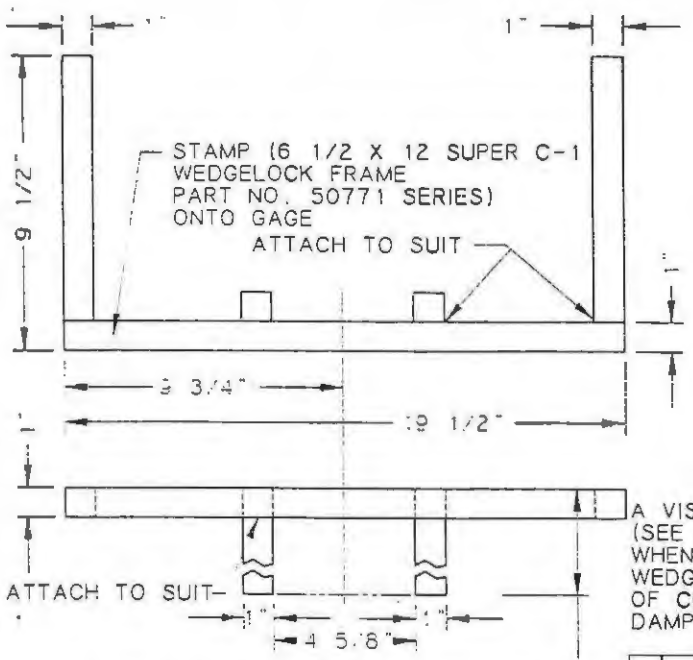
Figure 15

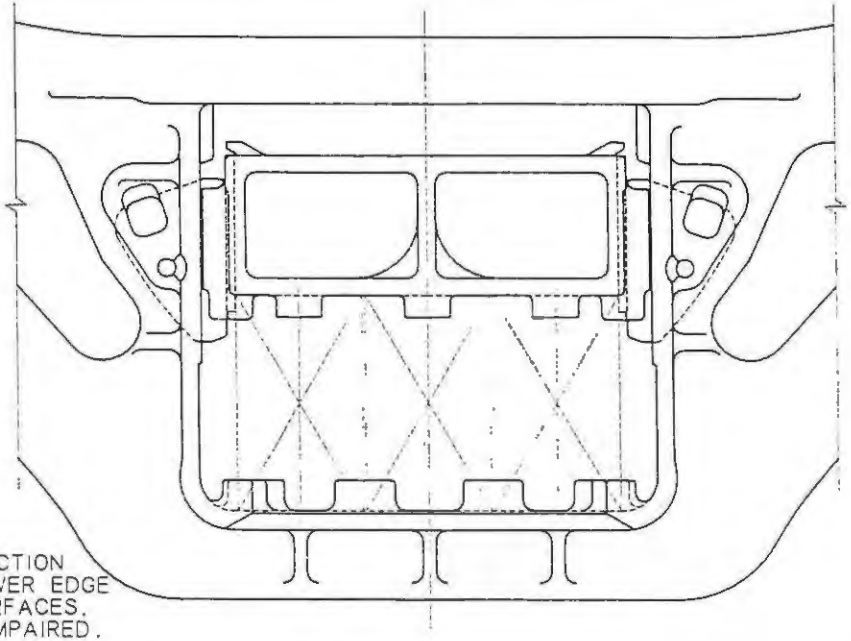


SECTION X-X

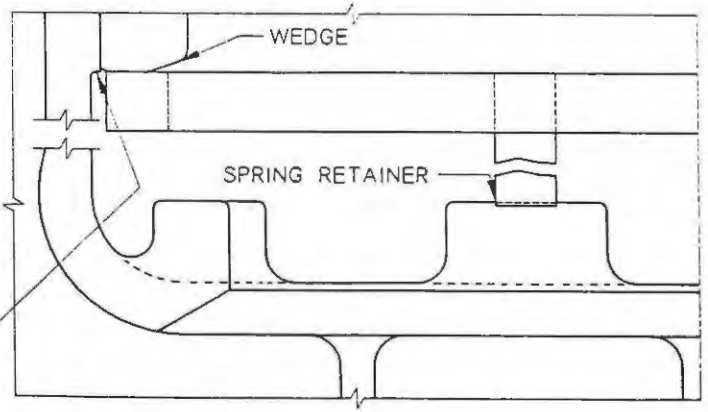
Bolster friction plates should be replaced when dimension "D" measures 17-3/16" or less on 6 x 11, 6-1/2 x 12 or 7 x 12 size bolsters.

Figure 11





A VISUAL INDICATOR
(SEE NOTE NO. 1)
WHEN TIPS OF BOTH FRICTION
WEDGES ARE BELOW LOWER EDGE
OF COLUMN BEARING SURFACES,
DAMPING OF TRUCK IS IMPAIRED.



MATERIAL: COLD DRAWN SQUARE

APPLICATION PROCEDURE

SEAT GAGE AGAINST BOTTOM PORTION OF WEDGE. WHEN BOTTOM LEG OF GAGE DOES NOT PASS OVER SIDE FRAME SPRING RETAINER REPAIR WORK IS INDICATED.

NOTE: 1) THIS APPLIES ONLY TO SUPER C-1 WEDGELOCK TRUCKS MANFG. WITH CAST DATE 1-86 OR LATER.

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FRACTIONS ±	
DECIMALS ±	
ANGLES ±	

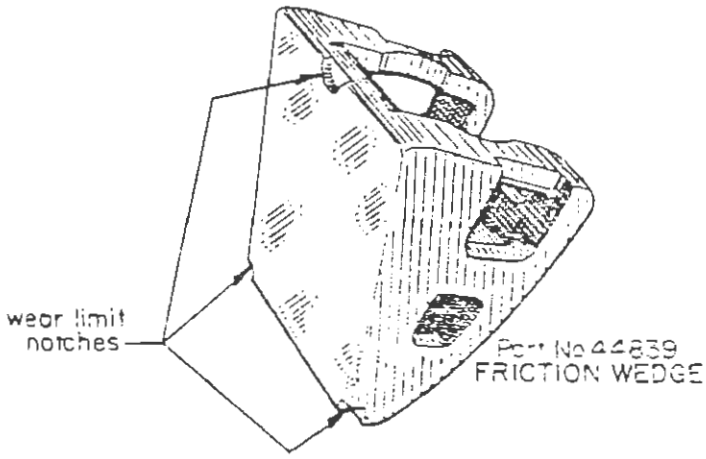
ISSUE	ORDER	REVISION	ZONE

NATIONAL CASTINGS INC.
1400 S. Laramie Ave.
Cicero, Illinois 60650

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NATIONAL SUPER C-1 WEDGELOCK TRK
6 1/2 X 12 SIDE FRAME
DAMPING CONTROL WEAR INDICATOR

ORD. 32288	SIZE	DWG. NO.	51553
DR. J.H. CK'D	B		
APPROVED		DATE	APRIL 30, 1990
	SHEET	OF	SCALE



Replace wedges when wear limit notches are 1/16" deep or less.

Figure 8

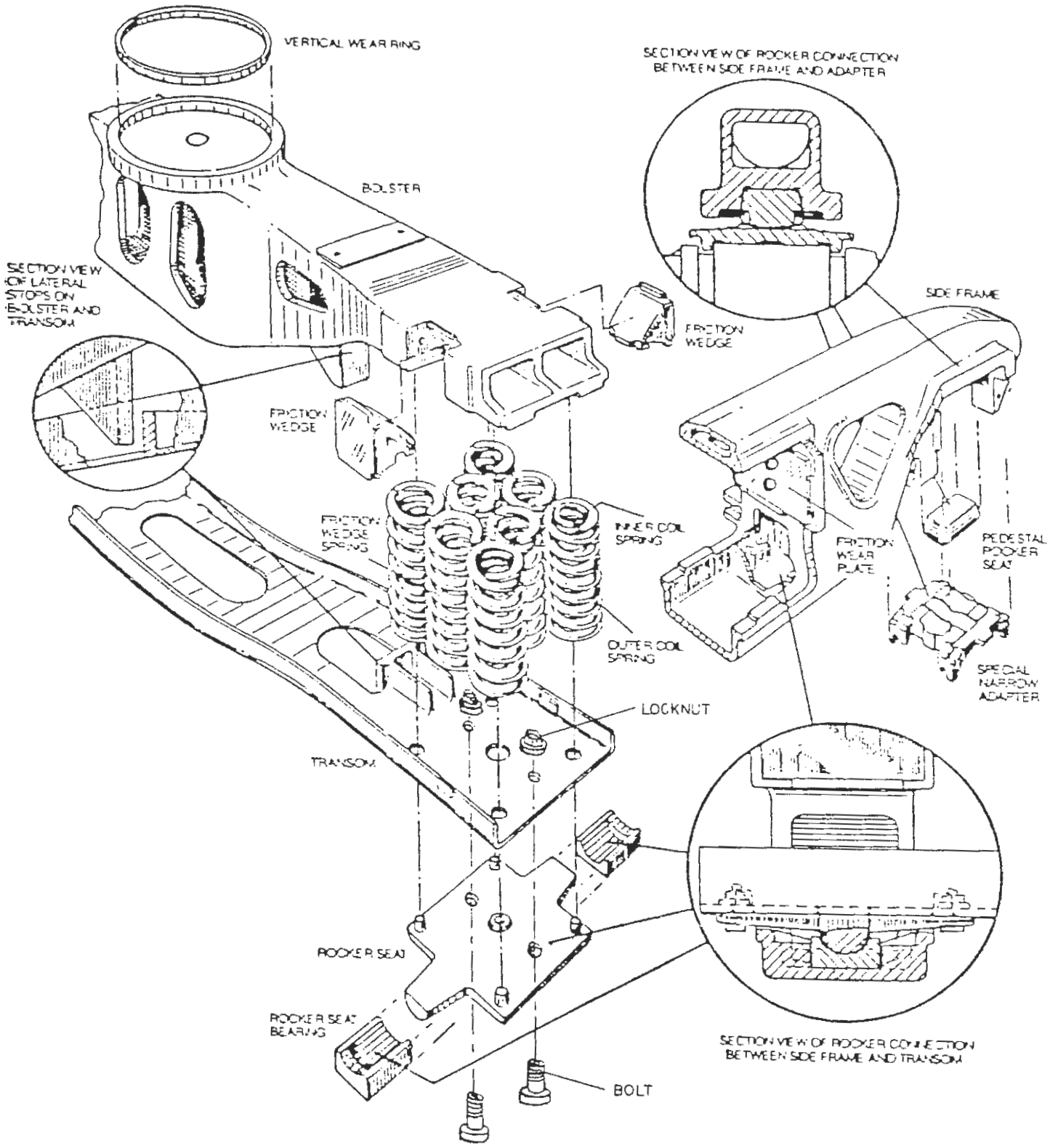
FRICTION WEDGE SPRINGS

Journal Size	Load Spring Deflection	Drawing No.	Outside Diameter	Free Height	Remarks
6x11	2-1/2	35166	3-1/8	9-1/4	See Note 1
6x11	3-1/16	35101	3-1/16	9-7/16	" "
6x11	3-11/16	35102	3-1/8	9-3/8	" "
6-1/2x12	All	43261	3-1/8	9-23/32	" "
7x12	All	44877	3-1/8	10-5/16	" "
6x11, 6-1/2x12 and 7x12	All	47664	3-3/8	10-17/32	" " 2
6x11, 6-1/2x12 and 7x12	All	47663	1-7/8	9-11/16	" " 3

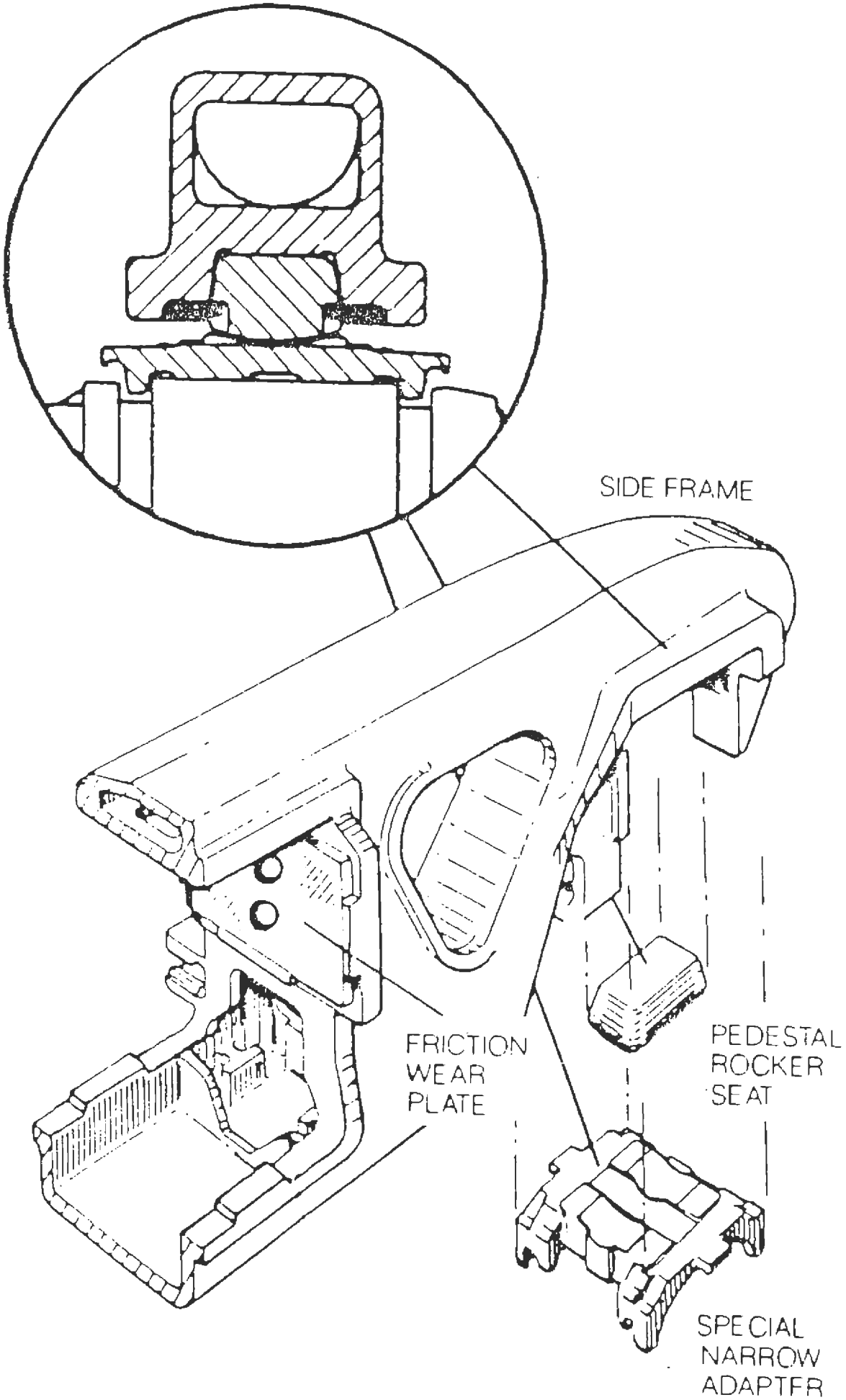
Notes:
1. These single coil wedge springs provide normal friction control.
2. This single coil wedge spring provides increased control. (Approximately 2x normal.)
3. This inner coil when used with No. 47664 outer coil provides increased control. (Approximately 3x normal) No. 47663 cannot be used alone.

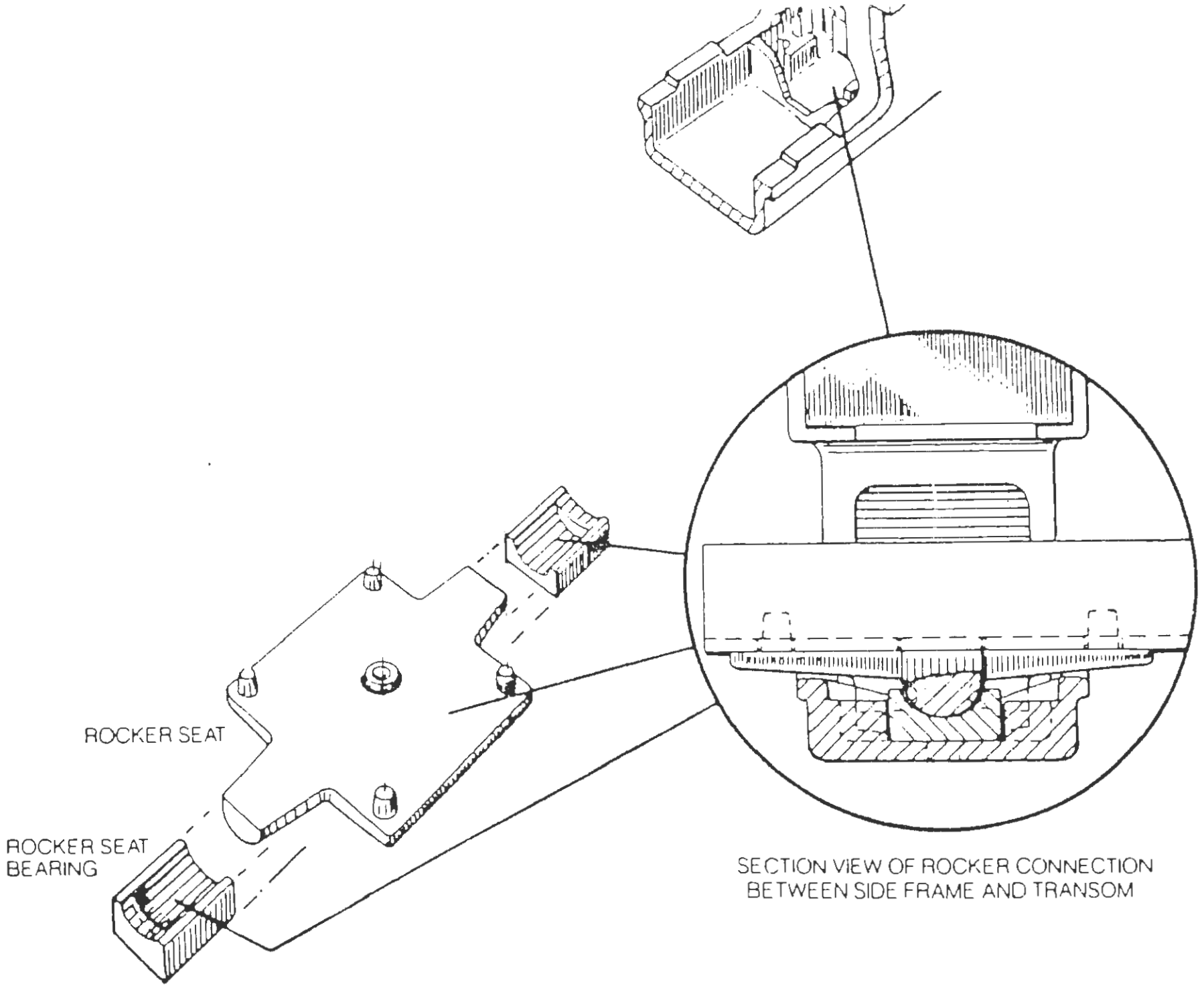
Figure 9

NATIONAL SWING MOTION TRUCK EXPLODED VIEW OF ASSEMBLY

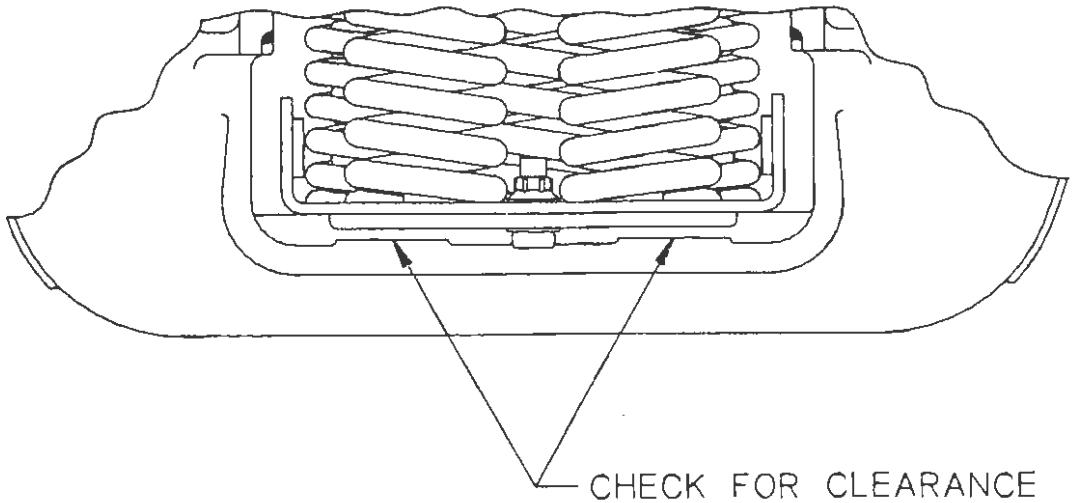


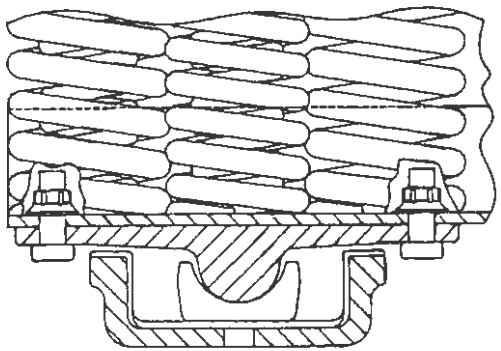
SECTION VIEW OF ROCKER CONNECTION
BETWEEN SIDE FRAME AND ADAPTER

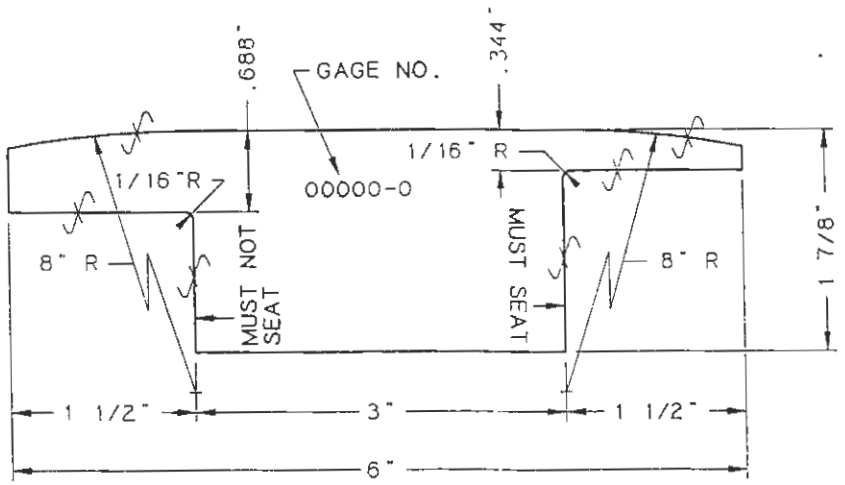




SECTION VIEW OF ROCKER CONNECTION
BETWEEN SIDE FRAME AND TRANSOM

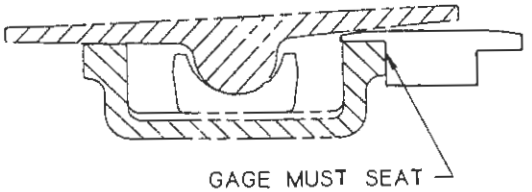

CHECK FOR CLEARANCE
AT PADS ON SIDE FRAME.

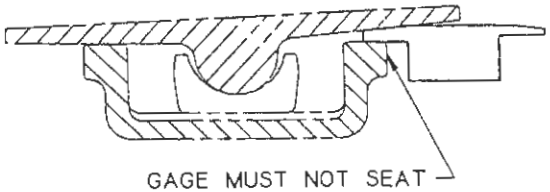




GAGE NO. 49706-1

1/4" STEEL-HARDEN $\sqrt{}$ SURFACES Rc 40-50





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FRACTIONS ± 1/32"	
DECIMALS ± .002"	
ANGLES ± 2 MIN.	

NOTES:
1) ALL MARKINGS STAMPED ON.
2) BREAK ALL SHARP EDGES.

ISSUE	ORDER	REVISION	ZON
A	28418	NO GO END THICKNESS CHANGED AND REDRAWN.	
8.21.91	DP JH		

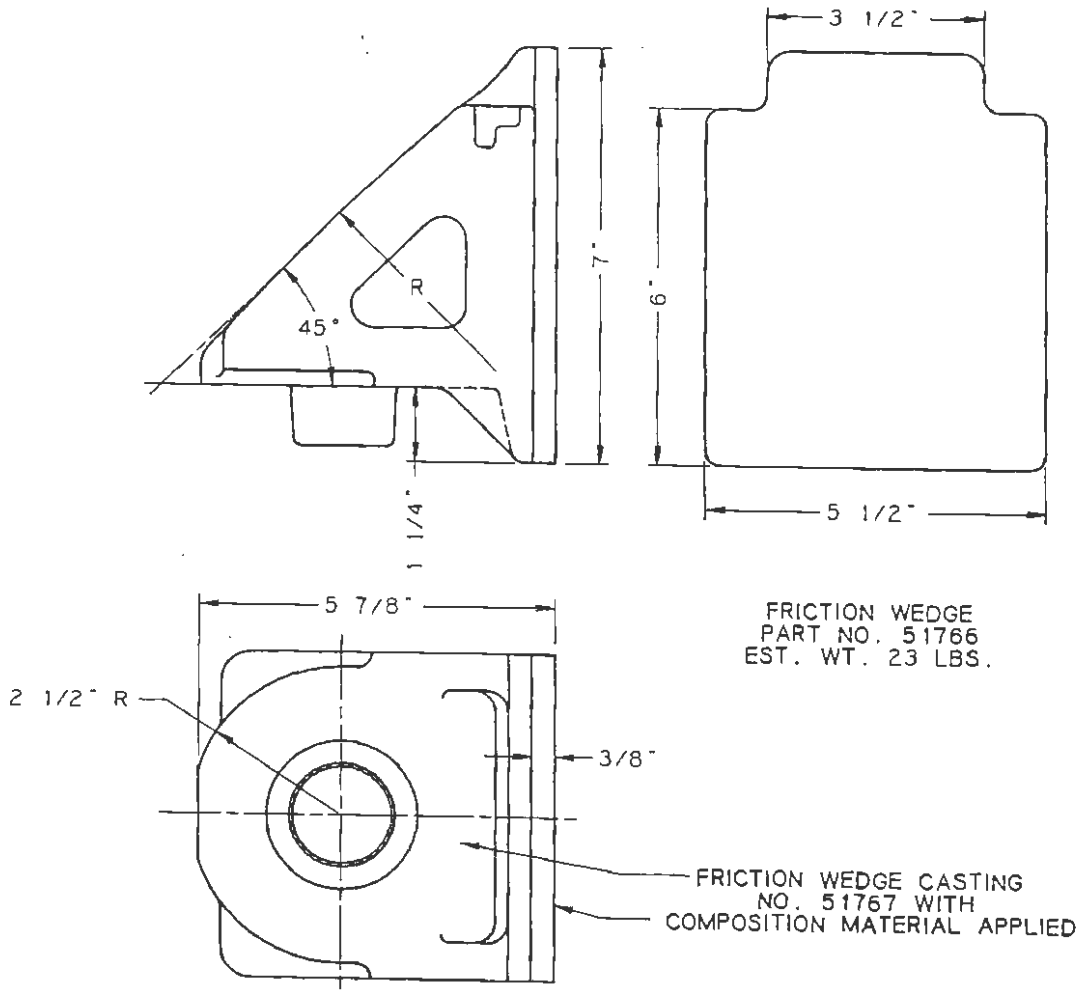
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Cicero, Illinois 60650

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NATIONAL SWING MOTION TRUCK

ROCKER SEAT CLEARANCE GAGE

ORD. 28418	SIZE	DWG. NO. 49706
DR. RWNCK'D R.	B	
APPROVED		DATE JAN. 30, 1975
	SHEET OF	SCALE



FRICTION WEDGE
PART NO. 51766
EST. WT. 23 LBS.

51767

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FRACTIONS ±	
L	
DECIMALS ±	
ANGLES ° ±	

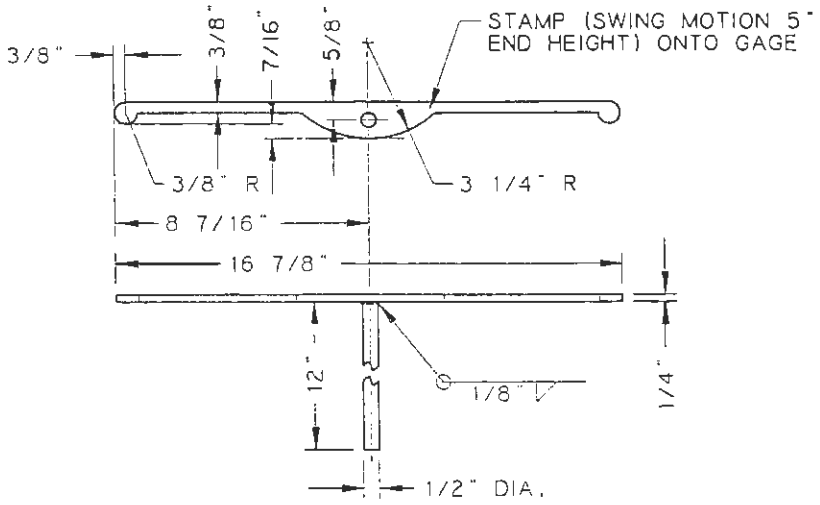
A	32123	VISUAL INDICATOR FOR LOSS OF SNUBBING ADDED.	
11-5-92	AH		
ISSUE	ORDER	REVISION	ZO:

NATIONAL CASTINGS INC.
1400 S. Laramie Ave.
Cicero, Illinois 60650

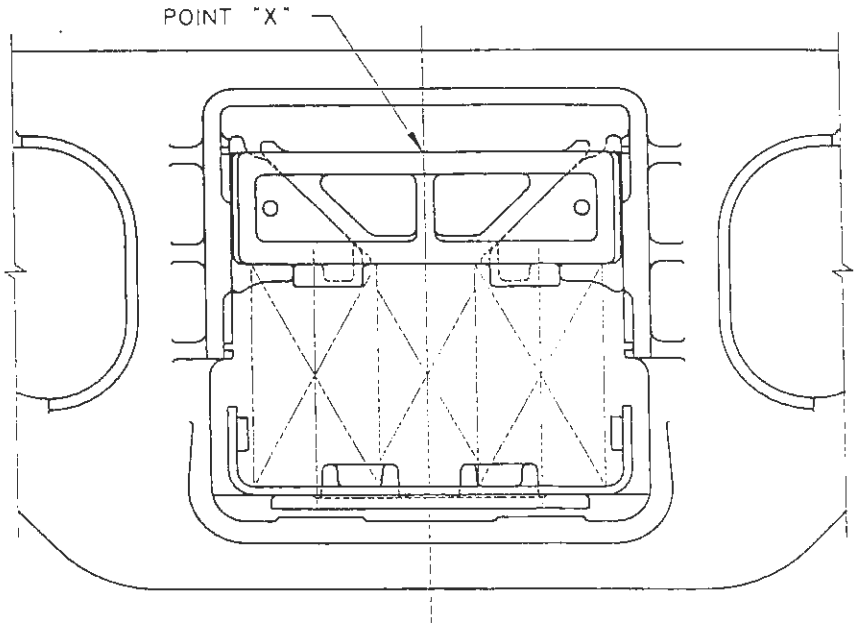
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NATIONAL SWING MOTION TRUCK
6 X 11 & 6 1/2 X 12 FRICTION WEDC
WITH COMPOSITION FRICTION MATERIA

ORD. 32123	SIZE	DWG. NO.	51766
DR. JH CK'D AH	B		
APPROVED		DATE	OCT. 21, 1991
	SHEET	OF	SCALE

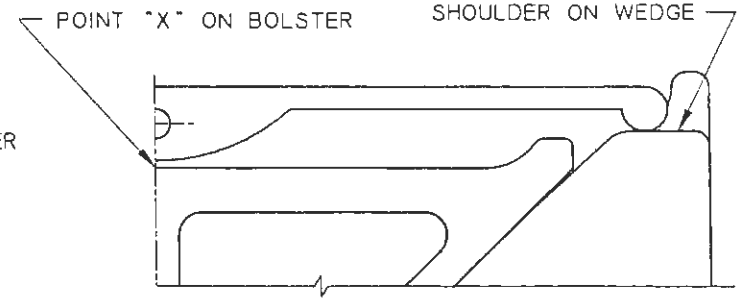


MATERIAL: COLD ROLLED STEEL



APPLICATION PROCEDURE

IF GAGE DOES NOT CONTACT BOLSTER AT POINT "X" WHILE RESTING UPON BOTH FRICTION WEDGE SHOULDERS REPAIR IS INDICATED.



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FRACTIONS ±	
DECIMALS ±	
ANGLES ±	

ISSUE	ORDER	REVISION	ZONE
A	32288	GAGE HEIGHT DIMENSION CHANGED	
2-27-12	JH	AH	

NATIONAL CASTINGS INC.
1400 S. Laramie Ave.
Cicero, Illinois 60650

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NATIONAL SWING MOTION TRUCK
6X11 & 6 1/2 X 12 BOL. 5" END HEIGHT
DAMPING CONTROL WEAR INDICATOR

ORD. 32288	SIZE	DWG. NO. 51555
DR. JH CK'D AH	B	DATE APRIL 30, 1990
APPROVED	SHEET	OF SCALE

ARMAND P. TAILLON

STANDARD CAR TRUCK COMPANY

Standard Car Truck Company.
865 BUSSE HIGHWAY • PARK RIDGE, ILLINOIS 60068
PHONE (708) 692-6050 TELEX 27-0651 FAX (708) 692-6299

September 22, 1994

Mr. R. Groskopf
Chicago Technical Center
3140 South Federal Street
Chicago, Illinois 60616

Send Via: First Class U.S. Mail.

SUBJECT: Truck Performance and Wear Presentation.

Dear Mr. Groskopf:

Enclosed please find a copy of the slides that I presented at the car truck conference in Champaign.

The purpose of these slides is to illustrate the basic recommended practices for determining the wear limits of the Barber S-2 freight car trucks. The slides illustrate the three primary criteria for determining when an S-2 truck is worn to the point where it should be reconditioned. In the first slide, a rendered version of one of our friction castings is shown in green with a recessed area shown in red indicating the wear indicators. These indicators are used in two ways. They are used to determine when a friction casting's column surface has worn out, and to indicate when the friction casting has risen to 3/4" from new condition.

The remaining slides depict the position of the friction casting(s) in new and worn condition respectively. The purpose of these frames is to illustrate the method of using the relationship of the lower edge of the wear indicator with respect to the bolster top surface as a means of gauging wedge rise. They also illustrate the method of using the "mustache gauge" for the purpose of determining when a pair of friction castings on one bolster end have reached an average height of 3/4".

The primary emphasis of the presentation is to illustrate that there is actually one method of determining when a truck is in need of reconditioning. The method illustrated is average wedge rise. When a pair of friction castings on one bolster end has reached an average height of 3/4", we recommend that the truck be reconditioned.

Furthermore, in my presentation I emphasized that Standard Car Truck Company is always available as a resource for information on the best practices of repair and maintenance of the Barber S-2 trucks.

Sincerely,

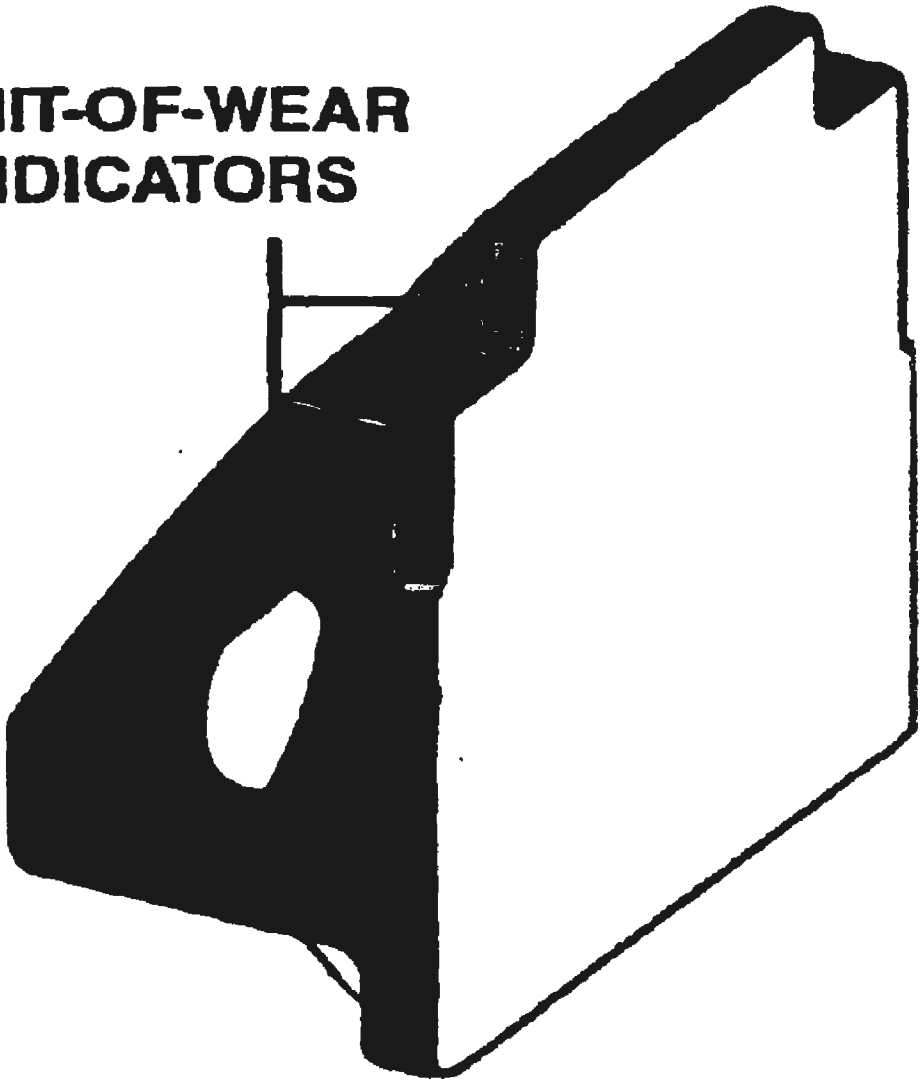
Armand P. Taillon
Senior Project Engineer

APT/wp

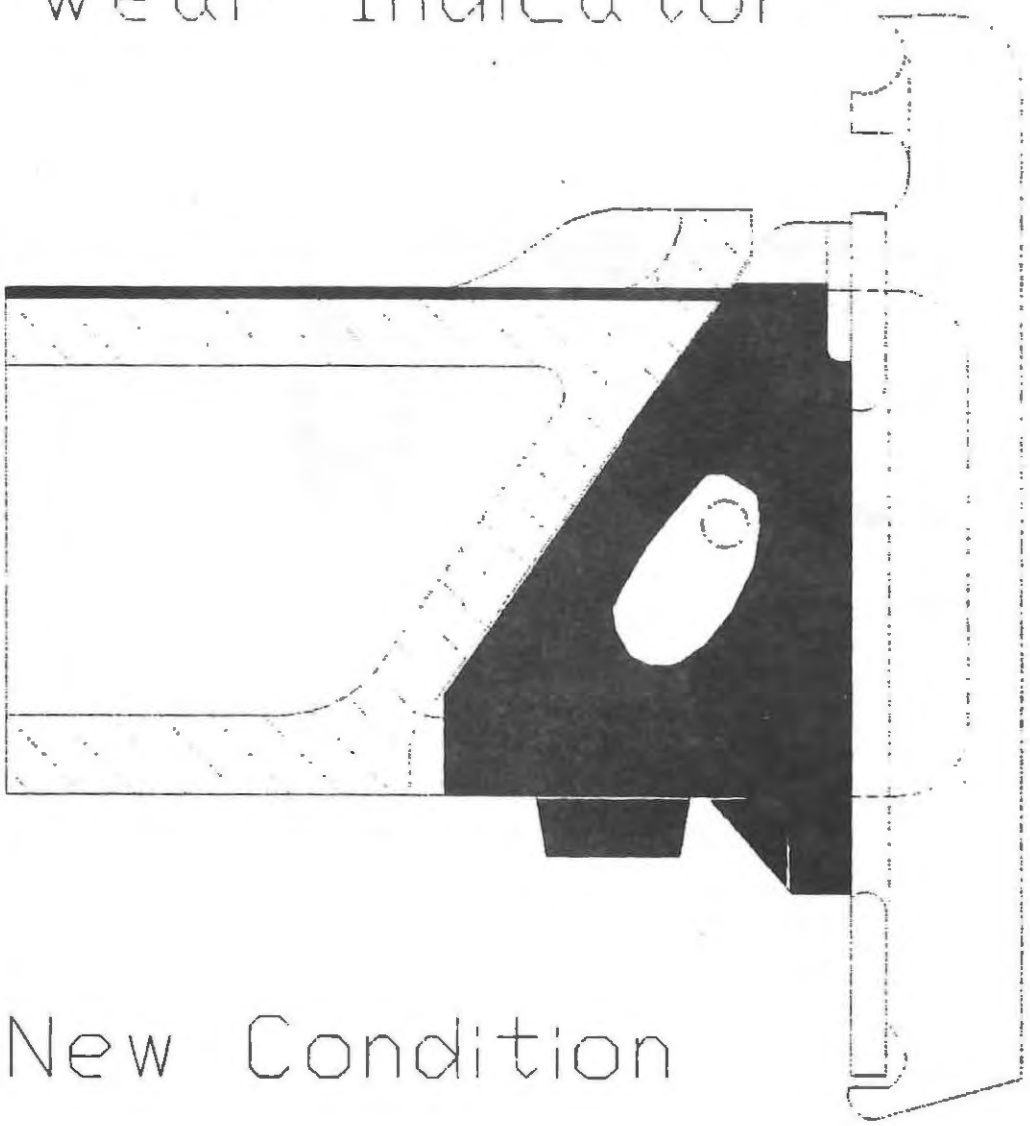
cc: R. L. Bullock

enclosures: Truck Wear Presentation.

LIMIT-OF-WEAR INDICATORS



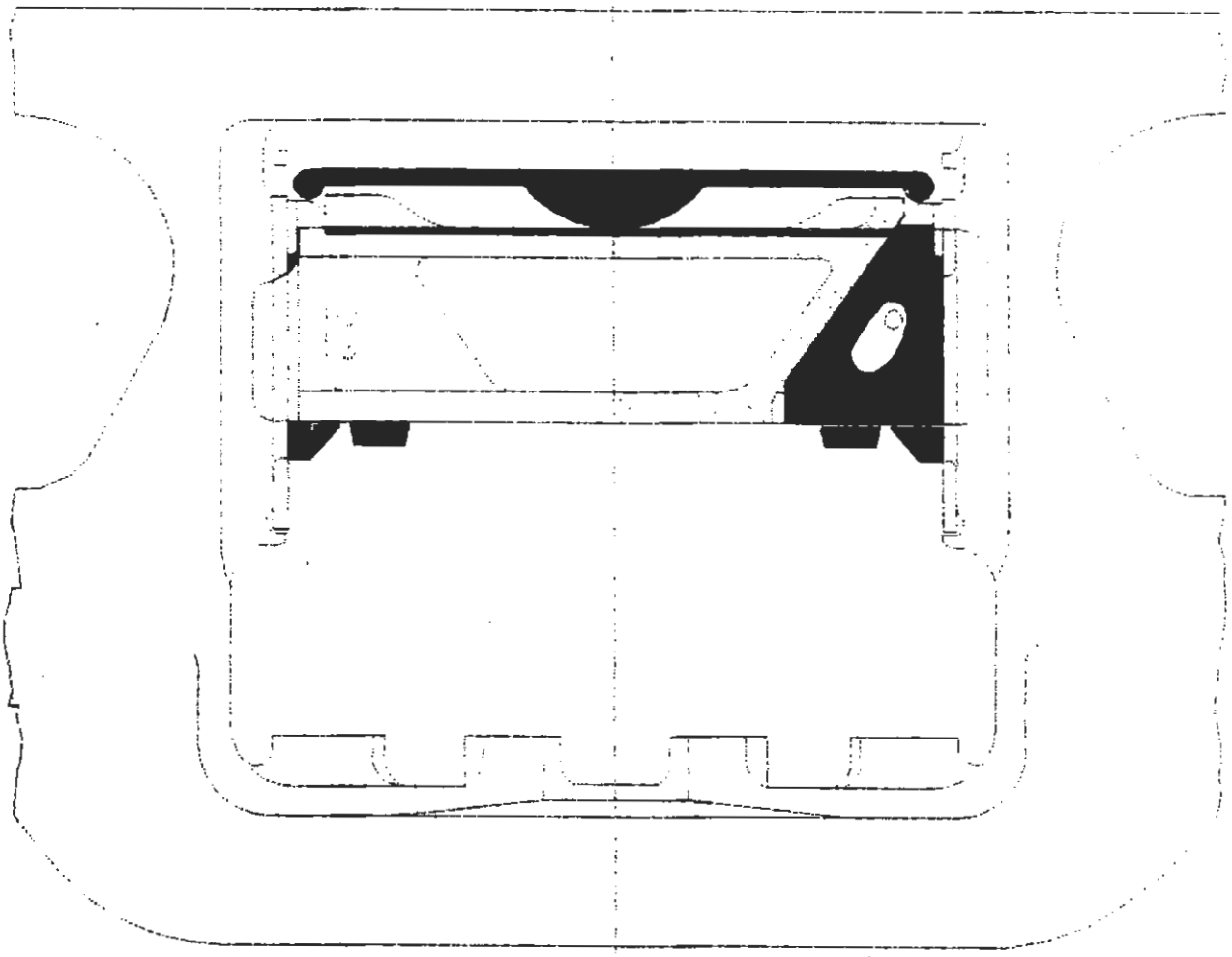
Friction Casting Wear Indicator



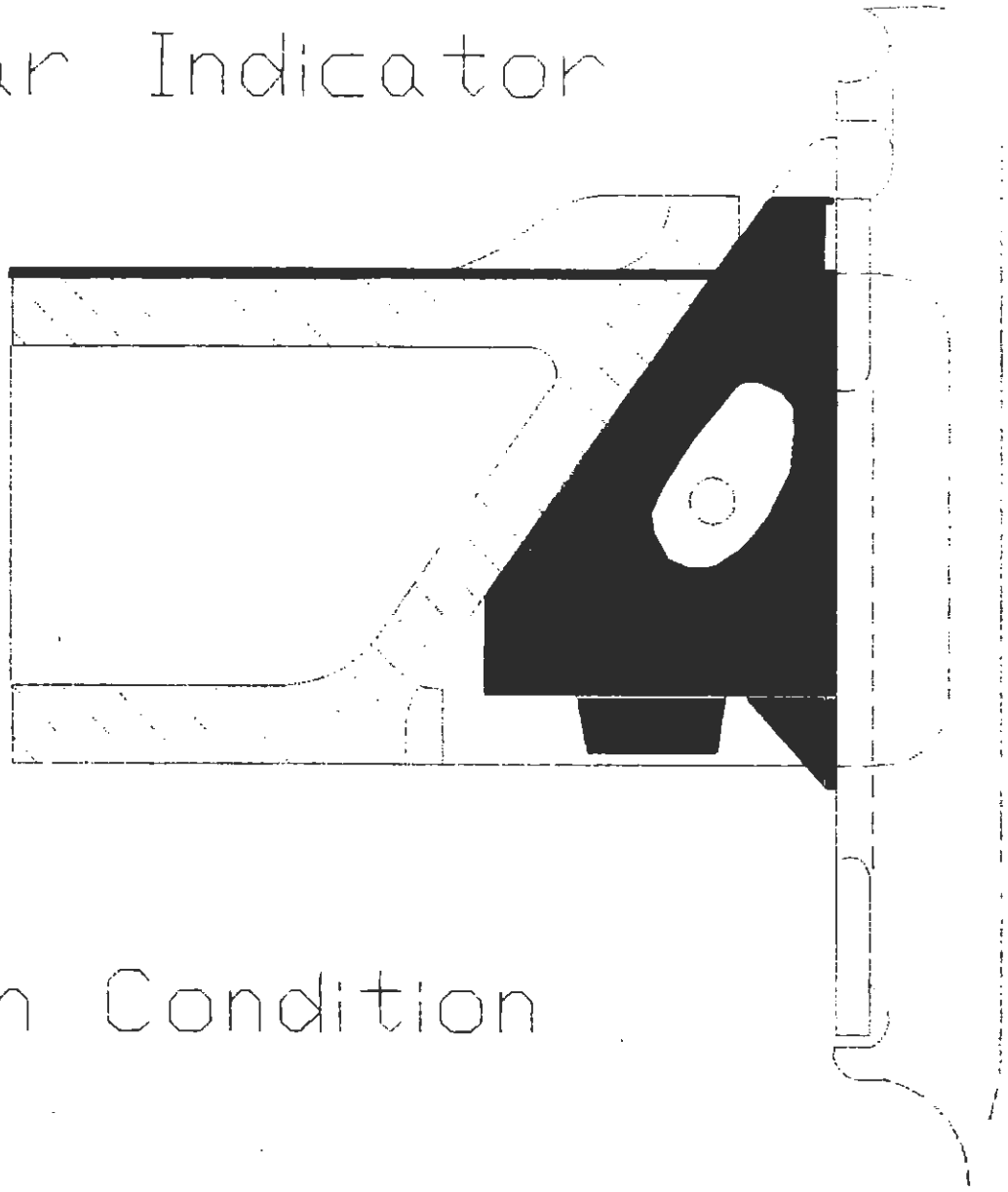
New Condition

Friction Casting Height Gauge

Pass Condition



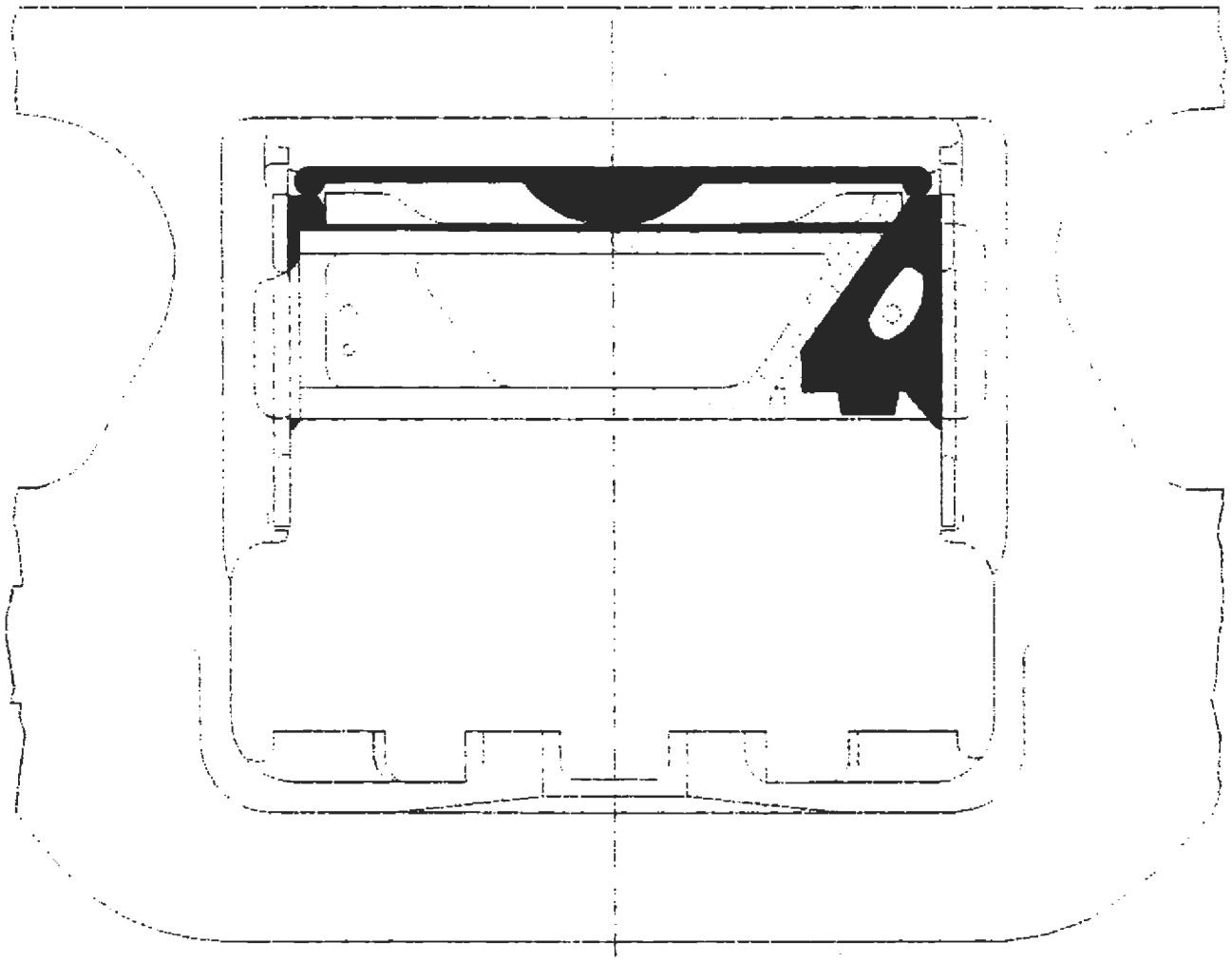
Friction Casting Wear Indicator



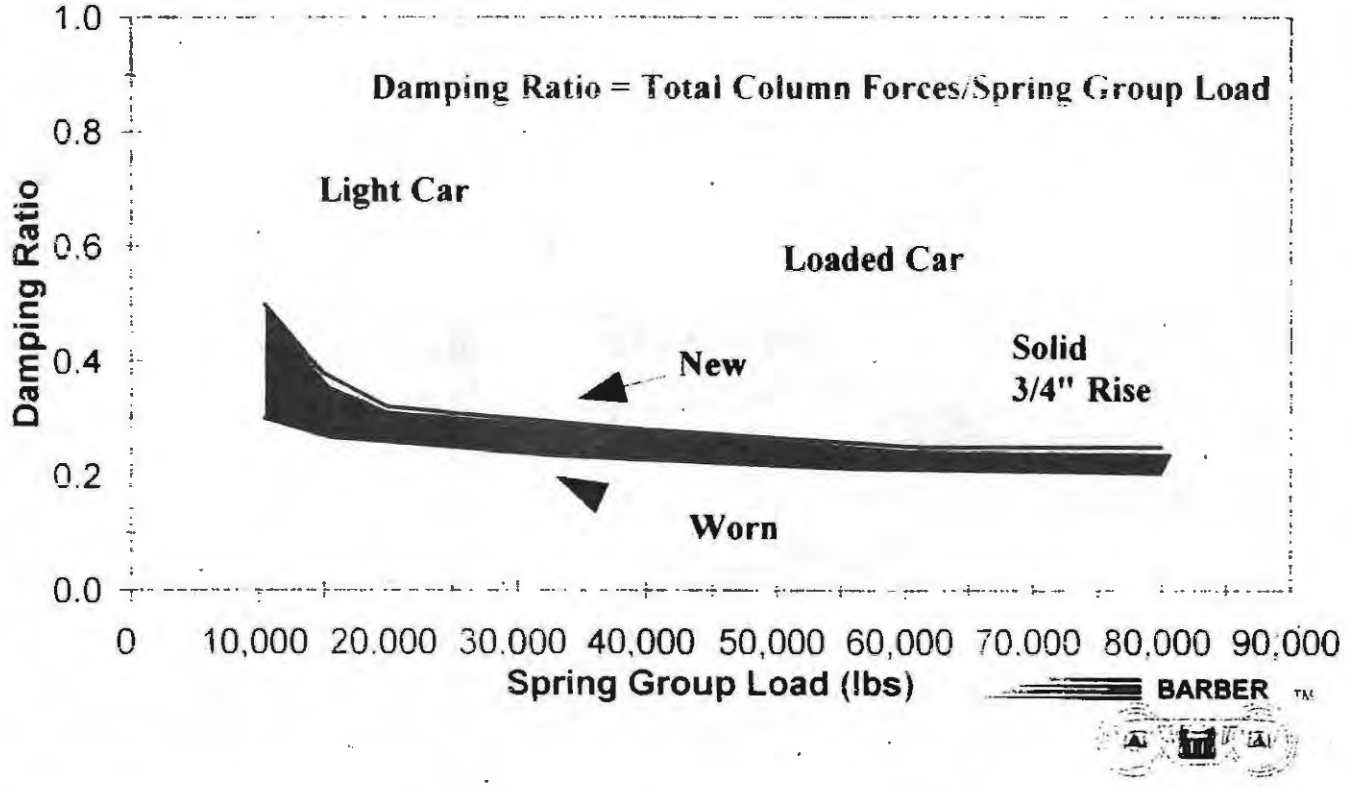
Worn Condition

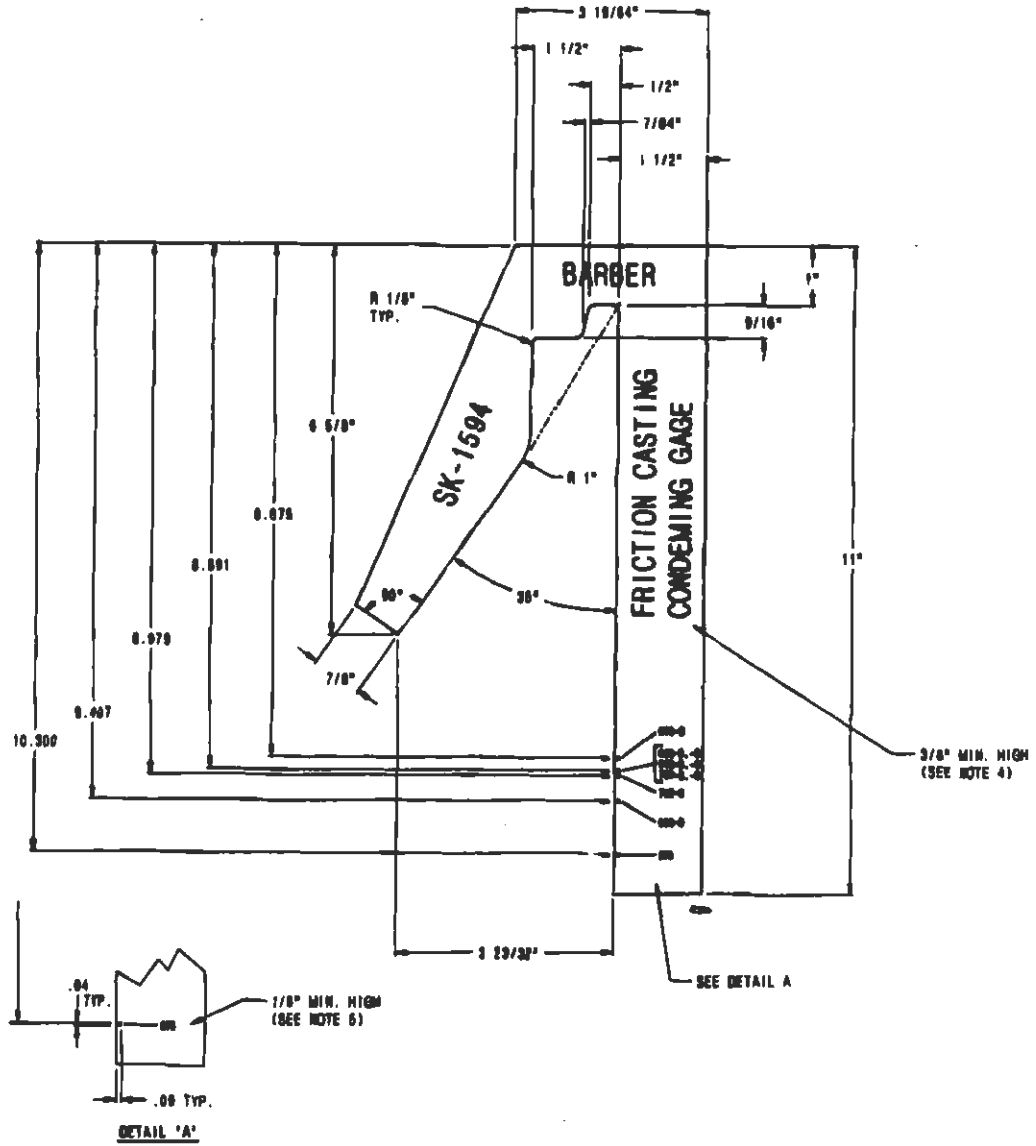
Friction Casting Height Gauge

No Go Condition



Damping Ratio
S-2-C Truck





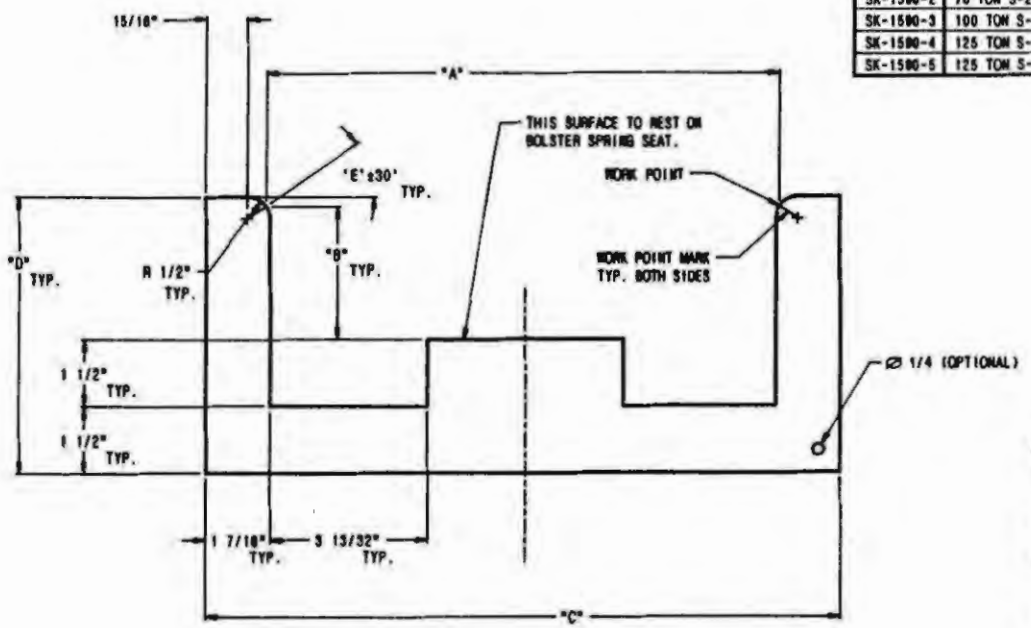
NOTES:

1. MATERIAL: 11 GA. AISI 304 STAINLESS STEEL.
2. FINISH: CLEAN
3. UNLESS OTHERWISE SPECIFIED FINISHED TOLERANCES:
   FRACTIONAL ±1/32
   DECIMAL ±.005
4. ENGRAVE CHARACTERS 3/8" MIN. HIGH x .040 DEEP AS SHOWN.
5. ENGRAVE NUMBERS 1/8" MIN. HIGH x .040 DEEP AND INDICATOR LINES .040 DEEP AS SHOWN.
6. PART TO BE FREE OF BURRS AND SHARP EDGES.

NOTE: THIS IS A CAD DRAWING. MANUAL REVISIONS SHALL NOT BE MADE.

		BARBER FRICTION CASTING CONDEMING GAGE - 36° ANGLE	
		STANDARD CAR TRUCK CO.	
A			
REV	DATE		SK-1594

PART NO.	CAPACITY	DIM. 'A' ± 1/64"	DIM. 'B'	DIM. 'C'	DIM. 'D'	DIM. 'E'
SK-1590-1	70 TON S-2-A,B,&C 100 TON S-2-A,B,&C 100 TON S-2-A LOW PROFILE	11 15/32"	3"	14 5/32"	6 7/32"	35°
SK-1590-2	70 TON S-2-B LOW CONVEYANCE	8 15/32"	3 3/8"	12 5/32"	6 19/32"	35°
SK-1590-3	100 TON S-2-HD	10 7/32"	3 1/2"	12 29/32"	6 23/32"	32°
SK-1590-4	125 TON S-2-A,&C	14 23/32"	3 1/2"	17 13/32"	6 23/32"	35°
SK-1590-5	125 TON S-2-HD	12 15/32"	3 1/2"	15 5/32"	6 23/32"	32°



NOTES:

1. MATERIAL: 11 GA. HIGH STRENGTH LOW ALLOY ABRASION RESISTING STEEL.
2. FINISH: ZINC PLATE .0004/.0006 THICKNESS.
3. UNLESS OTHERWISE SPECIFIED FINISHED TOLERANCES:
   FRACTIONAL ± 1/64
4. STEEL STAMP PART NO., AND CAPACITY ON FLAT SURFACE SHOWN WITH 5/16" MIN. HIGH CHARACTERS.
5. STEEL STAMP WORK POINT MARK 1/2" MIN. ON FLAT SURFACE AS SHOWN.
6. PART TO BE FREE OF BURRS AND SHARP EDGES.

NOTE: THIS IS A CAD DRAWING. MANUAL REVISIONS SHALL NOT BE MADE.

REV.	DATE	CHANGES
A	1/5/93	DIM. 3 15/16" WAS 3 1/2"; DIM. 'C' REDUCED BY 5/16" TYP.

WORK POINT NO-GO GAGE
BARBER S-2 BOLSTERS

PROPERTY OF
STANDARD CAR TRUCK CO.
PARK RIDGE, IL. U.S.A.

DR: M.P.
CH:

SCALE: 1:2

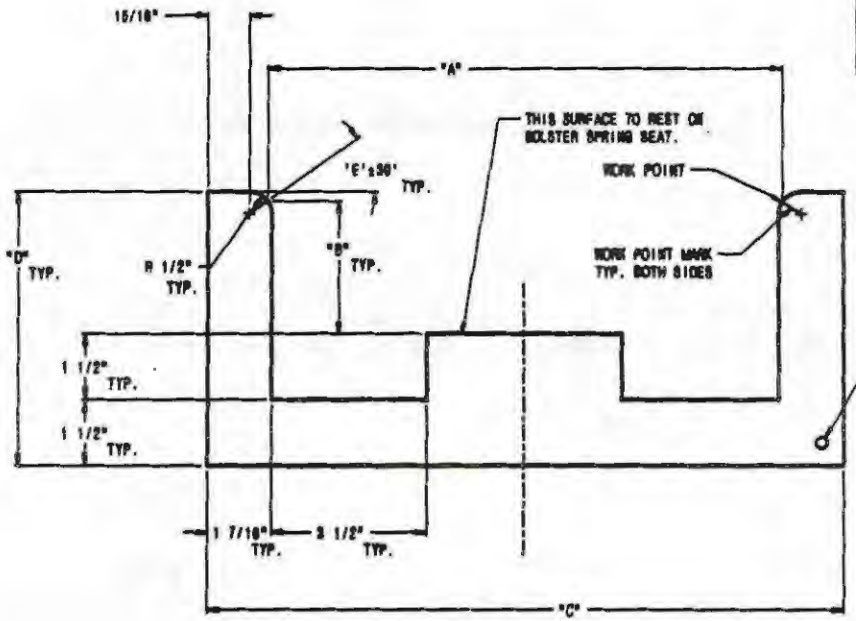
APPROVED BY:

JUNE 8, 1992

SK-1590

SHT. 1 OF 1

PART NO.	CAPACITY	DIM. 'A' +0 -1/64"	DIM. 'B'	DIM. 'C'	DIM. 'D'	DIM. 'E'
SK-1581-1	70 TON S-2-A,B,&C 100 TON S-2-A,B,&C 100 TON S-2-A LOW PROFILE	11 5/8"	3"	14 11/32"	6 7/32"	35°
SK-1581-2	70 TON S-2-B LOW CONVEYANCE	9 5/8"	3 5/8"	12 11/32"	6 19/32"	35°
SK-1581-3	100 TON S-2-HD	10 3/8"	3 1/2"	13 3/32"	6 23/32"	32°
SK-1581-4	125 TON S-2-A,&C	14 7/8"	3 1/2"	17 19/32"	6 23/32"	36°
SK-1581-5	125 TON S-2-HD	12 5/8"	3 1/2"	15 11/32"	6 23/32"	32°



NOTES:

1. MATERIAL: 11 GA. HIGH STRENGTH LOW ALLOY ABRASION RESISTING STEEL.
2. FINISH: ZINC PLATE .0004/.0006 THICKNESS.
3. UNLESS OTHERWISE SPECIFIED FINISHED TOLERANCES: FRACTIONAL ± 1/64
4. STEEL STAMP PART NO.,AND CAPACITY ON FLAT SURFACE SHOWN WITH 5/16" MIN. HIGH CHARACTERS.
5. STEEL STAMP WORK POINT MARK 1/2" MIN. ON FLAT SURFACE AS SHOWN.
6. PART TO BE FREE OF BURRS AND SHARP EDGES.

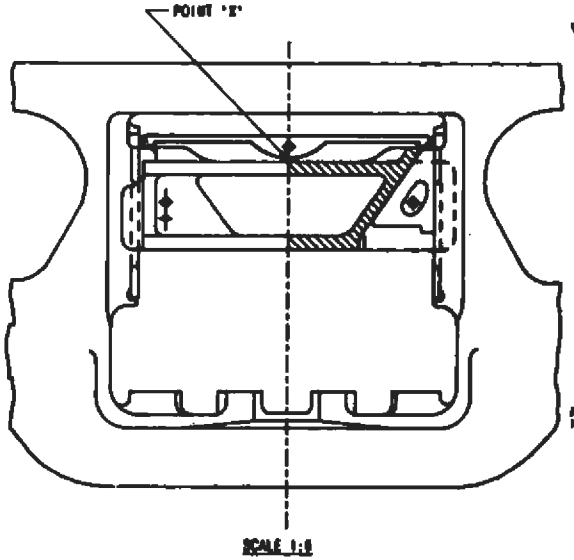
NOTE: THIS IS A CAD DRAWING. MANUAL REVISIONS SHALL NOT BE MADE.

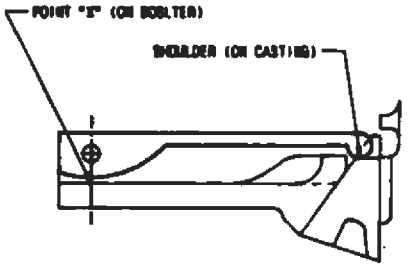
REV.	DATE	CHANGES
A	5-5-02	CHANGED DIM."A"& DISCRIPTION

WORK POINT GO GAGE
BARBER S-2 BOLSTERS

PROPERTY OF
STANDARD CAR TRUCK CO.
PARK RIDGE, IL. U.S.A.

DR:M.P.
CH:
SCALE: 1:2
APPROVED
MAY.14.2002
SK-1581
SHT. 1 OF 1





APPLICATION PROCEDURE

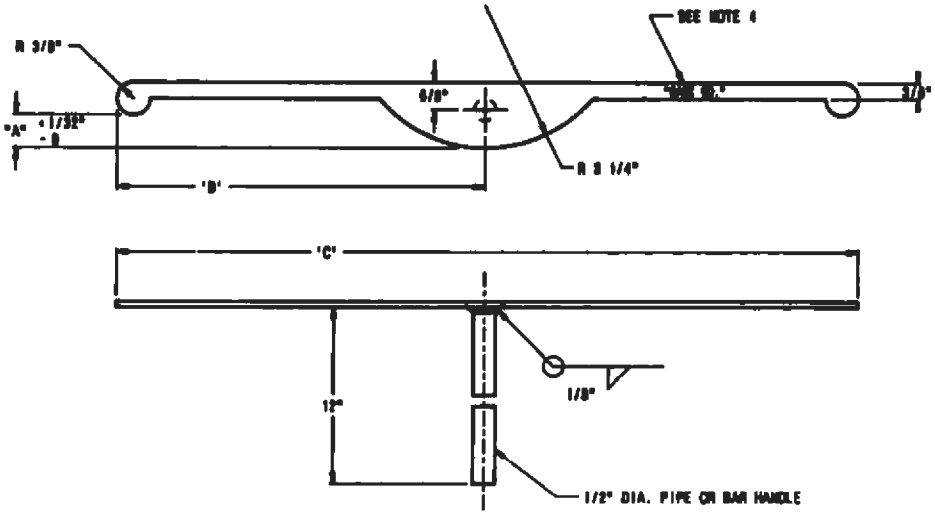
IF GAGE DOES NOT CONTACT BOLSTER AT POINT "X" WHILE RESTING UPON BOTH FRICTION CASTINGS SHOULDERS REPAIR IS INDICATED.

GAGE NO.	BEARING SIZE	A.A.R. ① SPRING TRAVEL	BARBER FRICTION CASTING PATT. NO.	DIM. "A"	DIM. "B"	DIM. "C"
SK-1540-1	6" x 11"	D-3	669-D	3/4"	8 7/16"	16 7/8"
	6" x 11"	D-4 or D-5	670-C, 670-B, ②787-C			
	6 1/2" x 12"	D-3	669-D			
	6 1/2" x 12"	D-5	670			
	6 1/2" x 12" ③	D-5	834-C			
	6 1/2" x 12"	D-7	670			
SK-1540-2	6 1/2" x 12"	D-4 or D-5	670-C, 670-B, ②787-C	1/2"		
SK-1540-3	6" x 11"	D-4	670-C	1 1/4"	7 3/16"	14 3/8"
SK-1540-4	7" x 12" ③	D-5 or D-6	834-C	1/2"	9 5/16"	19 1/8"
SK-1540-5	7" x 12"	D-3 or D-4	787-C	1/4"	10 3/32"	20 3/16"
SK-1540-6	7" x 12"	D-5	787-C	3/4"		

① STANDARD A.A.R. SPRING GROUPS FOR BARBER S-2-A, S-2-B, S-2-C, S-2-D & S-2-HD TRUCKS.

② EXTENDED TOE FRICTION CASTINGS FOR CANADA.

③ FOR S-2-HD



NOTE:

1. MATERIAL: 11 GA. AISI 304 STAINLESS STEEL.
2. FINISH: CLEAN
3. UNLESS OTHERWISE SPECIFIED TOLERANCE TO BE ± 1/32"
4. STEEL STAMP GAGE NO. ON FLAT SURFACE SHOWN WITH 1/4" MIN. HIGH CHARACTERS.
5. PART TO BE FREE OF BURRS AND SHARP EDGES.

NOTE: THIS IS A CAD DRAWING. MANUAL REVISIONS SHALL NOT BE MADE.

REV.	DATE	CHANGE
G	6-21-99	REVISED NOTES 1 & 2; CHANGED TABLE
F	3-30-98	CHANGED SK-1540-3 "B" & "C" DIM'S
E	9-24-97	ADDED SK-1540-4 GAGE; CHANGED NOTE 4;
		ADDED AND REVISED SCALE INFO.
D	10-09-96	REVISED AND REDRAWN BY CAD

STABILIZER WEAR GAGE

FOR STANDARD BARBER S-2-A, S-2-B, S-2-C, S-2-D & S-2-HD

PROPERTY OF
STANDARD CAR TRUCK CO.
PARK RIDGE, IL, U.S.A

DRAWN: MEP	QUAL ENGR:	OCT. 22, 1991
CHKR: AMP	APPD:	SK-1540
PROJ ENGR:	SCALE: 1:2	SHT. 1 OF 1

BRAD JOHNSTONE

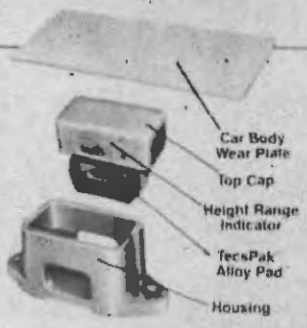
MINER ENTERPRISES INC.



1893-1993

MINER

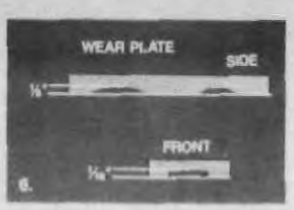
INSTALLATION OF STANDARD TCC SIDE BEARINGS ON CONVENTIONAL CARS



Miner's side bearings feature a TecsPak™ Alloy elastomer pad. The patented design includes a metal top cap that faces the car body wear plate. The large cap helps protect the TecsPak Alloy elastomer pad from abrasive wear and shear loads. It also provides a stable coefficient of friction. The results are reduced truck hunting and less wear on rail car parts for a longer performance life.

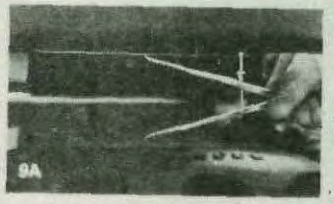
1. Position car on level track.
2. Check wear plate and bolster mounting surfaces for parallel. Must be within ⅛" end to end and 1/16" across width of plate. Check with caliper or gage.
3. Raise car. Remove old Side Bearings from bolster.
4. Check bolster top mounting surface for flat. Grind off high spots and bumps.
5. Inspect body side bearing wear plate. Mounting bolts must be tight, flush with plate surface.
6. Inspect wear plate for roller impressions by placing straight edge on wear plate. If depth of wear is more than 1/16", replace body wear plate.





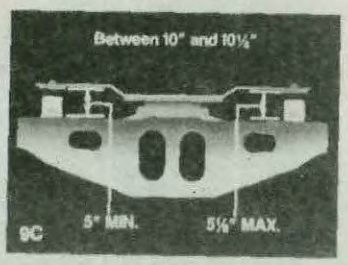
1893 - 1993

MINER









7. Clean truck bolster bowl. Solid bowl liner (plastic or steel) increases side bearing height readings— **Must be** in bowl for measurement and adjustment.
8. Paste or grease lube won't change set-up heights, can be added now. Graphite waters change set-up heights temporarily. Add after final adjustments.
9. Lower car. Measure (9A) and adjust side bearing height as needed by adding or subtracting shims between car body and wear plate to 5 1/16" ± 1/16" (9B). Sum of side bearing pairs must be between 10" and 10 1/8" (9C).



11.



10. Raise car. Bolt side bearing housings to bolster top. **Do not install shims under side bearing housing.** Use $^{7}/_{8}$"-9 Grade 5 hex bolts with self locking nuts, or equivalent. Torque (DRY) 375 to 425 ft.-lbs.

11. Install top caps into housings. Make sure slot(s) in top cap fit in housing. Lower car.

NOTE: Never expose TecsPak Alloy block to temperatures more than 200°F. Do not expose to temperatures above 175°F for more than a few minutes. If housings have been heated, do not install the top cap subassembly until housing is cool enough to touch.

1894 · 1994

MINER

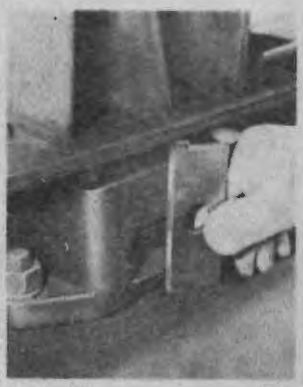
SHOP CHECKING FREE HEIGHT

Miner Gage No. 28740 is for quick check of top cap assembly free height when car is in shop.



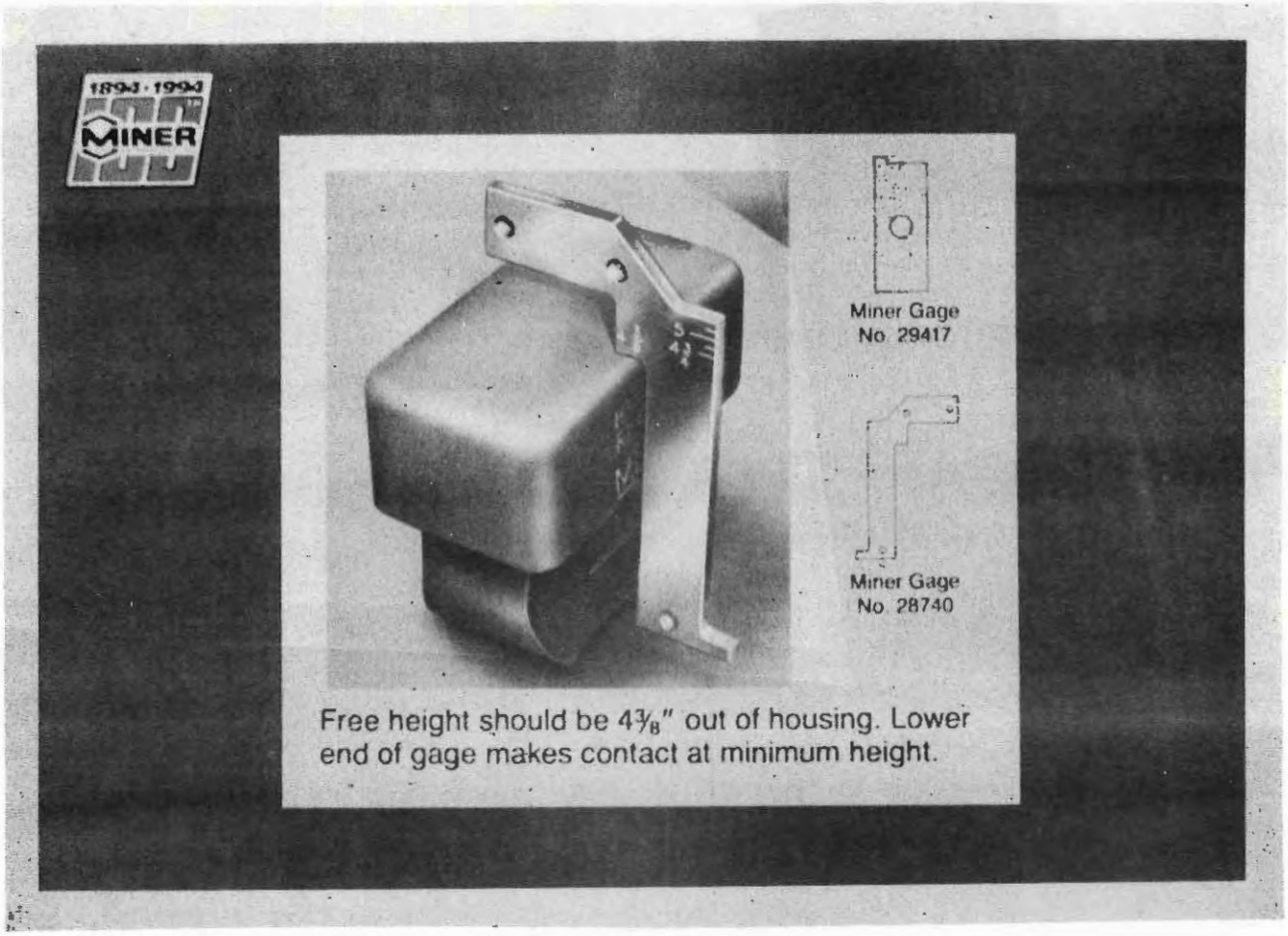
Free height should be $5\frac{3}{16}$" minimum in housing. Lower end of gage makes contact at minimum height.

FIELD CHECKING





Use Miner Gage No. 29417 for field checking. If long side (5⅛") of gage just fits, no adjustment is needed. If it won't fit, try short side (4⅞"). If short side doesn't fit, adjustment is also needed. Measurements greater than long side or shorter than short side require attention.


1893 · 1993
MINER
Miner Gage
No. 29417
Miner Gage
No. 28740
Free height should be 4⅜" out of housing. Lower end of gage makes contact at minimum height.

MILLION MILE TRUCK

C. N. SCOTT

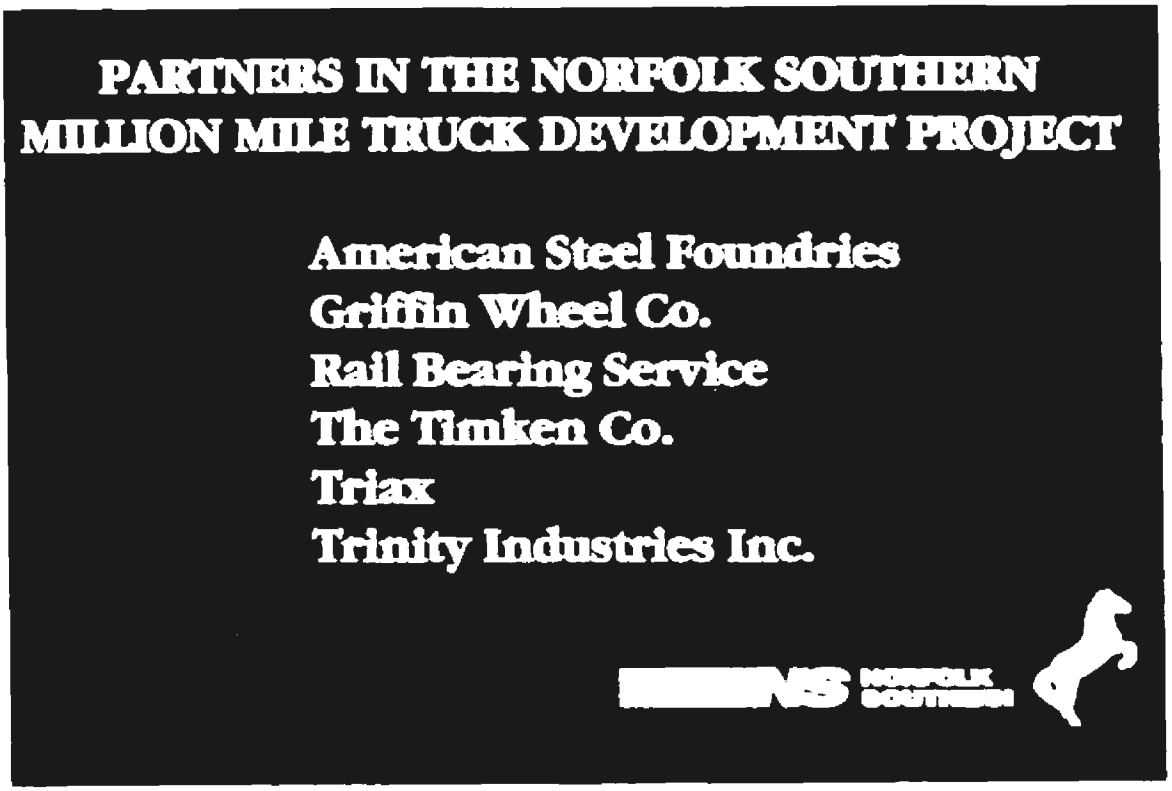
NORFOLK SOUTHERN

AAR Technology Forum

"Million Mile Truck"

By: C. N. Scott
Norfolk Southern Corporation

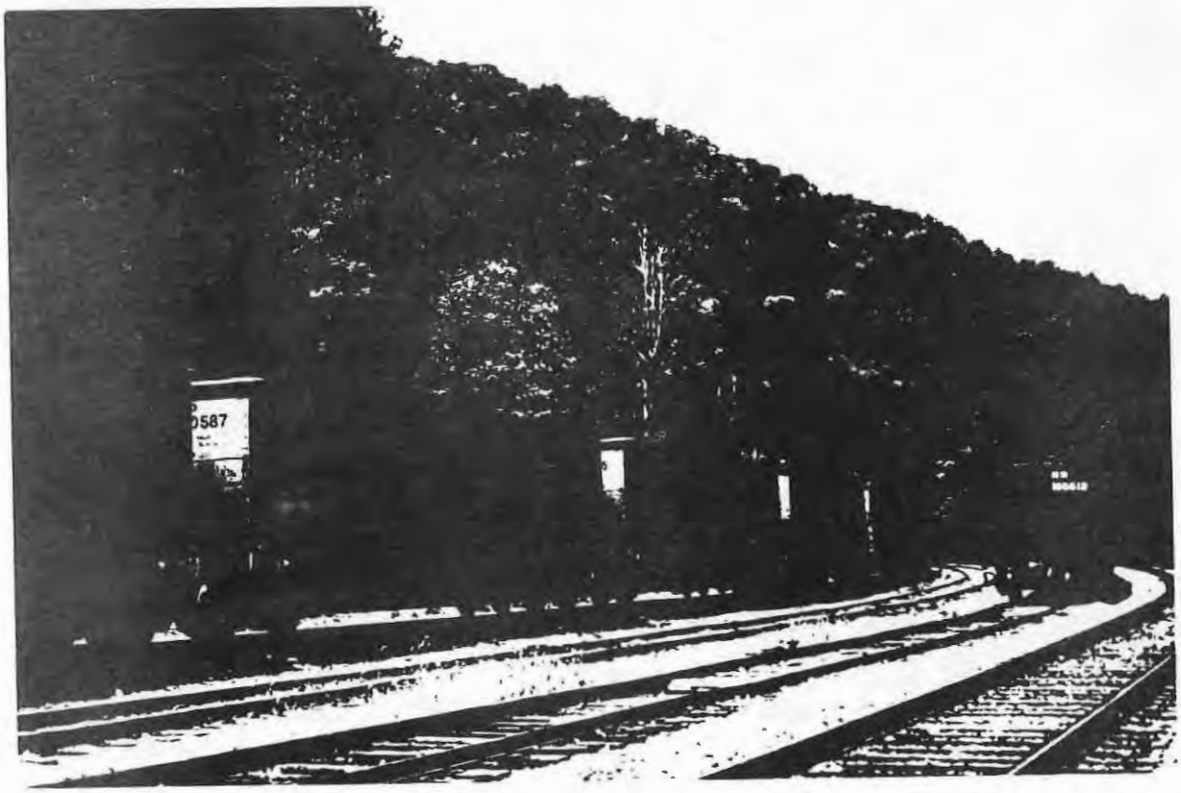




MILLION MILE TRUCK

In 1991 we realized that the high costs of truck maintenance including out-of-service time for cars was dictating a new and different technology for freight car trucks. We studied the repair records and found that wheels and roller bearings were the components causing the highest maintenance costs.

We formed a unique industry team with the goal of increasing truck maintenance intervals four fold to one million miles. We chose one of our unit coal trains for a Phase 1 test of improved wheels, brake beams, and brake shoes.



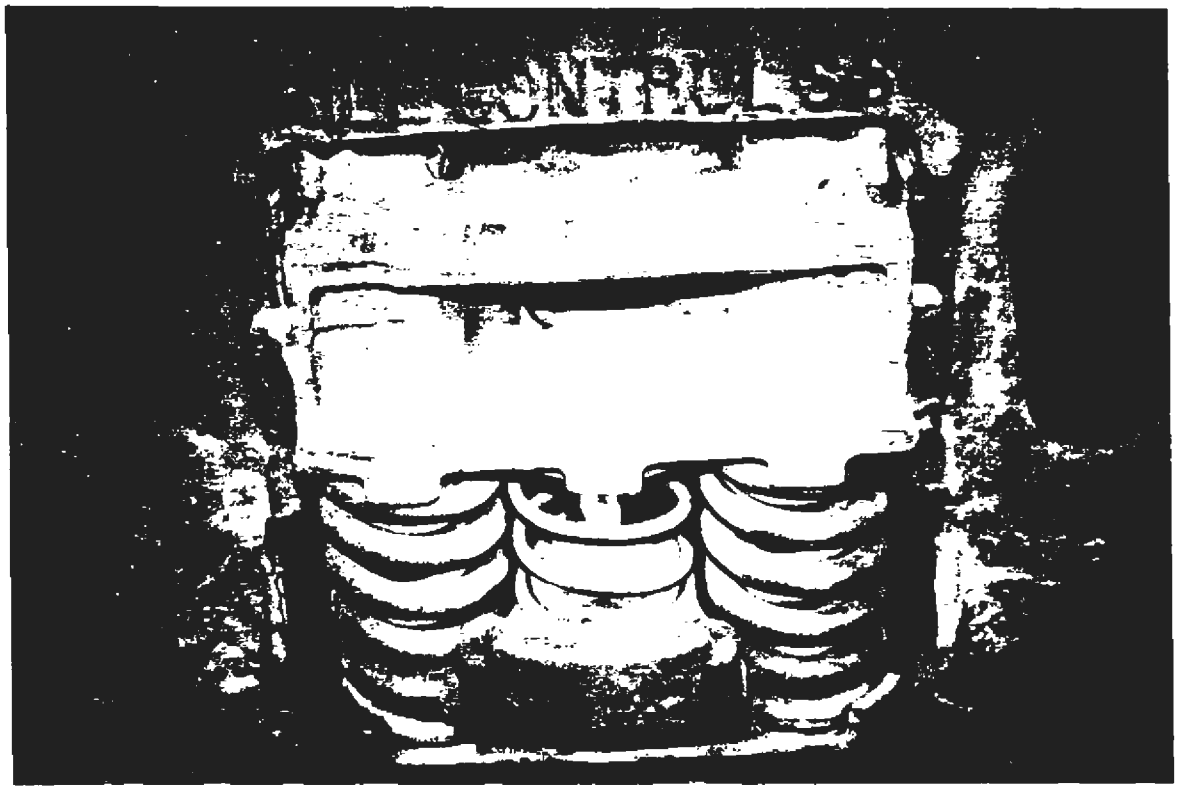
Unit coal train containing Phase I tests of harder wheels, redesigned brake beams, and Flange Saver brake shoes. There were 120 test cars and 50 control cars.



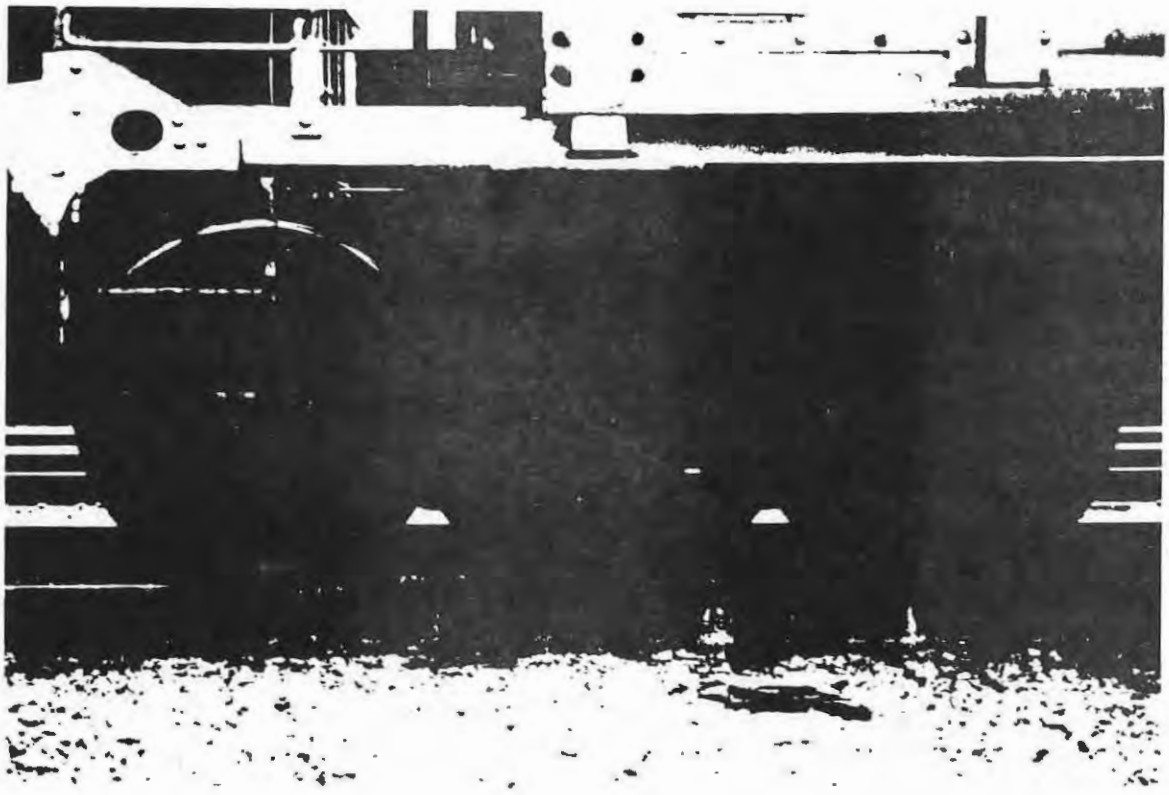
Aluminum Bethgon Coalporter used for Phase I testing. Car was stencilled with test information.



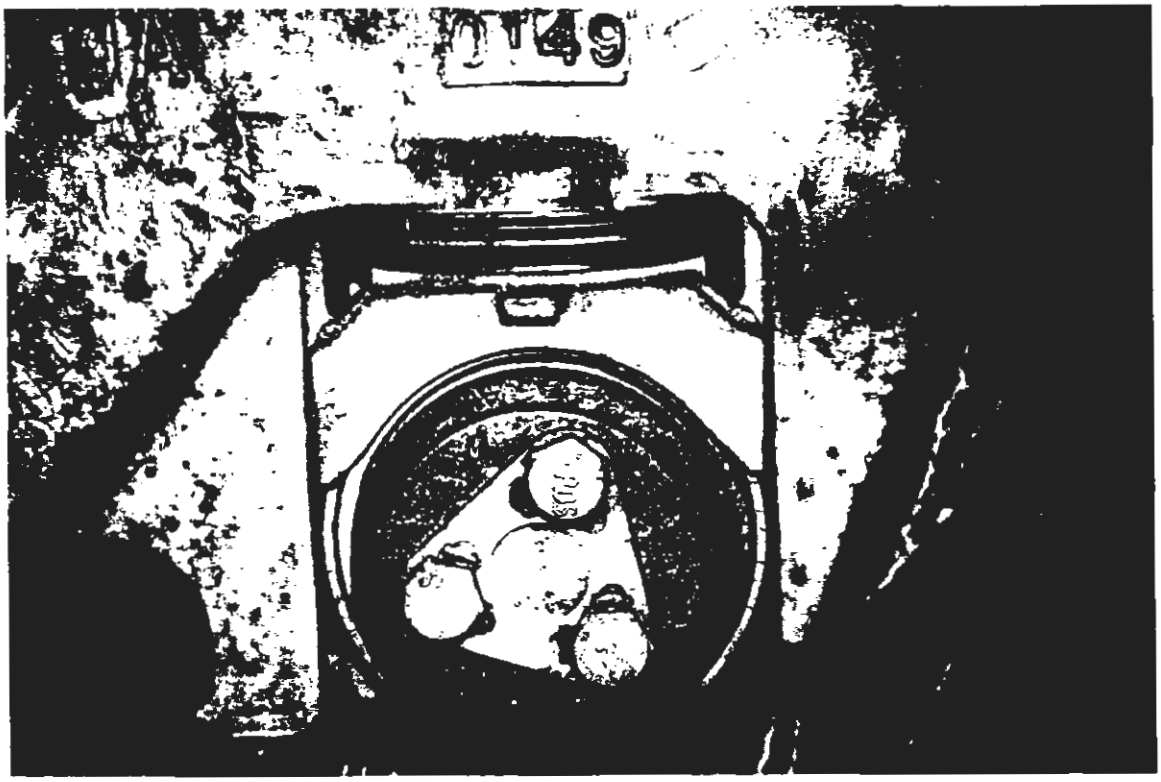
Phase I test of ASF Super Service truck.



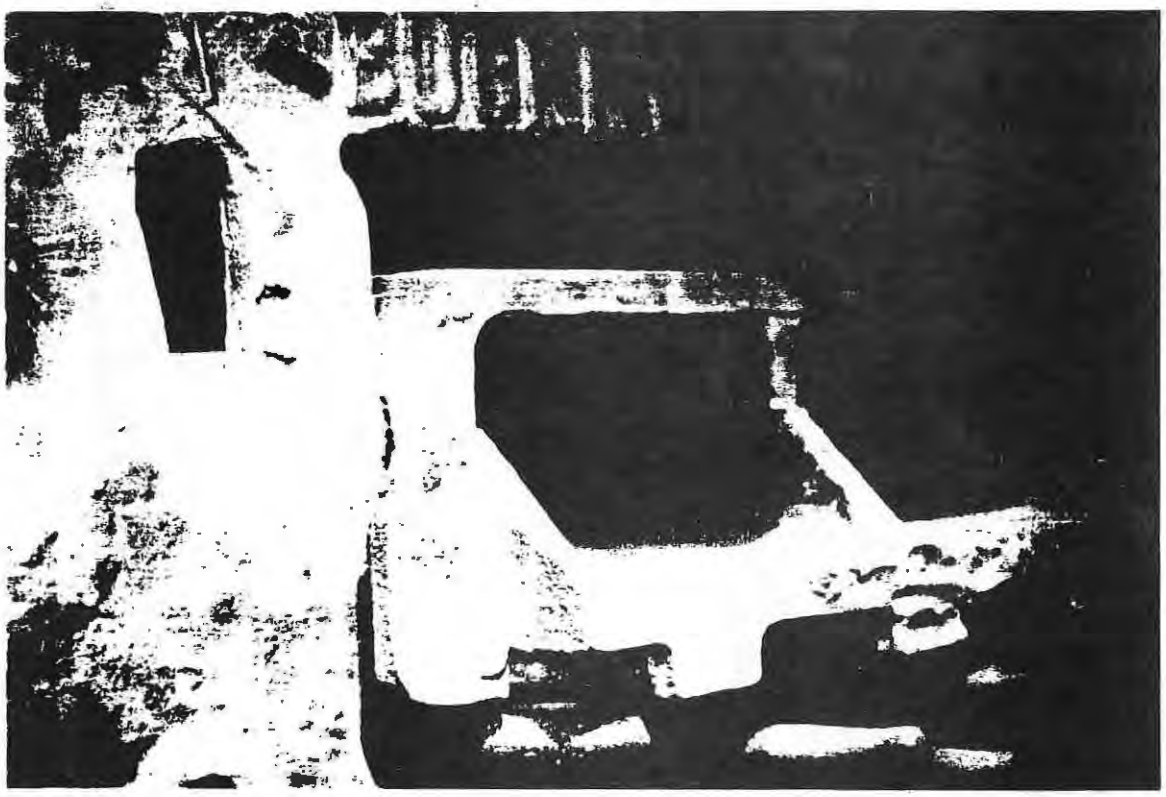
ASF Super Service truck with Stucki "million mile +" package.



ASF Roadmaster truck.



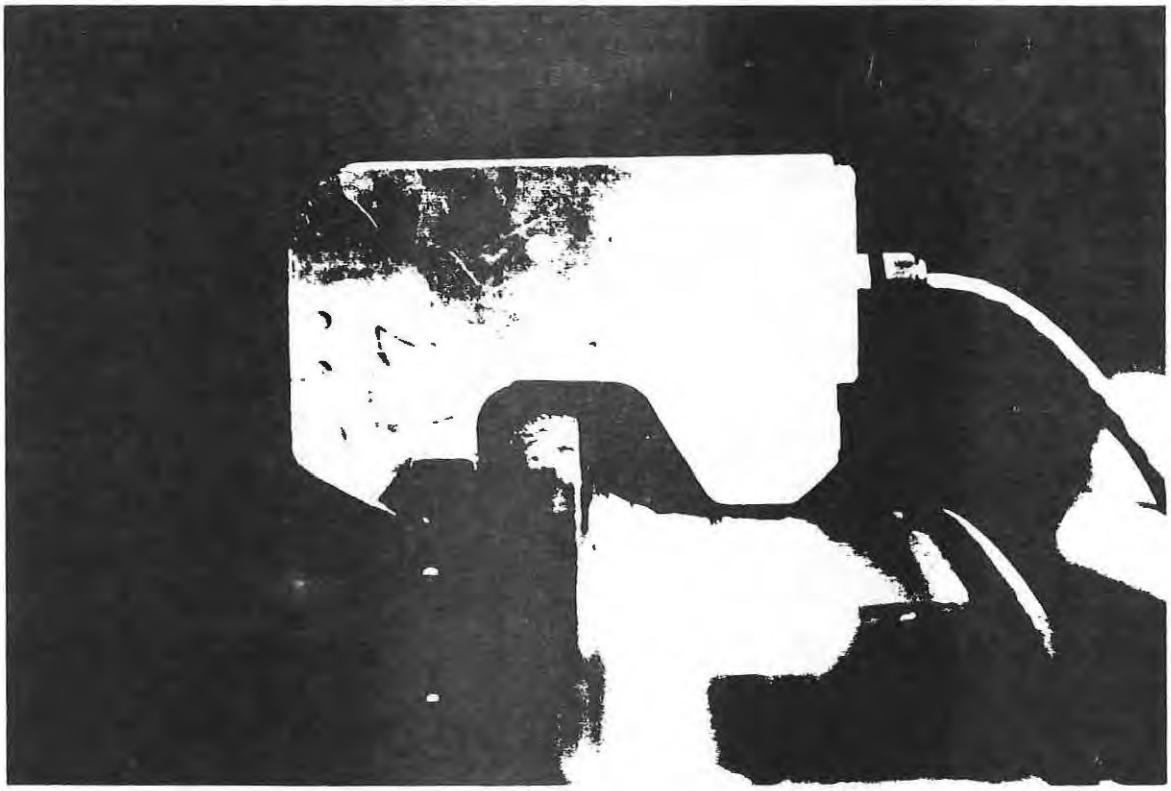
ASF Roadmaster truck showing special adapter.



Buckeye XC-R truck.



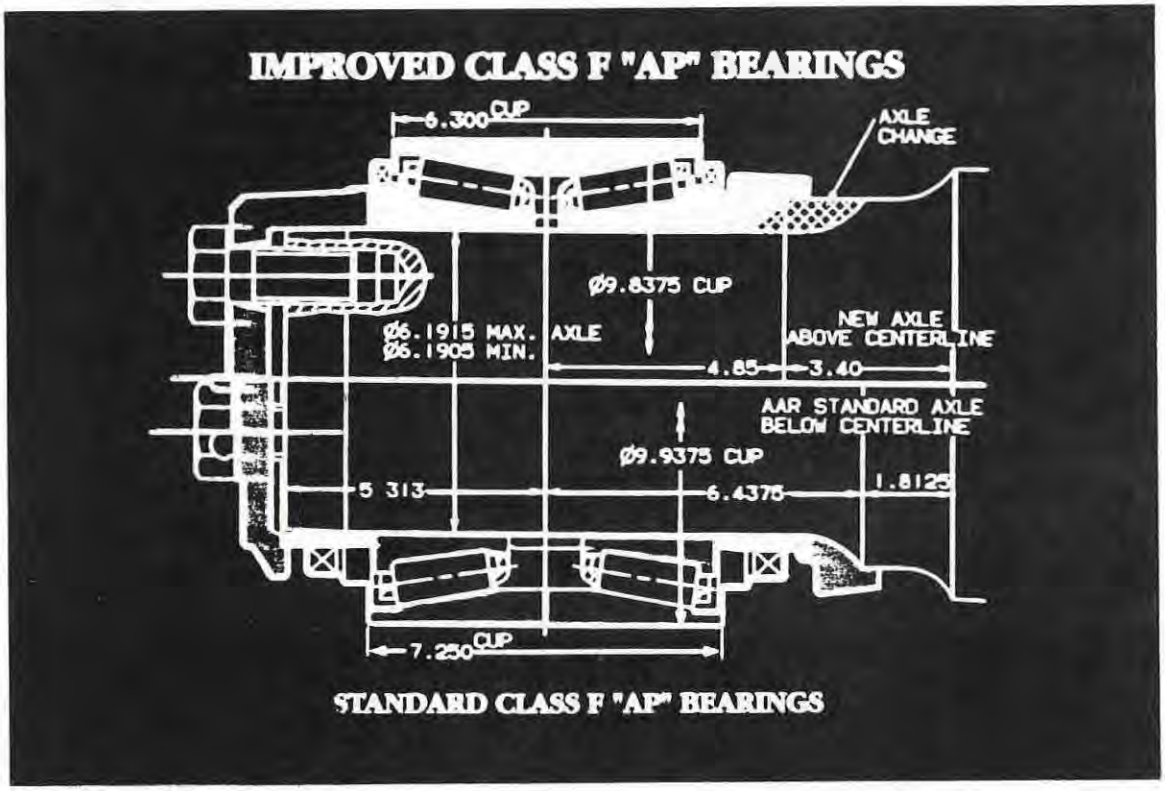
Slide 10 - The tread worn hollow wheel wear problem causes wheels to be removed prematurely as well as contributing to rail roll over derailments. We are installing a wayside detector which will allow long term, real time monitoring of rail forces as a function of wheel wear. Wheels will be trued at intervals more conservative than current AAR wear limits in order to study the effects on rail forces and economics based on possible increased wheel life.



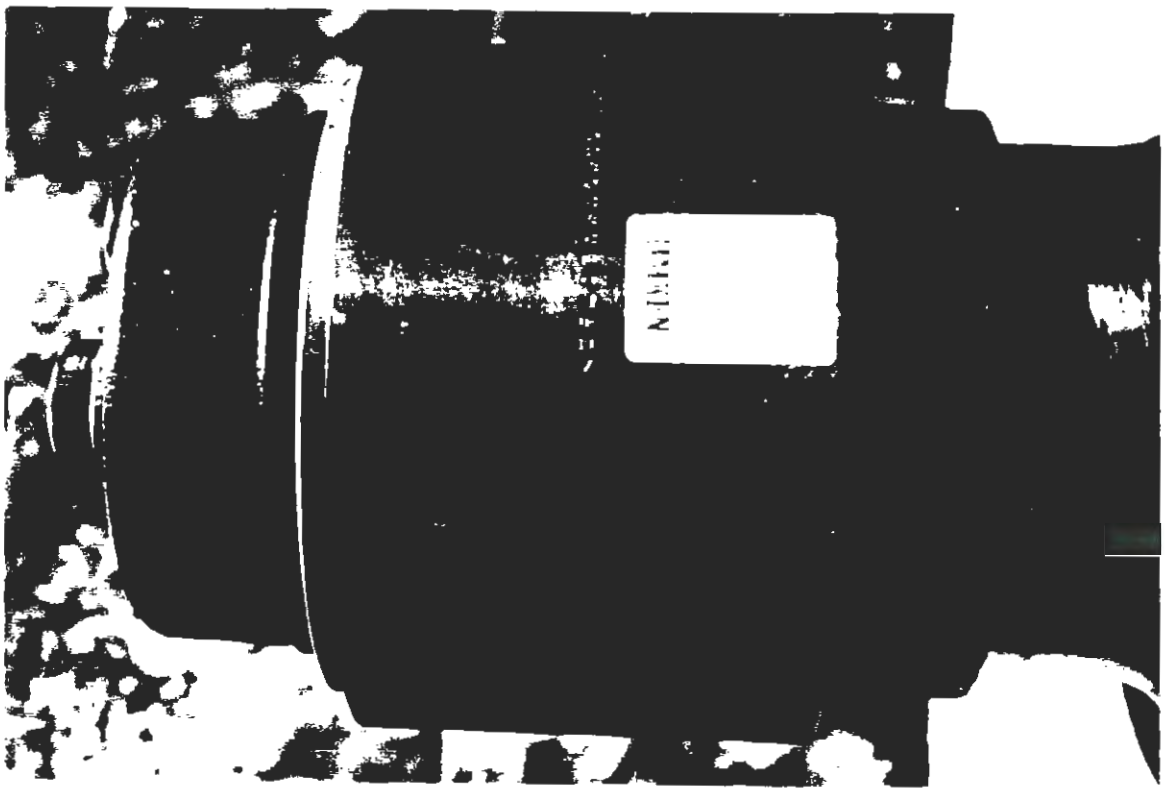
IEM electronic wheel gage used to make precise wheel measurements throughout the entire project.



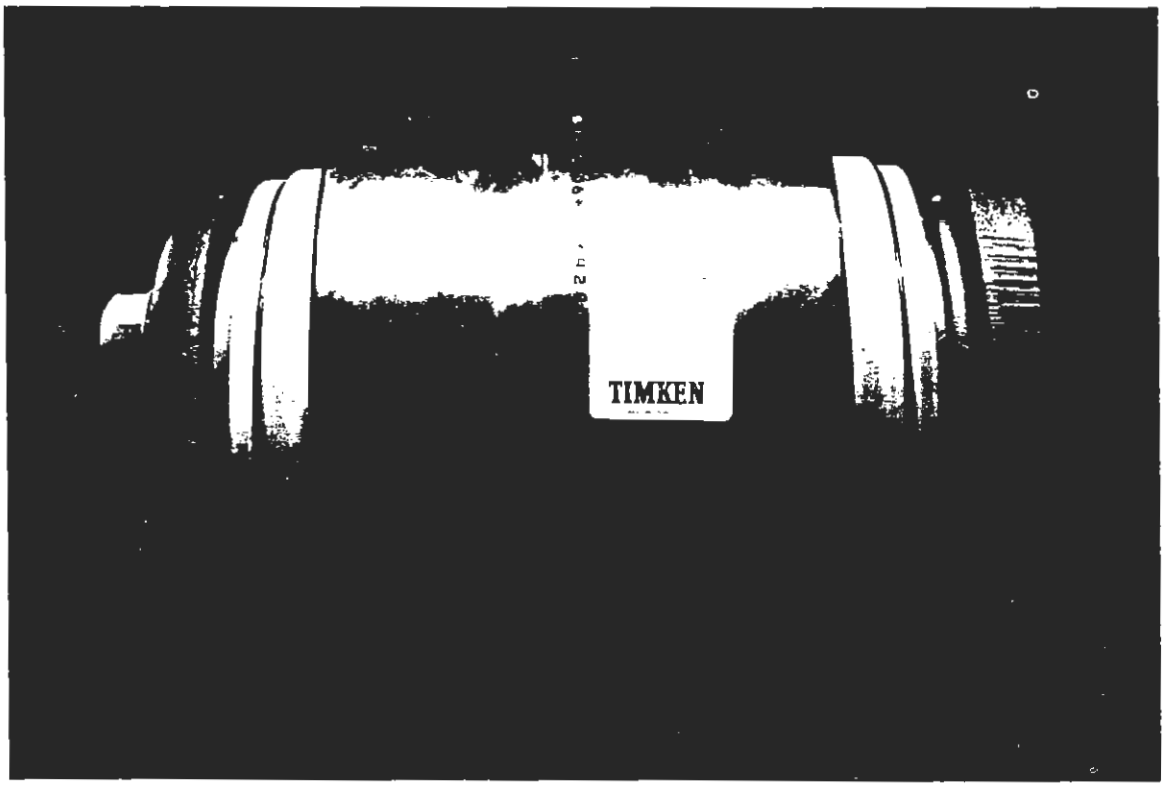
Operator using the IEM wheel gage.



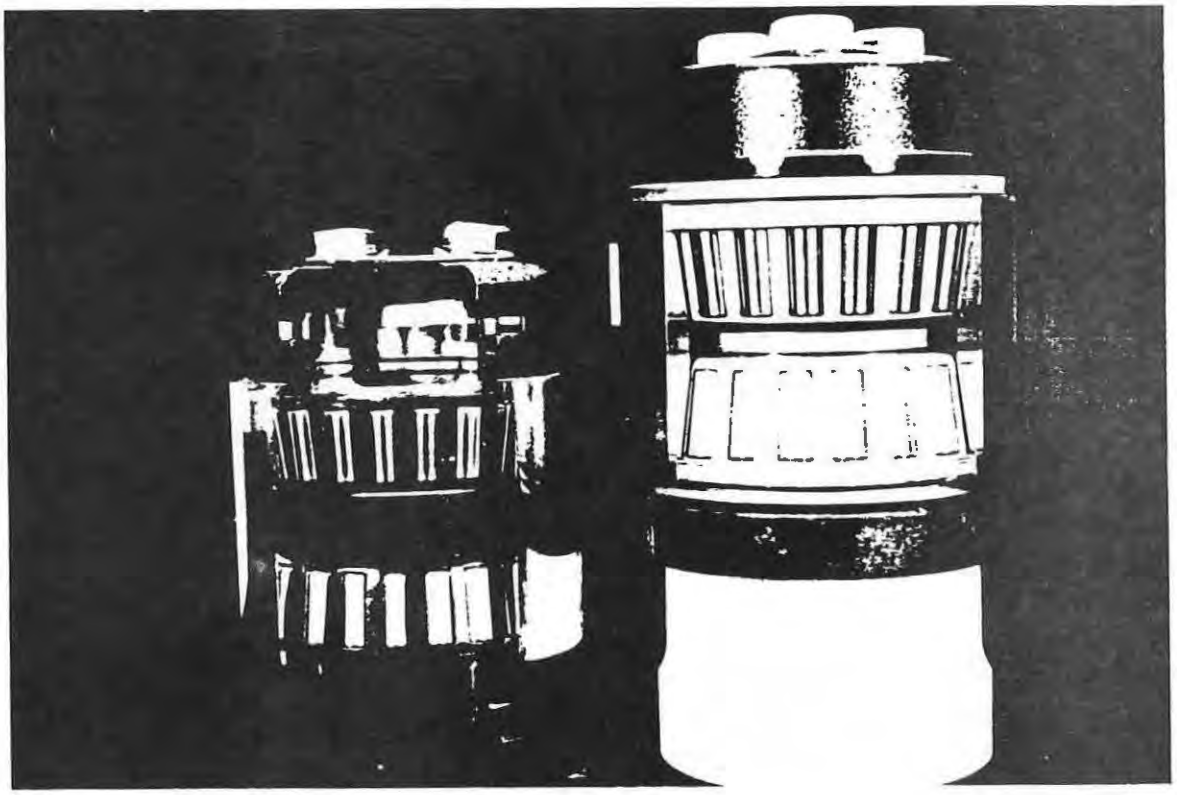
Slide 13 - Bearing manufacturers have always complained that the roller bearing design was less than optimum because they were told to fit the design within the dimensions of a journal box. We challenged them to design an optimal million mile bearing without constraints. One of our partners agreed to make whatever changes might be necessary to the side frame in order to accommodate the bearing. Another partner agreed to make any required axle redesign. The result was a roller bearing which promises to greatly reduce axle flexing, the major cause of bearing failure.



New AP 2 bearing.



Comparison standard AP 1 bearing.



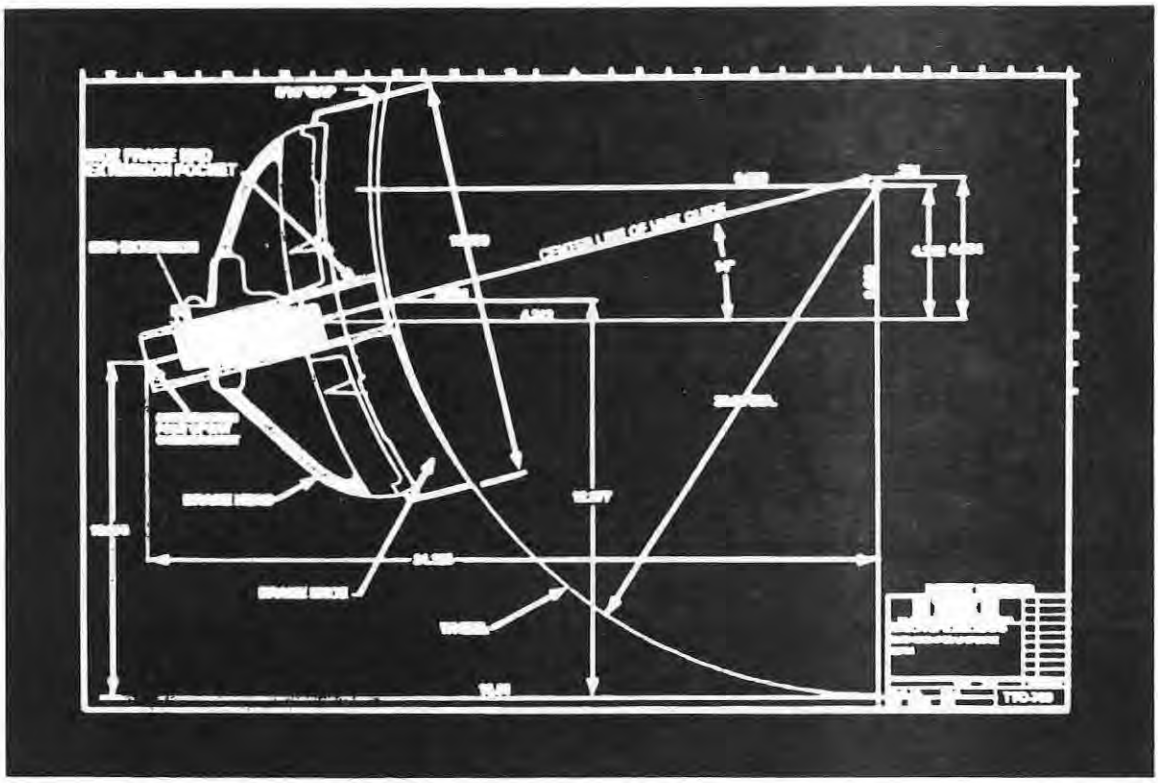
Cut away view of AP 1 (left) compared to the new AP 2 (right). Shortened bearing can be seen by comparing end caps and spacers.



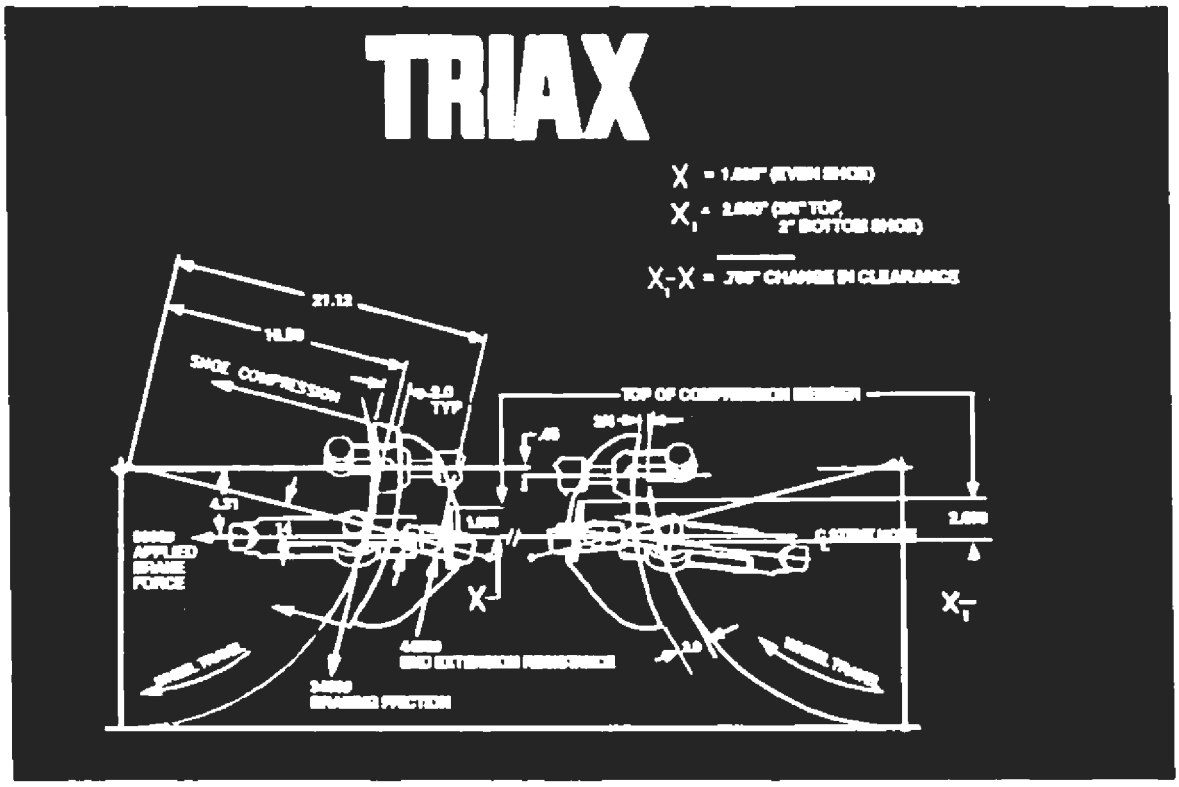
New design axle furnished by Allegheny Axle to accommodate new bearing.



Standard axle for AP 1 bearings.



A spin off to the project is the Triax redesigned brake beam. While we never set the goal of a million mile brake shoe, we were aware that most brake shoes wear on one end leaving plenty of brake shoe material on the other end when the shoe is scraped. Bob Jackson of Triax was challenged by the prospect that we would make changes to the side frame to solve the wear problem through a redesigned brake beam. His beam applies the load through the axle centerline and prototype testing has already shown that we are getting uniform brake shoe wear across the entire length of the shoe.

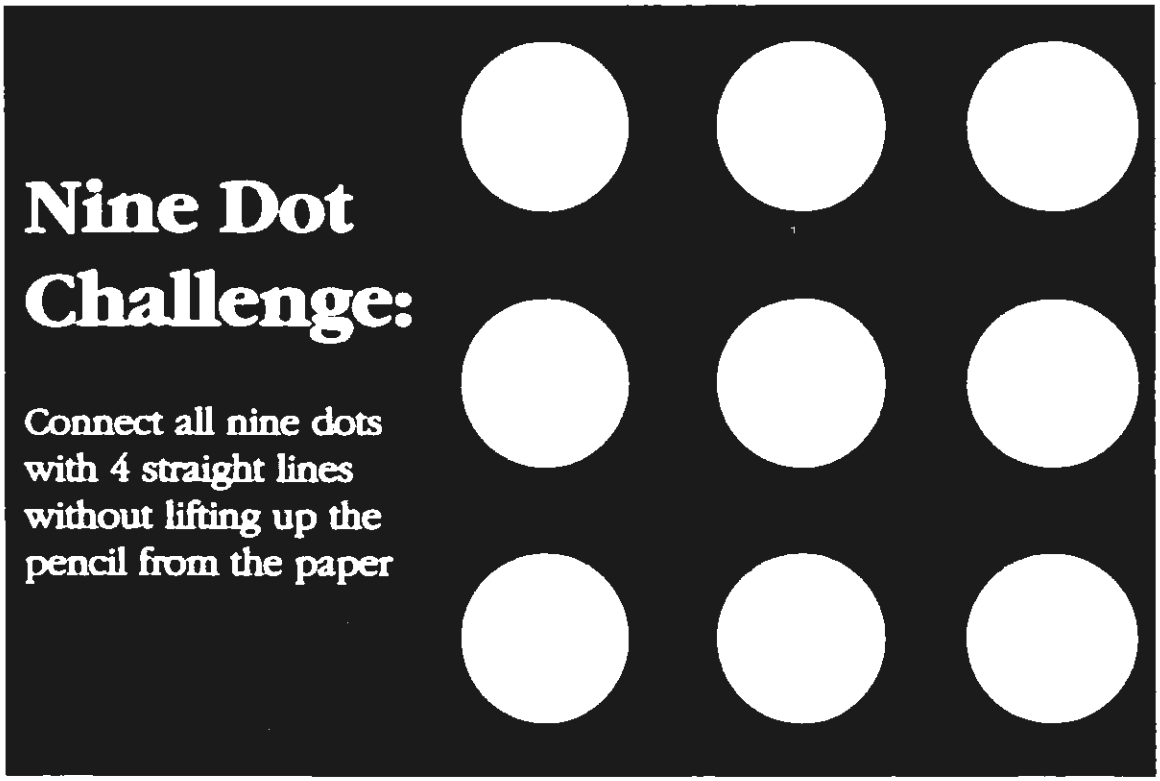


Triax diagonally corrected end extension brake beam vs. conventional AAR beam. We forgot to tell Bob Jackson that the brake beam design cannot be changed because "it's always been that way".

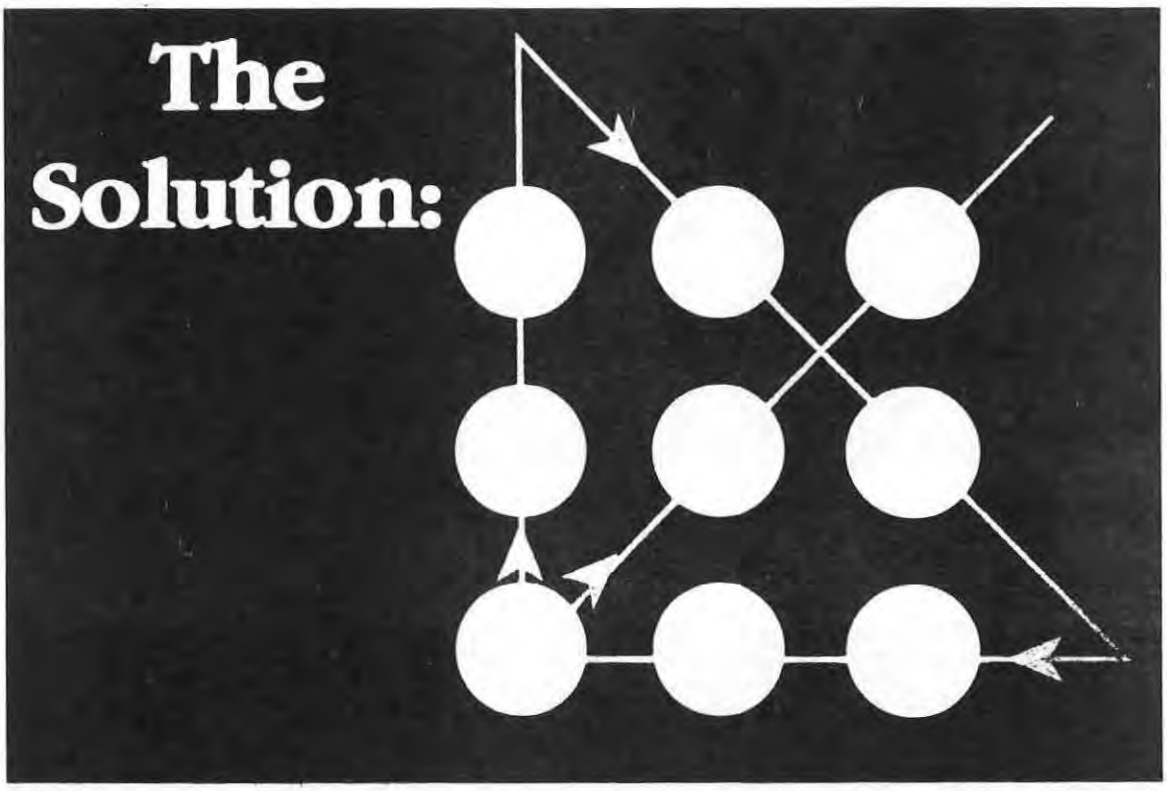


"Self aligning, spring plankless, double truss, 3 piece truck"

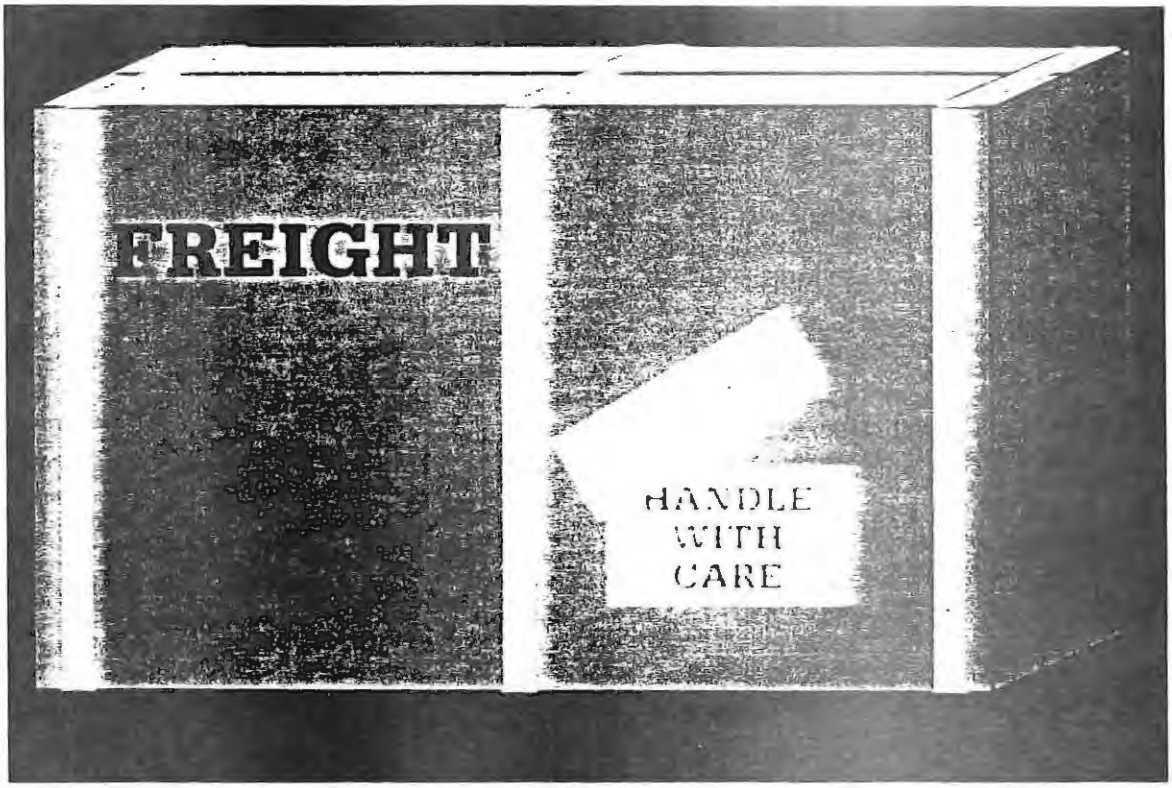
We will now change gears and talk about the truck of the future. It is obvious that the 3 piece truck developed by Ross Winans in 1831 has outlived its usefulness. If we are going to satisfy our customers and remain competitive in the future, we must remove our paradigms and find a better way. Surely there are some modern day Ross Winans' here today. The challenge is to inspire their innovative talents. The most brilliant engineers in the business haven't been able to significantly improve on Winans' 3 piece truck in 150 years. We have hung "jewelry" on it to make incremental improvements, but there have been no dramatic redesigns. The time for that dramatic improvement is NOW.



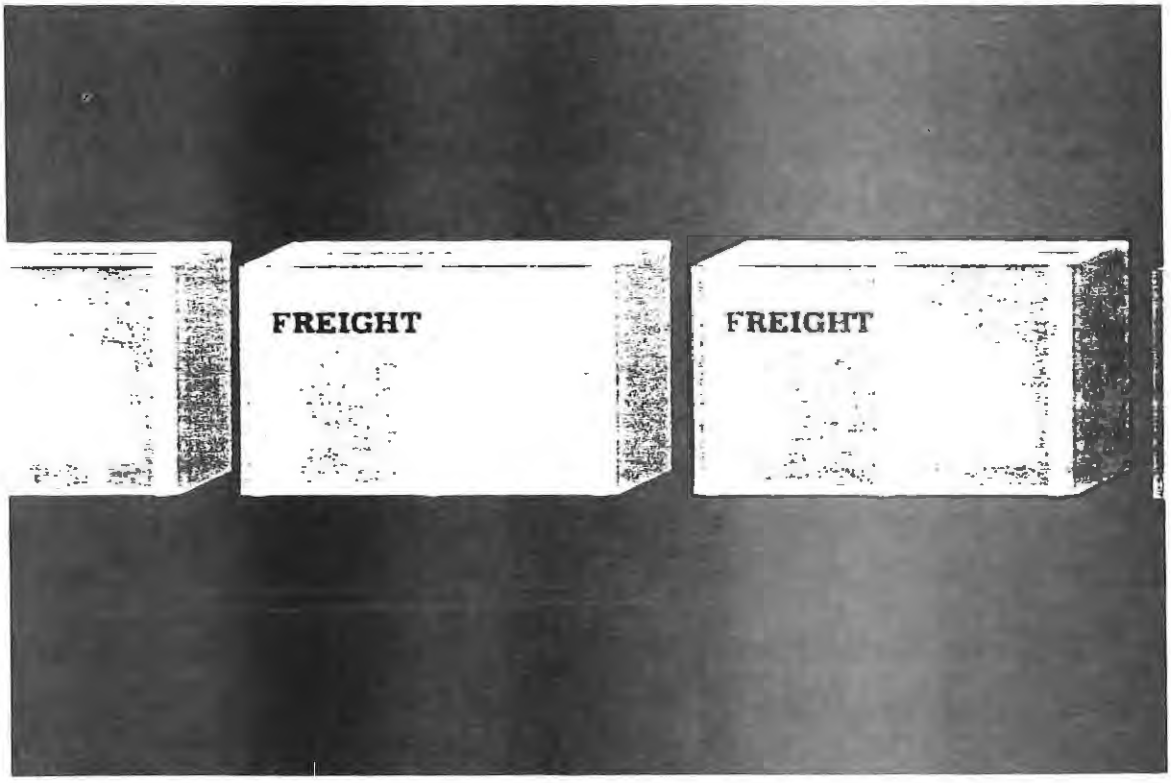
Clear out the paradigms. Remove the blinders. Forget the 3 piece truck. When you have done these things, you will be ready to become innovative. As a test, I will see if you can free your mind. Most of you have probably seen this puzzle. If you have, you know that it cannot be solved if you keep your mind closed (in a box).



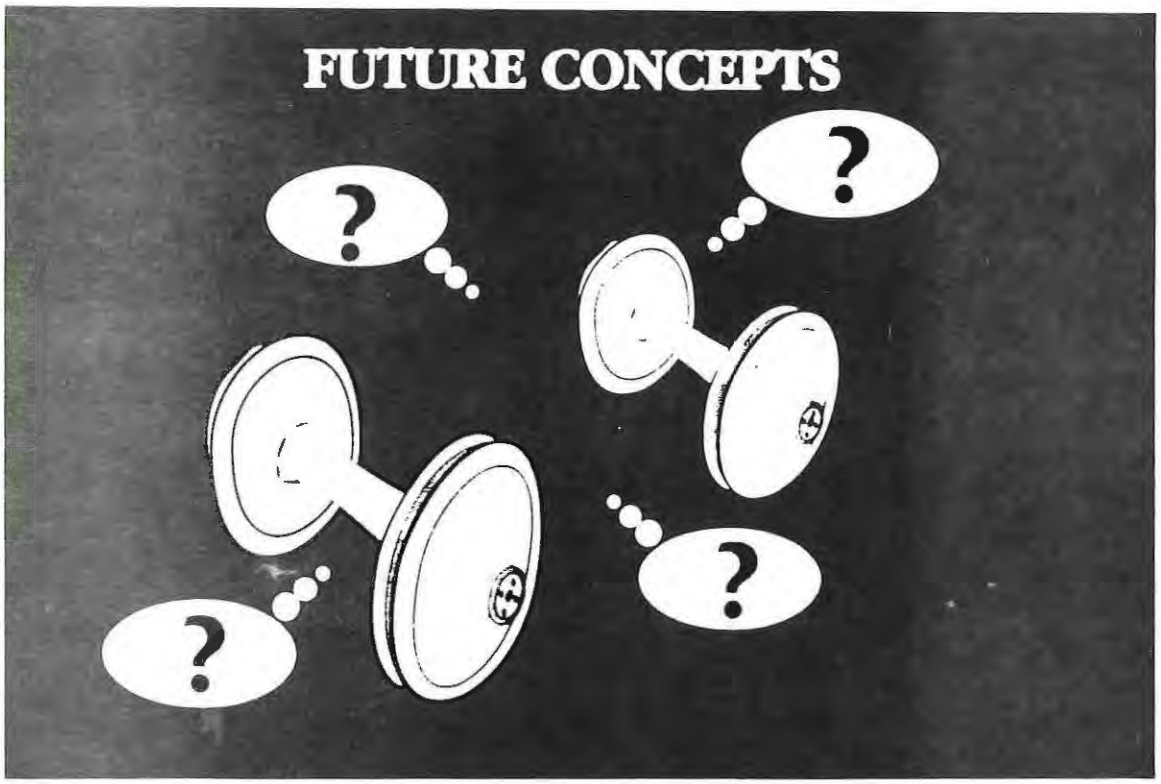
Notice that only when you free your mind and get outside the box, does the solution become obvious. Apply this technique to the conveyance system problem. Note that I didn't mention "truck" because I don't want you to remember the old ways.



Think of moving the package from point A to B. It must be handled carefully, without damage and be on time. The less it costs you to move it safety, the more money you will make.



Now visualize the need to move several packages simultaneously from A to B. This becomes more complicated and requires an efficient conveyance system!



You have learned from this conference the problems we are faced with. You know the performance we must have to meet our customers expectations and protect our financial situation. A new design is needed. We need new and different technology. New materials, new concepts, and new suspension systems will all combine to create the future conveyance system.

You don't have to single handedly create a whole new system. Write down ideas for sub-concepts. Discuss them with open minded associates. Don't discard any idea, no matter how absurd you may think it is.

If you need help in developing an idea or just to discuss a concept, call Keith Hawthorne or me. We will always be ready to assist or find the necessary help. Remember, each of you can be a Ross Winans.

INCENTIVES/PARTNERSHIPS

S. B. HARVEY

ASSOCIATION OF AMERICAN RAILROADS



Scott B. Harvey
Sr. Assistant Vice President

A Technology Transfer Forum

September 7-9, 1994

Levis Faculty Center, University of Illinois at Urbana-Champaign

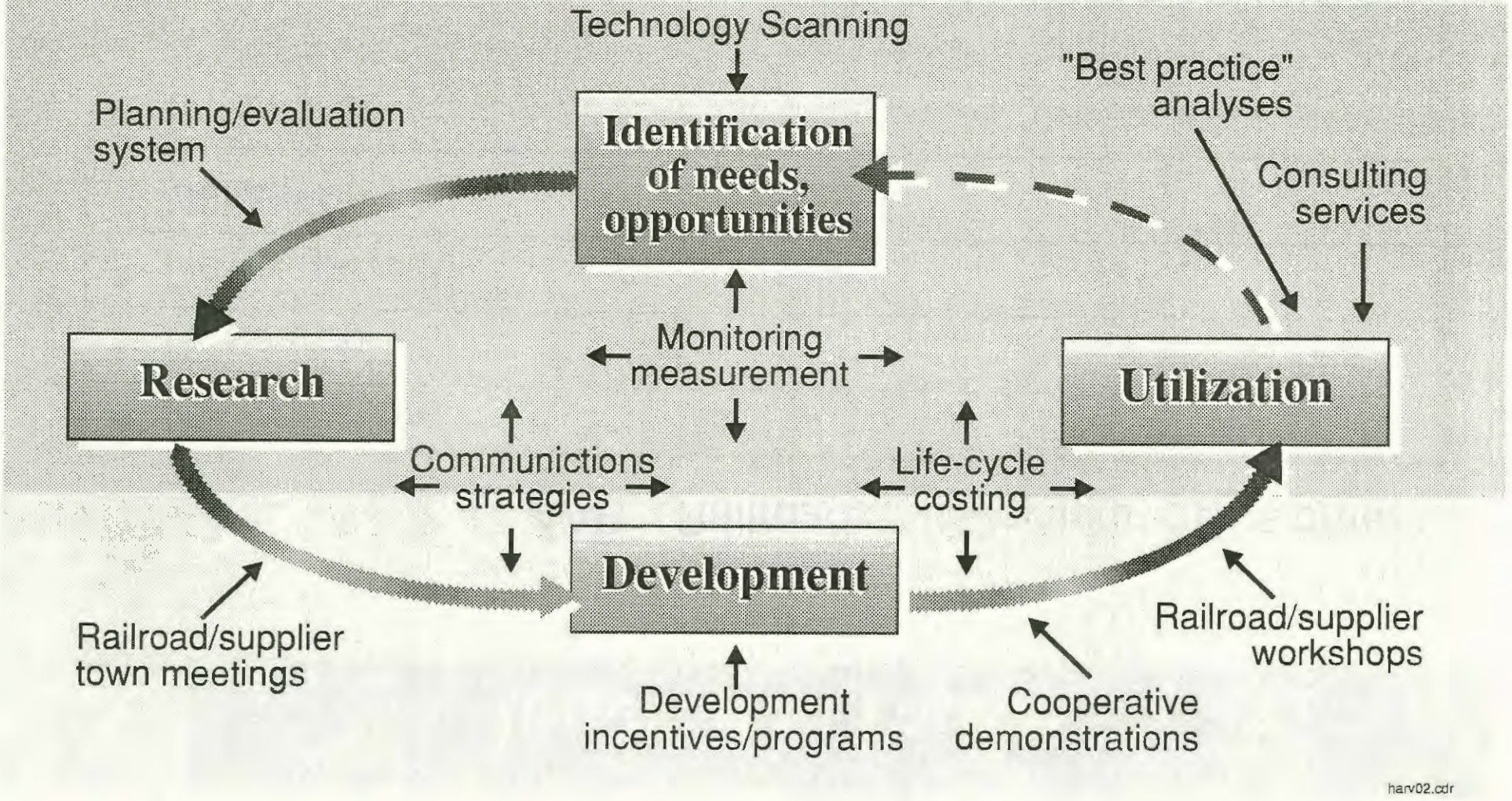
harv00.cdr

Some "Best Practices" in Innovation

- Innovation as a system
- System management to reduce time, minimize cost, increase probability of success
- Involvement of relevant system elements
  - Functions (R&D, marketing, manufacturing)
  - Suppliers, customers

harv01.cdr

Innovation Process



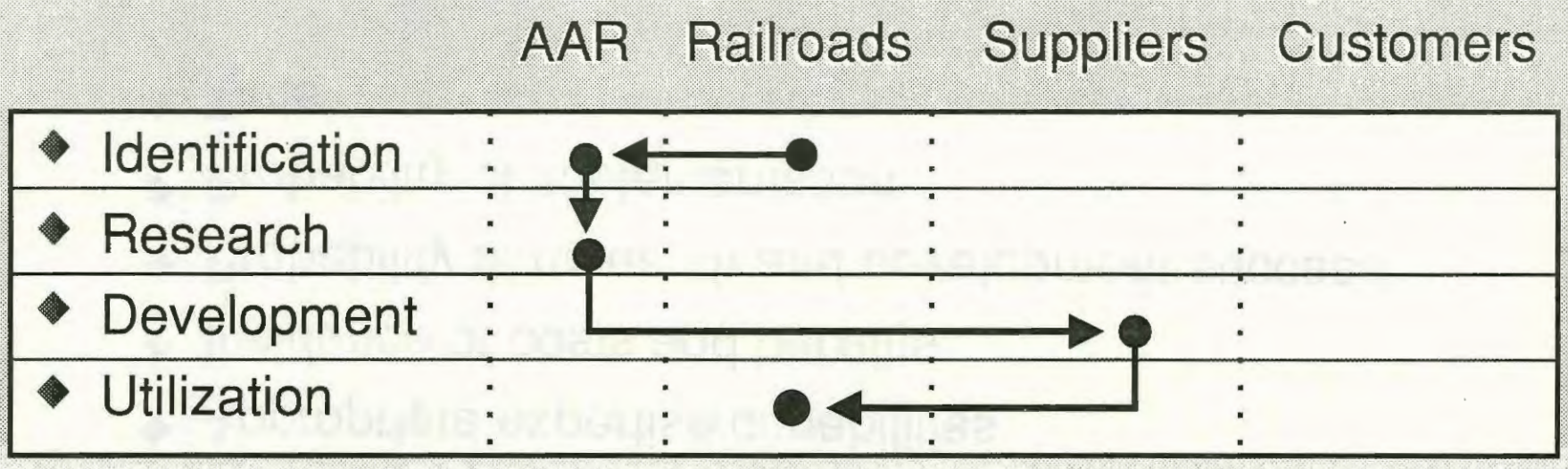
harv02.cdr

Innovation Parties

	AAR	Railroads	Suppliers	Customers
◆ Identification				
◆ Research				
◆ Development				
◆ Utilization				

harv03.cdr

Research and Test Paradigm

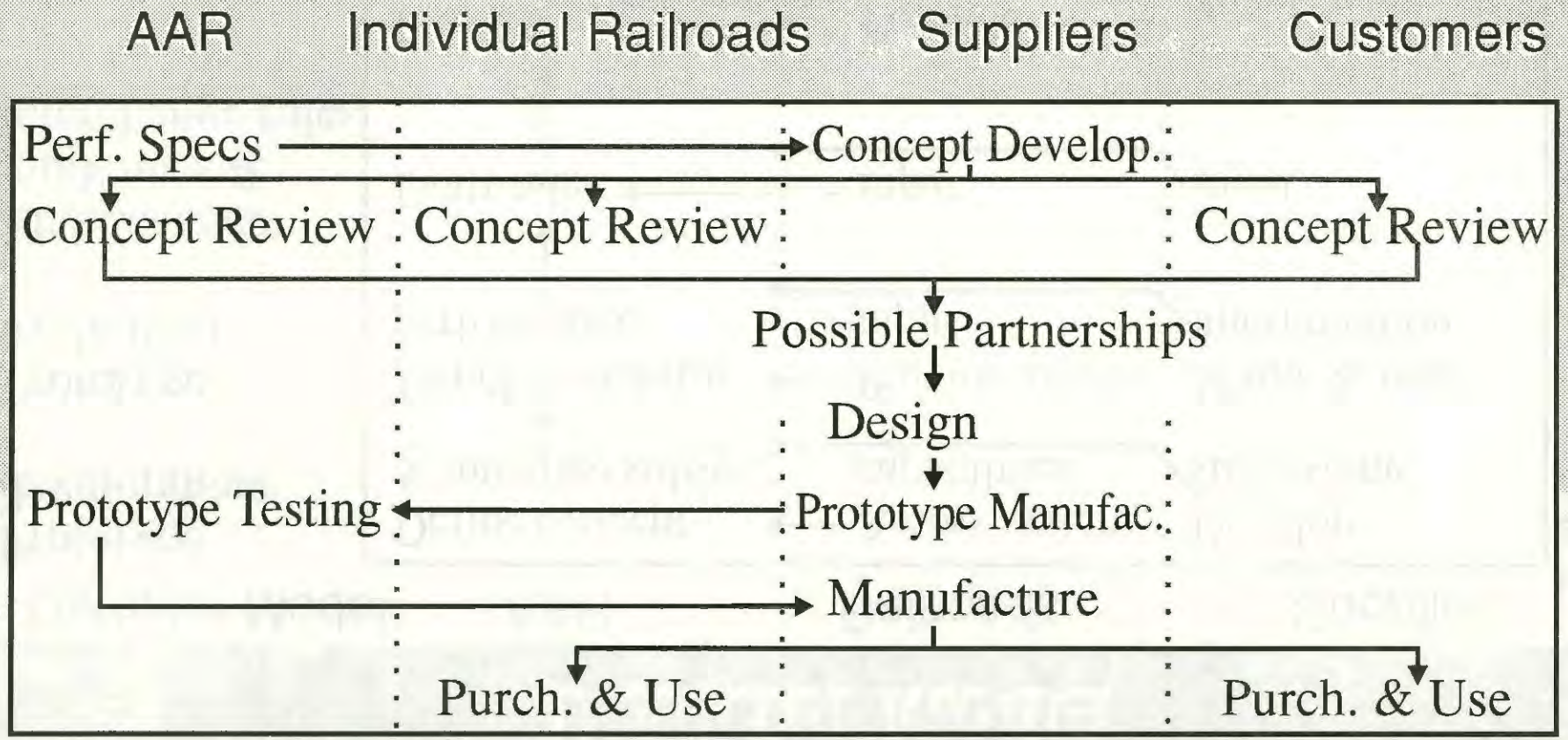


harv04.cdr

Need to Examine Alternatives

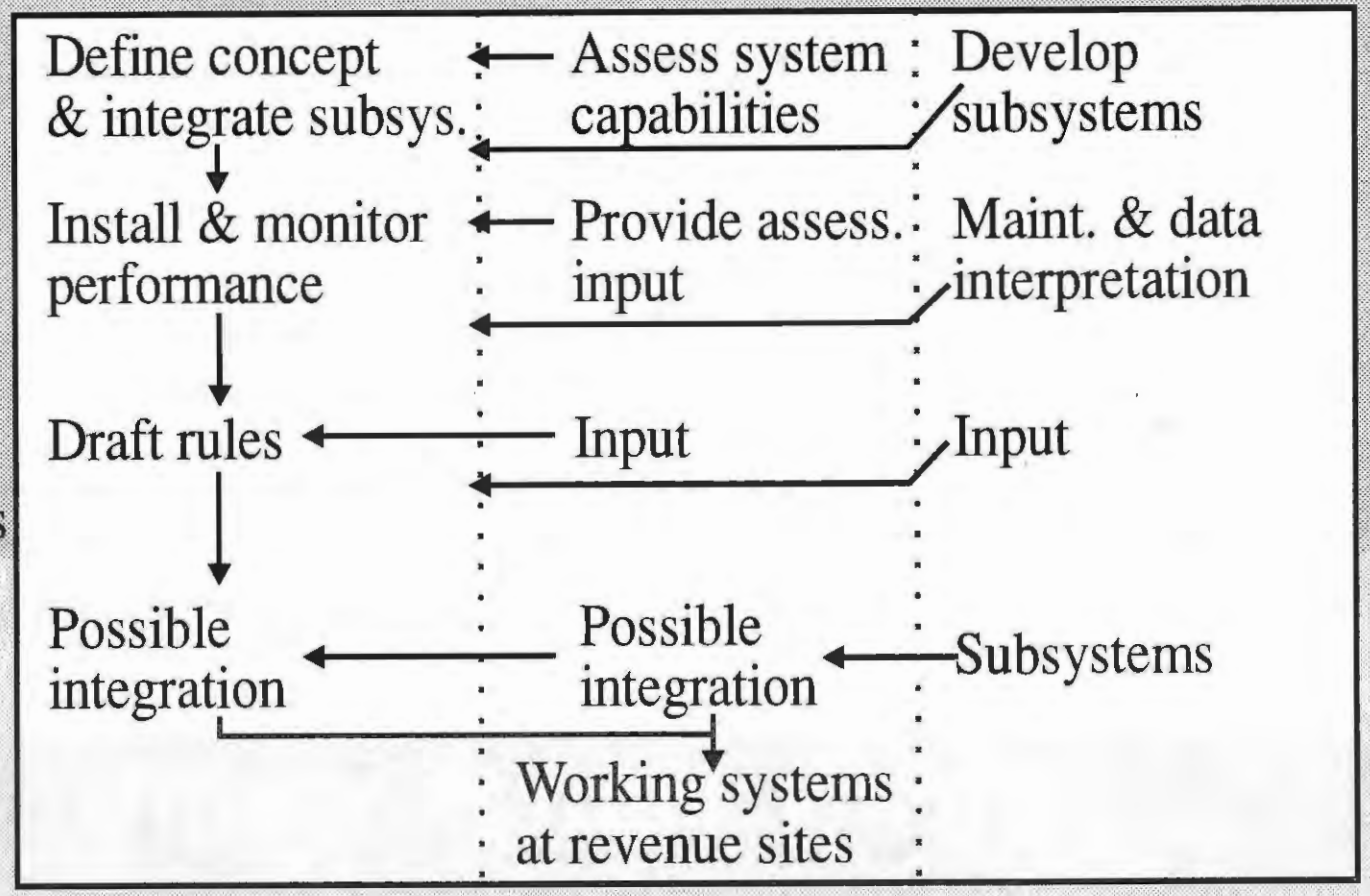
- Appropriate expertise/capabilities
- Incidence of costs and benefits
- Probability of research and development success
- Probability of implementation
- Time

Advanced Truck Model



harv06.cdr

Detector Model



harv07.cdr

Incentives--Impact on NPV

- Benefit of industry approach-market size, economies of scale
- Cost sharing
- Increased probability of research success
- Increased probability of implementation

harv08.cdr

Some Issues

- Organization/staffing/management
- Possible consortia/joint ventures
- Ownership
- Aligning costs/benefits

harv09.cdr

Opportunity

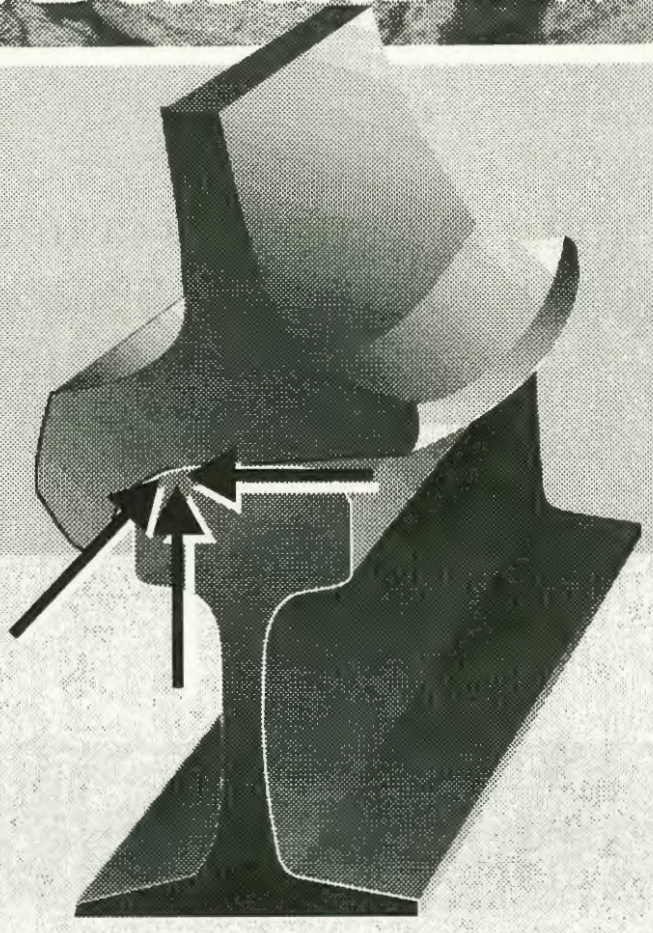
- Need
- Available expertise
- Railroads/other industry experience
- Support for railroad research and development

WAYSIDE DETECTION AND MONITORING

M. A. DEMBOSKY

ASSOCIATION OF AMERICAN RAILROADS

Wayside Vehicle Defect Detector--WVDD



madp00.cdr

Wayside Vehicle Defect Detector--WVDD

Main Purpose

- To identify vehicles in need of maintenance by observing dynamic response to track conditions

Wayside Vehicle Defect Detector--WVDD

Secondary Purposes

- To gather data for use in design of response based maintenance guidelines
- To provide a facility for in-service testing of detector & support systems
- To provide member railroads & researchers with fleet performance statistics
- To monitor effectiveness of rail lubrication
- New insights into design, wear & performance

madp03.cdr



WVDD

System Properties

- Capable of collecting & analyzing data between trains
- Capable of describing & identifying specific cars
- Capable of transmitting vehicle defect report over the phone & storing support information on removable disk
- Uses multiple sensors when possible



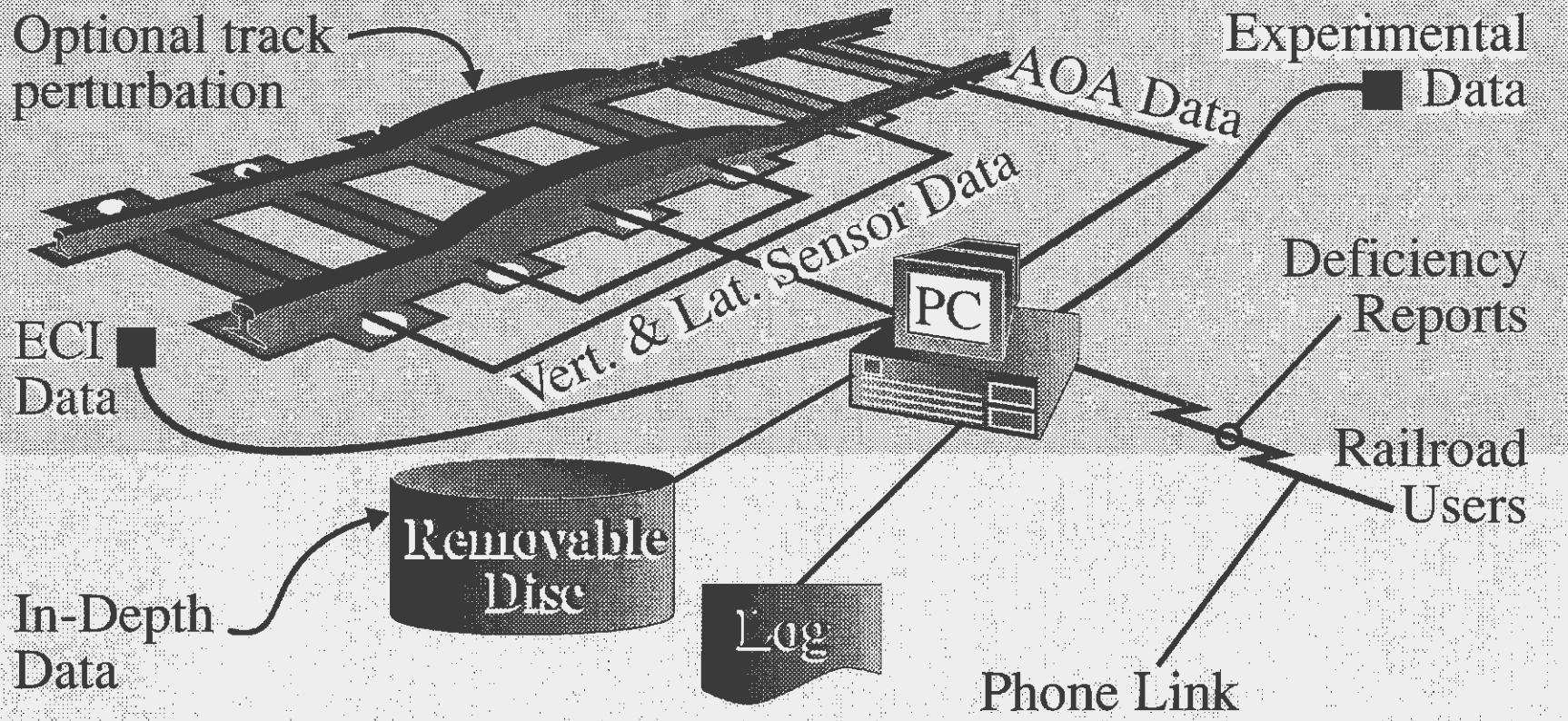
WWDD

System Properties (continued)

- Uses combination of simple rugged sensors & optional new devices
- Equipment small enough to reside in signal shack
- Power consumption low enough for solar electric use
- Suitable for portable or fixed installation

madp05.cdr

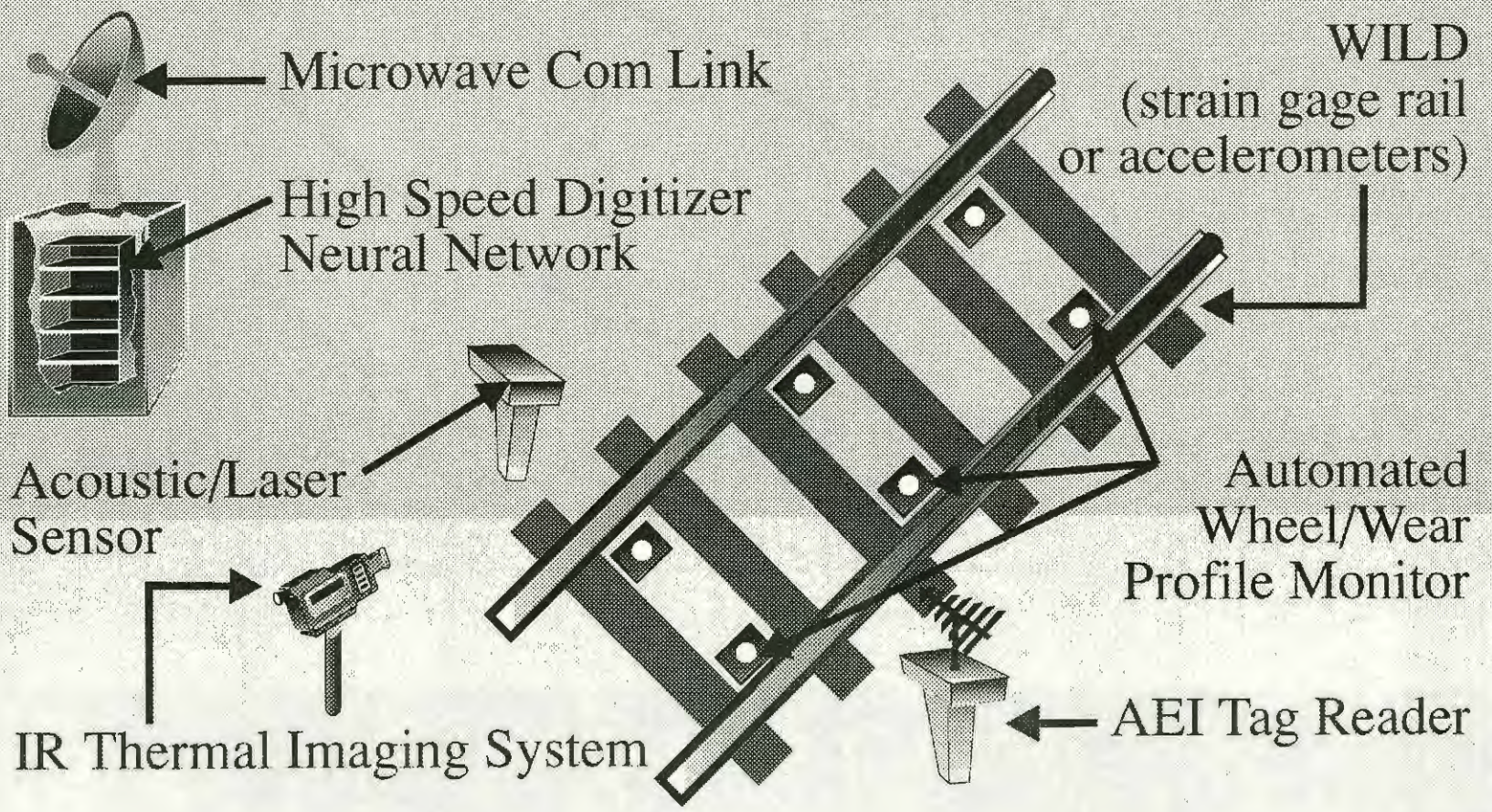
Typical Installation



madp06.cdr

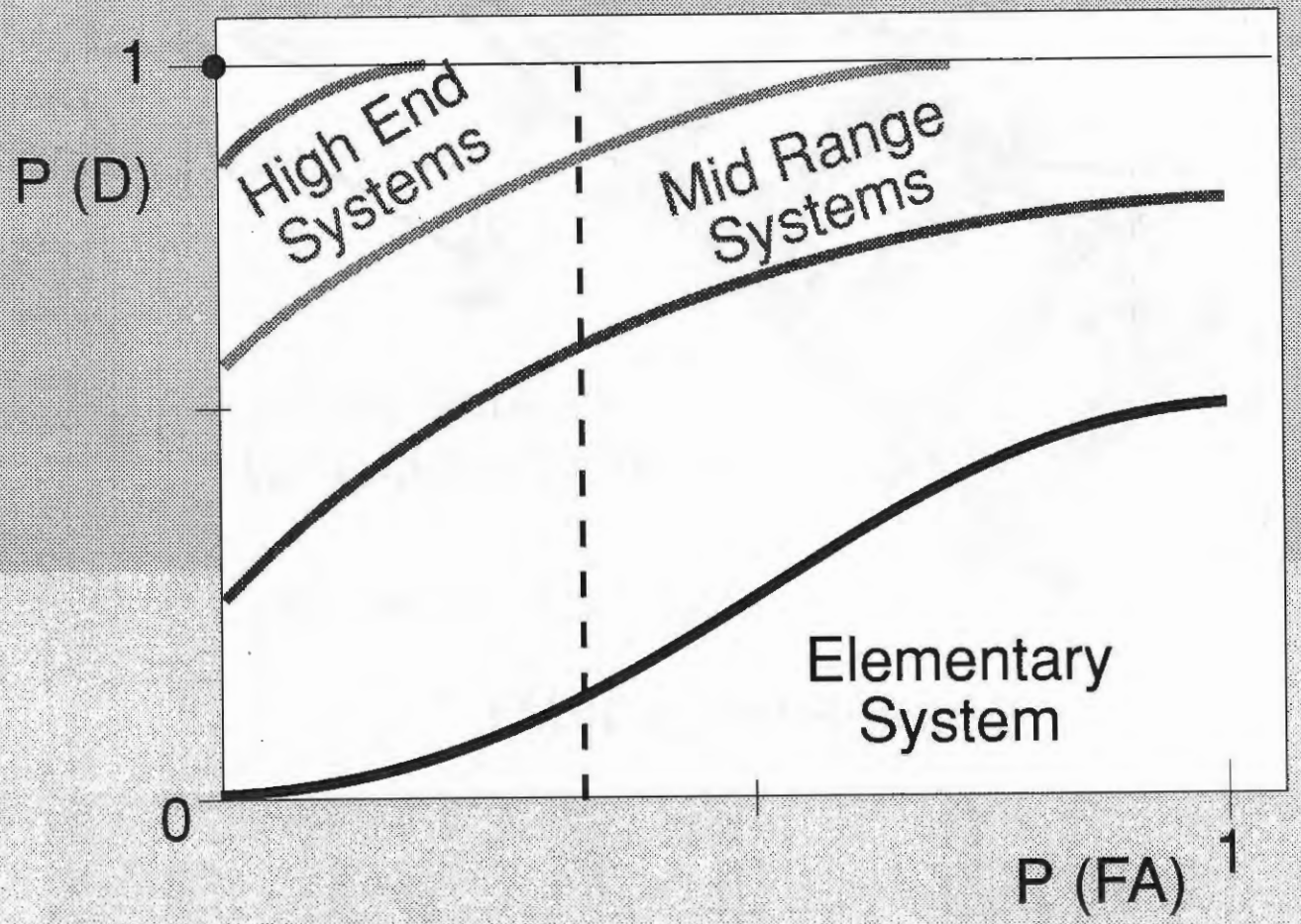
WILD

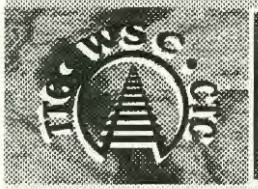
Typical Installation



madp06b.cdr

Approach to Project



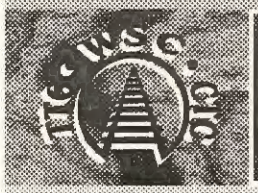


WVDD

Assessed Vehicle Characteristics

- Truck warp restraint
- Truck vertical & lateral damping
- Truck hunting tendency
- Wheel profiles
- Flat spots/out of round wheels
- Bad bearings

madp07.cdr



WVDD

Primary Measurements	Status
◆ Wheel vertical & lateral forces	Available
◆ Wheel Angle of Attack (AOA)	Available
◆ Wheel / rail contact point	Experimental
◆ Wheel profiles	Developmental
◆ Vehicle Identification	Available
◆ Acoustic data	Experimental
◆ Rail rollover moment	Experimental

madp08.cdr

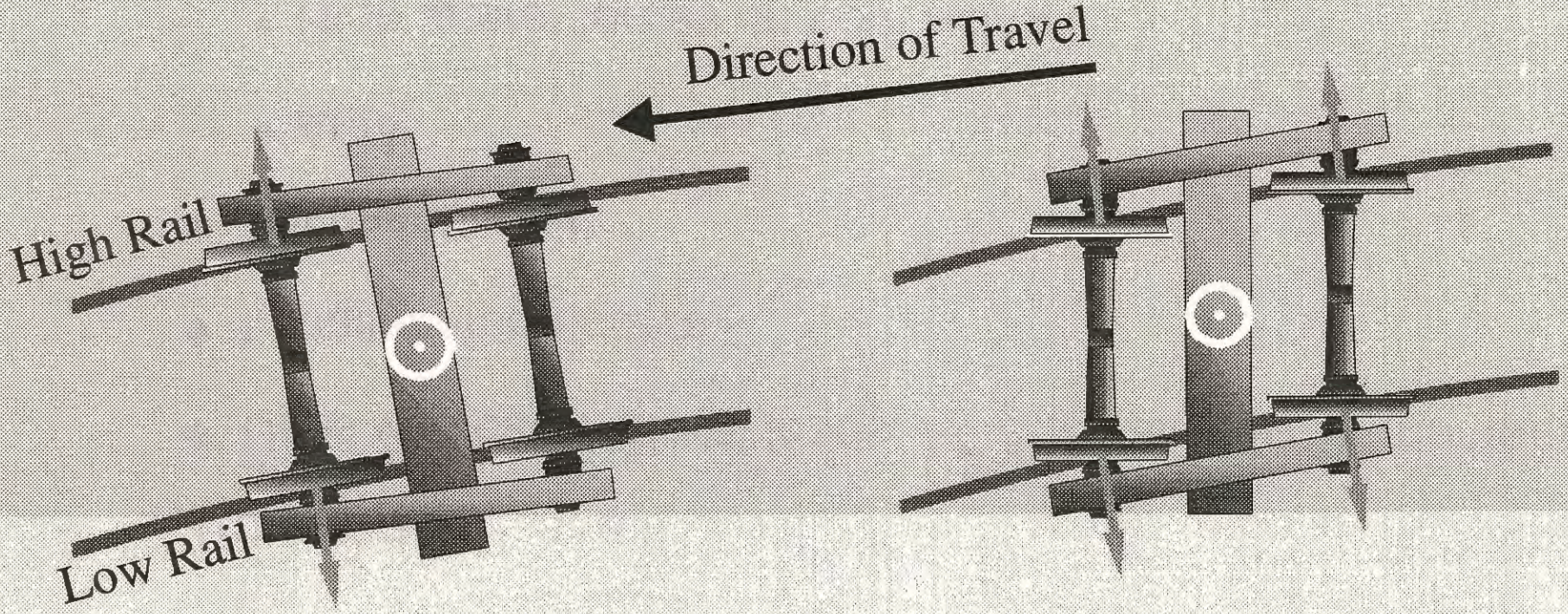
WVDD

Truck Warp Restraint

- Uses vertical & lateral loads, AOA, & rail rollover moment
- Installation on curve
- Assessable by:
  - AOA
  - High lateral forces on both axles
  - High rail rollover moment

madp09.cdr

WWDD



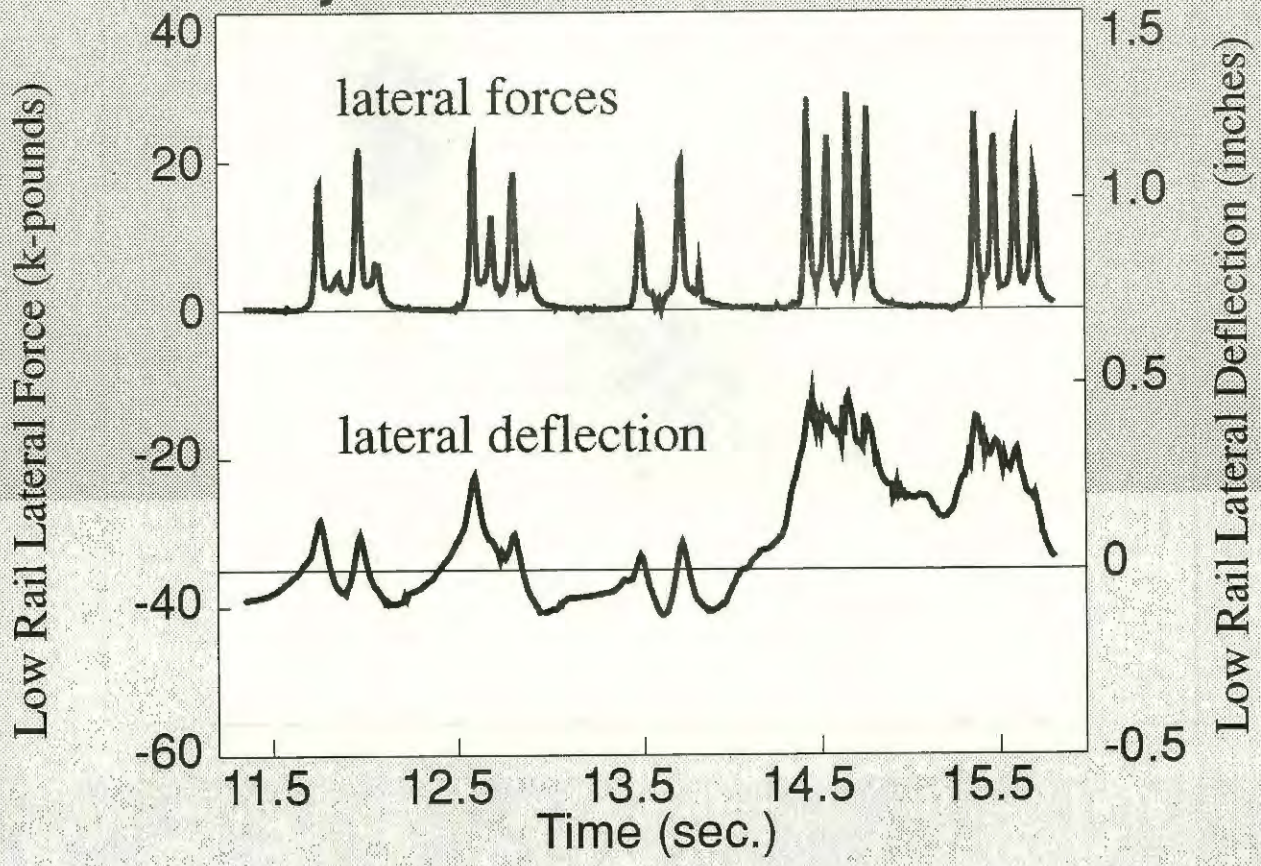
Turned Truck

Warped Truck

madp17.cdr

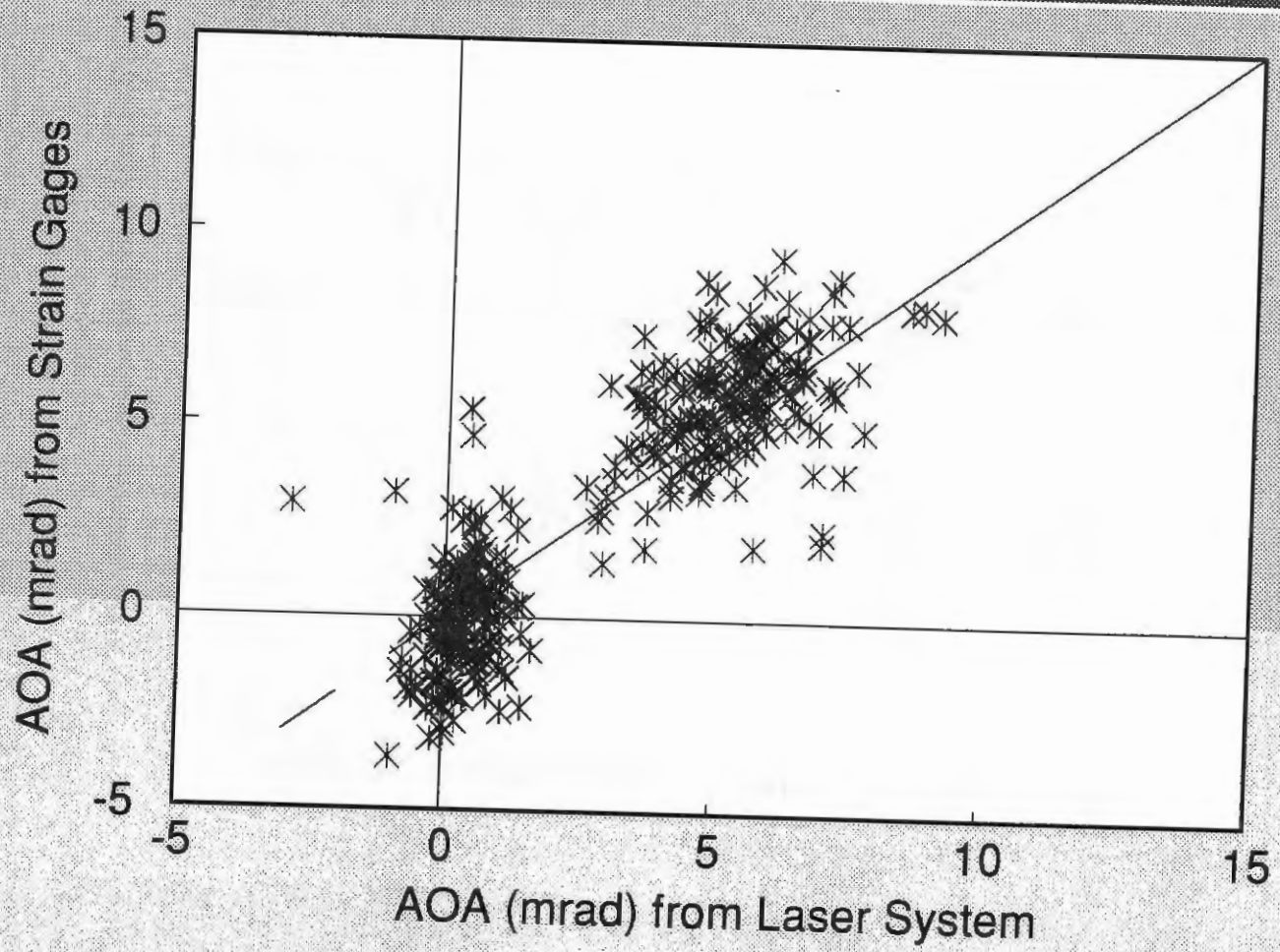
Effect of Adjacent Trucks

Heavy Axle Load Train Tested 1992



madp19.cdr

Wayside Vehicle Defect Detector--WVDD



MADP20.cdr

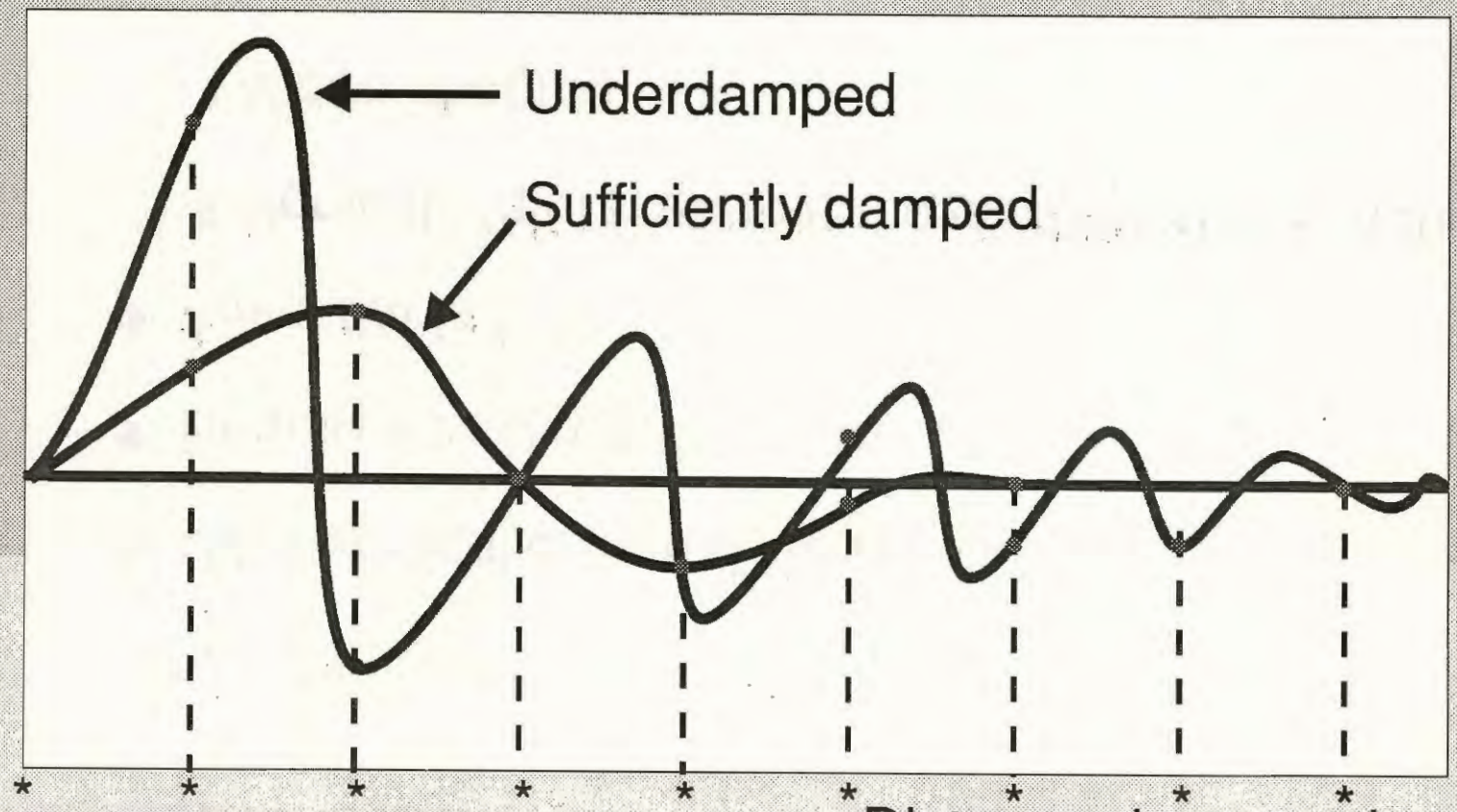
WVDD

Truck Vertical & Lateral Damping

- Uses vertical & lateral loads
- Requires track perturbation
- Assessable by:
  - Vertical / lateral augment over extended distance
  - Video imaging

madp10.cdr

Approach to Project



madp11.cdr



WVDD

Truck Hunting Tendency

- Uses vertical & lateral loads, AOA
- Installation on tangent
- May require perturbation
- Assessable by:
  - Lateral augment over extended distance
  - Video imaging

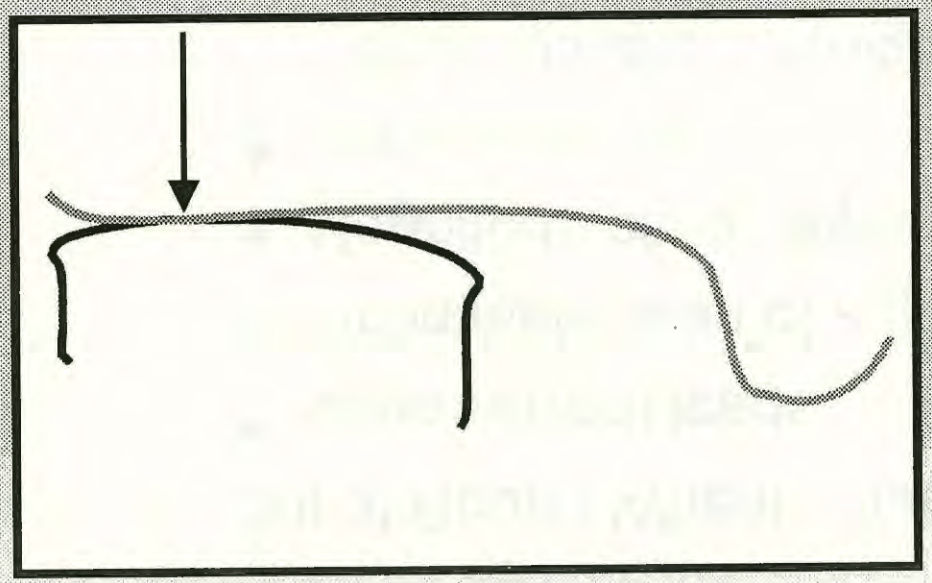
madp12.cdr

Wheel Profiles

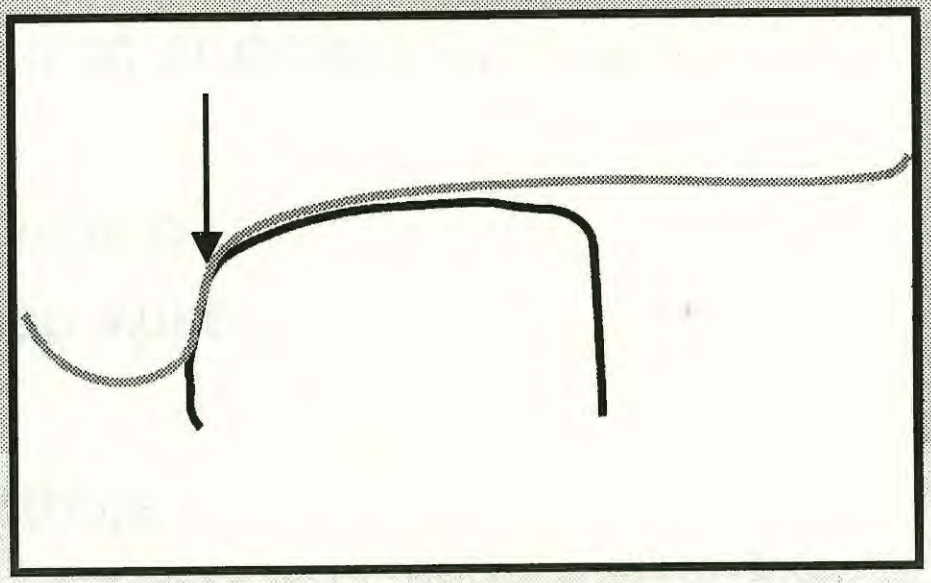
- Uses wheel/rail contact point sensor under development by AAR
- Requires known & controlled rail profile
- Limited to crude index of tread wear
- Assessable by:
  - Field side contact implies worn tread
- Could use new automatic profilometers under development by industry for direct reading of profiles

Wayside Vehicle Defect Detector--WVDD

Low Rail



High Rail



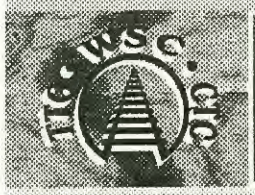
madp23.cdr

WVDD

Out of Round Wheel / Flat Spots

- Uses vertical loads
- Incomplete scan of a given wheel
- Magnitude of flat approximated
- Assessable by:
  - Short duration vertical load in excess of average for a given wheel
  - 100 kip limit (Rule 41S)
- Could use existing industry-developed systems

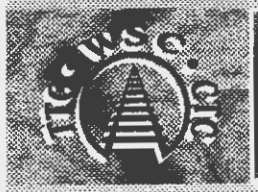
madp14.cdr



WVDD

Bad Bearings

- Uses acoustic emissions (audio frequency) analysis
- Uses thermal pattern analysis
- Assessable by:
  - Frequency content of acoustic spectra
  - Temperature distribution across bearing cup



WVDD

Bad Bearings (continued)

- Defect types:
  - Broken roll elements
  - Spalled raceways and roller elements
  - Grooved axle journal condition
- Could use existing industry-developed systems
- Integrate AEI technology

madp14C.cdr



WVDD

Summary

- Extensive development experience:
  - HAL revenue service
  - DDTF
  - AAR derailment investigations
  - Member railroads
- Modular & expandable
- Real time reports & research data
- Permanent or portable
- Site specific
- System integration

madp15.cdr

TECHNICAL EVALUATION OF PROPOSED TRUCK DESIGNS

S. P. SINGH

ASSOCIATION OF AMERICAN RAILROADS

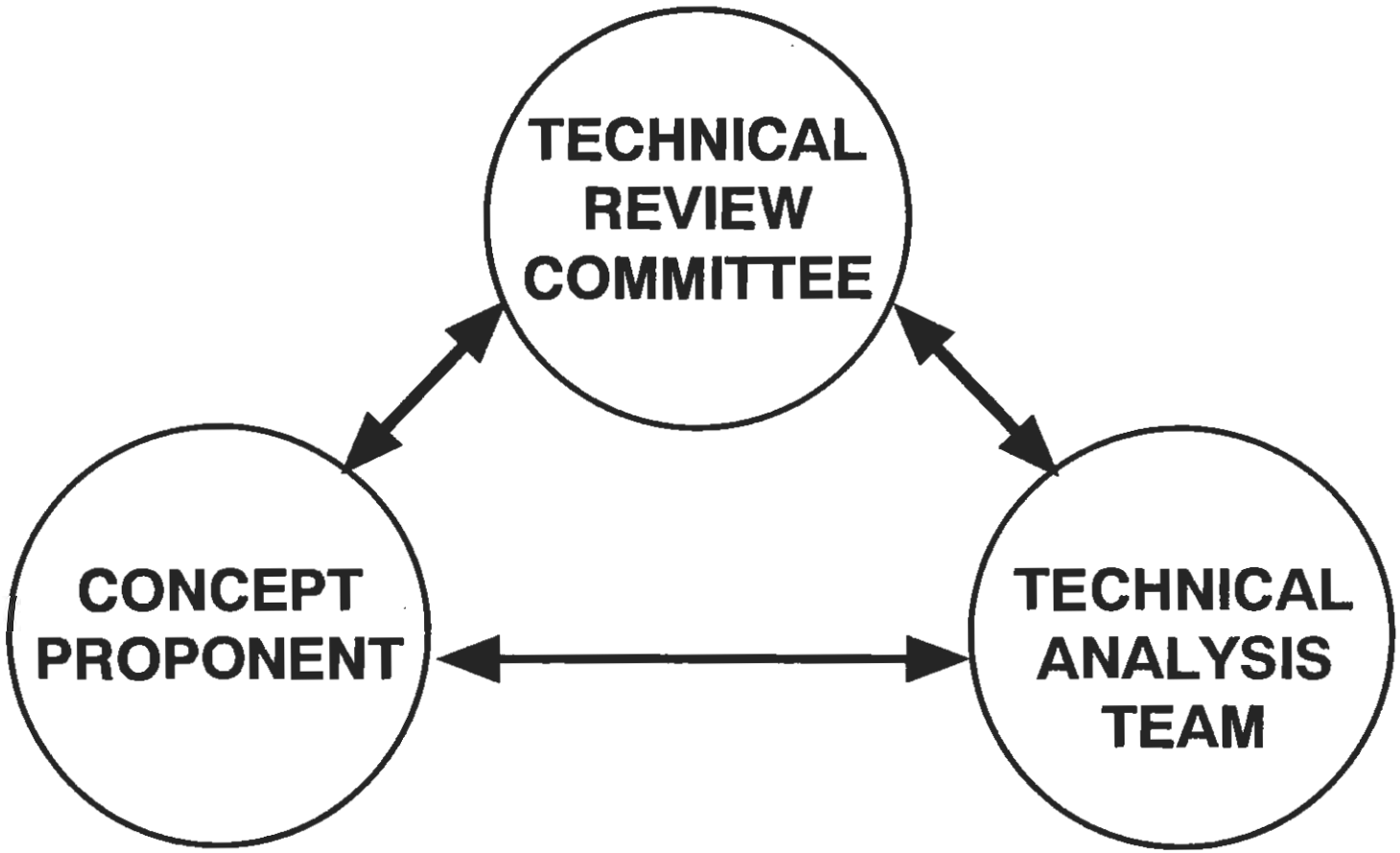
TECHNICAL EVALUATION OF PROPOSED DESIGNS

Sōm P. Singh
Association of American Railroads

An Assessment of Needs and Opportunities - Freight Car Trucks For The 21st Century A Technology Transfer Forum

September 7-9, 1994
Levis Faculty Center
University of Illinois at Urbana-Champaign

TECHNICAL EVALUATION OF PROPOSED DESIGNS



TECHNICAL EVALUATION OF PROPOSED DESIGNS

TECHNICAL REVIEW COMMITTEE

(AAR Member Roads Representatives)

- **Steer and approve the performance specification**
- **Evaluate truck designs as described by proponent(s)**
- **Advise/inform proponents of potential problems**
- **Recommend technical and economic evaluations**
- **Help develop input for economic analysis**
- **Offer suggestions?**



TECHNICAL EVALUATION OF PROPOSED DESIGNS

TECHNICAL ANALYSIS TEAM

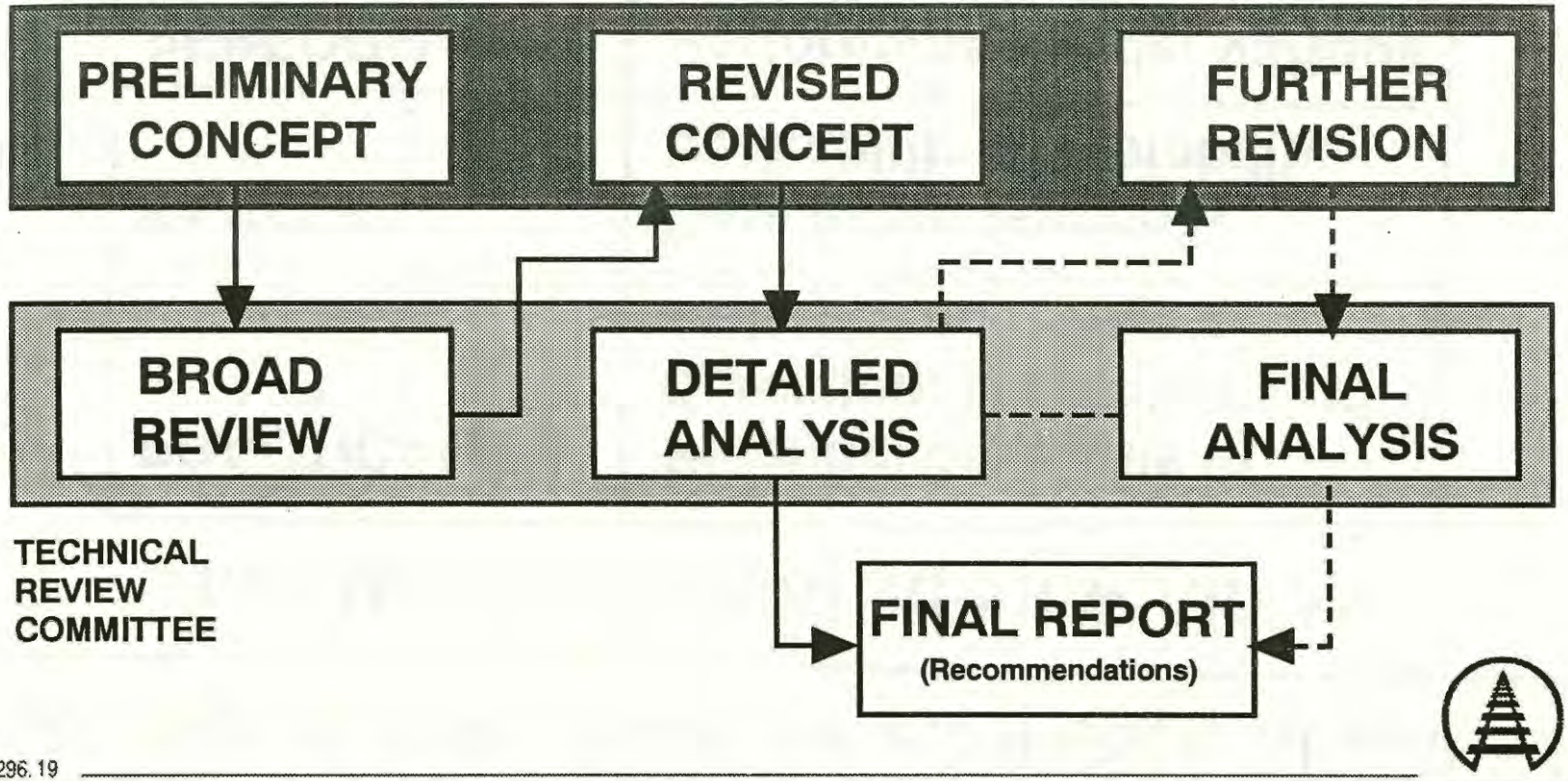
(AAR Staff)

- **Develop performance specification**
- **Define technical evaluation scope for recommended concepts**
- **Perform technical evaluation**
- **Report to the Technical Review Committee on evaluation**
- **Develop technical input for economic analysis**
- **Offer suggestions to proponents?**



TECHNICAL EVALUATION OF PROPOSED DESIGNS

TECHNICAL REVIEW CYCLE



TECHNICAL EVALUATION OF PROPOSED DESIGNS

PRIMARY TECHNICAL REVIEW AREAS

PRACTICALITY	**Potential problems in operation, inspection and maintenance, etc.**
SAFETY	**Derailment tendency, trackability and stability**
PERFORMANCE	**Performance under various operating conditions, economic implications**



TECHNICAL EVALUATION OF PROPOSED DESIGNS

PRACTICALITY

- **Operation considerations**
- **Inspection methods**
- **Detection considerations**
- **Brake system adaptability**
- **Maintenance**
  - **Modularity, disassembly, etc.**

TECHNICAL EVALUATION OF PROPOSED DESIGNS

INSPECTION, MAINTENANCE AND REPAIR

- **Component wear limits and inspection**
- **Brake system adaptability**
  - **Brake beam and brake shoe changes, disk brakes**
- **General replacement**
  - **Wheel set change, center bowl lubrication**
- **Inspection and maintenance cycles**
- **Repair procedure for major components**

TECHNICAL EVALUATION OF PROPOSED DESIGNS

SAFETY

Derailment Tendency

	Chapter XI Regimes
• Wheel Climb	• Pitch & Bounce, Twist & Roll
• Rail Rollover	• Curve Entry/Exit
• Wheel Lift	• Steady-State Curving
• Wheel Drop	• Dynamic Curving
	• Hunting, Yaw & Sway



TECHNICAL EVALUATION OF PROPOSED DESIGNS

SAFETY

Structural Integrity

- **Stress analysis**
- **Fatigue analysis**

Failure Modes

- **Catastrophic failures**

TECHNICAL EVALUATION OF PROPOSED DESIGNS

PERFORMANCE EVALUATION

SAFETY	**Chapter XI Regimes**
TRACK MAINTENANCE/ DAMAGE	**Chapter XI Regimes, Turn-outs and P1, P2 Forces**
RIDE QUALITY/ LADING DAMAGE	**Chapter XI Regimes and Revenue Track**
ENERGY CONSUMPTION	**Steady-State Curving and Misaligned Truck on Tangent Track**

TECHNICAL EVALUATION OF PROPOSED DESIGNS

PERFORMANCE RANKING

REGIME	WEIGHTED PARAMETER	CONDITION(S)
Twist & Roll	VL, WUL	Resonance
Pitch & Bounce	VL	Resonance/ Maximum Speed
Yaw & Sway	LL	Resonance/ Maximum Speed
Dynamic Curving	VL, LL, L/V	5 Degree Curve
Steady-State Curving	Wear, LL	3 & 10 Degree Curve
Curve Entry/Exit	LL, L/V, WUL	10 Degree Curve
Hunting	RMS Accn.	Maximum Speed
Turnout	VL, LL	Maximum Speed
Revenue Track	VL, LL RMS Accn.	25, 45, 65 MPH

LL: Lateral Load, VL: Vertical Load, WUL: % Wheel Unloading, L/V: Lateral/Vertical Load Ratio

TECHNICAL EVALUATION OF PROPOSED DESIGNS

SUMMARY

- Increased safety
- Reduced track maintenance
- Longer equipment/component lives
- Reduced energy consumptions
- Reliable and consistent performance
- Improved inspection/detection/maintenance

PERFORMANCE AND SERVICE TESTING

A. J. PETERS

ASSOCIATION OF AMERICAN RAILROADS

TESTING REQUIREMENTS FOR PROTOTYPES

A. J. Peters
Association of American Railroads

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TESTING REQUIREMENTS FOR PROTOTYPES

PRIMARY TECHNICAL REVIEW AREAS

Safety	**Chapter XI Regimes**
Track Maintenance/ Damage	**Chapter XI Regimes, Turn-outs and P1, P2 Forces**
Ride Quality Lading Damage	**Chapter XI Regimes and Revenue Track**
Energy Consumption	**Steady-State Curving and Misaligned Truck on Tangent Track**

TESTING REQUIREMENTS FOR PROTOTYPES

CONTROLLED ENVIRONMENT TESTING

- **Chapter XI testing**
  - **Critical areas identified by the Technical Evaluation**
  - **Others required by the Technical Review Committee**
- **Simulated worn component testing**
  - **Must ensure that simulated condition is representative of service-worn conditions**

TESTING REQUIREMENTS FOR PROTOTYPES

IN-SERVICE TESTING

- **Ride quality (for ride quality sensitive applications)**
- **Follow existing procedures for new car approval**
- **Closely monitor wear rates and resultant behavior**
- **Confirm recommended maintenance procedures applying refinements as necessary**

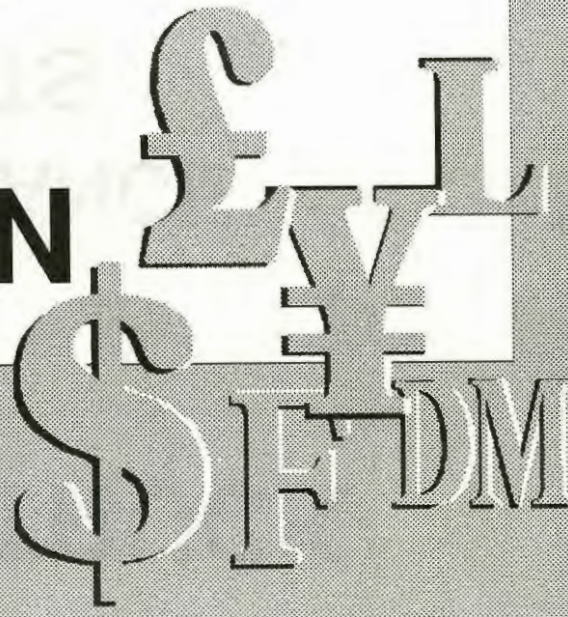
ECONOMIC EVALUATION OF ADVANCED TRUCKS

T. S. GUINS & M. B. HARGROVE

ASSOCIATION OF AMERICAN RAILROADS

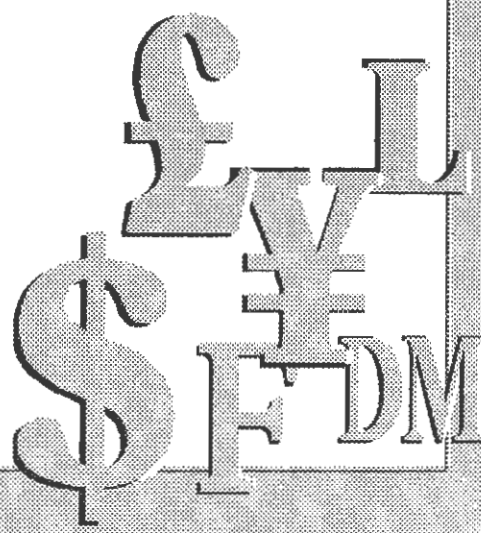
FREIGHT CAR TRUCKS FOR THE 21ST CENTURY

ECONOMICS OF IMPROVED TRUCK DESIGN



ECONOMICS OF TRUCK DESIGN

- ESTIMATE VALUE OF IMPROVED PERFORMANCE
- USE LIFE CYCLE COSTS



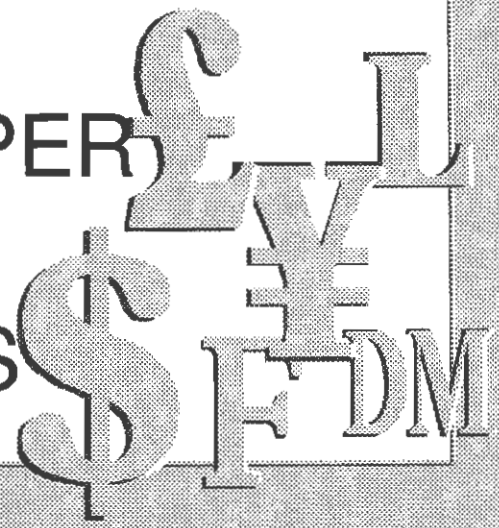
TOPICS

- ESTABLISH BASE CASE
- COSTS EFFECTED BY TRUCK DESIGN
- ESTIMATE POTENTIAL BENEFITS
- SPECIFY A "PREMIUM" TRUCK
- SAMPLE EVALUATIONS



BASE CASE

- STANDARD THREE-PIECE TRUCK
- TYPICAL BULK COMMODITY TRAIN
- 50% LOADED CYCLE
- 50,000 & 100,000 MILES PER YEAR UTILIZATION
- THREE SAMPLE ROUTES



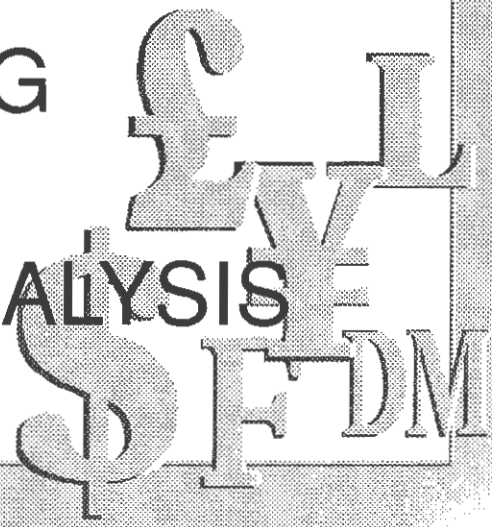
COSTS SENSITIVE TO TRUCK DESIGN

- ACCIDENT COSTS
- CAR MAINTENANCE COST
- TRACK MAINTENANCE COST
- ENERGY COST
- LADING DAMAGE



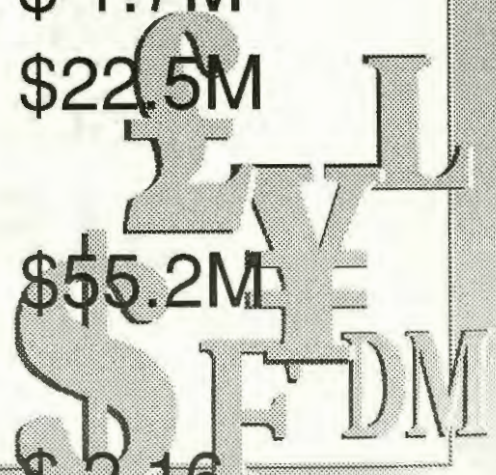
MODELS & DATA BASES

- CAR MAINTENANCE COST DATA BASE
- EQUIPMENT RELIABILITY ANALYSIS SYSTEM
- TRAIN ENERGY MODEL
- TOTAL RIGHT-OF-WAY ANALYSIS AND COSTING SYSTEM
- RAIL ENERGY COST ANALYSIS PACKAGE



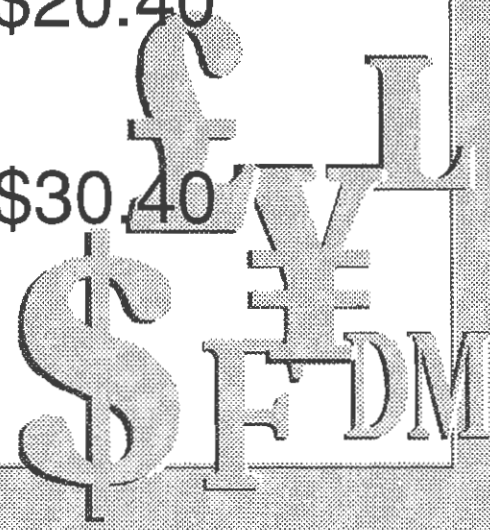
ACCIDENT COST

PERFORMANCE REGIME	ACCIDENT COST
ROCK AND ROLL	$ 3.5M
CURVE ENTRY	$10.3M
STEADY STATE CURVING	$17.2M
PITCH AND BOUNCE	$ 1.7M
LATERAL STABILITY	$22.5M
TOTAL	$55.2M
TOTAL/1000 CAR MILES	$ 2.16



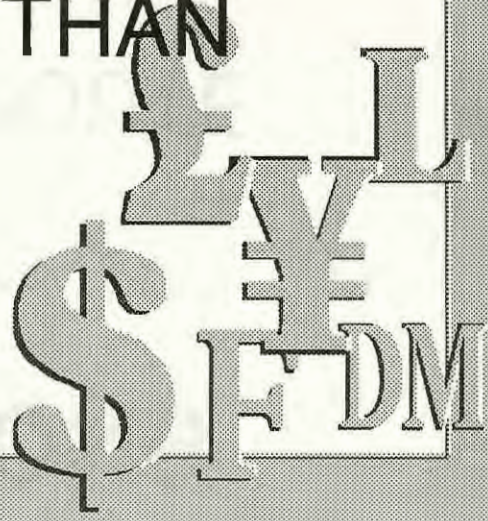
EQUIPMENT MAINTENANCE COSTS

PERFORMANCE REGIME	COST
CURVING	$15.60
VERTICAL FORCES	$ 4.80
TOTAL	$20.40
BASE CAR - TOTAL MAINTENANCE $'S PER 1000 CAR MILES	$30.40



"PREMIUM" TRUCK

- MAXIMUM 50% DYNAMIC AUGMENT
- CURVING PERFORMANCE BEST FAST IMPROVED TRUCK
- WEIGHS ONE TON LESS THAN BASE TRUCK

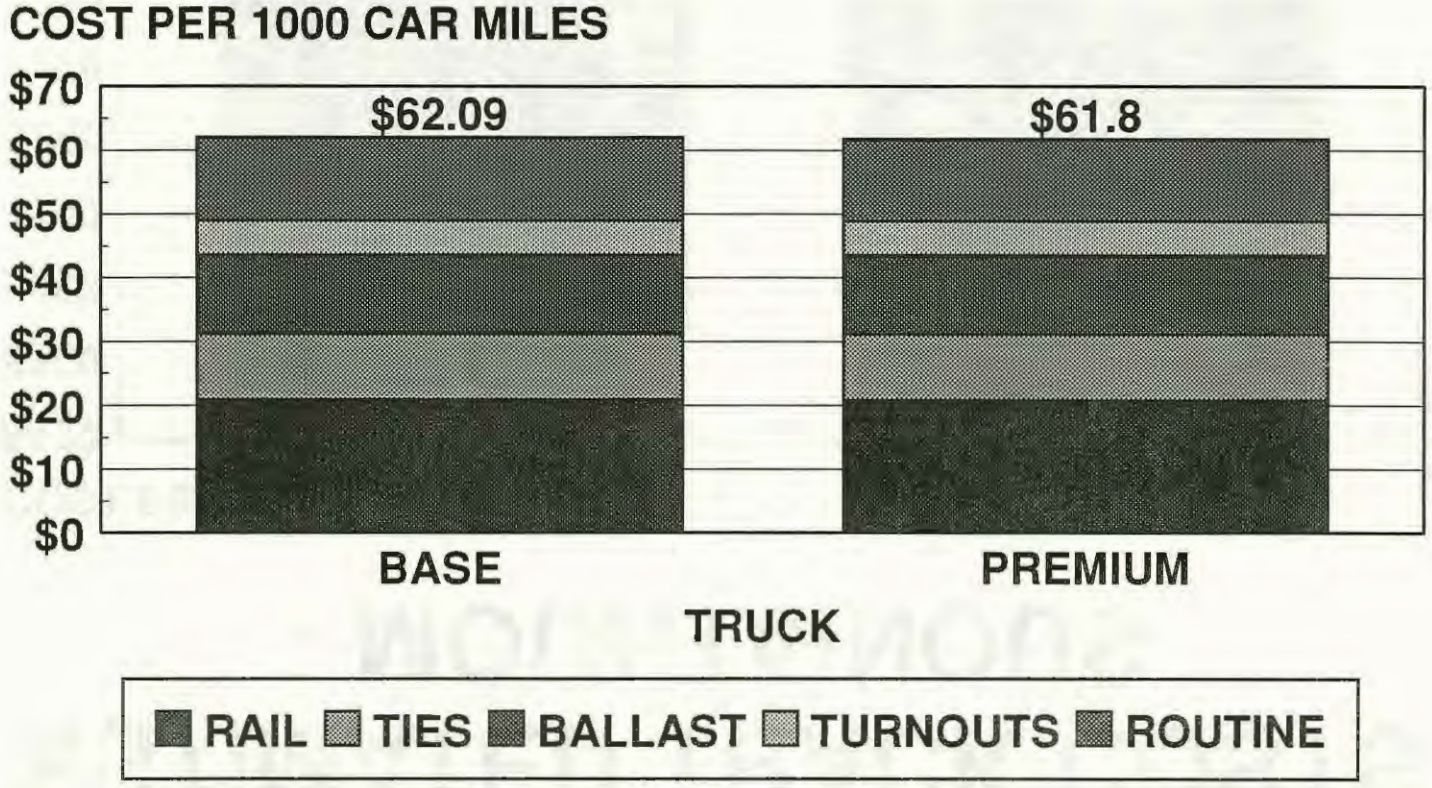


SAMPLE EVALUATIONS

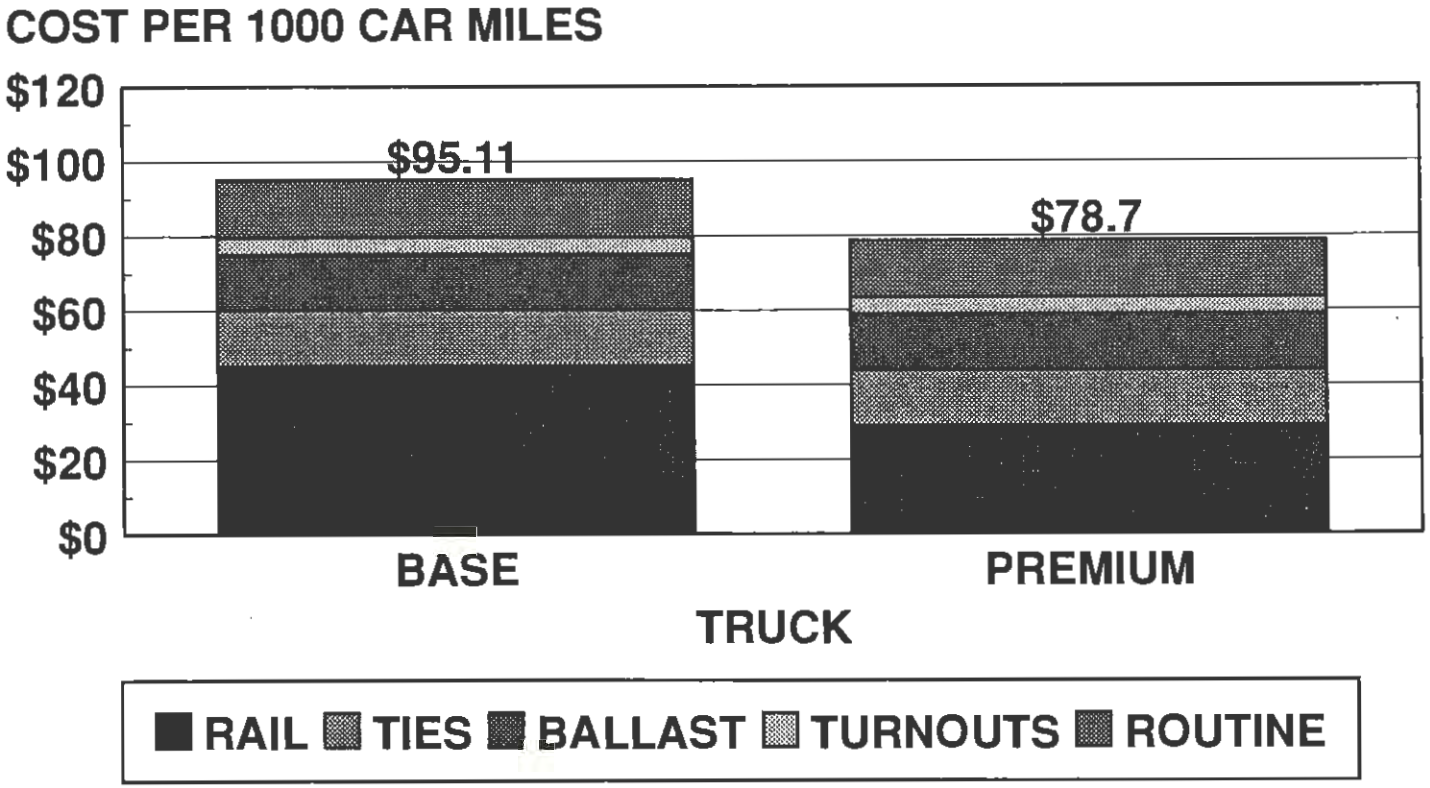
- STRAIGHT FLAT ROUTE
- CURVY, MOUNTAINOUS ROUTE
- TYPICAL BULK SERVICE ROUTE



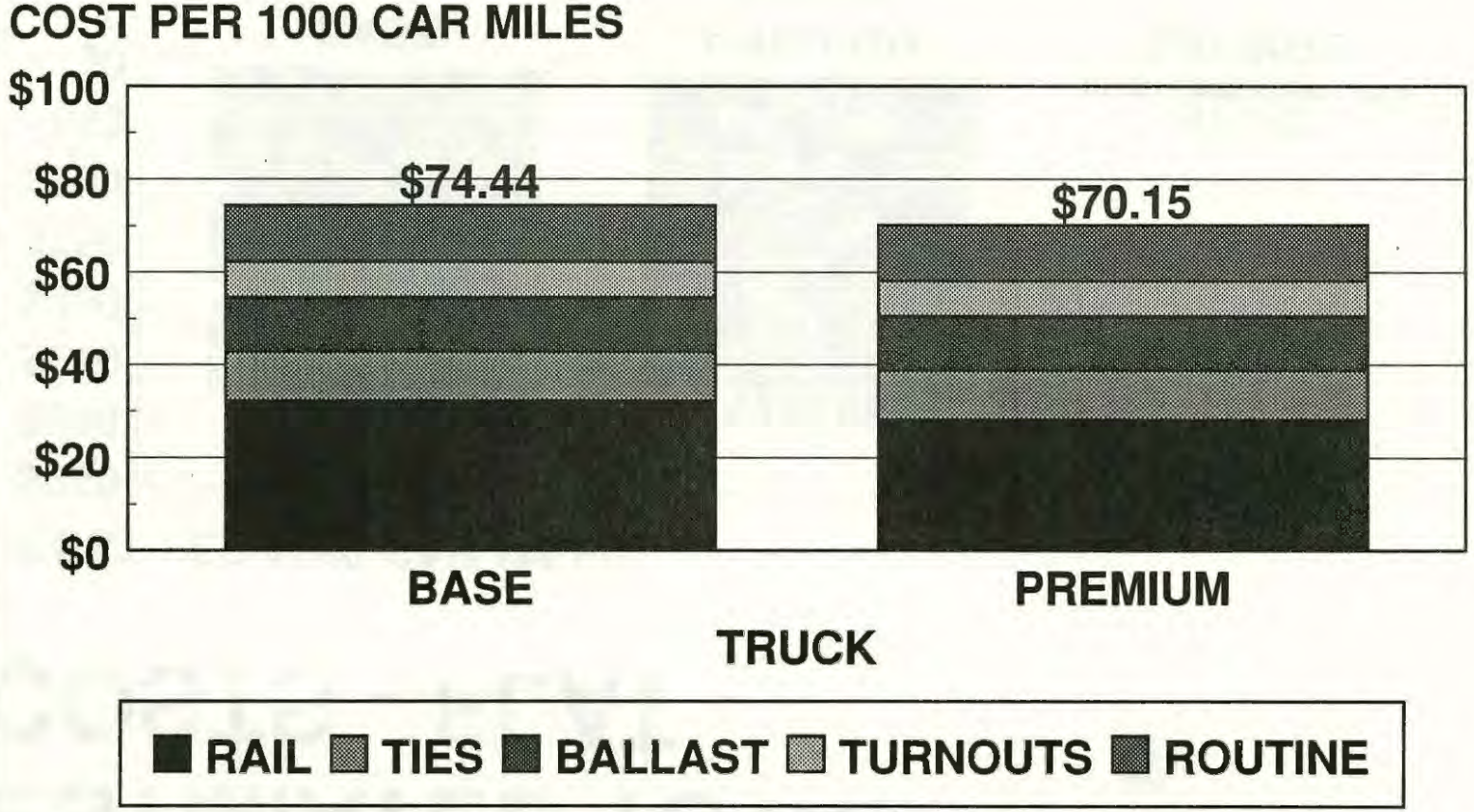
ESTIMATED TRACK COSTS - FLAT ROUTE



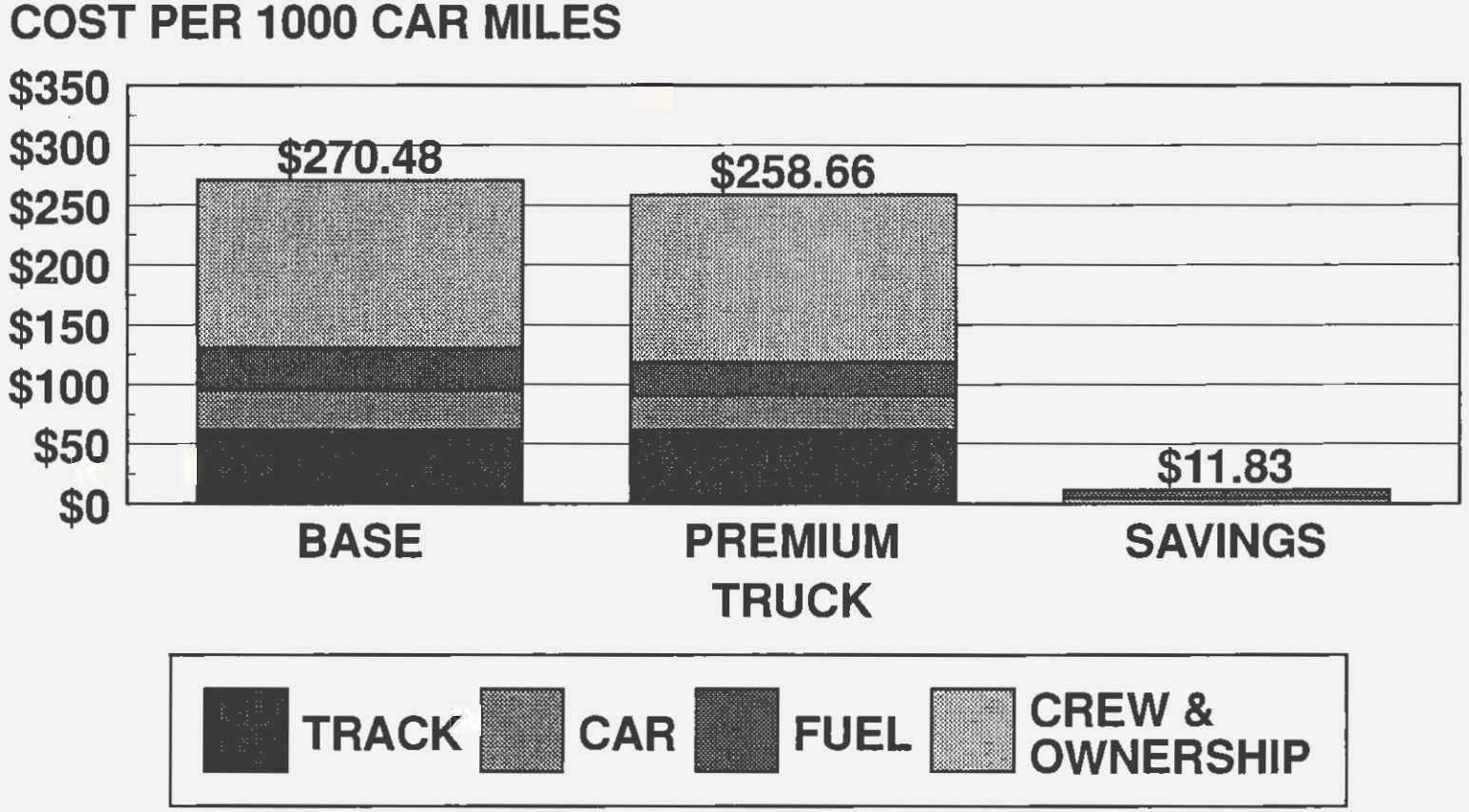
ESTIMATED TRACK COSTS - MOUNTAINOUS



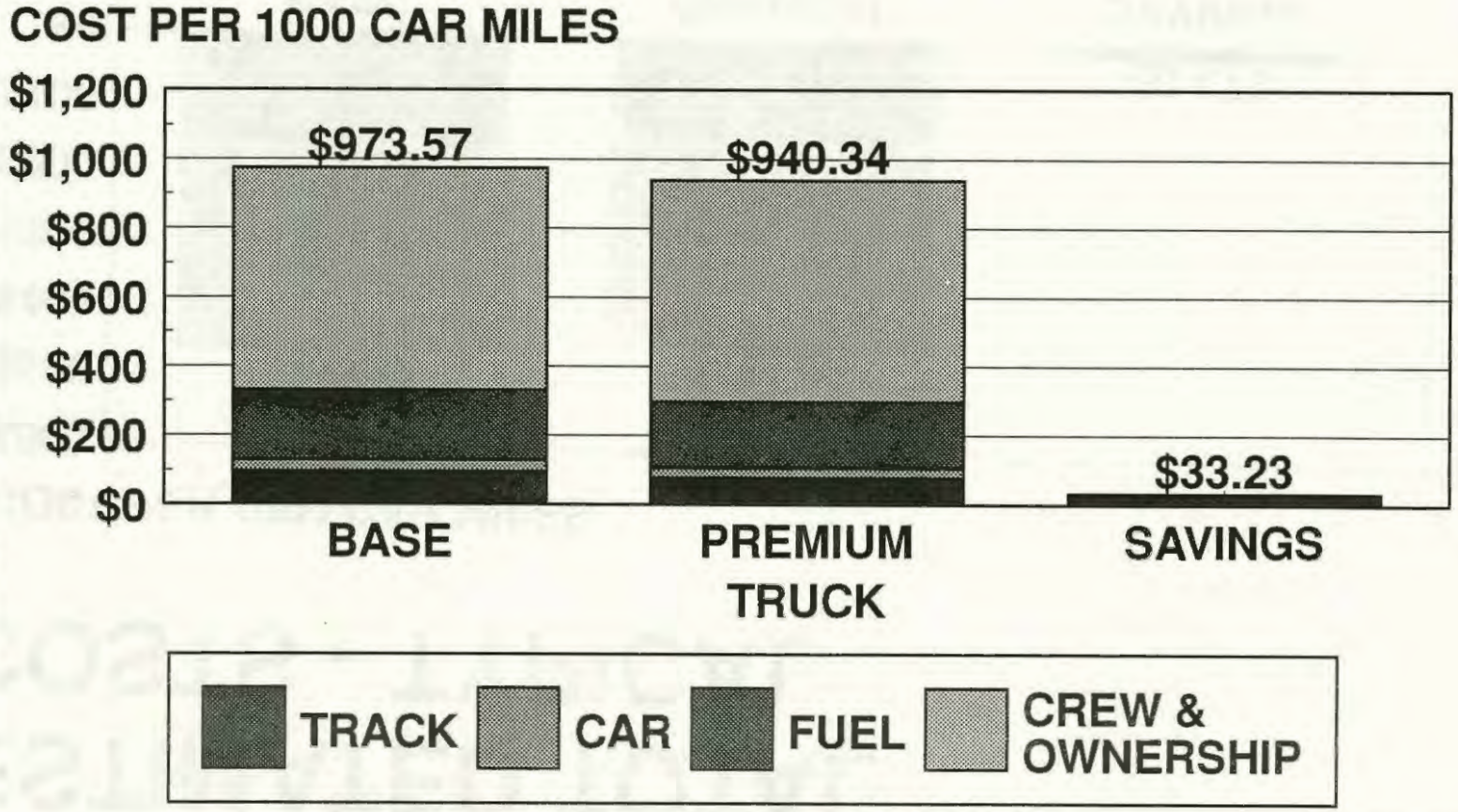
ESTIMATED TRACK COSTS - TYPICAL



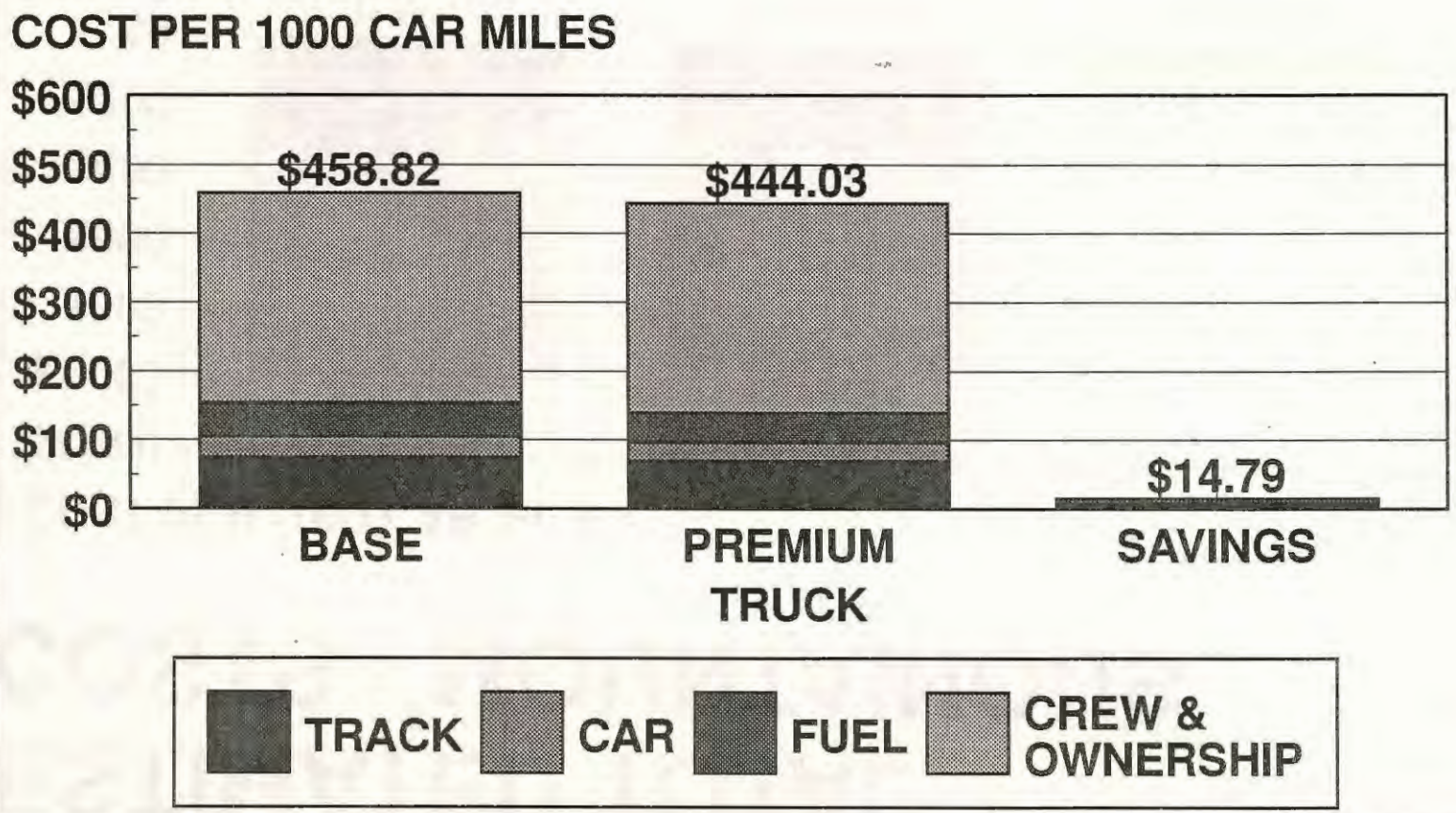
ESTIMATED TOTAL COSTS - FLAT



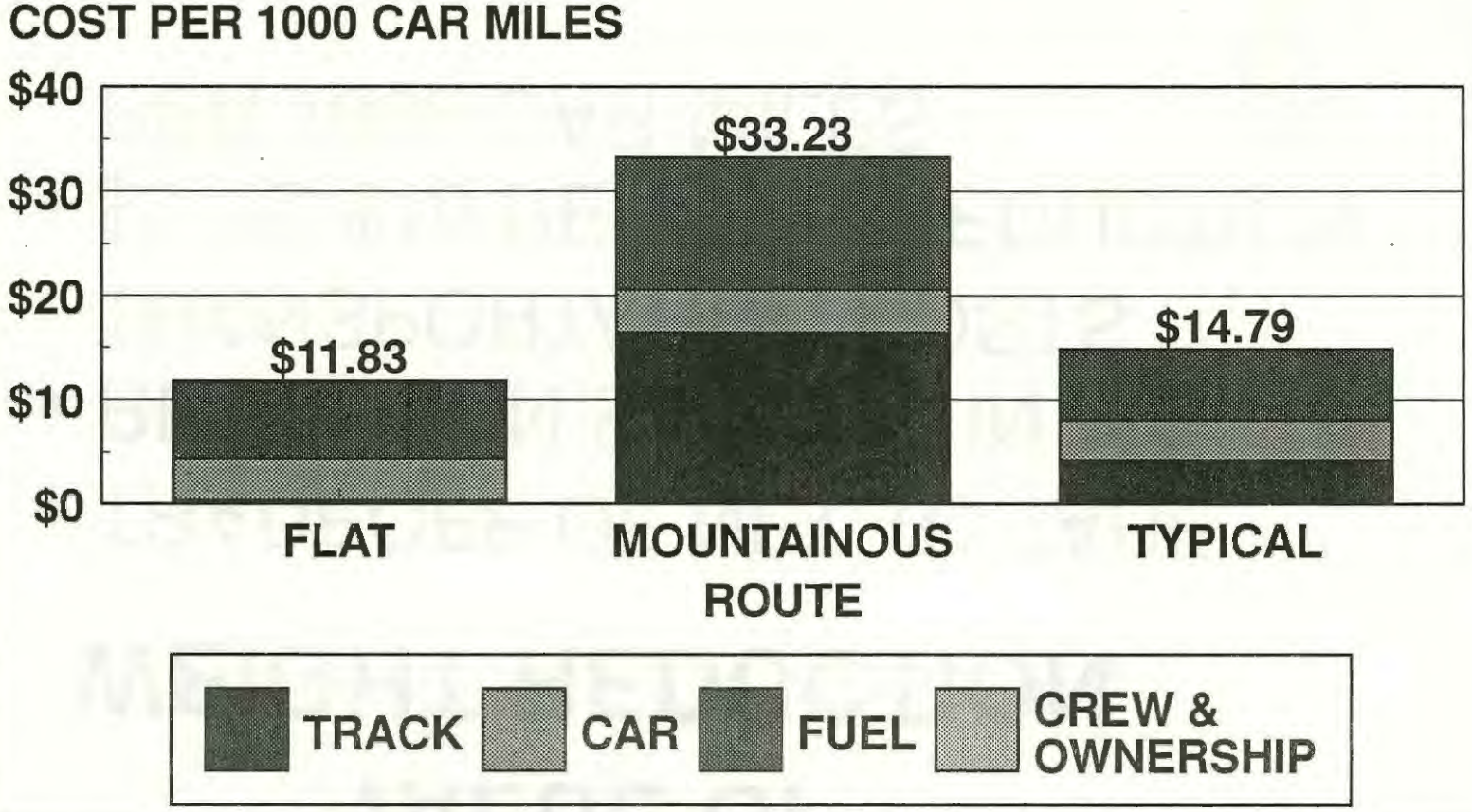
ESTIMATED TOTAL COSTS - MOUNTAINOUS



ESTIMATED TOTAL COSTS - TYPICAL

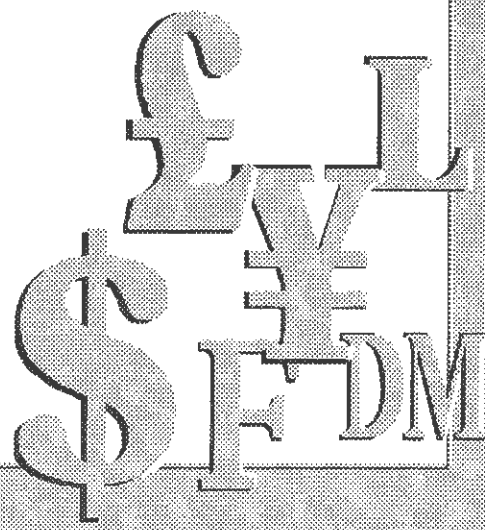


COMPARISON OF ESTIMATED SAVINGS



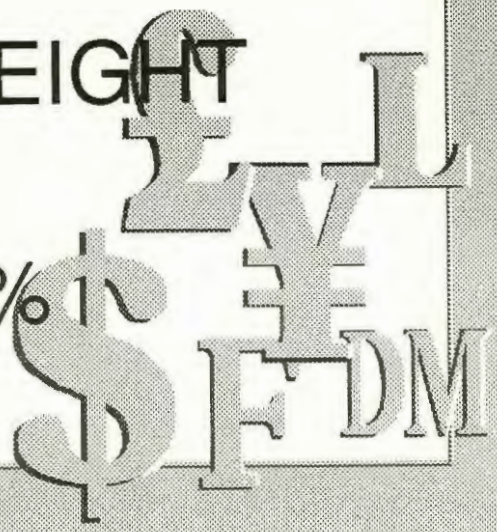
VALUE OF WEIGHT REDUCTION

- TRADEOFF OF NET TO TARE
- RESULTS IN SAVINGS IN ALL TRANSPORTATION COSTS
- $3 TO $10 PER TON REDUCTION PER 1000 CAR-MILES



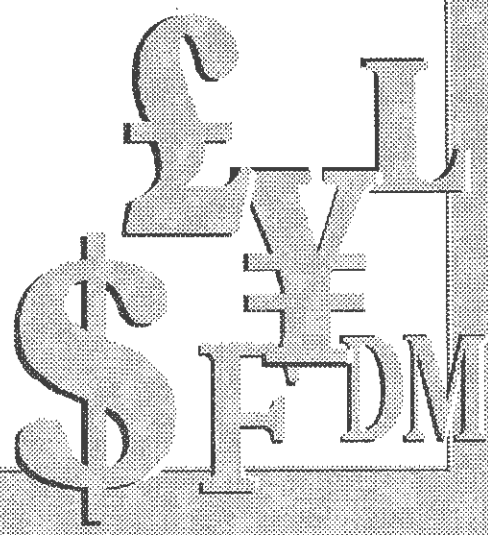
PREMIUM TRUCK EXAMPLE ANALYSIS

- "PREMIUM" TRUCK
  - 315,000 GRL CAR
  - 150% VERTICAL DYNAMIC AUGMENT
  - ONE TON PER TRUCK WEIGHT REDUCTION
  - REDUCE ACCIDENTS 50%

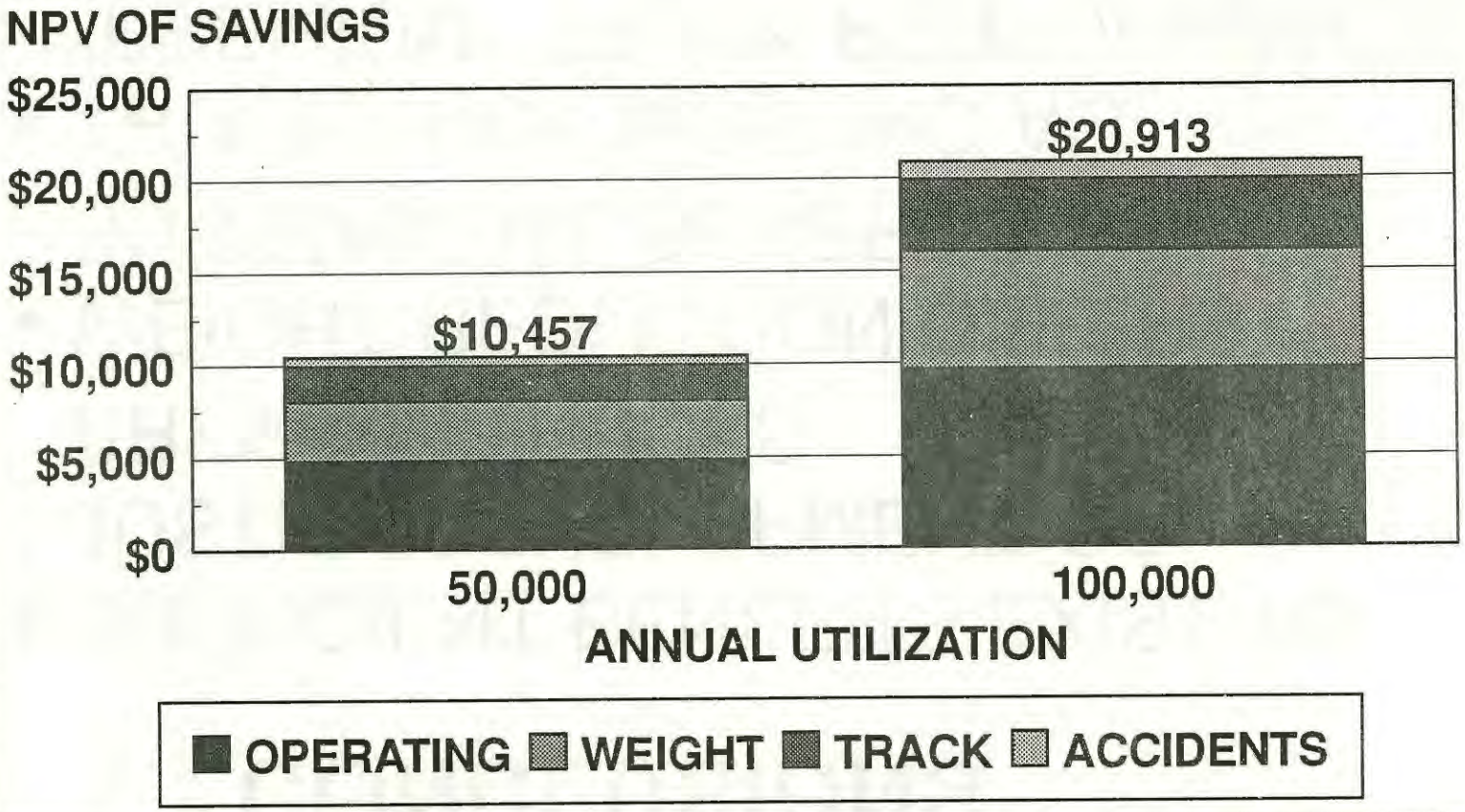


PREMIUM TRUCK EXAMPLE ANALYSIS

- TYPICAL ROUTE
- UNIT TRAIN SERVICE
- 9% COST OF CAPITAL
- TWENTY LIFE



ECONOMIC SUMMARY OF PREMIUM TRUCK



CONCLUSIONS

- SUFFICIENT BENEFITS EXIST TO JUSTIFY DEVELOPMENT OF PREMIUM TRUCKS
- WEIGHT REDUCTION OFFERS SUBSTANTIAL BENEFITS
- TRUCKS NOT MEETING ALL "PREMIUM" TRUCK PERFORMANCE LEVELS, MAY PROVIDE SIGNIFICANT SAVINGS



ADVANCED TRUCK PROGRAM

K. L. HAWTHORNE

ASSOCIATION OF AMERICAN RAILROADS


TTCI WSC CRC
Track / Train Systems
MR01.cdr

Track / Train Systems

Advanced Freight Car Trucks - Design

Objective

- To foster development of new innovative generation of freight trucks to replace the existing 3-piece trucks via performance-based design approach.

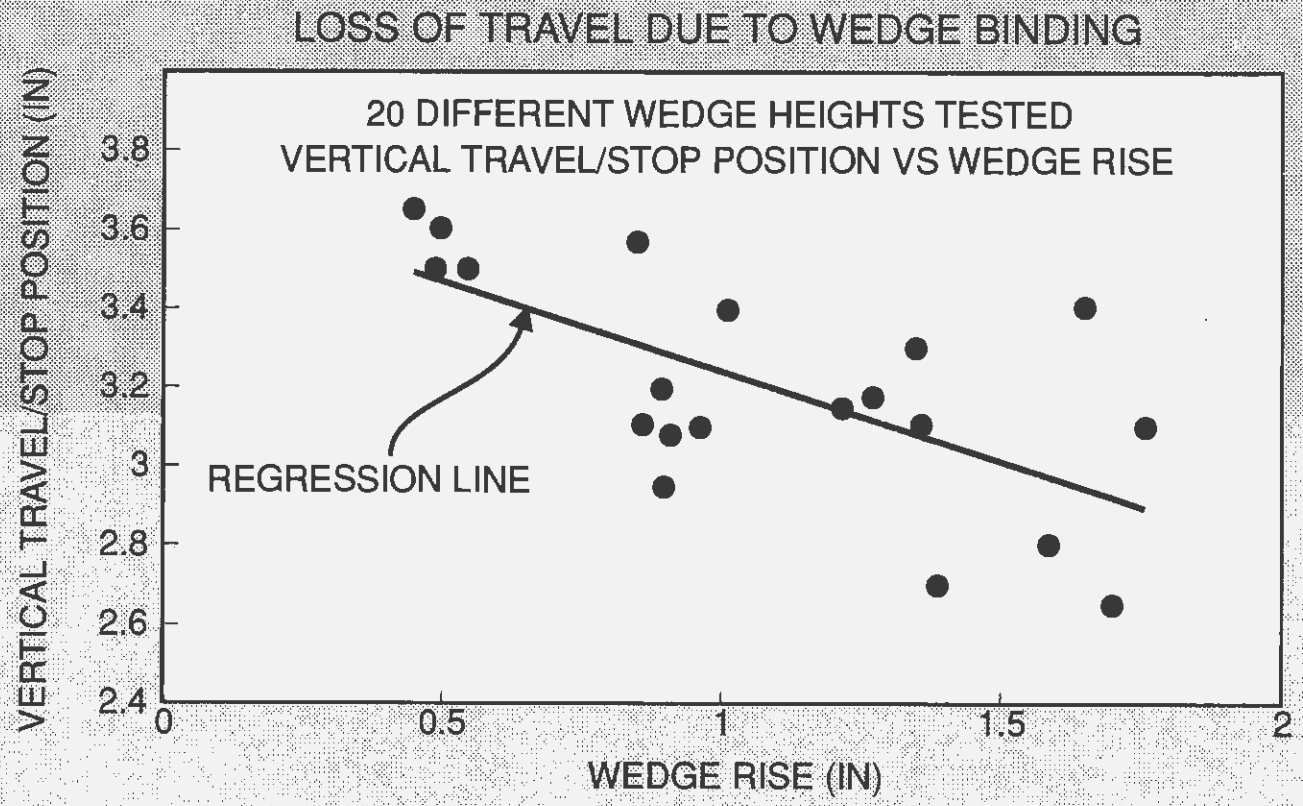
  - Problem Size: . . . . . $ 2.6 Billion
  - Research Costs: . . . . . 1995 . . . . . $ 570 K
    . . . . . 1996-99 . . . $ 1.4 Million
  - Potential Benefits . . . . . $ 95.6 Million/Yr
  - Net Present Value (NPV): . . . . . $ 20 Million

Track / Train Systems

Advanced Freight Car Trucks -Design Background

- Program 2 Research (Maintenance and Wear)
- Instrumented Truck
- Service Worn Truck Testing / Modeling

Advanced Freight Car Trucks - Design Background



Track / Train Systems

Advanced Freight Car Trucks -Design Deliverables

- 1995: Performance Criteria
  Economic Analysis Methodology
  Concepts Evaluation
- 1996: Prototype Design/Construction
- 1997: Prototype Test Evaluation/Refinement
- 1998: Endurance and Proof Testing of Successful Design(s)
- 1999: Revenue Service (Performance and Monitoring)

TT305.DRW

Track / Train Systems

Advanced Freight Car Trucks - Wayside Defect Detector

Objective

- To develop a system to detect freight car trucks with poor performance causing high wheel/rail wear and fuel consumption and likely to cause rail rollover and gage widening.
  - Problem Size: . . . . . $ 83 Million
  - Research Costs: . . . . . 1995 . . . . . $ 395 K
    . . . . . 1996-99 . . . . . $ 600 K
  - Potential Benefits . . . . . $ 26.3 Million/Yr
  - Net Present Value (NPV): . . . . . $ 60 Million

TTS06

Track / Train Systems

Advanced Freight Car Trucks - Wayside Defect Detector Background

- Program 2 Research (Maintenance and Wear)
  - Instrumented Truck
  - Service Worn Truck Testing/Modeling
- Gage Widening Derailments
  (Warped Trucks due to Wear and/or Dry Centerbowl)
- Rail Rollover Derailments
  (Worn/hollow Wheel, Grinding Practices, Differential Lubrication)
- Warped Truck
  (High Angle of Attack and High Lateral Load on Trailing Axle)

Track / Train Systems

Advanced Freight Car Trucks - Wayside Defect Detector Deliverables

- 1994: Establish AAR/Railroad/Supplier Consortium
- 1995: Technical Evaluation of Various Defective Truck Detection Systems

  Criterion/Criteria to Identify Poor Performance

  Software and Communication for Data Collection

  Design/Construction of a Prototype Detector System
- 1996: Revenue Service Installation and Monitoring
- 1997: Economic Evaluation of Detector Performance/Effectiveness Recommendation for Industry-wide Implementation