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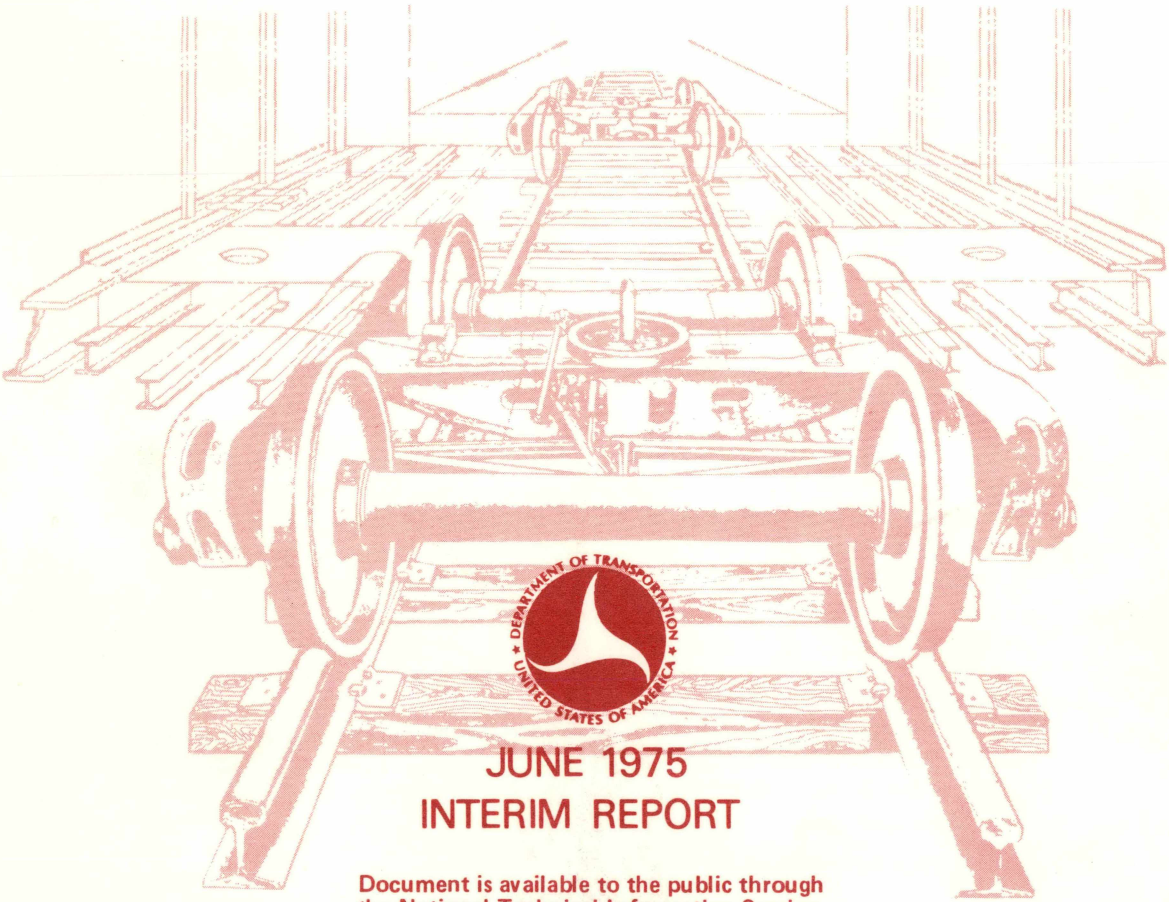
FREIGHT CAR TRUCK DESIGN OPTIMIZATION

LITERATURE SEARCH - VOLUME I

Southern Pacific Transportation Company
Technical Research and Development Group



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This interim report represents the first of a three-volume set. Volumes II and III bear the same report number with B and C suffixes.

16\. Abstract
This document serves as an introduction to the literature known to be available and relevant to rail freight car trucks, their components and performance characteristics. In connection with the Federal Railroad Administration sponsored research in Truck Design Optimization a literature search was conducted to review and assemble all relevant publications, papers, and articles. The collected documentation has been organized into five sections:

- The History of the Freight Car Truck
- Truck Design
- Truck Components
- Track-Train Dynamics as Related to Truck Performance
- Truck Performance

Each section contains: an introduction dealing with literature selected for reprinting, reprints of articles judged particularly representative or salient, and a bibliography alphabetized by author. The five sections have been organized into three-volumes. Volume I contains the sections entitled: "The History of the Freight Car Truck" and "Truck Design." Volumes II and III will complete the compilation. It is expected that supplements to the three initial volumes will be published at a later date as additional information becomes available.

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PREFACE

The Freight Car Truck Design Optimization Project (TDOP) was created to determine, through cost benefit analysis combined with technical considerations, the relative feasibility of improving truck performance either by mechanical modification of existing truck types or by technical introduction of new truck designs compatible with existing freight train systems. During the initial stages of the Project, it was necessary to review and assemble all relevant publications, papers, and articles dealing with the freight car truck. The three-volume, TDOP Literature Search is the result of that compilation.

The collected documentation has been organized into five sections:

- The History of The Freight Car Truck
- Truck Design
- Truck Components
- Track-Train Dynamics as Related to Truck Performance
- Truck Performance

Each section contains: an introduction dealing with literature selected for reprinting, reprints of articles judged particularly representative or salient, and a bibliography alphabetized by author. Certain bibliographic sections are further organized into subsections dealing with various aspects of the field under consideration. Thus, the bibliography covering truck design is organized into:

- Principles of Truck Design
- Freight Car Truck Design
- Passenger Car Truck Design
- Locomotive Truck Design

Every effort has been made to gather applicable source information for each document listed. Where available, Railroad Research Information Service (RRIS) references have been cited. A title index will be included in Volume III.

It is hoped that this literature search will be utilized as a reference guide by all those interested in the technical aspects of the freight car truck. In order to ensure both completeness and relevance, additional references will be added as they become available throughout the course of the TDOP.

CONTENTS

Volume I

Section 1

THE HISTORY OF THE FREIGHT CAR TRUCK

INTRODUCTION

The functions of the freight car truck may be categorized as follows:

- To support the weight of the car and its lading at desirable speeds
- To furnish guidance on both tangent and curved track
- To cushion forces applied between the freight car and the track
- To furnish braking effort for the freight car

Through technical alterations and refinements, truck designers and manufacturers have worked to produce trucks that perform these functions efficiently and safely under a variety of equipment and operating conditions. Another important objective in truck design has been standardization. While no industry-standard truck exists, components such as axles, wheels, and journal boxes (and their internal parts) are standardized, and the basic three-piece design is nearly universal.

The three-piece truck has been in use throughout a major portion of American railroad history. It has to recommend it: low first cost, relative ease of disassembly and repair, and, above all, simplicity. When the design was first developed, this simplicity was thought to extend to the truck's dynamic behavior as well (Ref. 1).*

* See page 1-9

The Early Trucks - From Wood to Iron

The first freight cars apparently were based on road wagons and utilized four wheels attached to fixed axles. These "burthen cars" used no springs, and the body was carried directly on the axle boxes. Later, when better riding cars were requested, flat, leaf springs were used. In an 1831 Baltimore and Ohio Railroad report, the chief engineer called attention to ride damage to a car and its lading and recommended: "that a number of burthen cars shall be furnished with springs in order to test their advantageous use on such cars" (Ref. 2).

The burthen cars were also eventually modified with pedestals attached to the side sills of the underframe. The axles were then set in boxes mounted in the pedestal jaws. However, the size of these cars was restricted by the upper length limits of the rigid wheelbase that could safely negotiate the maximum track curvature then in use (Ref. 3).

The center-bearing or swivel truck was introduced in America at some point during the first five decades of railroad development. These four-wheel trucks were remarkably similar to modern designs: "Two of these trucks were placed under each car body, several feet from the end. They were connected to the car body by means of stout 'king pins' centered in round bearing plates attached respectively to the car structure itself and the truck bolster," (Ref. 3). These trucks, through their shorter wheelbases, allowed freight cars to negotiate curves that would have been unsafe with the older wagons with rigid underframes.

The principal construction materials of the first trucks were wooden beams. At a later stage of development, these wooden beams were trussed for added strength, and some trucks were actually built up from iron bars (Ref. 4).

From Cast Iron to Pressed Steel

The "Diamond" or "arch-bar" truck was introduced in the latter part of the nineteenth century. This truck derived its name from its side frames, which were characteristically manufactured from bar iron in a bowed or "arched" design. The axleboxes were attached by means of tie bolts, and the sideframes were connected by means of a spring plank. The bolster unit was mounted in pedestal guides.

The arch-bar truck has been called the "father" of the modern four-wheel truck design (Ref. 1). At the time of its introduction, the truck was considered a great advance over existing designs in that it was easier to build and repair, and could be constructed entirely of metal (Ref. 5). Under the loads and conditions for which it was designed, the arch-bar truck performed well and was in wide service on American Railroads until the late 1920's. Over the years, the arch-bar truck was improved through the substitution of rolled steel beams for the wood originally used in the truck bolster. The spring planks were also modified, and the cast iron column castings were replaced with cast steel models (Ref. 4).

In 1890, the Fox truck was introduced. This truck was constructed of pressed steel components riveted together to form a rigid, one-piece structure. Since the bolster was integral with the side frames, springs were mounted in pedestals over the journal boxes.

The simplicity of the Fox truck contrasted sharply with the arch-bar design, and engendered initial industry interest. However, the Fox truck proved difficult to maintain in that it was too rigid, lacked adequate strength, and was subject to rapid corrosion. All of these defects were compounded by the impossibility of maintaining the integrity of the riveted construction (Ref. 4).

At this point, there was an obvious need for a more simple and rugged truck design. With the advent of high-quality, low-cost cast steel, the steel manufacturers provided the solution to the problem.

Cast Steel Trucks

The first use of cast steel in car construction came in the form of truck bolsters manufactured for the Missouri Pacific in 1893 (Ref. 3). This initial application utilized the inverted "U" section that later became standard in the industry. However, competing manufacturers, in the interest of avoiding weight, cost, and manufacturing difficulties, subsequently utilized "I," "T," and "L" section castings for bolsters as well as for side frames. It was not until the World War I era that the "U" section casting came into universal use (Ref. 1).

In 1903, William Bettendorf patented the first cast steel truck side frame with integral journal boxes (Ref. 3). The Bettendorf side frame gradually won acceptance over competing designs which had separate, malleable iron journal boxes. This design was so successful that Bettendorf was forced to "farm out" a significant portion of their work to other railroad foundries during World War I (Ref. 1).

Other early cast steel side frame designs included the Andrews type, which was introduced by American Steel Foundries (ASF) in 1904, and the Vulcan side frame, introduced by ASF in 1910. Both of these designs had separate journal boxes, but the Vulcan featured pedestal jaws that simplified the connection (Ref. 6).

In 1914, ASF began a freight car truck dynamics testing program. The subject car was a Pennsylvania Railroad H-21 hopper. Forces exerted on the side frames by the bolster and spring plank were measured as the car ran over tangent and curved track. By the early 1920's, both ASF and

Symington Gould Corporation were utilizing dynamic testing machines (Ref. 3). As the significance and complexity of the dynamic forces at work on the freight car truck became known, manufacturers began to make significant improvements in their cast steel designs. When the integral journal box patents expired in the 1920's, all side frame manufacturers developed units utilizing the Bettendorf concept. During the same period, the "U" section design also became standard.

In 1927, the major cast steel side frame manufacturers, at the suggestion of the General Counsel for the Western Railway Association, agreed to pool existing patents pertaining to conventional truck designs. The result was the Four-Wheel Truck Agreement through which the double-truss, self-aligning, inverted tension member, spring-plankless truck was evolved. This truck was utilized extensively through the 1930's. The self-aligning feature of this truck was thought to be essential at the time of its introduction but was later rejected in favor of the rigid "trammed" truck (Ref. 1).

Another significant development in 1927 was the first in a series of steps taken by the American Railway Association to eliminate the arch-bar truck from service. The increasingly rigorous service conditions were, by this time, exacting a heavy toll in arch-bar truck maintenance and failures. The first official action took the form of a rule requiring cast steel side frames conforming to A. R. A. codes on all cars constructed on or after July 1, 1928. By 1930, the hazards posed by running the old arch-bar design under "new" service conditions demanded further action.

Further action was taken in the form of a regulation banning the arch-bar design from all service by January 1, 1936. A report filed in 1935 states concerning the 1930 regulation: "The move was not made hurriedly or unadvisedly, for the records of the Interstate Commerce Commission reveal that for the four years ending with 1931 an average of 588 wrecks per year took place on American railways due solely to arch-bar trucks," (Ref. 4).

Throughout the 1930's and 1940's, extensive spring and snubber testing was conducted in an effort to improve ride quality and to curb the incidence of spring failures and tank car derailments. In 1931, for example, both ASF and the T. H. Symington Company were performing studies on ride quality. In an article published in Railway Mechanical Engineer in January of 1931, T. H. Symington stated that: "Spring oscillation is probably the largest factor in damage to perishable freight and to truck springs," (Ref. 3).

In 1939, the Association of American Railroads (A. A. R.) conducted a series of tests to evaulate the high-speed performance characteristics of a number of contemporary freight car truck designs. The trucks were run under varying loads at speeds up to 85 miles per hour (137 kilometers per hour). None of the designs tested performed adequately at speeds above 60 miles per hour (96.8 kilometers per hour). Among the study conclusions were the following:

- "...Snubbers, which have proven helpful in solving the problem of oscillation at normal freight train speeds, are of little if any value at the high speeds reached in these tests."
- "These tests show that soft springs are one of the requirements for a satisfactory high speed truck."
- "Proper lateral stability is very important from a standpoint of safety of operation at high speed (Ref. 7)."

In 1951, the A. A. R. published the Summary Report for the Cooperative Freight Truck and Snubber Research Program. Like the 1939 study, this program had a significant impact on freight car truck development. During a three year period, a number of trucks and snubber units were road tested utilizing American Steel Foundries' "5-Car Service Laboratory" on the Illinois Central Railroad. The report recommended the following:

- "All existing freight car trucks which are not equipped with snubbing devices should be improved by incorporation of those devices found to provide satisfactory performance."
- "All new freight cars should be equipped with the High-Speed Trucks found to provide satisfactory performance."
- "Each railroad should set up a schedule for examination and replacement of worn friction parts of the snubbing devices in the trucks (Ref. 8)."

In designs such as the previously mentioned "double truss" truck, coil-elliptic spring groups were used in an effort to combine the load capacity of the coil spring with the damping properties of the elliptic type. As more performance data became available through tests and studies, friction dampers or snubbers were applied to various truck designs, and, during World War II, trucks appeared with long-travel springs and built-in snubbers. These models were the forerunners of trucks now in general use, such as the Barber S-2, ASF Ride Control, and National C-1 models (Ref. 1).

Among the more recent developments is the use of tapered roller bearings, which gained wide acceptance in the late 1950's and early 1960's. While these devices have reduced both maintenance costs and "hot box" problems, some authorities feel that the roller bearing represents a trade-off between lower maintenance requirements and impaired truck performance through the elimination of necessary wheelset lateral (Ref. 1).

The Dynamic Challenge To Modern Truck Designs

Since the 1930's, significant progress has been made in freight car truck design and development. But the basic method of proving new models has been, and continues to be, trial and error (Ref. 1). While some new models have been tried and proven in field tests by interested railroads, many others have fallen short of general acceptance.

Meanwhile, the railroads have introduced many new car designs including long, flat cars for loading automobiles, truck trailers, and containers; 100-ton (90.9-mt) and 125-ton (113.6-mt) bulk commodity hopper cars; all welded, center sill-less tank cars; and high-cubic-capacity cars for hauling low-density commodities. These new mechancial designs have brought about higher wheel loadings, higher centers of gravity, greater curving forces, and increased train action. The introduction of unit trains and truck on flat car/container on flat car services have significantly increased the use and mileage of some truck types. New freight car designs also have made greater use of their cubic capacities, thereby achieving higher net-to-tare ratios. All these factors have conspired with various track and service conditions to push the conventional truck to the limits of its performance capability.

Mr. Loren Smith, in a presentation entitled The Freight Car Truck "Capability" Gap, states: "...we appear to be reaching a point where the gamble of the cut and try method of new truck development is no longer acceptable due to the high costs involved and the increasing uncertainty of the outcome. Speaking perhaps for most truck designers we are not at all certain as to the dimensions of the problem. So many changes have taken place and each interacts with and affects the others to the extent that a renewed understanding of today's basic design parameters is now necessary," (Ref. 1).

As Mr. Smith points out, it seems the time has come to write another chapter in the history of the freight car truck, either through the modification of existing designs or through the substitution of compatible alternative designs.

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7. Association of American Railroads, Tests of Trucks for High-Speed Freight Service, Operations and Maintenance Department, Mechanical Division, Office of Mechanical Engineer, April 1940.

8. Association of American Railroads, Summary Report of Road Tests - Cooperative Freight Truck and Snubber Research Program and Related Freight Car Research Programs, Report No. f3800, Operations and Maintenance Department, Mechanical Division, Mechanical Research Department, June 30, 1951.

Reprint and Bibliography

The following subsections are comprised of a reprinted article, which was selected from the assembled literature concerning the history of the freight car truck, and the TDOP truck-history bibliography.

A portion of the report entitled, A Brief History of The American Freight Car, has been selected for reprinting. This chapter provides an overall history of the American freight car truck with special emphasis on the contributions of American Steel Foundries.

The bibliography, which follows the reprint, is arranged in alphabetical order by author. Supplemental pages will be added to the "Bibliography--History," as new articles become available throughout the course of the TDOP.

CHAPTER 2

Development of the modern freight car truck

The modern high-speed freight car truck is one of the most efficient and economical structures in the transportation field, its efficiency and economy representing the result of more than half a century of development work. In the early 1900s a rugged yet simple basic cast steel structure was evolved; and in recent years so much progress has been made in improving the riding qualities of the truck, that today goods may be transported in railroad freight cars at speeds exceeding 90 miles per hour with relatively little damage.

Objectives in Truck Development

Over the years, a number of major objectives have been sought in the development of the 4-wheel swivel truck. The first objective was *strength combined with simplicity,* for early freight car trucks made of a multiplicity of parts were no match for the growing burdens they had to carry. The second objective was *durability,* as early trucks were short-lived and suffered frequent break-downs. The repair problem became serious, not only in expense, but in delays and interruptions of the freight service. A third objective was *safety*—as trucks were needed which could operate safely at continually higher rates of speed.

A fourth objective was *standardization.* While no "standard truck" design has been adopted, certain components such as axles, wheels, journal boxes (and their contained parts) have been standardized. Furthermore, each freight car truck must conform to certain standard physical requirements and dimensions *so as to be interchangeable as a whole* with any other truck meeting the same requirements.

Finally, a fifth objective has been to improve *riding quality,* for as strength, durability, safety and standardization were built into the truck, it was found desirable to improve the riding quality especially at the high speeds that have become common in the last few years in railroad freight service. Hence, the last episode in the present chapter presents the concentrated attack on riding quality which has been dramatized by road testing of freight cars involving speeds of 95 to 100 miles or more per hour.

While seeking all of these objectives, a most important consideration has been to keep the cost down. As has been noted in the previous chapter, the great strength of railroad freight transportation in the face of growing competition is its inherent *economy.* Hence, it is important that continued progress in freight car truck efficiency, riding quality and ease of maintenance be achieved without excessive cost.

Much effort has also been directed toward reducing the dead weight of the freight car and the freight train. There is still much room for progress here; and while there is undoubtedly greater opportunity for weight reduction in the car body than in the "running gear"; nevertheless, lighter weight is a goal gradually being achieved in the production of freight car trucks. At present, the two 4-wheel trucks which support a typical 50-ton freight car weigh about 14,000 pounds, which represents about 1/3 of the total weight of a box car. Wheels and axles alone account for about 8,000 pounds of the total. Some of the possibilities for reducing the weight of a freight car truck include the use of tubular, instead of solid, axles and substitution of lighter-weight, high-tensile steel for so-called "Grade B" steel in the side frames, bolsters, and other vital parts of the truck. Again, the question of economy enters, for high-tensile alloy steels are more expensive than the conventional "Grade B" steel.

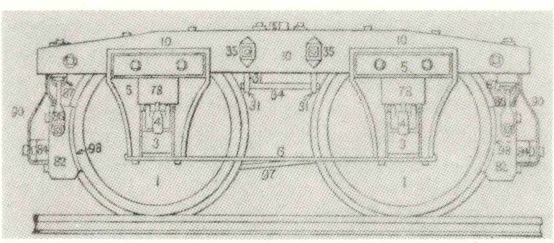
The efficiency of the modern freight-car truck is evident when it is realized that the trucks of a 50-ton car weighing about 14,000 pounds are capable of transporting a total of 155,000 pounds of *gross weight,* made up of the car body weighing 45,000 pounds, plus the weight of the lading, which can be as high as 110,000 pounds. Hence, the truck including the wheels and axles, carries 11 pounds of load per pound of weight. The distribution of weight on various parts of the truck is even greater. It will be found that the cast steel truck bolster in a modern 50-ton freight-car truck actually supports about 90 pounds per pound of its own weight and the cast steel side frame about 65 pounds per pound of weight.[1]

From Fixed Axles to Swivel Trucks — 1825-1850

The earliest railroad freight cars were apparently directly patterned after road wagons and as such used only four wheels per car, attached to fixed axles. The freight cars on the early American railroads, like those in Europe, had running gear very similar to the 4-wheel English colliery wagon with inside journals and without springs. Subsequently, pedestals were attached to the side sills of the underframe and the axles rotated in boxes fitted in the jaws of the pedestals. Before the introduction of springs, the shocks picked up from the track surface were transmitted directly into the underframe. Furthermore, the length of the original car bodies was limited by the length of rigid wheel base that could be safely accommodated by the maximum curvature of the track.

H. S. Haines pointed out in 1919 that one of the most important American contributions to the art of railroad transportation was the adaptation of the center-bearing or swivel truck "from the long-coupled wagon swiveling on a king bolt". (See Chapter I, *supra.*) By means of this device, which entered American railway practice during the first half century of railroad development, a great advance in efficiency was accomplished since the swivel truck permitted the construction of freight cars of greater length and loading capacity than was possible with the previous fixed-axle design.[2] The swivel truck consisted of two side frames placed parallel to the tracks and separated by a member called a truck bolster. Two pairs of wheels mounted on axles and confined at adjacent ends of the side frames gave mobility to the truck. Two of these trucks were placed under each car body, several feet from the end. They were connected to the car body by means of stout "king pins" centered in round bearing plates attached respectively to the car structure itself and the truck bolster. In this way, the previous, long, rigid wheel base was shortened sufficiently to traverse curves which were unsafe for the older type of running gear.

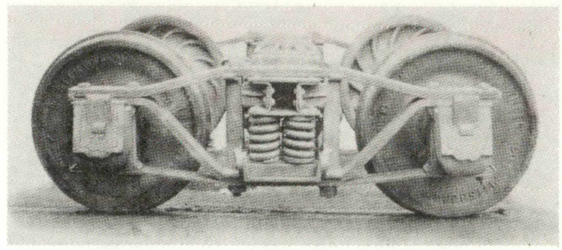
The principal structural portions of the first freight car trucks were heavy wooden beams which were later trussed for added strength and, in some cases, later were built up from wrought iron bars.[3]



This early type of wooden four-wheel swiveling truck was first used in 1831.

Arch-bar or "Diamond" Truck — 1876

About 1876 the so-called "Diamond" or "Arch-bar" truck was introduced. According to J. H. Ames, Chief Engineer of American Car and Foundry Company, writing in the year 1906—the "Diamond" truck was easy to build and was easy to repair and could be constructed entirely of metal. According to Ames, "The main point to be observed in designing a truck of this type is to keep the rise of the top arch-bar within the limits of 1½″ and 3½″ as, if made too high, there is great tendency to buckling between the bolts, the bar being wholly in compression; this also increases the strain on the column bolts, which may be termed the keystone of the side-frame, since, when they fail, the truck goes down."[4]

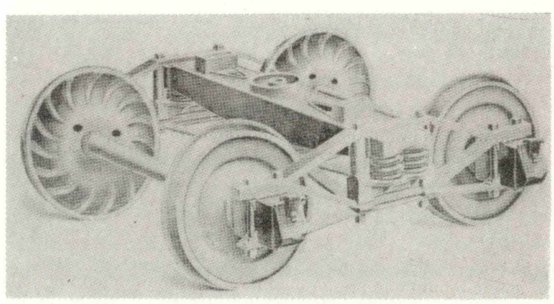


Arch-bar truck with fabricated bolster.

Refinement of the arch-bar truck consisted principally in the substitution of rolled steel beams for the wooden beams of the truck bolsters and the substitution of angles or channels, or the combination of such shapes, with wooden planks for the spring planks. Column castings, originally made entirely of cast iron, were gradually changed over to malleable iron and later to cast steel.[5]

L. F. Loree, in his book *Railroad Freight Transportation,* published in 1922, commented: "The arch-bar truck was an essential type in the early days of railroading when most track was in, what we would now deem, impossibly bad surface, and when much depended upon it to accommodate the movement of the truck thereto. With the modern heavy car, trouble resulted from frequent breakage and difficulty of maintenance."

As early as 1884, at the Convention of the Master Car Builders Association, the "Committee on a Standard Passenger and Freight-Car Truck" suggested three types of freight-car trucks. Two out of the three designs proposed the use of steel castings for the side

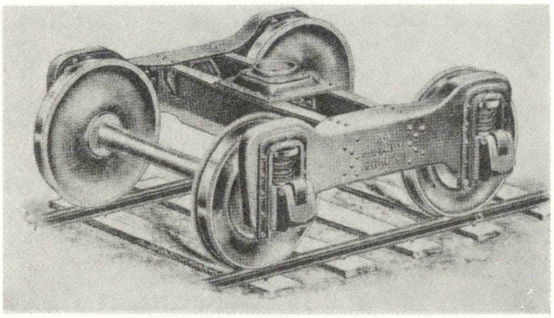


Arch-bar truck with cast steel bolster.

frames, with the committee concluding that "the use of cast steel having a tenacity superior to wrought iron and equal ductility makes it possible to design a truck which shall have correct shape for strength with minimum weight and least number of parts." In spite of this idea expressed by the Master Car Builders' Committee, it was a great many years before steel foundry practice was sufficiently advanced to produce commercially satisfactory side frames. In 1890, the typical freight-car truck had arch-bar side frames and a wooden bolster. Wood was also employed in the underframe and the car body, although the body bolster was frequently steel-plated.[6]

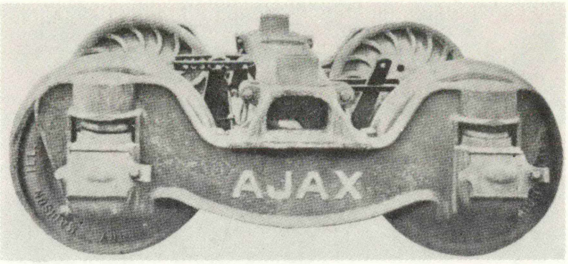
Pressed-steel Trucks Introduced in 1890

About 1890, the Fox pressed-steel truck was introduced and attracted considerable attention. In this truck the side frame members and the bolster were rigidly riveted together, making the entire truck of a single structure. The Fox truck was a pedestal type of truck with the springs over the journal boxes. There was appeal in the simplicity of this truck in comparison with the arch-bar type. However, it had a number of serious defects, including lack of flexibility, the impossibility of keeping rivets tight, rapid corrosion and lack of adequate strength. These defects ultimately caused its elimination.[7]



The Fox Pressed Steel Truck.

The steel freight car was beginning to be actively discussed and, at the Master Car Builders' Annual Convention in June, 1903, T. E. Adams of the St. Louis Southwestern commented with regard to some of the difficulties he had experienced. It was his opinion that the steel car of the time was far from satisfactory and that one of the main difficulties was the roll-pressed steel truck used on many of the cars. To repair such a truck required taking it all to pieces, straightening and reriveting.[8] Obviously the railroads in 1903 were in need of more substantial freight car trucks of simple design. The steel foundries were soon to meet this need.



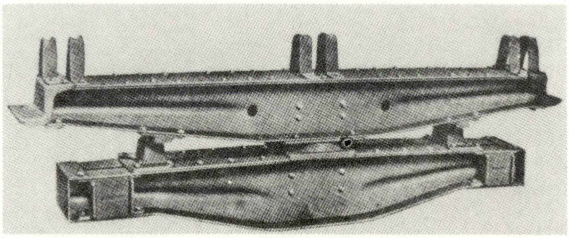
The Ajax Truck, introduced about the same time as the Fox Truck.

Early Use of Cast Steel in Freight-car Trucks

The first steel castings used in railroad cars were truck bolsters for Missouri Pacific cars built in 1893. The first truck bolsters offered in cast steel were of a design known as the *inverted "U" section.* The advantages of cast steel construction were quickly demonstrated.

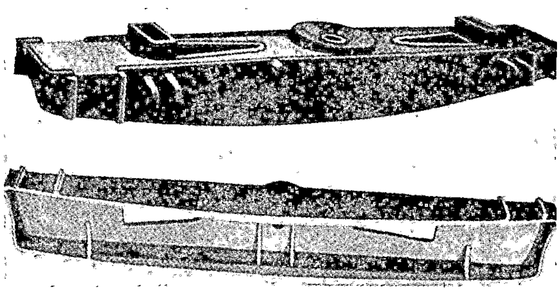
In 1906, J. H. Ames, of American Car and Foundry Company, stated that:

> In cars constructed wholly of wood it will be found that cast steel is the best material for bolsters, permitting them to be made of a single piece; whereas, when built up of plates and castings, the rivets or bolts become loose, permitting them to deflect and come down on the truck side-bearings. The result is that the trucks will not pass easily around curves, producing excessive wear on the flanges of the wheels and heads of the rails, and increasing the resistance to be overcome by the locomotive, thus causing a greater consumption of fuel.[9]



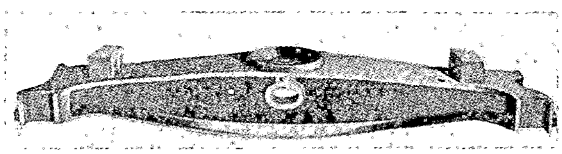
Structural steel body bolster and truck bolster.

The first inverted "U" section truck bolsters made of cast steel were designed with the top and sides of the bolsters cast solid with large bulbs along the bottom inside arch of the side members. An oak block set into the open bottom side of the end of the truck bolster was generally used for the spring seat. This design represented a compromise between foundry practice and the existing pressed steel or built-up structural bolster design.

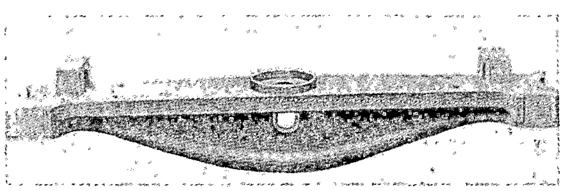


"U" section cast steel truck bolster.

The idea of a single piece bolster was attractive to several manufacturers and it is reported that competition between these manufacturers as well as competition between the cast steel bolsters and pressed steel bolsters caused the early designers to attempt unwise extremes in the reduction of weight. They also were forced to adopt sections which could be manufactured easily in order to accomplish a price advantage. This is the explanation of the early "I" section truck bolster and a subsequent "T" section, neither of which designs were entirely satisfactory. It was not until the *box section* became generally used in the production of cast steel truck bolsters that the full possibilities of the use of steel castings in this particular item were to be realized.[10]



"I" section cast steel truck bolster.



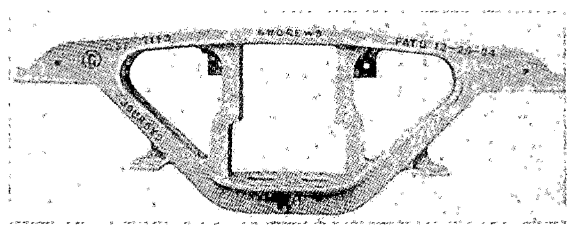
"T" section cast steel truck bolster.

The development of the cast steel side frame dates from about 1900. At first the separate journal boxes used with the arch-bar frame were retained, together with the tie bar or bottom member of the truss. The next structural development brought the elimination of the full-length bottom tie member and substituted short tie straps extending from under the journal boxes to brackets cast on the tension member of the side frame and riveted onto it. However, the separate journal boxes were a source of annoyance and also increased maintenance cost because of the tie bars and necessary attaching bolts.

The integral-box side-frame design was also proposed about 1900, but the art of steel casting had not, by that time, progressed sufficiently to make a satisfactory casting.[11]

Bettendorf and Andrews Cast Steel Side Frames — 1903-1904

The Leighton and Howard Steel Company, one of the original companies included in the organization of American Steel Foundries, had actively proposed the use of cast-steel freight-car truck side frames and bolsters as early as 1901. William P. Bettendorf patented the first cast-steel truck frame with integral journal boxes in 1903, and in 1904, American Steel Foundries introduced the original cast-steel "Andrews" type truck side frame.[12]



The "Andrews" type cast steel side frame.

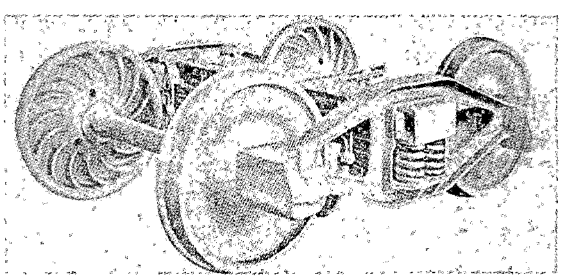
The "Andrews" type of side frame made use of separate journal boxes. This design was considerably easier to produce and this advantage, coupled with the desire of the railroads to retain the idea of separate journal boxes, promoted the use of the "Andrews" type frame more rapidly than the integral-box type. For the sake of greater convenience in the foundries, the sections of the members were changed from "I" to "L" or angular sections, and next to "T" sections. Finally, because of design superiority, the members were changed to "U" or channel sections. Early cast steel bolsters and side frames sometimes did not give a service life greatly in excess of the types they replaced. This was due to the lack of specific knowledge of what side frame and bolster structures were called upon to do in railroad service. There is no question, however, that these simple structures had appeal through the elimination of innumerable small parts, bolts, nuts and rivets, which had been constant sources of maintenance expense as well as safety hazards.[13]

American Steel Foundries' Truck Tests Begin in 1910

American Steel Foundries was only eight years old when it was decided to establish a product engineering research and development department at the Granite City Works to explore the behavior of existing freight-car trucks and try out new improvements in design.

At that time, nothing was known of the forces existing in the various truck members as the trucks negotiated curves. To measure these forces and in order to determine the frictional resistance of "loose" and "square" trucks, American Steel Foundries' engineers built a unique inclined track arrangement containing both curved and straight tracks on the grounds of the Granite City Works. This machine was referred to as the "scenic railway" and the manner of operation was simply to pull trucks to the top of the incline and observe their performance as they rolled down the slope and around the curves.[14]

Also in the year 1910, American Steel Foundries introduced a very successful type of truck side frame with the trade name "Vulcan". This side frame was similar to the "Andrews" and "Bettendorf" types except that it had pedestal jaws engaging the journal boxes, thereby eliminating the vertical journal box bolts which were commonly used in the arch bar and Andrews trucks. The popular A. S. F. "Vulcan" side frame was simply supplied with a single small bolt to secure the journal box to the side frame by means of projecting lugs on the top of the box.[15]



Truck with "Bettendorf" side frames designed with integral journal boxes.

A pronounced forward step in the design of side frames and bolsters came in 1912, when American Steel Foundries first adapted the Berry Extensometer or strain gauge for *static testing* of side frames and bolsters. This instrument was a great help in the determination of stresses since it permitted measurements as exact as two ten-thousandths of an inch in the deformation of the steel casting under various loading conditions.[16]

Dynamic Road Testing of Trucks Begun by American Steel Foundries in 1913 and 1914

While static vertical loads borne by truck members were easily measured and generally understood during the early stages of cast steel development, there was little definite information on the extent of dynamic forces acting on these parts. *The measurement of dynamic loads* on side frames and bolsters in actual road service actively began at American Steel Foundries in 1913. It was soon found that while designs were amply strong for static loads, failures occurring in actual service could only be attributed to fatigue from dynamic loads which were considerably higher.

In cooperation with the railroads, American Steel Foundries provided special truck castings and adequate testing apparatus for what has been generally recognized as the first of all test cars—a Pennsylvania Railroad standard H-21 hopper. This car was first tested in 1913 in the Allegheny yards of the Pennsylvania Railroad and in 1914 was placed in service on a local freight train running from Pittsburgh, Pennsylvania, to Alliance, Ohio. Later many test runs were made in a train to Altoona, Pennsylvania, where very sharp curves were encountered. The purpose of these tests was to determine the forces on a truck side frame during actual operating conditions. Various instruments were also employed to make a record of the force exerted by the bolster as it contacted the side frame, and of the spring plank as it reacted to twist the side frame as well as the variations in the vertical load as the car rolled along the track. These first road tests conducted in 1913 and 1914, were the forerunner of the present five-car test train—the American Steel Foundries' "Service Laboratory" which, of course, represents a great advancement over the measurement techniques used in 1913 and 1914.[17]

As a result of the knowledge gained from *dynamic tests* and the use of the Berry Extensometer, American Steel Foundries in 1914, were the first to introduce cast steel side frames having channel or "U" sections throughout, except for a box section under the spring seat. This major improvement made its advent as an "Andrews type" side frame. However, this was considerably ahead of its time as the "U" section was not adopted in the integral journal box side frame until much later when American Railway Association test specifications made its adoption in new designs almost imperative.

Static test and strain gauge analyses made between 1914 and 1918 clearly indicated the superiority of the "U" section side frame. Various designs began to appear in which the tension member as well as the compression member were made in "U" sections, although this increased the cost of manufacturing due to additional coring and other complications involved in production. In 1914, the Andrews side frame incorporated "U" section members and in 1915, Vulcan side frames were also produced incorporating "U" section members in place of the former "I" and "T" sections. By 1917, the demand for integral-box side frames had increased and following the early action taken in the case of Andrews and Vulcan types, the manufacturers adopted "U" section members to improve the strength of the design.[18]

Railroads Invite Truck Manufacturers to Cooperate for Standardization

While all of the testing described in the foregoing was going on between 1914 and 1918, the subcommittee on Designs and Specifications of the American Railway Association's Committee on Standard Freight Car Equipment took steps to bring about more active cooperation between the railroad committees and the various truck manufacturers in the direction of truck standardization. Some time during the fall of 1915, the Railroad Subcommittee invited four leading manufacturers of cast steel side frames to cooperate in designing a standard side frame for proposed standard box cars. The companies involved were American Steel Foundries of Chicago, Illinois, the Buckeye Steel Castings Company of Columbus, Ohio, Gould Coupler Company of Depew, New York, and Scullin Steel Company of St. Louis, Missouri. During 1916 and 1917, a committee consisting of the chief mechanical engineers of these companies met frequently with the A. R. A. subcommittee and carried on extensive correspondence with respect to railroad design specifications and with respect to the ability of the truck manufacturers to meet these specifications. When the Government took over the American railroads in 1917, the United States Railroad Administration joined in a three-way cooperation along with the A. R. A. and the committee representing the truck manufacturers, with the result that for many years thereafter the "U. S. R. A. truck design" was regarded as something in the nature of a standard.

It is said that probably to this time, the best examples of constructions resulting from static tests and service trial tests were the full "U" section Andrews side frames and box section truck bolsters used on 100,000 cars built by the U. S. R. A. in 1918, when the railroads were under Federal control. These designs represented the first joint effort of the truck manufacturers cooperating as a group with railway mechanical committees—the designs having been approved by both an A. R. A. committee and a U. S. R. A. committee. The specifications for these designs were worked up jointly and were the first general specifications covering cast steel side frames and truck bolsters.[19]

Fatigue Testing of Side Frames Begins in 1921

The year 1921 marks the introduction of a revolutionary new technique in the development and testing of cast steel side frames. Experience in road service tests, which had been conducted since 1913, indicated the desirability of designing fatigue testing equipment that would impose on the side frames repetitive vertical, lateral and torsional loads as determined in service testing, but increased in magnitude to produce a rapid failure in the laboratory in a few days equivalent to many years of actual service. American Steel Foundries' engineers designed and installed the first testing machine for scientifically studying the effect of repetitive dynamic loadings on full-sized side frames. The machine subjected the side frame to loads, reversals and compoundings of stresses experienced in service. This machine, the first of its kind ever built, has been in constant use since its installation in the Granite City Works of American Steel Foundries in 1921. The results of the program were immediate and revolutionary and were followed by a period of rapid progress in the art of designing stronger and more serviceable cast steel truck side frames.[20]

In a paper prepared for the American Society of Mechanical Engineers on the development of side frames and bolsters for freight car trucks, D. S. Barrows, Vice President, Symington-Gould Corporation, commented on the accomplishments of the dynamic testing machine as follows:

> In due course it began to be appreciated that it was not possible to establish a fundamental design of a truss side frame simply by means of a static test. Side frames do not fail in service because of the slow imposition of a load several times as great as they will ever be called upon to carry. When they fail, it is while carrying static loads well within their theoretical capacity under conditions of constant vibration and impact. In order to test side frames under conditions more nearly approximating actual service, the American Steel Foundries in 1921 built a dynamic testing machine at its plant in Granite City, Illinois.
>
> By the use of this machine, it was possible to determine definitely on specimen frames the location and progress of the first crack, the first critical crack, and the crack leading to failure.[21]

As data were accumulated from the study of existing designs of side frames, the product was rapidly improved, providing a balance of conditions dictated by service requirements and foundry techniques with due consideration of the most economical use of material.[22]

Samuel O. Dunn, Editor of *Railway Age,* in an article dealing with research in railway development paid tribute to the contribution of American Steel Foundries and other manufacturers to both the safety and economy of railway operations:

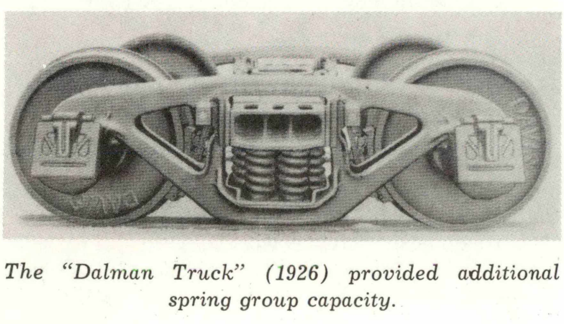
> In the important field of car trucks and parts, individual manufacturers have done most of the research and experimental work to date. For example, the cast-steel side frame was developed by the use of dynamic testing machines in the laboratories of two manufacturers [American Steel Foundries and Symington-Gould Corporation]. With integral journal boxes, this type of frame has effected a striking increase in safety, combined with an equally important reduction in truck maintenance expense.[23]

Another development reported in 1921 was the cooperative effort of a subcommittee of the American Railway Association Car Construction Committee and the Truck Committee of the Manufacturers Association engaging in the design of cast steel side frames and bolsters. It was apparent that the time was not ripe for adopting a single truck design as standard for uniform application on the part of the railroad industry. Consideration was given to various previously recognized M. C. B. dimensions of side frames and bolsters and two designs that were put into effect by the U. S. R. A. during government operation of the railroads—1918-20. The issue of the *Railway Age*, August 20, 1921, stated that "The manufacturers' committee has met with the subcommittee [of the A. R. A. Committee on Car Construction] on two occasions, each time submitting for consideration designs in detail representing various ideas in view, but up to this time no single design has been brought out that could be presented as a recommendation for standard. It is possible that more than one design will have to be considered, with alternates."[24]

Metallurgical Research and the Dalman Truck — 1926

Another important milestone in the development of freight car running gear was the establishment of the metallurgical research laboratory of American Steel Foundries in 1926 at East Chicago, Indiana. Since its establishment, this laboratory has contributed important improvements in the metallurgy of steel for railway use—either alone or in cooperation with other manufacturers, private research organizations and the railroads. Specific achievements of this laboratory will be discussed in the concluding chapter.

Also in 1926, American Steel Foundries offered the "Dalman" truck — a new design incorporating increased spring group capacity by increasing the number of, but without changing the dimensions of the existing A. R. A. standard bolster spring coils. The "Dalman" truck was designed to conform with the recommendations of the American Railway Association's Committee on Car Construction as to strength, service and safety. Notwithstanding improved features, the Dalman truck was interchangeable as a complete truck assembly with respect to center-plate height, side bearing spacing, and wheel base with A. R. A. standard recommended truck designs.[25]



The "Dalman Truck" (1926) provided additional spring group capacity.

Another 1926 development was the adoption as standard of the American Railway Association's acceptance test specifications for cast steel side frames and truck bolsters. This action preceded the adoption of more severe test specifications in 1929.[26]

Truck Manufacturers Agree to Pool Patents for Standard Freight Car Truck — 1927

As previously noted, the railroads, in 1915, had invited four leading truck manufacturers to cooperate in the development of standard specifications for the design of trucks and, although this arrangement had been very helpful in resolving the matter of technology and design, there was still one serious obstacle in the way of standardization—the problem of patent infringement. The individual truck manufacturers competing with each other over the years in developing cast steel side frames and bolsters had taken out many patents. The railroads wanted the benefit of the best design features that had been developed by all of the manufacturers, but whenever a standard design was proposed by the A. R. A. subcommittee, it was found that such standard design infringed various patents held by certain of the companies. The railroads obviously wished to be free from possible liability of infringement and yet the individual manufacturers were unwilling to hand over their patent rights which represented many years of research and development efforts. So it was that late in 1926, the General Counsel for the Western Railroad Association suggested in a letter that the leading manufacturers of truck side frames might mutually interlicense or release each other under their patents on side frames at least as far as the conventional types of cars were concerned. He thought this idea might be the best way to achieve standardization and still permit side frames to be bought by the railroads on a competitive basis from any one of the companies entering into such an agreement.

After considerable discussion during the early part of 1927, the truck manufacturers executed what was known as the "Four-wheel Truck Agreement" on June 9, 1927. This agreement enabled the railroads to use the best design, incorporating features patented by various manufacturers without fear of patent infringement.[27]

Nine years later at a meeting of the American Society of Mechanical Engineers, D. S. Barrows, of the Symington-Gould Corporation, summarized the circumstances surrounding the four-wheel truck agreement as follows:

> Ever since the formation of the original Master Car Builders Association, the American railroads have progressed toward the standardization of im-

portant car details, and a few years ago the then American Railway Association sponsored a detailed design of side frame. This was later withdrawn as a definite required standard, because the art of side-frame design had not then reached its limit of development. The present requirements are merely that a freight truck shall conform to certain general dimensions and be interchangeable as a whole with any other truck meeting the same requirements.

When the then American Railway Association tentatively adopted the specific side-frame design referred to previously, there were available a number of different types of side frames, as well as hundreds of thousands of arch-bar trucks in service, representing an unknown number of variations from an earlier established standard for this particular type of truck. It was natural therefore that the American Railway Association should desire to discourage new truck designs before the chaotic conditions which existed prior to the adoption of the standard automatic coupler were duplicated in another field. [See Chapter III of this report.]

Long before these specifications had reached their present form, certain conventional details of side-frame design were identified as points of weakness. As research proceeded, it was found that the establishment of a design, which would give the best static and fatigue results, and be susceptible of economical manufacture through the elimination of the maximum number of foundry difficulties, required the use of a considerable number of detail patents taken out over a period of years by the various side-frame manufacturers in the course of their independent research work. Therefore, in 1927, the manufacturers, in cooperation with the then American Railway Association, cleared the path toward the development of a future side-frame standard through the formation of a manufacturers' association based on an agreement under which all detail patents on side frames and bolsters were made available to all the members as soon as any of these patents were commercially utilized. This manufacturers' association through its engineering committee develops the standard designs which are available to each manufacturer-member and through continued research and cooperative test work carries on the refinement of these designs . . .[28]

Action Taken to Eliminate Arch-bar Trucks—1927

The freight equipment of the railroad was being subjected to increasingly severe service which was exacting a heavy toll in maintenance expense and resulting in many dangerous failures of arch-bar trucks. The problem became so serious that, in 1927, the American Railway Association adopted a rule governing the interchange of freight cars to the effect that cast steel truck side frames conforming to A. R. A. specifications would be required on all cars built on or after July 1, 1928. This measure led to gradual elimination of arch bars under freight cars, but there was still great hazard and excessive maintenance expense entailed by continued use of arch-bar trucks under old equipment.

One authority describes the action as follows:

> Fully cognizant of the unsuitability of the arch bar truck structure comprising some thirty-five separate pieces relatively loosely held together and, consequently, impossible to maintain as a unit structure, and with the background of excellent service for the cast steel side frame comprising only one integral member, the American Railway Association in 1930 adopted a regulation to totally wipe out arch bars on all equipment . . . [by] January 1, 1936.
>
> The move was not made hurriedly or inadvisedly, for the records of the Interstate Commerce Commission reveal that for the four years ending with 1931 an average of 588 wrecks per year took place on American railways due solely to arch bar trucks, with a casualty list of 28 persons killed and 77 injured resulting. No attempt has been made to appraise the cost of these wrecks which, together with the bare maintenance requirements of arch bar trucks, has unquestionably been a tremendous sum. It may be confidently stated that no measure since the obligatory use of the automatic coupler and automatic air brake will have produced greater ultimate benefit to the railways than the rule prohibiting the interchange of cars equipped with arch bar trucks.[29]

When the Mechanical Division of the American Railway Association met in convention in June, 1929, the Car Construction Committee strongly advised against perpetuating arch bar trucks by using old or reconditioned arch bar trucks under new car bodies. The committee pointed out that freight trains were operating at much higher speeds and the carrying capacity of the smaller axles had been increased over the loads for which the trucks were originally designed. The increasing age of arch bar trucks resulting in the loosening of the multiplicity of parts, was making it increasingly difficult to maintain these old type trucks in satisfactory condition to give dependable service. The Committee, in its final decision, stated: "It is felt that any added cost by reason of the application of cast steel side frames would be more than compensated for by (a) reduction of accidents, (b) shortening of time in handling trains through yards, (c) less delays of loaded cars, and (d) saving in cost of maintenance and inspection.[30]

Springs Studied for Better Riding Quality — 1928-1930

In reporting the year's trend in equipment development, C. B. Peck, Mechanical Department Editor of *Railway Age* stated January 7, 1928, that: "The Car Construction Committee is working with spring manufacturers to bring about development of springs possessing qualities sufficiently superior to those of the present standard types, so that satisfactory service may be insured in trucks with the present standard type of bolster opening." Increasing freight train speeds had revealed that spring performance was not entirely satisfactory and that something would have to be done to increase cushioning action and improve the riding qualities of freight cars. At that time, efforts to improve the riding qualities of freight cars usually consisted of applying some type of friction device, but in certain cars more flexible springs were used and the maximum permissible load reduced. In 1929, tests were being conducted to measure spring performance, both by the railroads and by some of the manufacturers. As a result of such tests, springs gradually improved. Interest in the subject of riding quality also led to the development of better instruments for measuring the riding qualities of a freight car.[31]

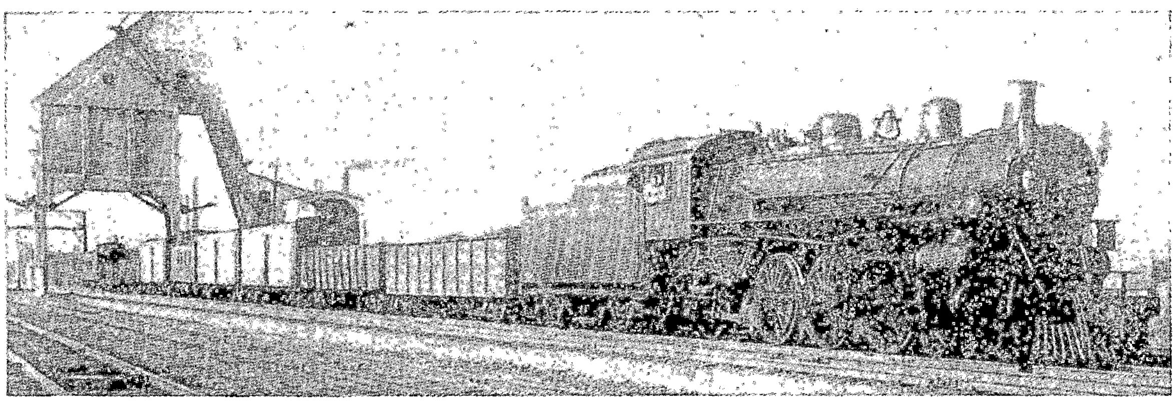
In 1929, more severe acceptance test specifications were adopted as standard by the A. R. A. for cast steel side frames and truck bolsters, these increasingly severe requirements being realized with a reduction in weight through improvements in design and foundry practice. Further refinements of details led to greater simplicity, reduction of parts, reduction of truck maintenance and longer life of the freight car running gear.[32]

American Steel Foundries' Road Tests of Riding Quality — 1931

While, as has been noted, American Steel Foundries began road testing freight car trucks as early as 1913 and 1914 to measure the stresses set up under dynamic loading of side frames and bolsters, effort in those days was primarily directed toward improving the strength and durability of cast steel members of the truck. In 1931, the Company began its first really comprehensive program of road service testing to explore various possibilities for improving the riding quality of freight car trucks. There had been great improvements in instrumentation and much new information was thereby made available to the design engineers. Also in 1931, the trend toward faster freight train speeds was evident and this meant that riding quality would need to be studied more intensively.

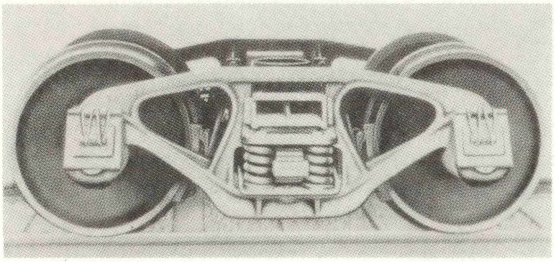
Road test equipment used by American Steel Foundries for its 1931 tests was a nine-car train consisting of four loaded coal cars, a caboose for the accommodation of the regular train crew, three test cars and an instrument caboose equipped with indicating and recording instruments. The train was operated on the Chicago and North Western Railway at speeds up to 60 miles per hour, between Chicago and Clinton, Iowa, and between Chicago and DeKalb, Illinois. It is believed that up to that time there had never been a more complete train specifically equipped for testing freight car trucks, and as a result of its operation a great many new facts about truck performance were made available to the manufacturers of freight car running gear and to the railroad industry.[33]

About the same time, the T. H. Symington Company made several tests which revealed the need for some changes in spring conditions. In an article published in *Railway Mechanical Engineer,* January, 1931, T. H. Symington concluded that "Spring oscillation is probably the largest factor in damage to perishable freight and to truck springs." He explained that cars become subject to violent spring oscillation at certain critical speeds and the effect, inside the freight car, is serious damage to lading. For instance, he said, (1) beef is found on the floor of refrigerator cars; (2) fat cattle suffer 10 to 15 per cent loss in weight during a long shipment; (3) certain fruits must be picked so green in order to stand the oscillation in shipment, that they never reach perfection at a distant market; (4) spring failures occur on cars loaded to the rail limit, and (5) such cars do much damage to the rail. In his opinion, all of these problems could be solved by breaking up the harmonic truck spring action.[34]



American Steel Foundries' test train on the Chicago and Northwestern Railway (1931).

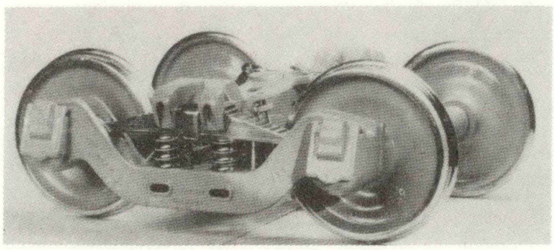
The results of extensive studies made by American Steel Foundries and the Symington Company along slightly different lines led the truck manufacturers to an improved "Double Truss" truck design, employing what were called "coil elliptic" spring groups. These spring groups were made up of a combination of coil springs and elliptic springs such that the substantial carrying capacity of the coil springs would be combined with the damping properties of the elliptic springs.[35]



"Double Truss" truck design employing "coil elliptic" spring groups.

In connection with its road testing of trucks for riding quality, American Steel Foundries introduced a truck of entirely new design principles during the year 1931, called the "Simplex Truck". This was a high-speed truck for new equipment. In general appearance it closely resembled an equalized 4-wheel passenger truck, and differed from the usual freight car truck design by the employment of leaf springs in combination with coil springs. In this arrangement the leaf spring extended longitudinally of the frame and rested in support pockets slightly inward of the journal boxes. The coils were near the center of the frames and were deflected by the leaf spring. Another feature was that this truck contained no spring plank.

The new truck was tested at maximum speeds of over 65 miles per hour and average speeds of 40 miles per hour, in which it was found that its shock absorbing characteristics provided soft cushioning action and freedom from undesirable oscillation while, at the same time permitting variable capacity for light and heavy loads. A variation of the Simplex truck employed the series spring arrangement and the "Type C" bolster, permitting conversion of old freight car trucks. In this case, conversion could be accomplished by applying new bolsters and leaf springs. Both of these two truck designs yielded distinct improvements in riding qualities. It was found in the road test on the Chicago and North Western Railway that the Simplex-Type truck rarely developed forces exceeding an increase of 30% in dead weight, while conventional freight car trucks were found to develop forces exceeding an increase of 120% or more in dead weight.[36]



American Steel Foundries' "Simplex Truck"—a high-speed truck for new equipment (1931).

American Railway Association Tests — 1933

In 1933, following appointment of a special subcommittee to study the merits of various snubbing devices, the A. R. A. supplemented road tests of the manufacturers by conducting similar tests of riding quality. Tests were conducted on a number of railroads using several types of trucks and a number of friction devices designed to minimize the harmonics of truck springs, to provide easy riding qualities and reduce damage to fragile lading. At the time it was reported that five types of complete trucks were being developed and tested along with some 14 different types of springs and spring arrangements. The numerous tests conducted by the American Railway Association revealed that a friction snubber greatly improved the riding qualities of the freight car.[37]

Light Weight Alloy Steel Considered for Truck Frames and Bolsters — 1934

To this day the usual material of which truck side frames and bolsters are made is what is known as "annealed or normalized" cast steel (described as "Grade B" steel in the A. A. R. specifications). Laboratory research and improved foundry techniques by 1934, however, had led the American Steel Foundries and other truck manufacturers to jointly explore the possibilities of making side frames and bolsters of high-tensile alloy steels. By 1934, this research had proceeded to the point where it had been definitely established that the use of alloy steels could accomplish as much as a 25% saving in weight over "Grade B" steel designs without sacrificing strength. Of course, the problem again was economic, for there was considerable doubt whether the saving in operating costs and other benefits due to lighter weight of equipment could justify the increased cost entailed by the use of weight-reducing material.

F. G. Lister, Superintendent of Motive Power, St. Louis and San Francisco Railway, discussed the problem as follows in the *Transactions of the American Society of Mechanical Engineers*:

> The freight-car truck has been the subject of a great deal of thought for a good many years, and more so in the last few years since the railroads

have speeded up their trains, requiring more attention to the bolsters, truck frames and brake rigging than ever before. The service is more severe. Heavier and more frequent braking has thrown more strain on the bolsters and side frames.

The truck frame, in its evolution from the old arch bar to the present U-section cast-steel frame, has undergone many changes, and not until the dynamic-testing machines were developed by the manufacturers of steel frames and research work commenced in connection with the design of the frames was it possible to distribute the metal through the structure to uniformly control the stresses encountered in actual road service. However, this meant a cast-steel side frame of heavy sections, making the extra weight undesirable. . . It is admitted that a worth-while saving in weight at no sacrifice of strength can be made by use of alloy steels in both truck side frames and bolsters . . . The problem resolves itself into the selection of an alloy steel which has high strength, ductility and good casting properties and which at the same time is reasonable in cost.[38]

High-Speed Truck Tests Conducted by the A.A.R. — 1939

It will be recalled that the American Railway Association had conducted freight car road tests at speeds up to 60 miles per hour in 1933. In 1934, the American Railway Association and several other railroad organizations were combined under a new organization known as the Association of American Railroads. The A. A. R. promptly began to concern itself with the problems arising from still higher freight train speeds.

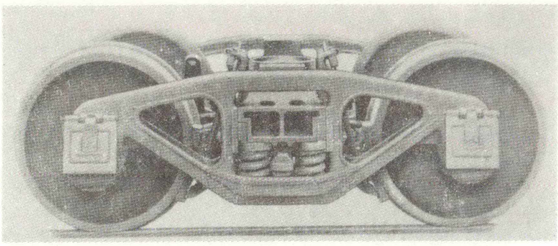
In the summer of 1939, the A. A. R. conducted a series of 85 mile-per-hour tests on the 54-mile Bald Eagle branch of the Pennsylvania Railroad between Tyrone (near Altoona) and Lockhaven. On this branch line track, maintained for heavy freight service, tests were run with a five-car train consisting of a baggage car followed by a freight car, another baggage car, another freight car and a coach. A dozen different truck designs were tested with varying loads of pig iron, ranging from 58,000 pounds to 145,000 pounds at rail. After eight round trips with each design, trucks were changed under one of the box cars, while under the other identical box car, the same trucks were kept throughout all the runs—providing a method to check the performance of the first box car. Electrical recording instruments installed in and on the two box cars measured the shocks received by the car body, action of the truck springs, lateral stability of the car and other factors of performance. The principal conclusion reached from these A. A. R. tests of 1939, was that none of the freight truck designs tested was yet satisfactory for 85 mile-per-hour service under freight cars.[39]

One of the outstanding facts revealed by the A. A. R. tests was that snubbers which could solve the problem of oscillation and resonance at speeds below 60 miles per hour, are of little if any value above 60 miles per hour. The reason for this was discussed in a *Railway Age* article entitled "Truck Requirements for High-Speed Freight Service", published in September, 1940. Oscillation below 60 miles per hour builds up from small impulses regularly repeated. When speeds increase to above 60 miles per hour, each track irregularity produces a very damaging shock unless the spring system is capable of modifying it. The reason for the failure of snubbers above 60 miles per hour in conventional trucks tested in 1939 was given as follows by the *Railway Age*:

> The usual snubber is something which is placed under the truck bolster to absorb energy during movement of the spring group. Its very presence in the spring system makes the springs more sluggish in their operation and this is the last thing desired in high-speed trucks. Spring systems for high-speed trucks must be lively and capable of making the wheels follow the rails with as little variation as possible in the wheel pressure on the rail. . . The first action of a spring system in passing over a low spot at high speed is to open rather than to close, and any interference with this action is detrimental.
>
> These tests show that *softer springs are the real solution of the high-speed truck problem,* and the softer they are the better the results secured. The limitations in coupler-height variation and the large variation between empty and fully loaded weights of freight cars place limitations on the maximum spring travel which can be incorporated in freight trucks, but very few of the trucks in this test took full benefit of the spring travel which could be used. [Emphasis supplied.]

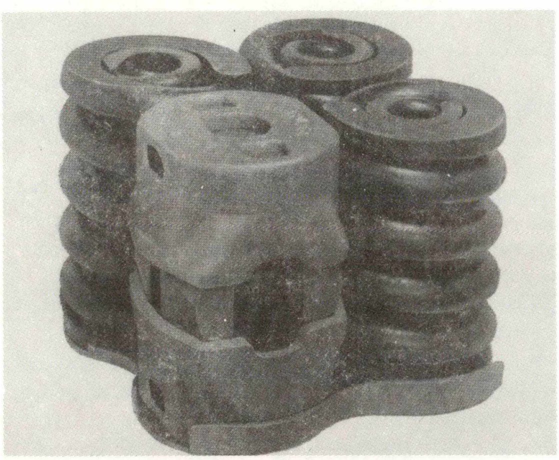
Another factor in truck improvement — very important from the standpoint of safe operation at high speed—was the question of proper horizontal stability. The A. A. R. tests revealed that lateral action of the wheels became very pronounced at high speeds resulting in "unsquaring forces" on the truck. Failure to provide sufficient resistance to unsquaring action was found to result in objectionable lateral shocks to the car body, which might become great enough to cause difficulty.[40]

It is interesting to note that among the trucks tested by the A. A. R. was American Steel Foundries' Spring Plankless Self-Aligning Truck, which differed from conventional trucks by the omission of the spring plank and the use of long travel springs (springs of 2½" travel). This truck permitted the use of longer springs than could be inserted in the conventional truck due to the fact that the springs were installed before engaging the bolster in the side frame and thus the springs could be set down in a pocket in the bottom of the side frame. The truck was also equipped with the improved American Steel Foundries' "Simplex Snubber", consisting of external *rubber* springs with friction surface angles changed to give longer travel. (This unique snubber, introduced in 1936, is believed to have been the first freight truck snubber to employ rubber for the spring action.) Another feature of the truck was that it permitted quick wheel change as the ends of the side frames could be spread enough to remove a pair of wheels after the bolster had been raised sufficiently high, without completely tearing down the truck.[41]



The American Steel Foundries' "Spring Plankless Self-Aligning Truck."

In 1939, American Steel Foundries introduced its new *"Simplex Unit-Type Snubber"* employing a single internal rubber spring instead of the external rubber springs of the earlier Simplex Snubber.[42] This economical unit snubber can be conveniently inserted in place of one double-coil spring in any conventional freight car truck. It contributed substantial improvement to the riding quality of thousands of freight cars prior to the advent of even more effective ride control developments in later years.



The "Simplex Unit-type Snubber" as it is applied to an A.A.R. conventional truck spring group.

The A. A. R. 1939 tests proved that the secret of an easy riding freight car truck involved long, soft-travel springs properly controlled. They also proved that the separate snubber or friction truck spring which gave good satisfaction at average speeds became erratic in performance at speeds above 60 miles per hour. The theoretical reasoning for the latter phenomenon was that at the higher speeds the snubber failed to follow the bolster movements. This theory appeared to be sound when, in supplementary tests at Altoona, American Steel Foundries welded the snubber to the top spring plate and this resulted in improved riding characteristics.[43]

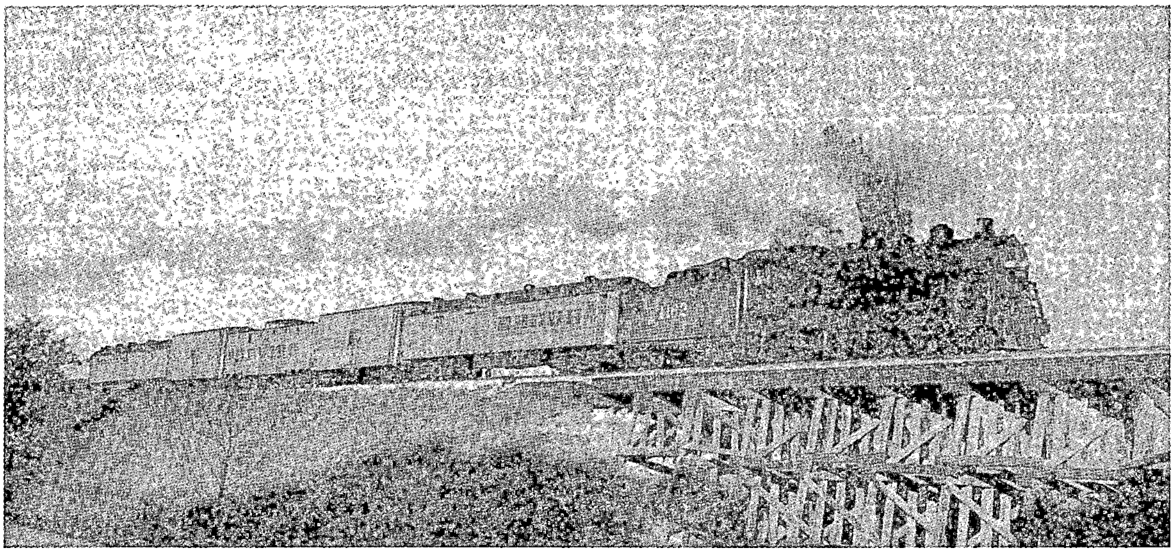
"Basic Freight-Car Truck" — 1940

In the spring of 1940 American Steel Foundries offered a simplified low-weight, low-cost truck design designated as the "A. S. F. Basic Truck". This truck had no spring plates or spring planks, and being of simple design, rugged construction and few parts, it was of particular interest to the War Department. This truck was used under all of the many thousands of freight cars and locomotive tenders built for the Government during World War II for overseas service. It was also used in interchange service on the American railroads.

"Service Laboratory" Tests — 1940-1950

Following the work of the Association of American Railroads in its high-speed truck test program begun in 1939, American Steel Foundries decided to build a completely equipped modern "Service Laboratory" to be used in the intensive study of freight car truck riding qualities. The A. A. R. tests had made it clear that the solution of problems involved in higher speed freight service demanded far more detailed information than was then available on all phases of truck performance under all conditions of road and lading. The Company, therefore, decided to continue the high-speed testing that had been done by the A. A. R. This could best be done by designing and equipping test cars that would become a permanent 90 mile-per-hour "rolling laboratory". In 1940, the first two cars of the American Steel Foundries "Service Laboratory" were completed and placed in special train operation on the Chicago, Milwaukee, St. Paul and Pacific Railroad. A description of the "Service Laboratory" equipment and tests will be found in the concluding chapter.

The maximum speed reached by the "Service Laboratory" on the Milwaukee road between Milwaukee and Portage, Wisconsin, was 96 miles per hour in a series of 56 test runs totaling nearly 10,000 miles run between August and November, 1940. Other tests were run in 1941 and 1942. During the spring of 1943 and the spring and fall of 1944 and 1945, a series of tests were conducted by a western railroad in which the American Steel Foundries' cars reached a maximum speed of 102 miles per hour.[44]



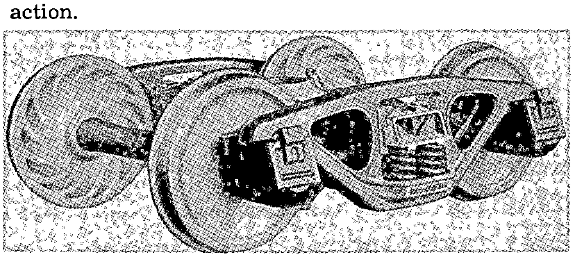
American Steel Foundries' "Service Laboratory for Freight Car Truck Development" on the Illinois Central Railroad.

Introduction of the New A-3 "Ride Control Truck" — 1943

In 1943 the American Steel Foundries' A-3 "Ride Control Truck" was introduced which combined two essential requirements for good riding quality—(1) *long travel springs* and (2) *proper control of spring action.* Repeated tests by American Steel Foundries and by others had proved that short travel springs could not completely cope with the higher speeds then being demanded by the freight service. Another fact which had been revealed by tests was the importance of having adequate *reserve* spring travel after the car had been loaded to its maximum permissible loading on rail. Simplicity of design was also featured in the A-3 truck. Moreover, the new truck demonstrated its ability to satisfy a number of fundamental requirements of a modern freight car truck:

(1) Safety—the truck must be safe for speeds up to 100 miles per hour.

(2) Easy riding—the truck must eliminate harsh, sharp shocks and reduce other impacts so that the lading in the car will not suffer damage from truck action.



American Steel Foundries' A-3 "Ride Control Truck" (1943).

(3) Low first cost—an important factor in keeping down railroad freight rates under competitive conditions.

(4) Low maintenance—the truck must be sturdy and inexpensive to maintain.

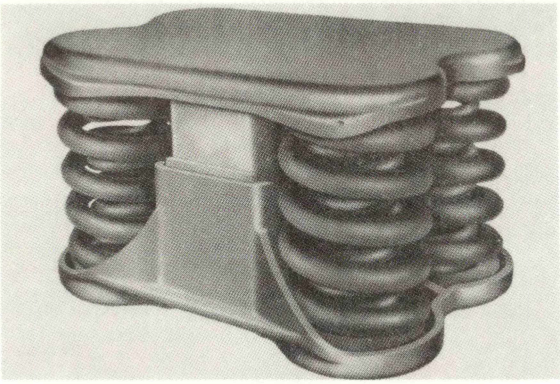
(5) Light weight—an important consideration in holding down the ratio of dead weight to revenue load in railroad freight service.[45]

Basically the A-3 truck consists of integral-box truss type cast steel side frames and box section truck bolsters with provision for long-travel coil springs, built-in spring control and without the use of spring planks or spring plates.

In discussing "Modern Freight Car Trucks for the Present and Future" before the Southern and Southwestern Railway Club in July, 1945, an executive of American Steel Foundries stated that the A-3 Truck produced an excellent ride at any speed up to 100 miles per hour and called specific attention to 150 box cars assigned to "box express" service, in which the service frequently called for speeds exceeding 100 miles per hour. (Incidentally, one lot of these box express cars had attained over 250,000 miles of service per car by June 1953. After examination of cars that had accumulated more than 250,000 miles, it was found that the wear on the side frame friction plates was negligible and that there was no appreciable wear on the friction shoes.) Another officer of the Company stated that in developing the new truck there had been considerable concern with the tracking characteristics at high speeds "because when you get up to those high speeds you are playing with fire if anything goes wrong with your running gear." However, no difficulties had been encountered in the tracking characteristics of the A-3 truck, although at that time over 15,000 car sets were in service.[46] (This is further borne out by the fact that by June 1953 there were more than 300,000 car sets in service and no difficulties had been encountered in the tracking characteristics of the A-3 truck.)

"Ride Control Package" — 1946

Three years after introducing the A-3 Ride-Control Truck, American Steel Foundries offered the railroads a means of modernizing *existing* freight car trucks under older cars. This was the new "Ride-Control Package" designed to prevent spring resonance by employing long spring travel with proper friction control—the same basic principles which had made the Ride-Control Truck a success.



American Steel Foundries' "Ride-Control Package" (1946).

The effectiveness of the Ride-Control Package was dramatically proved to railroad men in June, 1953, by demonstration runs made by the A. S. F. test train in connection with railroad meetings held at Atlantic City, New Jersey.

Modern Railroads, July, 1953, (page 144) commented:

> Though it has been one of the most important railroad research laboratories for years, it wasn't until the Atlantic City meetings that many railroaders actually got a chance to ride the American Steel Foundries train. It made two demonstration runs daily. In addition to showing railroaders the test setups and instrumentation, the demonstration runs gave those who rode the train an opportunity to compare the service performance of present-day high speed freight car trucks with older-type trucks. At maximum speed of 50 miles per hour, one had to hang on for dear life to stand in the car with conventional trucks. At the turnaround point a modern spring and snubbing package [A. S. F. Ride Control Package] was inserted and on the return trip the ride was bearable even though the speed was increased to 80 miles per hour.

Ride Control Trucks and Snubbers Reduce Damage to Freight

Interest in the use of snubbers to dampen the harmonic vibration of spring groups in existing trucks developed rapidly after the road tests of the Association of American Railroads in 1939. G. A. Lewis, Chief Engineer, W. H. Miner, Inc., writing in May, 1943, in the *Railway Mechanical Engineer,* discussed the idea of improving riding quality by incorporating snubbers on freight car trucks. There were, in fact, four advantages which snubbers could contribute according to Mr. Lewis: (1) reduction of freight claims, (2) prevention of displacement and damage to machined projectiles, as well as breakage of floor boards in high speed freight service, (3) reduction of stresses and fatigue on parts of the car structure, and (4) reduction in the number of waste grabs, which would mean both reduced cost of maintenance and increased operating safety.[47]

As an illustration of what excessive harmonics set up by spring action will do, it was reported that, during the war, spring action was sufficient to unlock couplers on tank cars resulting in so many train separations that the Office of Defense Transportation required all tank cars to be equipped with snubbers.[48]

The Freight Claim Division of the A. A. R. Operations and Maintenance Department, in a special bulletin issued to railroad freight claim officers in May, 1947, cited the problem of fires breaking out in carloads of *cotton.* It appeared that vertical shocks caused buckles on cotton bales to come in contact with each other in such a way as to strike a spark. Another case cited dealt with a train carrying a flat car loaded with 16 or 18 *telephone poles.* This car was properly stacked and wired, but the wires were not tightened sufficiently to the top of the load. When the train reached a long down grade, the speed of its movement caused the springs to vibrate and close, with the result that every time the springs went solid, one of the poles not held down by the wire bounced and moved forward a little. By the time the train reached the foot of the hill this pole had slipped off between the cars and derailed 14 of them. This case was cited as an indication of accidents and damage that may result from vertical vibration not properly controlled. The bulletin concluded that cars equipped with snubbers were producing far less damage than the cars equipped with A. A. R. standard short travel springs. The results of one study cited indicated average damage to cars equipped with snubbers was only $1.69 per car as compared with $18.72 per car in those equipped with A. A. R. standard springs.[49]

Also in 1947, reduction in damage claims in connection with *egg shipments* were reported by the Northwestern Refrigerator Line. Test shipments were made from Sioux City, Iowa, to Chicago and 10 cars equipped with conventional trucks showed an average damage of 17.2 cases of eggs per car, while no damage was reported in the case of 5 cars equipped with ride control trucks having spring travel of 3¹³⁄₁₆". While admittedly the evidence in a case such as this is insufficient to be conclusive, the size of the annual freight claim bill paid by the railroads on egg shipments indicates the importance of controlling vertical vibration damage on this type of traffic. In the year 1946, for instance, a total of 38,530 carloads of eggs originated on the Class I railroads and on these shipments the railroads paid out claims of $2,095,974, or an average of $54.40 per car—this representing a loss of 18.2% of the revenue received on this class of traffic. The railroad claim officers were even more concerned with this problem in 1947 when it was estimated that the damage to eggs amounted to about $70 per car.[50]

In 1948, the Freight Claim Division of the A. A. R. Operations and Maintenance Department issued another special report on "Claims for Damage to Dressed Beef". Out of the claims submitted by 11 packers on dressed beef shipments, it was noted that Armour & Company had an extraordinary record with only a single claim amounting to $26.89, of a total of 762 claims reported amounting to $90,572.98. The report called attention to the fact that Armour & Company had equipped its entire fleet of refrigerator cars with snubbers and, since that had been done, had experienced virtually no beef down.[51]

J. B. Scott, Vice President of the Armour Refrigerator Lines was quoted as saying "The cases of meat off hooks were very, very few and the situation is getting better all the time." All of the Armour fleet of refrigerator cars had been equipped with spring snubbers of the "Simplex" type made by American Steel Foundries, and in 1948, the company had started construction of 2,000 new refrigerator cars all of which were to be equipped with A. S. F. Ride Control trucks.[52]

In 1950, total claims for freight loss and damage for United States and Canadian railroads were approximately $89,000,000, of which $67,000,000 was for damage of undetermined cause. The nature of the commodities involved in the "unlocated damage" claims indicates that ineffective draft gears and truck springs caused a large part of the damage. Fragile commodities were found to be particularly susceptible to damage from rough riding cars and lifeless draft gears. Use of various types of snubbers and long travel springs were reported to be reducing breakage as much as 50% in the transportation of such items as *household appliances, plumbing fixtures, glassware, bakery goods, newsprint goods* and *livestock*. In an editorial published September 10, 1951, entitled "High Cost of Impact" the *Railway Age* stated:

> Largely as a result of freight-truck spring and snubber research conducted by the Association of American Railroads in conjunction with railway supply companies, interest in this subject has been greatly augmented. It is expected that, roughly, 95% of the freight cars ordered this year will have long travel friction-snubbed springs. The problem which confronts railroads is to equip the present inventory of older cars still used in high-speed service with modern trucks as rapidly as practicable, not forgetting that many of these cars, up to perhaps 25%, have draft gears which are obsolete or suffer from lack of maintenance, and that snubbers, worn out or broken, are not protection against damaging vertical shocks.[53]

The *Proceedings* of the A. A. R. Freight Claim Division's annual meeting in June, 1952, contained a report of an extensive field study of the transportation performance of 17,480 cars in dressed beef service over a period of 18 months ending December 1, 1951. The seriousness of the problem of beef damage was indicated by the fact that 101,880 quarters of beef were damaged. The encouraging result of the study was that only 3.85% of the damage occurred in 4,055 cars equipped with high-speed trucks. These cars represented 23.2% of the total of 17,480 cars in beef service. The damage per car was 7.32 quarters for 13,018 cars with conventional 1⅝" travel springs as compared to .97 quarters per car for the 4,055 cars equipped with long travel spring groups (2½" or more) controlled by built-in snubbing devices. The study concluded that "excessive vertical vibration of cars not equipped with high-speed trucks caused all but a small fraction of damage estimated at $400,000." The special committee concluded that "claims paid by the railroads for damage to dressed beef can be reduced 85% by equipping all beef-rail refrigerator cars with modern, high-speed trucks."[54]

Progress in Freight Car Trucks

This review of the historical development of the freight car truck by American Steel Foundries and other manufacturers has disclosed that initially the objective was to obtain an all-metal truck of great strength and simplicity. Obtaining the modern, efficient cast steel side frames and bolsters, accepted today as commonplace, was the result of many years of research and experimentation facilitated by special testing apparatus and actual road tests in railroad service. It is also clear that no small part of the refinement and improvement in truck design has been due to advances in metallurgy and foundry practice.

Over the years the freight car truck has developed strength and simplicity, durability, safety at high rates of speeds, a large degree of standardization for interchangeability of parts, and finally improved riding quality. While all of these characteristics have been built into the truck, its cost has been held down and its dead weight considerably reduced in relation to its capacity for performance.

Indicative of progress in the development of freight car trucks is the reduction of train accidents due to defective trucks. The annual *Accident Bulletins,* published by a bureau of the Interstate Commerce Commission since 1922, provide evidence of the achievement of safer and more dependable railroad trucks. As shown by the tabulation below, the total number of freight and passenger train accidents due to defective trucks declined from 2,341 accidents in 1923 to 465 accidents in 1950. In spite of increased speed and volume of railroad traffic from 1923 to 1950, *the annual number of accidents due to truck failures has been reduced more than 80%.*

TRAIN ACCIDENTS DUE TO DEFECTIVE TRUCKS

Year	*Accidents*	*Year*	*Accidents*
		1936	862
1922	1,853	1937	777
1923	2,341	1938	315
1924	1,903	1939	275
1925		1940	350
1926	1,589	1941	375
1927	1,512	1942	555
1928	1,280	1943	736
1929	1,342	1944	725
1930	949	1945	743
1931	578	1946	598
1932	490	1947	698
1933	547	1948	558
1934	647	1949	397
1935	687	1950	465

SOURCE: *Accident Bulletin,* Interstate Commerce Commission, Bureau of Transport Economics and Statistics, 1921-1950.

Throughout this chapter, particular reference has been made to the efforts and achievements of American Steel Foundries, which may be briefly summarized as follows:

(1) In 1910, American Steel Foundries established what was probably the first specialized department for freight car truck research and development at Granite City, Illinois. This organization pioneered in developing ingenious tests used in the development of improved cast steel side frames.

(2) In 1913 and 1914, the Company provided test apparatus and special truck castings for what has been generally recognized as the first of all test cars —a Pennsylvania Railroad standard H-21 hopper.

(3) In 1914, American Steel Foundries introduced the first cast steel side frames having channel or "U" sections throughout, with a box section under the spring seat. This design has been generally adopted by the railroads.

(4) In 1921, the Company designed and installed, in its Granite City Laboratory, the first testing machine for scientifically studying the effect of dynamic loadings on full-size side frames. (The side frame testing machine at Granite City has been used by American Steel Foundries and some railroads for over 30 years and in 1947 the use of this machine for acceptance testing of side frames was made part of the specifications by the Association of American Railroads.)

(5) In 1926, a metallurgical research laboratory was established by American Steel Foundries at East Chicago, Indiana. This laboratory has been instrumental in bringing about improvements in the quality of cast steel used in truck side frames and bolsters.

(6) In 1931, the Company began its first really comprehensive program of road service testing to improve the riding quality of freight car trucks. A nine-car train was operated on the Chicago and Northwestern at speeds up to 60 miles per hour.

(7) By 1934, American Steel Foundries and other manufacturers, after extensive studies, concluded that the use of high-tensile alloy steels could reduce truck weight materially without the sacrifice of strength.

(8) In 1940, American Steel Foundries assembled the first two cars of a fully equipped permanent "Service Laboratory"—*now a five-car set—to* be used in scientific road testing of freight car trucks and other equipment components at speeds of 90 to 100 miles per hour. The value of these test cars is attested by the fact that the cars have been leased by the Association of American Railroads for special tests on several occasions since 1948.

(9) In 1943, the American Steel Foundries' A-3 "Ride Control Truck" with its long-travel coil springs and built-in spring control was introduced. Important new principles of the A-3 design were: (a) "long travel" springs—with 2½" to 3¹³⁄₁₆" travel as compared with the 1⅝" travel of conventional "A. A. R. 1936" springs, (b) constant frictional damping of springs under all loads by means of friction shoes built into recesses in the bolster and (c) elimination of spring planks and spring plates. Since 1943, this efficient and economical high-speed *truck has been more widely adopted than any other* design and has clearly set the "pattern" for postwar American freight car trucks. As of May 1, 1953, there were about 325,000 car sets of the A-3 truck in service on the freight cars of some 200 different railroads and private car lines.

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[9] Ames, J. H., *op. cit.*

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[16] Ernst, F. B., *op. cit.,* pp. 4-5; also: American Steel Foundries' brochure, "Road Testing Freight Car Trucks," p. 8.

[17] Reck, F. M., *op. cit.,* pp. 28-9; also: American Steel Foundries' "Road Testing Freight Car Trucks", p. 8.

[18] Morley, C. D., "History of Types of Truck Side Frames for Freight Cars", memorandum dated April 27, 1943, A. S. F. Product Engineering Department.

[19] Ernst, F. B., *op. cit.,* pp. 6-7.

[20] Morley, C. D., "The American Steel Foundries and Canadian Railroad Operations" (unpublished paper), p. 6; also: A. S. F. "Road Testing Freight Car Trucks", p. 9; and R. B. Cottrell, memorandum, June 8, 1953.

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[51] A. S. F. Sales Dept. Files, *Damage Lading Claims, 1946-48,* A. A. R. Operations and Maintenance Department, "Report on Claims for Damage to Dressed Beef", 1948.

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American Steel Foundries, A Brief History of The American Freight Car, by S. Berge, Chicago, Illinois, 1953, pp. 25-41 (see Reprint--History).

American Steel Foundries traces its experiences, history, and the evolution of its modern freight car truck design in this paper (see History - Reprint).

TDOP: 01-001

Ash, J. F. G., "The Freight Bogie," Railway Engineering Journal, November 1973, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England, pp. 30-37.

In Europe, and particularly in the United Kingdom, industrial areas had developed before the advent of railways, whereas, in the United States, railways were in operation before the formation of major industries. As a result, bogie freight wagons were introduced at an early date in America because of their capacity to operate on rough tracks hurriedly laid. Collieries and docks also were built with loading and unloading bays arranged to accommodate high-capacity bogie vehicles. The early American bogie was of extremely simple design with axle-boxes attached to its solebars by means of laminated springs pinned at one end in a bracket and free to slide in a shoe at the other end of the spring. On this side of the Atlantic, only a few vehicles of the bogie type were in service around the middle of the 19th century. The bogies had laminated springs and their general configuration was little different from many still in use today. At this stage, UK railway engineers were of the opinion that the bogie wagon was dangerous because of its inability to turn at right angles in the event of a derailment. In the United States, however, as early as 1834 a patent was granted for "cars intended to run on railways," disclosing a design which incorporated a bogie "bolster." The latter-mentioned word was thus used for the first time to describe an important component of the modern bogie.

RRIS: 051376

TDOP: 01-002

Association of American Railroads, Status Report of The Ad Hoc Committee on Truck Design, 1971.

In 1971 an Ad Hoc Committee reported to AAR Research Committee on the status of freight car truck design. This was an attempt to make a coordinated attack on truck design problems. The report is a summary of field tests and major findings. A performance specification for trucks was drawn up as well as a test specification. This work contributed to development of the basis for a contract between DOT/FRA and Southern Pacific Transportation Co. regarding the Freight Car Truck Design Optimization Project.

TDOP: 01-003

General Steel Industries, Freight Trucks: Yesterday, Today and Tomorrow, Granite City, Illinois, pp. 1-41.

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TDOP: 01-004

Hoather, S. J., "A Review of the First Three Years Operation of the New Tokaido Line," Railway Division Journal, Vol. 1, No. 6, 1970, Institution of Mechanical Engineers, 1 Birdcage Walk, London SW1, England, pp. 664-677.

Discusses the aspects of the operation of the New Tokaido Line in Japan. The history of the line and reasons for the construction are included. The technical description of multiple train units, the design of catenary and collection systems, and aerodynamic considerations are also included. Finally, the maintenance of the train and all of its systems is discussed. The maintenance schedule of the units is detailed as well as the revenue and expenditures of the operation.

RRIS: 033453

TDOP: 01-005

Keller, W. M., "Cars and Locomotives in Japan," ASME, 63-WA-151, 345 East 47th St., N. Y., N. Y. 10017.

The author presents a historical summary of passenger and freight equipment in service on Japan's railroads. Of interest is his description of Japan's treatment of hot-box and wheel-hunting phenomena.

TDOP: 01-006

Keller, W. M., "Mechanical Engineering Progress on Russian Railroads," ASME, 60-WA-283, 345 East 47th St., N. Y., N. Y. 10017.

The author presents a historical view and summary discussion of the status of mechanical engineering on Russian railroads. The author found that considerable progress has been made. Although the equipment is not yet up to U. S. standards, it is improving, especially with regard to four- and six-wheel trucks, automatic couplers, automatic air brakes, and larger-capacity cars. Electrification and dieselization also represent progressive steps. Ongoing education and research programs are also discussed.

TDOP: 01-007

Law, E. H. and N. K. Cooperrider, "Literature Survey of Railway Vehicle Dynamics Research," Surveys of Research in Transportation Technology, AMD - Vol. 5, 1973, pp. 49-78 (abstract of paper presented at 1973 Winter Annual Meeting of ASME, Nov. 11-15, 1973).

A survey of the research concerned with the dynamics of single conventional railway vehicles is presented. Attention is concentrated on analytical research and experimental research performed in conjunction with analytical efforts. The often conflicting objectives for railway vehicle suspension design and the research done to understand the design implications of these objectives are discussed.

RRIS: 057160
TDOP: 01-008

Manos, W. P., "Progress in Railway Mechanical Engineering (1968-1969: Report of Survey Committee)," ASME, 69-WA/RR-3, 345 East 47th St., N. Y., N. Y. 10017.

This report reviews technical developments in freight and passenger car equipment, locomotives, and operating procedures for a yearly period from 1968 to 1969. Of interest is the mention of hydraulic snubbers and preassembled, pre-set, and prelubricated bearings, and the introduction of the Symington XL 70 truck and Buckeye Steel's elasto-cushion truck.

TDOP: 01-009

Manos, W. P. and M. G. Marshall, "Progress in Railway Mechanical Engineering (1969-1970 Report of Survey Committee) - Cars and Equipment," ASME, 70-WA/RR-10, 345 East 47th St., N. Y., N. Y. 10017

This document represents the cars and equipment portion of the 1969-1970 two-part report on Progress in Railway Mechanical Engineering. Engineering improvements covered include freight car components, passenger cars and facilities, containerization, and bulk materials handling.

TDOP: 01-010

Manos, W. P. and M. G. Marshall, "Progress in Railway Mechanical Engineering (1970-1971 Report of Survey Committee) - Cars and Equipment," ASME, 71-WA/RT-7, 345 East 47th St., N. Y., N. Y. 10017.

This part of the Annual Report by the ASME Survey Committee reviews various improvements made in freight car equipment with emphasis on TOFC/COFC modifications. Some freight car truck improvements are discussed. Engineering on passenger cars and facilities, intercity passenger systems, and commuter transit cars is also covered.

TDOP: 01-011

Manos, W. P., "Progress in Railway Mechanical Engineering (1972-1973) Report of Survey Committee) - Cars and Equipment," ASME, 73-WA/RT-4, 345 East 47th St., N. Y., N. Y. 10017.

This survey reviews some major developments during 1973 in freight, passenger, and transit equipment. The trend toward higher cubic capacity and more rapid unloading is covered. New truck designs are discussed with regard to loading safety and high speed operation. New freight car and rapid transit developments are also treated.

TDOP: 01-012

Smith, Loren, The Freight Car Truck "Capability" Gap, Pueblo, Colorado, 1974, (paper presented at the 11th Annual FRA Railroad Engineering Conference).

The author presents his views on the nature of truck performance as opposed to what should reasonably be expected from freight car trucks.

TDOP: 01-013

Wallace, L. W., "Freight Car Design" (Course Notes) 1920, Purdue University.

This document comprises a 1913-1920 set of notes for administering a course of freight car design at Purdue University that was intended to guide students through truck, underframe, car assembly, and brake system phases. These notes are based on Master Car Builders' Association's publications, railroad and classroom experiences over period of several years, and have been used by several colleges. Some basic formulae on car axle diameters, beam truck bolsters, trussed bolsters, center plate and side bearings, springs, cast steel truck side frames, and truck assemblies are presented.

TDOP: 01-014

Section 2

TRUCK DESIGN

INTRODUCTION

The following subsections are comprised of reprinted articles, which were selected from the assembled literature concerning truck design, and the TDOP truck design bibliography.

The reports selected for reprinting are: "Design Features and Performance Characteristics of The High Traction, Three-Axle Truck," by H. A. Marta, K. D. Mels, and G. S. Itami, "Progressive Development of A High Speed Truck Design," by E. J. Warnock, and "Calculations On Hunting of High Speed Railway Truck - Problems of Truck Design For SANYO SHIN KANSEN," by K. Yokose. The article by Mr. Marta, Mr. Mels, and Mr. Itami describes a locomotive truck design providing improved adhesion performance. Mr. Warnock's article describes the development of the XL freight car truck concept, which provides for high-speed roller bearing trucks. Mr. Yokose's article deals with the fundamental characteristics of a hunting truck, and explains the theoretical foundation of the test truck for SANYO SHIN KANSEN.

The bibliography, which follows the reprints, is divided into the following subject groupings:

- Principles of Truck Design
- Freight Car Truck Design
- Passenger Car Truck Design
- Locomotive Truck Design

References under each subsection are listed alphabetically by author. Supplemental pages will be added to the "Bibliography--Truck Design," as new articles become available throughout the course of the TDOP.

Design Features and Performance Characteristics of the High Traction, Three-Axle Truck

H. A. MARTA K. D. MELS G. S. ITAMI

INTRODUCTION

With the increasing demands of Railroad Operations in the United States, as well as around the world during the past ten years, EMD has been extensively involved in the development of mechanical and electrical components to support the high performance requirements. To this end, significant improvements have been accomplished in a number of areas including the diesel engine, generators and traction motors, electrical controls, and the running gear—i.e., trucks or bogies.

Considerable EMD development and testing has been done on the running gear in the related fields of wheel-rail adhesion, curve negotiation mechanics, and truck design; some of this work has been published in ASME and other literature (1-8).[1] As a result, high adhesion efficiency, three-axle trucks were developed for domestic and export applications on EMD diesel-electric locomotives. The high traction, three-axle truck, Model HT-C, was designed to replace the SD type truck in locomotives starting in production January 1972 (Fig. 1).

[1] Numbers in parentheses designate References at end of paper.

DEVELOPMENT BACKGROUND

The initial development at EMD of a high traction truck dates back to the early 1960's when engineering work was done on a lightweight export truck design for locomotive applications where stringent weight and size, high adhesion support, and limited wheel-rail loading requirements were among the primary aspects of consideration. In addition to these primary purposes for developing a high adhesion efficiency truck, it was considered essential to maintain the simplicity of design for ease of maintenance, and to improve component integrity for longer service life.

The first high traction truck was a Model "GL-C" which refers to general purpose, lightweight, three-axle truck arrangement. The GL-C was made available in 1964 for multiple gage applications covering a locomotive weight range of 156,000 to 210,000 lb. The GL-C truck was

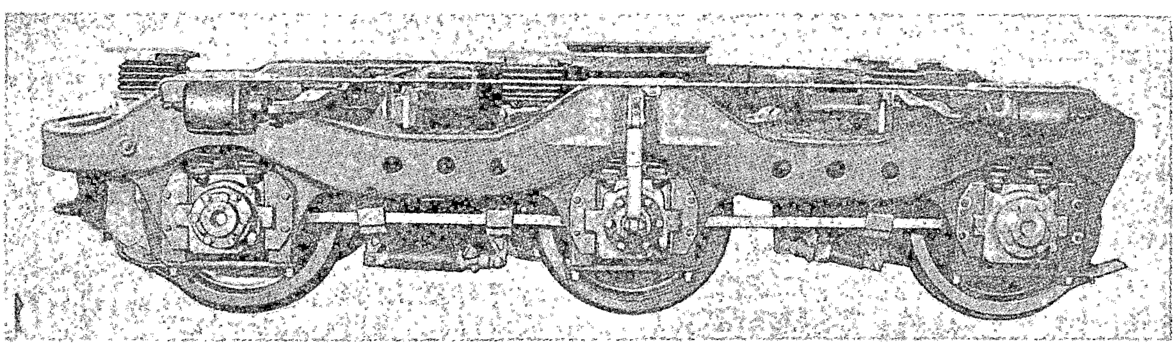


Fig. 1 High traction, three-axle truck model HT-C

Reprinted from The American Society of Mechanical Engineers, The Railroad Transportation Division, 345 East 47th Street, New York, N. Y. 10017

used on 1500-hp GL-22C locomotives for Angola, Africa in 1966.

In 1964, an associate locomotive builder in Australia became interested in the same truck concept. With EMD's cooperation, this builder developed a high traction truck for 55,000-lb axle load and used it in 1966 and 1967 on 3000-hp GT-26C, six-axle locomotives for Australian Railroads. The reported operating performance of these locomotives in regard to support of adhesion was very encouraging.

Engineering and test work in the laboratory and the field were done in 1967 and 1968, which investigated weight transfer between axles and wheel-rail adhesion of a special 3600-hp SD-45 locomotive which was modified to have higher adhesion efficiency in one direction of operation compared to the opposite direction (4). The test results confirmed that the improved weight transfer in the one direction of operation supported a higher tractive effort in the same proportion as the reduction in weight transfer. This was necessary to determine if the other variables affecting tractive effort would nullify the expected improvement derived from low weight transfer.

In the United States, EMD had previously refrained from introducing a high traction truck due to standardization considerations which appeared to be of highest priority during the 1960's. Although this aspect is still a very imporatnt one, the pressing Railroad operating requirements and the test experience gained between 1966 and 1968 provided the incentive to develop and introduce the HT-C truck on American Railroads. Simultaneously, a second export three-axle truck (Model GH-C) using the same concept was developed for multiple gage applications in locomotives ranging in weight from 190,000 to 315,000 lb.

The designs of the HT-C and GH-C trucks were developed during the latter part of the 1960's. Early models of the HT-C truck were used on seven SD-45X experimental locomotives placed in service in 1970. A considerable amount of testing and experience has been gained on these experimental locomotives since their introduction. The export Model GH-C truck has been used since June 1971 on fifty 3 ft-6 in. gage, 2650 hp GT-26MC locomotives built by EMD for the South African Railways. A slightly modified version of the same truck will also be used on 80 locomotives to be built during 1972 for the Argentine Railways.

The following sections of this paper will present the design features of the HT-C truck and a summary of test results comparing the performance of the HT-C truck to that of the conventional, three-axle truck.

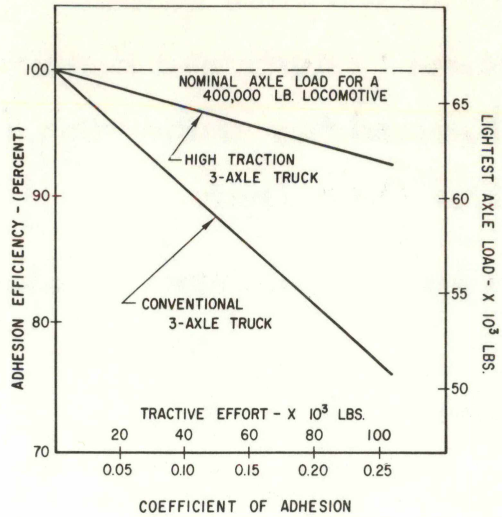


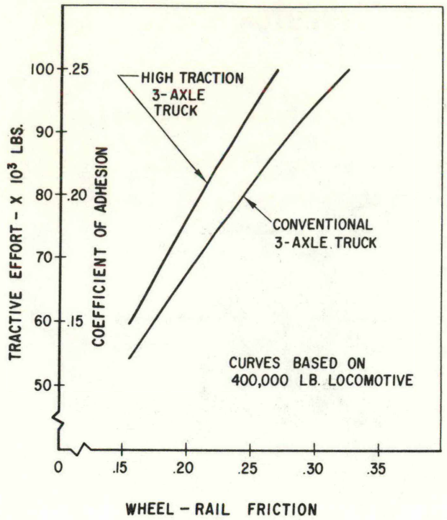
Fig. 2 Adhesion efficiency and axle load as a function of tractive effort for 400,000-lb locomotive

LOCOMOTIVE WEIGHT TRANSFER

The goal at the outset of the HT-C program was simple: design a highly reliable truck with improved traction capability which would adequately support a 420,000-lb locomotive under current operating conditions such as track profile. The HT-C design was developed using many of the concepts of the earlier traction export truck. During the development of the HT-C truck, every effort was made to retain interchangeability of parts with the domestic truck design.

To achieve the highest possible tractive capability from a locomotive, the truck must be designed to utilize the maximum amount of available weight for adhesion. If the power is equally distributed to each axle, which is common, then the weight at each axle should be kept as nearly equal as possible, since the lightest axle will determine the locomotive's pulling capacity as limited by wheel-rail adhesion. Control of the power supplied to individual axles can further affect the adhesion capability of a locomotive; however, that does not involve the truck design and, therefore, will not be discussed in this paper.

A large amount of weight transfer can occur between axles as a result of heavy tractive forces,



Coefficient of adhesion = $\frac{\text{Locomotive Tractive Effort}}{\text{Locomotive weight}}$

Fig. 3 Locomotive tractive effort and coefficient of adhesion as a function of wheel-rail friction

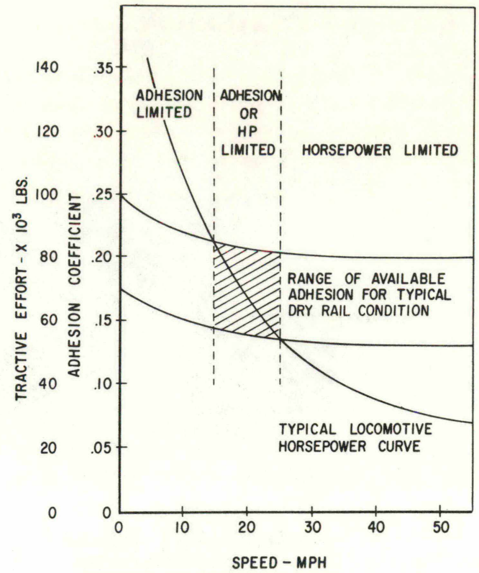


Fig. 4 Typical locomotive operating conditions: Sanded conditions will raise the band by 50 percent; wet or oily conditions will lower the band by 50 percent

Fig. 2. When accelerating from a stop, an automobile will "lift off" at the front wheels redistributing part of its weight to the rear wheels. This is also true of a locomotive, with three major differences.

1 In a locomotive with all driven axles, the wheels which get lighter are powered and will, therefore, tend to slip.

2 Significant weight transfer usually takes place for only a short time in an automobile. However, a locomotive operates for extended periods of time at high tractive effort; such conditions occur during acceleration and grade operation.

3 Since there are more than two reaction points (such as two axles on an automobile), the resultant distribution of weight transfer between axles depends on truck geometry and not merely the unloading and loading of the lead and trailing axles. (The phenomenon of weight transfer occurs during both driving and braking.)

PREDICTED ADHESION CAPABILITY

The ability of a truck to minimize weight transfer under traction is measured in terms of adhesion efficiency, the efficiency being equal to the ratio of minimum axle load for a specific tractive effort to the nominal static axle load.

Calculations have shown that the adhesive capacity of the HT-C truck would be 15 to 20 percent greater than existing six-wheel designs used domestically. For example, Fig. 2 shows that the SD truck, which is similar in adhesion performance to existing designs, offered 77 percent efficiency while the HT-C truck provides 93 percent efficiency at 25 percent adhesion demand. Thus, for a 400,000-lb locomotive, the lightest axle of a SD truck would be 51,000 lb compared to 61,000 lb for the HT-C truck. In essence, a locomotive pulling with a drawbar of 100,000 lb (25 percent adhesion) would require a coefficient of friction between the wheel and rail of 0.32 (i.e., 0.25/0.77) if equipped with the SD type trucks, and only 0.27 (i.e., 0.25/0.93) if equipped with the HT-C trucks. It is apparent that the conventional type truck would require a 21 percent higher friction coefficient at the wheel-rail interface than the new truck to prevent wheel slip under these heavy drawbar loads.

Fig. 3 shows that at a given value of wheel-rail friction, the HT-C truck can pull greater loads before slipping than the previous design. For example, at 0.25 available friction level, the HT-C would pull 93,000 lb before slipping, while the older SD would pull only 81,000 lb.

A locomotive operating under normal running

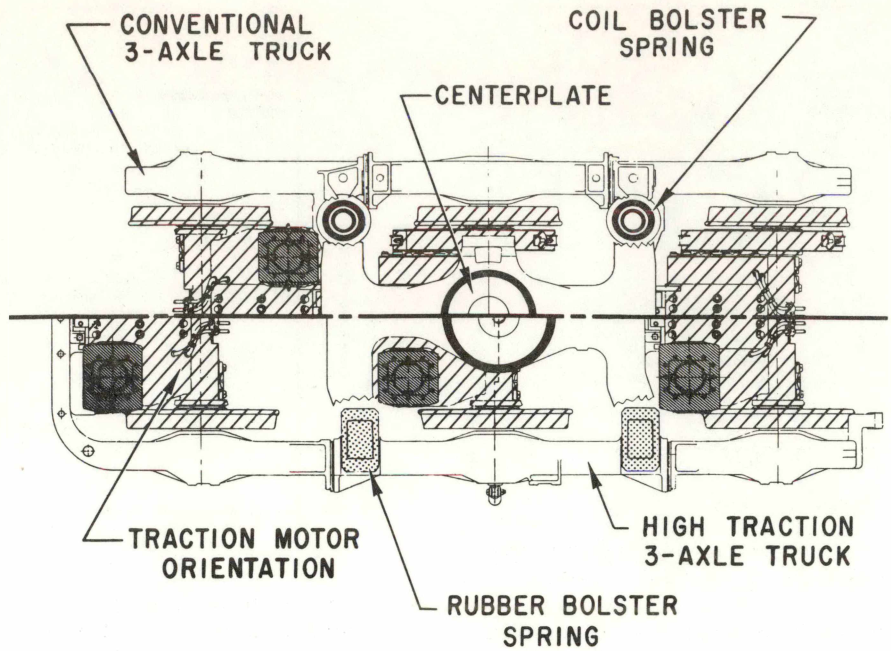


Fig. 5 Half-sections of the HT-C and Conventional three-axle trucks showing the major differences:
(a) Motor orientation
(b) Centerplate diameter
(c) Bolster suspension

conditions will encounter varying levels of tractive effort demand. The maximum locomotive tractive effort will either be limited by horsepower capacity or by the available wheel-to-rail adhesion as shown in Fig. 4. A tractive effort versus speed curve for a high horsepower locomotive has been drawn along with a band representing a range of available adhesion which the track will support under dry unsanded rail conditions. The upper limit is typical of the adhesion level attainable on slightly contaminated track with good rail joints, while the lower limit represents the available adhesion on moderately contaminated track with poor rail joints. Since the available coefficient of adhesion between the wheel and rail is a function of such things as atmospheric conditions, track contamination, and rail irregularities, this band will shift as indicated in Fig. 4.

For the conditions given in Fig. 4, at speeds below 15 mph, the usable tractive effort is adhesion limited, since the locomotive can develop more horsepower than the rail can support. However, at speeds above 25 mph, the delivered tractive effort is horsepower limited, because the locomotive cannot develop the horsepower that the rail can support. Between these speeds is a grey area where the locomotives may be adhesion or horsepower limited, depending upon the given wheel-to-rail conditions. In order to properly evaluate the adhesion capability of a truck, the testing must be done in the adhesion limited areas; if not, only the comparative locomotive horsepowers can be measured.

DESIGN FEATURES

The HT-C truck was designed for maximum performance under extreme driving and braking conditions by using specific truck geometry, component orientation, and suspension characteristics to reduce weight transfer between axles. This was accomplished while providing good riding qualities, overall simplicity of truck arrangement, and ease of maintenance.

The minimal weight transfer between axles, which provides the improved adhesion capability of the new truck, was achieved by using the following concepts, Fig. 5:

1 A relatively stiff secondary suspension between the bolster and truck frame, and a soft spring primary suspension between the truck frame and journal boxes: The secondary suspension is over 10 times stiffer than the primary, which tends to transmit a large portion of the moment reaction due to the driving forces to the carbody rather than to the wheels.

2 A large diameter centerplate between the truck bolster and the carbody underframe, to accept the large tractive reactions which result from the high traction design.

3 Traction motor orientation in one direction with each motor resting on a separate truck frame transom: This allows similar torque reactions at each axle which promotes equal axle loads.

4 Lower driving faces which contact at the trailing interfaces between the truck and bolster providing increased adhesive stability.

<u>Ease of Maintenance</u>

It is important to stress the accomplishment of high adhesion efficiency with design simplicity in the new truck. This was essential in justifying the adoption of the new design for domestic application. Some foreign designs use traction bars between the truck and carbody, or axles which are interconnected by gears; these arrangements are cumbersome to maintain and more expensive to build.

Effort has been made to extend the service time between maintenance requirements on all truck component parts other than those expected to provide service for the life of the locomotive. One of the HT-C trucks from a prototype locomotive was completely dismantled after one year's service and 114,000 miles to inspect for possible problem areas, such as abnormal wear, stress fatigue, or deterioration of any kind. The truck was found to be in very good condition.

<u>Suspension</u>

Journal coil springs provide a softer primary suspension for the locomotive sprung mass which results in good riding qualities. These coil steel springs offer improved equalization when compared to the suspension of the previous three-axle truck. This is a result of the 38 percent higher static deflection at nominal wheel loads. The new journal springs will produce smaller wheel load variations at a given vertical rail irregularity which helps reduce wheel slip, as well as minimize the loading of truck components.

New rubber springs used at four locations between the truck frame and bolster perform a vital role in the high adhesion design. These springs have undergone an extensive development program in the laboratory and in actual field service. This secondary suspension provides about 5/8-in. static vertical deflection and allows 1 1/4 in. of lateral deflection between the carbody and truck frame. The rubber pads isolate track noise, and they also serve as a secondary damping medium for the suspension system.

The primary damping is provided by hydraulic shock absorbers which are accessibly located between the journal bearing and the truck frame at both ends of each center axle, Fig. 6. A simple but rugged adaptor bolts on the standard journal box to accept the lower mounting pin of the shock absorber. The top of the shock absorber is bolted to the center journal spring pocket. The primary damping system is designed to control undesirable resonant motions, such as bounce, body pitch, and roll. The shock absorbers maintain a stable and comfortable ride for safe operation at all speeds.

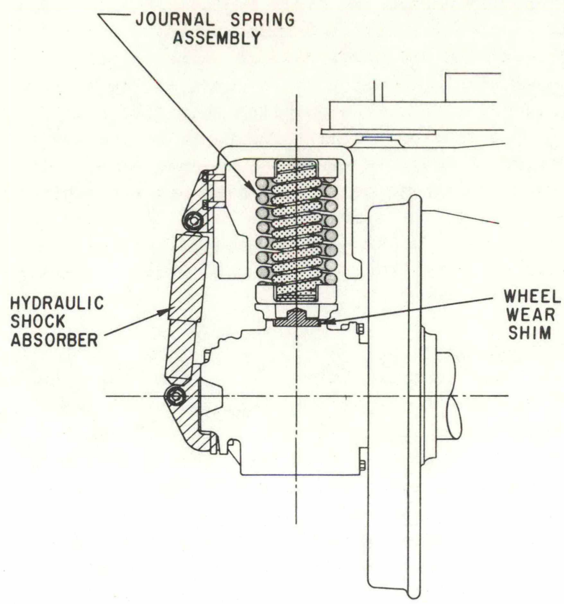


Fig. 6 Primary suspension system

<u>Wheels</u>

The standard taper contour cast or wrought steel, 2 1/2-in. rim multiple wear wheels are basic on the truck, with cylindrical contours available and recommended for high-speed operation above 95 mph. The HT-C truck has been designed to accept a special 42-in. wheel which is presently undergoing field evaluation. Provision is made to maintain the same carbody and coupler height with any wheel size which may result from wheel wear. Shimming can be made at the journal suspension area to accomplish this feature, Fig. 6. The lack of proper shimming to compensate for wheel wear can result in unequal axle loads severe enough to completely overshadow the high adhesion features of this truck. (It should be noted that similar adhesion losses will occur with existing truck designs when wheels are mismatched.)

Brake Rigging

With one single exception, the brake rigging of the HT-C truck is identical to the SD type, three-axle truck. The hand brake is located on the outside of the truck frame at the left side of the No. 3 axle, rather than on the inside of the truck at the left side of the No. 4 axle. Single shoe (per wheel) brake rigging using composition brake shoes with screw-type slack adjusters is basic. Clasp brakes or pin-type slack adjusters are also available.

Journal Boxes

Many improvements have been incorporated into the journal box components to extend service periods. A new rear cover provides an improved labyrinth seal which extends intervals between necessary oil additions to three times compared with the older boxes. A new oil fill cup shortens the time required to inspect or maintain the oil level, as no tools are necessary for its removal or replacement. The journal box housing has been modified to provide improved support for the wear plates. High strength bolts and special lock-washers have eliminated the need for lockwiring both front and rear cover bolts. A modified front retainer ring provides improved oil flow to the thrust block bearing surface. Crowned rollers reduce the contact stresses between the rollers and races, thereby extending the service life. There is also an improved non-sticking rear cover gasket.

Truck Frame Strength

Structurally, the truck frame and bolster casting have been designed to adequately support a 420,000-lb locomotive for unlimited service under maximum driving and braking conditions while negotiating jointed track with a vertical loaded rail profile of 3 in. in one rail length. Static stress tests have been performed in the laboratory, and dynamic stress tests have been run under actual service conditions which have confirmed that the design criteria have been met.

LOCOMOTIVE APPLICATION CONSIDERATIONS

The three-axle high adhesion truck will not be interchangeable with any previous three-axle truck assembly. This is primarily due to the following differences between trucks:

1 Traction motor air duct locations
2 Centerplate diameter
3 Carbody-to-truck safety interlock systems
4 Overall truck dimensions.

Truck Weight

The truck assembly will weigh from 54,600 lb with single shoe brakes and a hollow bolster casting to 60,300 lb for clasp brakes and a solid bolster. The bare truck frame casting weighs approximately 11,400 lb, and the bolster is 4500 lb if hollow or 8200 lb if solid. The new truck assembly can weigh up to 1500 lb more than the older assembly.

Truck Dimensions

Wheelbase—overall------------------163 3/8 in.
Axle 1 - 2 spacing-----------------79 5/8 in.
Axle 2 - 3 spacing-----------------83 3/4 in.

Truck Casting
Overall length-----------------19 ft-1/8 in.[2]
Overall width-----------------8 ft-4 3/4 in.
Overall height-------------------37 1/4 in.

Locomotive Application Considerations

Truck assembly
Overall length--------------19 ft-3 5/8 in.[3]
Overall width----------------9 ft-8 5/8 in.[4]
Overall height-------------------50 5/8 in.[5]

[2] Single shoe brake design shown. Add 7 3/4 in. for clasp brake truck.

[3] Distance shown for sander guide to end transom. Add 10 in. for clasp brake assembly.

[4] Add 5 in. at outside hand brake crossover lever.

[5] Height shown to top of bolster with 1/2 variable supplies. Add 3 1/8 in. when standing free.

LABORATORY AND FIELD WEIGHT TRANSFER TESTS

In July 1970, a prototype SD-45X locomotive built with the HT-C truck was available for experimental work. It was a six-axle diesel electric locomotive rated at 4200 hp and weighing 398,000 lb, or slightly more than 66,000 lb per axle. In order to determine the increased adhesive capacity of the HT-C truck, a series of laboratory

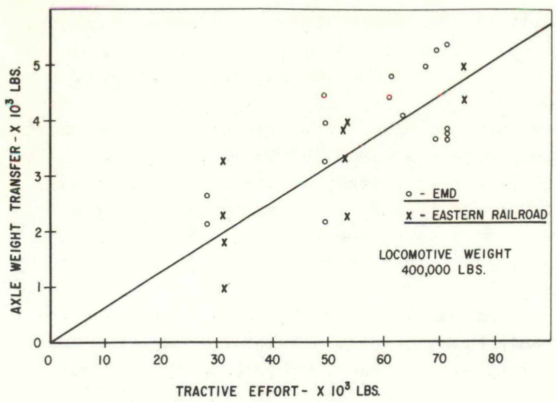


Fig. 7 Experimental weight transfer for the leading and trailing axles of the HT-C truck summarized from EMD and field tests

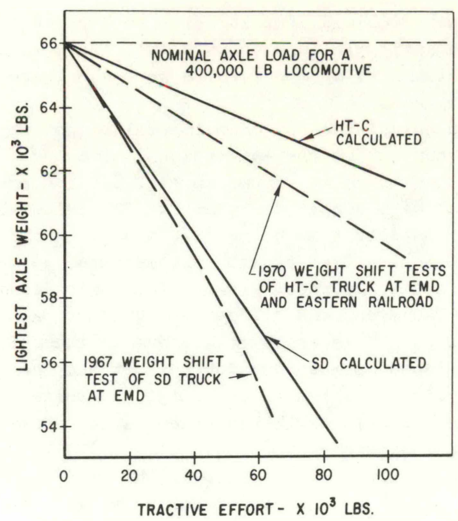


Fig. 8 Comparison of experimental and calculated axle weight transfer versus locomotive tractive effort for the HT-C and conventional three-axle trucks

and field tests were conducted on this unit. Static weight transfer tests were performed at EMD and on an Eastern Railroad. Then in early 1971, adhesion tests comparing the HT-C truck to the conventional SD truck were completed on a Midwestern Railroad. The results of these tests confirmed the practical value of the high adhesion design concept which, in turn, led to the adoption of the HT-C truck in the 1972 SD model locomotive.

The initial testing of the HT-C truck took place at EMD to establish a relationship between the weight transfer at each axle to the tractive effort of the locomotive. This was accomplished by recording the following quantities:

1 Change in wheel load
2 Generator current.

Track load cells were used to monitor the individual wheel loads by locating the axle to be tested directly over the load cells. Since locomotive tractive effort is a function of main generator current, the current was measured by placing a shunt in series with the main generator.

In this stationary test, the locomotive was held in place by blocks welded to the rail and by application of the brakes to the truck which was not to be tested. As the tractive effort increased, the change in wheel loads occurred, and the resultant axle loads were recorded on an oscillograph. Under the conditions involved, a locomotive tractive effort of 74,000 lb, equivalent to a coefficient of adhesion of 0.18, was usually developed before the wheels started to slip. The axle weight transfer was measured in both the forward and reverse direction of travel.

Static weight transfer tests were also run on an Eastern Railroad as part of the first road test of the SD-45X locomotive. The unit was taken to an electronic scale where individual axle weights could be measured. Again, the tractive effort was determined as a function of the main generator current. The locomotive was held stationary by braking the remaining units in the consist. As in the previous tests, no braking was applied to the truck being tested. The total axle load was recorded for each axle with the locomotive being powered in both forward and reverse directions for throttle positions 2, 3, and 4. A maximum locomotive tractive effort of 77,500 lb was delivered before the braking force of the consist was overcome and the locomotive started to move.

The results of the laboratory and field weight transfer tests were summarized, and Fig. 7 was developed. The data points represent the change in axle load (in absolute value) for the end axles (No. 1 or No. 6) in either the forward or reverse direction as a function of tractive effort. The end axles were chosen, since they exhibit the largest change in axle load, and the adhesive capacity of the locomotive is related to the lightest axle. Since the scatter of data was between 2000 and 3000 lb, or 3 to 4 percent of the nominal axle load (66,000 lb), the accuracy and repeatability of the results were considered to be very good.

The calculated and experimental weight transfer curves for the conventional SD truck and the HT-C truck are shown in Fig. 8. The experimental data compared closely to the calculated weight transfer for both the HT-C and SD trucks. At 60,000-lb tractive effort, the measured weight transfer was 1.9 percent greater than calculated for the HT-C truck and 3.1 percent greater for the SD truck. This increase was due to the friction developed between the journal box and the pedestal liners, which is present to a greater degree in a static test than under actual dynamic operating conditions.

The improved performance of the HT-C truck is made evident by the difference in weight transfer between the two trucks. Thus, at any given tractive effort, the HT-C truck effectively produces a higher adhesive weight locomotive. This means that a locomotive with an HT-C truck can deliver the same tractive effort at a lower locomotive weight; conversely, at the same locomotive weight, the HT-C truck requires a lower available coefficient of friction between the wheel and rail to maintain a given tractive effort, resulting in a lower slip risk.

ADHESION TESTS

To develop data on the effectiveness of the HT-C truck, adhesion tests were performed on a Mid-Western Railroad to compare the SD-45X locomotive to the SD-45 locomotive under simulated operating conditions. The SD-45 locomotive was the same unit which had been used during adhesion tests conducted in 1967, and it weighed 397,000 lb, virtually the same weight as the SD-45X. The SD-45 was a six-axle diesel electric locomotive rated at 3600 hp and was equipped with conventional SD type trucks. Both units were thoroughly checked to insure that they were operating properly and were producing the rated horsepower.

A consist comprised of the two SD locomotives, the EMD Test Car, and two dynamic brake units operated as a test train at two different test sites. The first location contained level tangent track, which was primarily welded rail. The second test site had different degree curves constructed with jointed rail.

Since the purpose of these tests was to verify the improved performance of the HT-C truck, a primary objective was to maintain similar operating conditions between the two locomotives, especially the available coefficient of friction between the wheel and rail. Therefore, numerous test runs were made over the same section of track by each unit. An attempt was made to conduct tests on each locomotive within a few hours to minimize the variation in rail surface due to such things as weather conditions, time of day, and passing trains. (For example, a passing train could cause the track to become oily, and a 50 percent loss in available friction would result.)

The remainder of the consist provided the dynamic braking load, simulating the train load, for the unit in power. This braking load was controlled from the Test Car and was adjusted according to the testing conditions. Hump control (variable throttle control) was installed on each locomotive, so that maximum tractive effort could be applied at the wheels at all times. Initial testing was conducted without wheel-slip protection; however, it was soon found that wheel slip occurred too frequently and wheel overspeed increased beyond control. Manually regulating the power was too slow; therefore, to prevent damage to the traction motors and the wheels, the wheel slip systems were used. Both units were equipped with similar IDAC wheel slip systems, and the automatic application of sand by the IDAC system was deliberately prevented.

By combining the IDAC with the variable throttle control, it was possible to operate at maximum tractive effort. The variable throttle control could establish a level of tractive effort about which the IDAC could modulate. The maximum tractive effort was obtained by increasing the power to the motors until small wheel slips occurred. At this point, IDAC reduced this power, and by watching the traction motor amperes, it was possible to determine the frequency and magnitude of the reduction. After the slip was corrected, small amounts of additional power were applied to the traction motors until it was obvious that IDAC was in continuous operation and the maximum tractive level had been reached.

Numerous tests were run with each locomotive under varying operating conditions:

1 Stop to 18 mph: Full dynamic brakes and air brakes were applied on the load units. Power was gradually applied by the test unit to prevent gross slip and to get the test train moving. As the variable throttle control was being operated to obtain the maximum tractive effort, the braking force was slowly reduced to build up the speed of the consist.

2 18 to 2 mph: With the consist traveling 18 mph and the test locomotive at maximum throttle, the dynamic braking load was slowly applied to

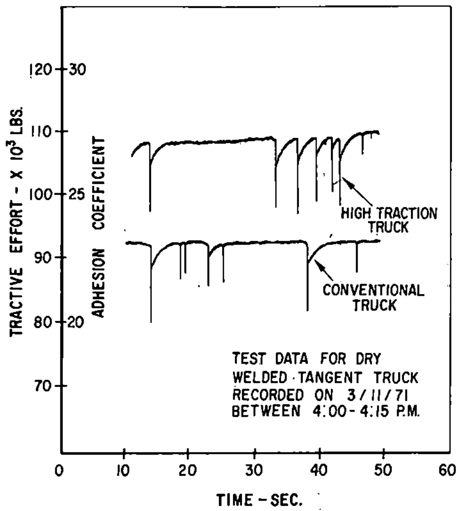


Fig. 9 Actual test traces comparing the HT-C truck to the conventional truck under adhesion-limited conditions—maximum tractive effort versus time for constant speed—5 mph

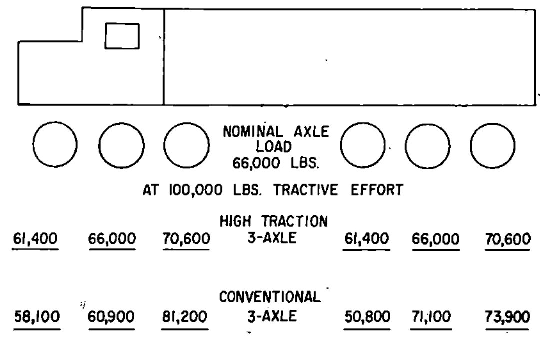


Fig. 10 Axle load distribution for a 400,000-lb locomotive at 25 percent tractive effort

decrease the train speed. Once under 10 mph, the air brakes were used to bring the consist down to 2 mph. While reducing the speed, the variable throttle control was adjusted to maintain maximum power to the wheels without gross slip.

3 Constant speed—5, 10, 15 mph: With the throttle control at the maximum level of tractive effort without gross slip correction, the braking load was adjusted to maintain a constant speed.

The data was recorded on an X-Y plotter located in the EMD Test Car—a fully equipped mobile laboratory containing modern data acquisition and recording instrumentation. The first two testing procedures resulted in tractive effort versus speed curves, while the last procedure provided tractive effort versus time plots at a known constant speed. Since one end of the Test Car had a calibrated drawbar, the tractive effort of the SD-45X was measured directly in drawbar pull, as well as from the main generator current. The tractive effort of the SD-45 was determined from the main generator current. The individual axle speeds of the SD-45X, the train speed, and the track profile were recorded for each test run on an oscillograph record.

Table 1 is a summary of the results obtained on tangent track. The level of the maximum available coefficient of friction was low for dry welded rail; however, it may have been slightly contaminated. The relative values of adhesion between the SD-45X and SD-45 may be considered typical for operation when both locomotives were limited by adhesion and not by the rated horsepower. The HT-C constantly provided 12 to 24 percent higher adhesion than the SD truck, which confirmed the theoretical calculations shown in Fig. 3. At 8 mph, the SD-45X developed 83,000-lb tractive effort, while the

Table 1 Summary of Adhesion Limited Test Data Under Dry Welded Rail Conditions on Tangent Track

	SD-45X With HT-C Truck		SD-45 With Conventional Truck	
Miles per hour	Tractive effort (lb)	Adhesion coefficient (percent)	Tractive effort (lb)	Adhesion coefficient (percent)
8	83,100	20.8	73,900	18.5
10	83,000	20.7	71,700	17.9
12	82,900	20.7	69,500	17.4
14	82,800	20.7	67,300	16.8
16	82,000	20.5	65,100	16.3

SD-45 could deliver only 74,000 lb; this represents a 12 percent increase in drawbar pull. Under conditions in which a conventional three-axle truck can deliver 74,000-lb tractive effort, the predicted value for the HT-C truck is 83,800 or a 13 percent increase in drawbar pull.

A sample of the actual test data for tractive effort versus time at a constant speed of 5 mph is shown in Fig. 9. The available adhesion during this test run was very high, and test results above 7 mph would have been affected by horsepower limitations. However, at 5 mph, both locomotives could deliver the horsepower necessary to develop the maximum tractive effort that the rail could support. The HT-C truck provided 16 percent greater tractive effort than the conventional truck. These test results confirm the validity of the theoretical calculations and the predicted increase in adhesion capability.

One method of checking these test results can be made by using Fig. 2. A conventional truck developing 93,000-lb tractive effort requires a coefficient of friction between the wheel and rail of 0.29 (COF--AD/AE), and at 180,000-lb tractive effort, the HT-C requires a coefficient of friction of 0.29. Thus, both trucks were developing the expected tractive effort for the given track condition.

CONCLUSIONS

It has been shown that the HT-C truck offers improved adhesion capability over the more conventional SD flexicoil truck or other domestic three-axle trucks. Calculations and field evaluations have shown an improvement in the range of 10 to 20 percent. This means that a locomotive with HT-C trucks requires a lower coefficient of friction at the rail or needs less total locomotive weight to produce the same tractive effort than the previous three-axle truck. This also means that the slip risk of a locomotive equipped with high traction trucks should be lower than a locomotive having the same weight but using the conventional design.

An additional benefit of the low weight transfer is the lower maximum axle loads on rails during operation at high tractive effort, Fig. 10. Furthermore, the soft primary suspension provides reduced wheel-rail load variation and lower twisting moments on the truck castings during operation. These aspects are significant in that they offer improved reliability of performance on present commonly encountered track condition.

ACKNOWLEDGMENT

The experimental data presented in this article are the results of tests conducted with the cooperation of our customer railroads in the field, and EMD Experimental Test Instrumentation and Field Engineering Sections. The authors also wish to express their appreciation to L. Buchholz and his group for preparation of the illustrations and to L. F. Koci for his continued advice and encouragement in supporting this work.

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trucks on adjacent cars were farther apart. These were problems that were not given enough considaation before the cars were built and offered in service.

Chairman Smith: Was that a six-wheel car, Mr. Sapp?

Mr. Sapp: Yes it was.

Chairman Smith: I think most of the problem that Mr. Parker is referring to are the four-wheel truck, 100-ton cars. Is that right, Charlie? The 125-ton truck and its effect on the boad bed?

Mr. Parker: C. W. Parker, Canadian Pacific Railway Co. We have gone quite heavily into 100-ton covered hoppers, Loren, and the track people were telling me last week that the shelling has increased and certainly the rail replacements and the rate of rail wear have increased in that territory where these cars operate.

Chairman Smith: Well, I think you all know that the industry has a number of 125-ton cars out, and I still believe that permission is needed in order to go on another line. I just wonder how this has been working out during the last year. Can anyone make a comment on this? Has anybody restricted any use of 125-ton cars on their lines?

Mr. Parker: *Do you mean some of the roads that are taking them or restricting them, or, do you mean those that won't take them at all?*

Chairman Smith: The roads that won't take them at all.

Mr. Parker: We wouldn't touch one at present.

Chairman Smith: There's many who won't, but there's some who will. I imagine you have in your own lines some restrictions on this in order to save your rails. This is about the point that you were bringing out, though, Parker, isn't it? Has anyone else made further studies on this maintenance versus heavier car usage? I guess not. Further questions? Elmer.

Mr. Sebrell: E. H. Sebrell, Seaboard Air Line Railroad Co. Certainly, due to rail conditions and bridge constructions, there are limitations which we on the Seaboard control the cars weight. We tried to use 263,000 lbs. weight on the rail and in some cases, we will go to 275,000 lbs. We do recommend and suggest that these cars that would go to that limit be watched carefully while they are in transit on the road. As far as the six-wheel truck is concerned, we feel that our railroads and the construction of cars be such that you will be getting longer cars, of course, but to get a heavier commodity in a car you will necessarily have to go shorter and probably higher, especially in a hopper car. We have decided and we do put in practice the use of the 100-ton truck with six-wheel centers and the use of 38″ wheels in order to satisfy our Engineering Department and their recommendations of loads to be carried on rails.

Chairman Smith: Further questions or comments from the Pullman presentation?

(APPLAUSE)

Chairman Smith: Because Mr. Hart has not arrived yet, we will just bypass his paper for the time being and go right down to the next paper.

The next paper is "Progressive Development of a High Speed Truck Design" and will be given by Mr. E. J. Warnock, Chief Mechanical Engineer of the Symington Division. Ernie.

Mr. Warnock: My subject is the "Progressive Development of a High Speed Truck". At the 1964 Engineering Conference, I described the development of the XL* Truck concept to provide a modern high speed truck adaptable to roller bearings. Perhaps a short review of this development, as first applied to 50-ton capacity plain bearing trucks and subsequently projected to 70-ton roller bearing trucks will set the stage for a report of an XL-70 Truck application after one year of service.

We reviewed in 1964 the "built-in" snubber truck, pointing out that this design concept using two side frames with a connecting sprung bolster had not been changed since the acceptance of the four-wheel truck by the American Railroads.

True, the springing was changed from "All-Coil" groups to "Coil-Elliptic" groups, to coil-snubber (unit) groups and finally to the current long travel spring, "built-in" friction device trucks. All these changes were at the centrally located truck spring groups to control synchronization of the springs from input shock loadings.

The advent of roller bearings eliminated the ⅜″ or better lateral movement available in the plain journal box assemblies. It made it necessary to provide lateral freedom between the frame and bolster by allowing the bolster to float laterally on the spring groups.

The change of lateral movement location from the journal box to the bolster and frame connection does not effectively control hunting of the wheels and axles in high speed service and results in repeated nosing conditions with excess wear of the truck parts, particularly between adapters and pedestal jaw roofs.

The XL design concept was originally developed in 1946, for 50-ton plain bearing trucks. Over 100 car sets were placed in service subsequent to this date. In many of these applications assigned cars were designated as test cars, in which pre-service measurements were taken to determine the extent

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of wear and serviceability. One group of cars in mail baggage service had units which accumulated mileage exceeding 800,000 miles without replacement of XL parts.

Based on the knowledge that roller bearing trucks in high speed service should have lateral movement and control such as provided in the XL concept at the journal location and that current conditions require maintenance free trucks for high mileage service, the 70-ton capacity XL Truck was designed and furnished.

We now have a few slides to review the concept of the "XL" Truck.

Fig. 1. XL-70 Truck.

The XL-70 Truck shown is for application to 70-ton cars. The truck frames, bolsters and journal housings are manufactured in high tensile cast steel to AAR Spec. M-201 Grade "C". It has separate journal housings, each with supporting coil and friction elements.

This construction reduces the unsprung weight and prevents transmission of uncushioned road shock to the side frame by absorbing these shocks as closely as possible to their point of origin.

Fig. 2. Journal Housing.

Showing load spring, and friction element assembly.

Each journal housing carries a separate set of expanding elements mounted between flanges on each side of the housing. There is ⅝″ lateral movement allowed inboard and outboard of the journal housings with respect to the frames. This provides lateral movement at the journals similar to the plain type bearing. The same friction elements that control vertical oscillations control lateral movements. The friction assembly moves vertically with the housing, but stays with the frame pedestal in the lateral direction so that there is lateral friction between the faces of the shoes and the housing liners and vertical friction control between the sides of the shoes and the liners in the pedestal.

Fig. 3. Sectional View Through Friction Elements.

The legs of the pedestal jaws of the frame form recessed channel walls. The inside vertical walls have hardened steel liners.

Fig. 4. Sectional View of Truck Frame and Bolster Assembly.

The side frames function as free acting equalizers, rocking fore and aft relative to the bolsters, either simultaneously or independently. This contributes to equal deflection of pedestal load springs and insures equalization of the truck.

The rocking feature is provided by a large radius on the end of the bolster where it contacts the side frame center member.

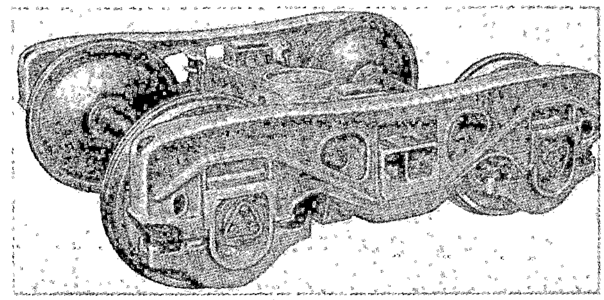

Fig. 1. XL-70 Truck.

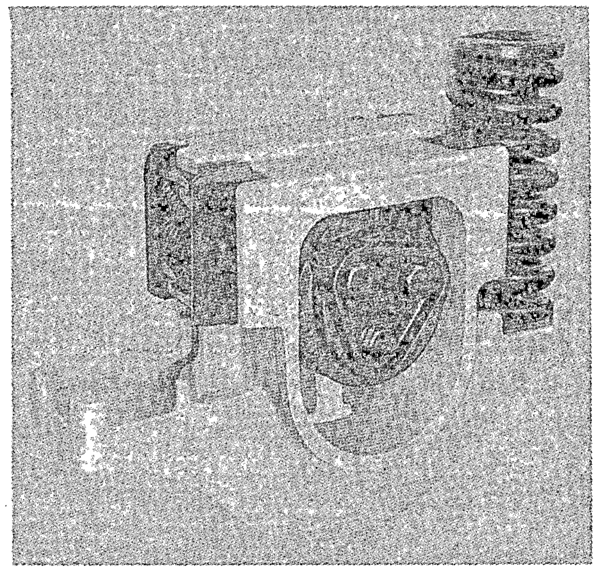

Fig. 2. Journal Housing.

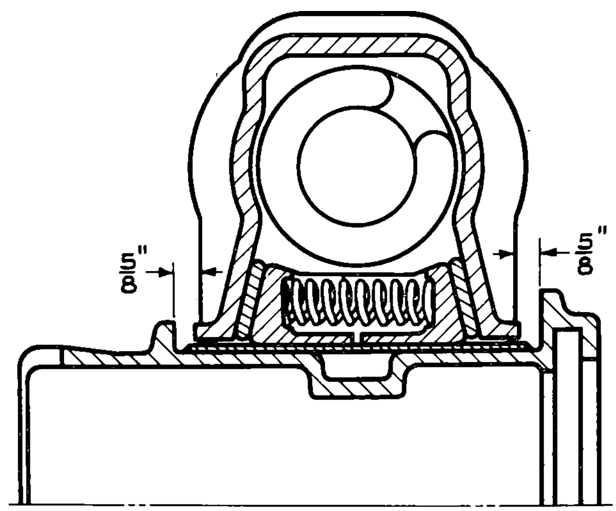

Fig. 3. Sectional View Through Friction Elements.

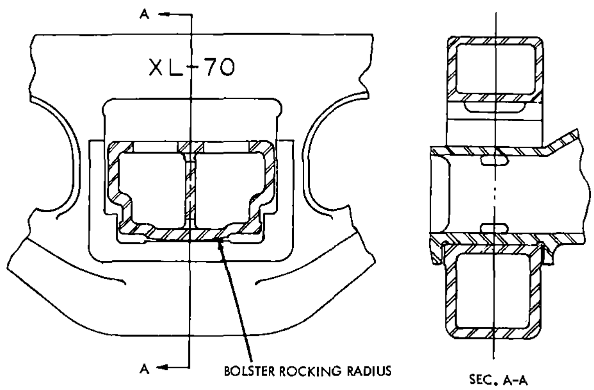

Fig. 4. Sectional View of Truck Frame and Bolster Assembly.

The side frames and bolsters are held in square alignment by providing close fitting flanges on the bolster which seat on the side frame center member.

The brief rundown above was to refresh your memory if you attended the 1964 Conference, or to acquaint you with the XL design concept if you did not.

During January, 1966, 35 car sets of XL-70 Trucks were applied to NYC Flexi-Van Cars equipped for head end operation. Preliminary inspections of these cars indicates satisfactory conditions. A complete inspection is scheduled after the cars have accumulated additional mileage.

Nearly a year earlier, in May, 1965, ten car sets of XL-70 Trucks were applied to PFE 70-ton Mechanical Refrigerator Cars built by Pacific Car & Foundry.

These cars were assigned car numbers PFE 452741 to 452750 inclusive. Of the ten cars, two were designated as having test trucks. The trucks under these cars, numbers PFE 452745 and PFE 452746, were carefully measured at contact surfaces and this pre-service data recorded. During July, 1966, after approximately one year's service and 40,000 miles, car PFE 452745 was sent to Depew for inspection.

The car and trucks were carefully examined as received, the trucks were rolled out and dismantled, and measurements taken for comparison with the pre-service data. The XL-70 trucks under the PFE have the following descriptions:

6 x 11″ A.A.R. Hyatt Roller Bearings and Standard A.A.R. Adapters.

A.A.R. Standard Hung #18 Brake Beams and Hangers.

Truck Levers with connection rod under bolster, foundation rigging standard.

Truck center plate height 25¾″ light car.

14″ dia. truck bolster center plate, with 11 14 per cent Manganese wear rings applied and horizontal bearing surface machined.

Stucki, single roller type 656-C Side Bearings.

Truck spring load group designed for 220,000 lbs. maximum rail load, with 3¾″ effective travel, coils shot peened and end-ground for parallelism.

Slack Adjuster Elcon National Peacock #1340.

The next series of figures show the car and measurements of components of the trucks inspected.

Fig. 5. Photograph of the PFE 452745 Car As Received at Depew for Inspection In July, 1966.

Fig. 6. Photograph of a Journal Box Assembly before Dismantling of Trucks. Note that this housing, although it is free to lateral ⅝″, does not show heavy contact between the journal box flanges and the frame pedestal jaw.

Fig. 7. Wheel Numbers Record showing that the original wheels applied to the car as built had not been changed out. Note the record of flange thickness taken with the AAR steel wheel gauge showing an average of 3/32″ wear.

Fig. 8. Roller Bearing Record sheet showing that original bearings were on car at this inspection indicating that the original wheels, axles and bearings had not been changed out.

Fig. 9. Wheel Contours from the A-End and B-End Trucks.

The full line represents the theoretical wheel contour as copied from the A.A.R. Manual. The dotted line shows the wear contours taken with a wheel contour gage at this inspection.

Fig. 10. A.A.R. Roller Bearing Adapter Dimensions. Showing location of dimensions taken as applied and at this inspection. The average shown is wear for a truck set of adapters.

Fig. 11. Journal Housing Dimensions: Average shown for carset.

Fig. 12. The Snubber Shoe Dimensions.

Fig. 13. Side Frame Pedestal Dimensions.

Fig. 14. Side Frame Center Connection Dimensions.

Fig. 15. Bolster End Dimensions.

In summation, the results of this inspection indicate that all components of the XL-70 Trucks applied to PFE Mechanical Refrigerator Cars are functioning as designed. After approximately 40,000 miles initial service, the wear is a minimum. There is in some locations slightly greater wear than at others. We believe this condition to be the result of adjustment of new assembled parts to establish alignment positions. Another inspection of this car will be made next year after accumulation of additional mileage.

Specifically, we would emphasize the following:

1. Contact surface wear for both friction elements

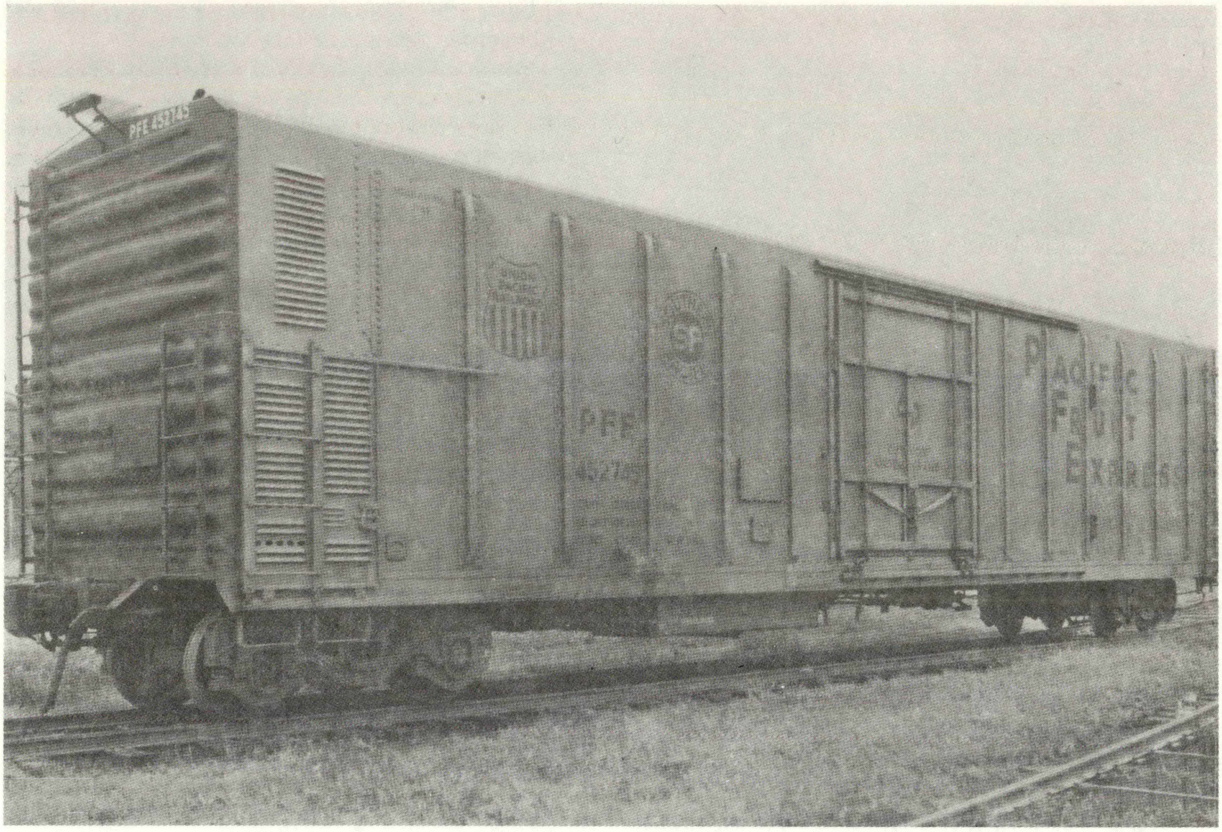

Fig. 5. PFE 452745 Car As Received at Depew for Inspection in July, 1966.

and alignment control areas are minimum.

2. Wear on friction shoes, journal housings and pedestal liners, negligible, in most cases consistent with establishing full contact of the bearing surfaces. We expect continued service of these components for long life before replacement.
3. The wear patterns of the squaring control surfaces, ie., truck frame and bolster squaring connection and rocker feature, are a minimum and due to adjustment of parts to establish alignment contact.
4. Wear of the roller bearing adapters was not measurable. Contact patterns in the journal housing roof from the adapters indicates that the adapter is not wearing into the housing roof, but is moving with the journal housing during lateral displacement. We believe this will minimize the excessive wear conditions now occurring on conventional narrow pedestal truck frames between the jaw roof and the adapter.
5. The wheel wear contours taken at this inspection indicate that lateral and vertical control at the journals is effectively minimizing wheel wear.

In final summation we are satisfied that this 70-ton truck, designed for use with roller bearings, indicates from this inspection, after the initial 40,000 miles of service, that it will perform as satisfactorily as the previous 50-ton plain bearing design truck in attaining high mileage with minimum maintenance.

Chairman Smith: Questions? Dr. Manos.

Dr. Manos: *W. P. Manos, Pullman-Standard. I have a question. How does this truck perform at low speeds over rough track?*

Mr. Warnock: Your reference to performance at low speed and rough track we assume is related to the rock problem now encountered on 100-ton hopper cars. We have not tested the XL-70 in this type of service. We are now considering making such a test. The design concept, with coil springs at each journal box and the freedom of each frame to rock, vertically at the center bolster connection could very well break up the harmonics causing the rock condition.

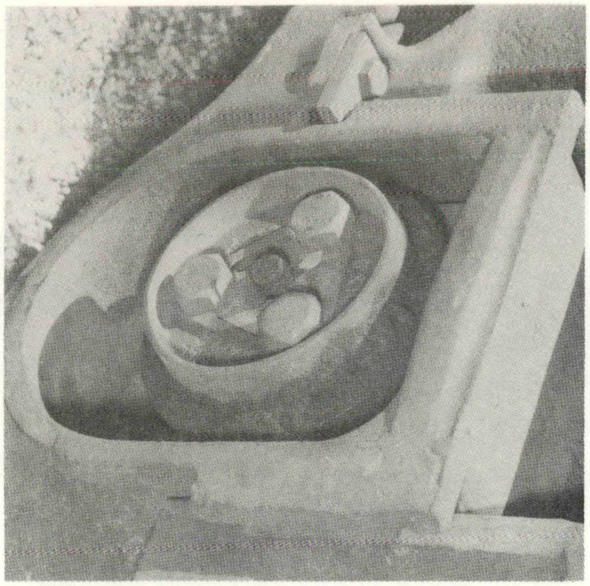

Fig. 6. Photograph of a Journal Box Assembly before Dismantling of Trucks.

SYMINGTON WAYNE CORPORATION
SYMINGTON DIVISION

"XL-70" TRUCK SERVICE TEST

WHEEL RECORD

RAILROAD PFE TRUCK ASSY. R-8382-A

CAR NO. 452745 *GB - Griffin Wheel Co. WHEEL TYPE CS2-8R (CM-33)

POSITION		ORIGINAL	INSP. DATE	INSP. DATE	INSP. DATE	INSP. DATE
		5-18-65	7-19-66			
L-1	WHEEL NO.	59944	59944			
	MFR.	GB *	GB *			
	DATE	1-65	1-65			
	RIM. THICK.	2-3/32	2"			
L-2	WHEEL NO.	61772	61772			
	MFR.	GB *	GB *			
	DATE	2-65	2-65			
	RIM. THICK.	2-3/32	2"			
L-3	WHEEL NO.	61729	61729			
	MFR.	GB *	GB *			
	DATE	2-65	2-65			
	RIM. THICK.	2-3/32	2"			
L-4	WHEEL NO.	58307	58307			
	MFR.	GB *	GB *			
	DATE	1-65	1-65			
	RIM. THICK.	2-1/16	1-15/16			
R-1	WHEEL NO.	61284	61284			
	MFR.	GB *	GB *			
	DATE	1-65	1-65			
	RIM. THICK.	2-1/16	1-31/32			
R-2	WHEEL NO.	64199	64199			
	MFR.	GB *	GB *			
	DATE	2-65	2-65			
	RIM. THICK.	2-3/32	2-1/16			
R-3	WHEEL NO.	60068	60068			
	MFR.	GB *	GB *			
	DATE	1-65	1-65			
	RIM. THICK.	2-1/16	1-31/32			
R-4	WHEEL NO.	68450	68450			
	MFR.	GB *	GB *			
	DATE	2-65	2-65			
	RIM. THICK.	2-3/32	2"			

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Fig. 7. Wheel Numbers Record.

SYMINGTON WAYNE CORPORATION
SYMINGTON DIVISION

"XL-70" TRUCK SERVICE TEST

ROLLER BEARING RECORD

RAILROAD PFE TRUCK ASSEMBLY R-8382-A

CAR NO. 452745 BEARING TYPE AAR 6 (6 x 11)

		ORIGINAL	INSP. DATE	INSP. DATE	INSP. DATE	INSP. DATE
POSITION		5-18-65	7-19-66			
L-1	SER. NO.	65C2797	65C2797			
	MFR.	HYATT	HYATT			
L-2	SER. NO.	65C2800	65C2800			
	MFR.	HYATT	HYATT			
L-3	SER. NO.	65C416	65C416			
	MFR.	HYATT	HYATT			
L-4	SER. NO.	65C423	65C423			
	MFR.	HYATT	HYATT			
R-1	SER. NO.	65C2920	65C2920			
	MFR.	HYATT	HYATT			
R-2	SER. NO.	65C2913	65C2913			
	MFR.	HYATT	HYATT			
R-3	SER. NO.	65C379	65C379			
	MFR.	HYATT	HYATT			
R-4	SER. NO.	65C378	65C378			
	MFR.	HYATT	HYATT			

REPORT NO. TE-1965-1, Supplement #1

Fig. 8. Roller Bearing Record.

Chairman Smith: Mr. McGregor.

Mr. McGregor: *L. S. McGregor, Canadian National Railways. I have a few comments to make, and I would like to ask a few questions. One, what is the actual place of the XL-70 Truck in our railway industry? I mean it is obviously more expensive.*

Mr. Warnock: That is right, sir.

Mr. McGregor: You commented on high speed. I am assuming by that the truck is designed for the 80 to 85 miles an hour level.

Mr. Warnock: The cars in NYC's Flexi-Van service are running 80 to 90 miles an hour in head end service, as John Reehling, who is here, can verify. Is that right, John?

Mr. Reehling: J. R. Reehling, New York Central System. Right.

Mr. McGregor: With conventional freight trucks we are operating on trains that are operating at 70 to 75 miles per hour without too much difficulty. As far as wear is concerned, the present truck is within what we may call economic maintenance limits. With this new truck we are getting into a controversial area deviating with some of the points raised by our speaker last night on minimizing cost of cars. Actually, I believe that railway cars are going to be equipped with roller bearings throughout in the near future, and this XL-70 Truck offers

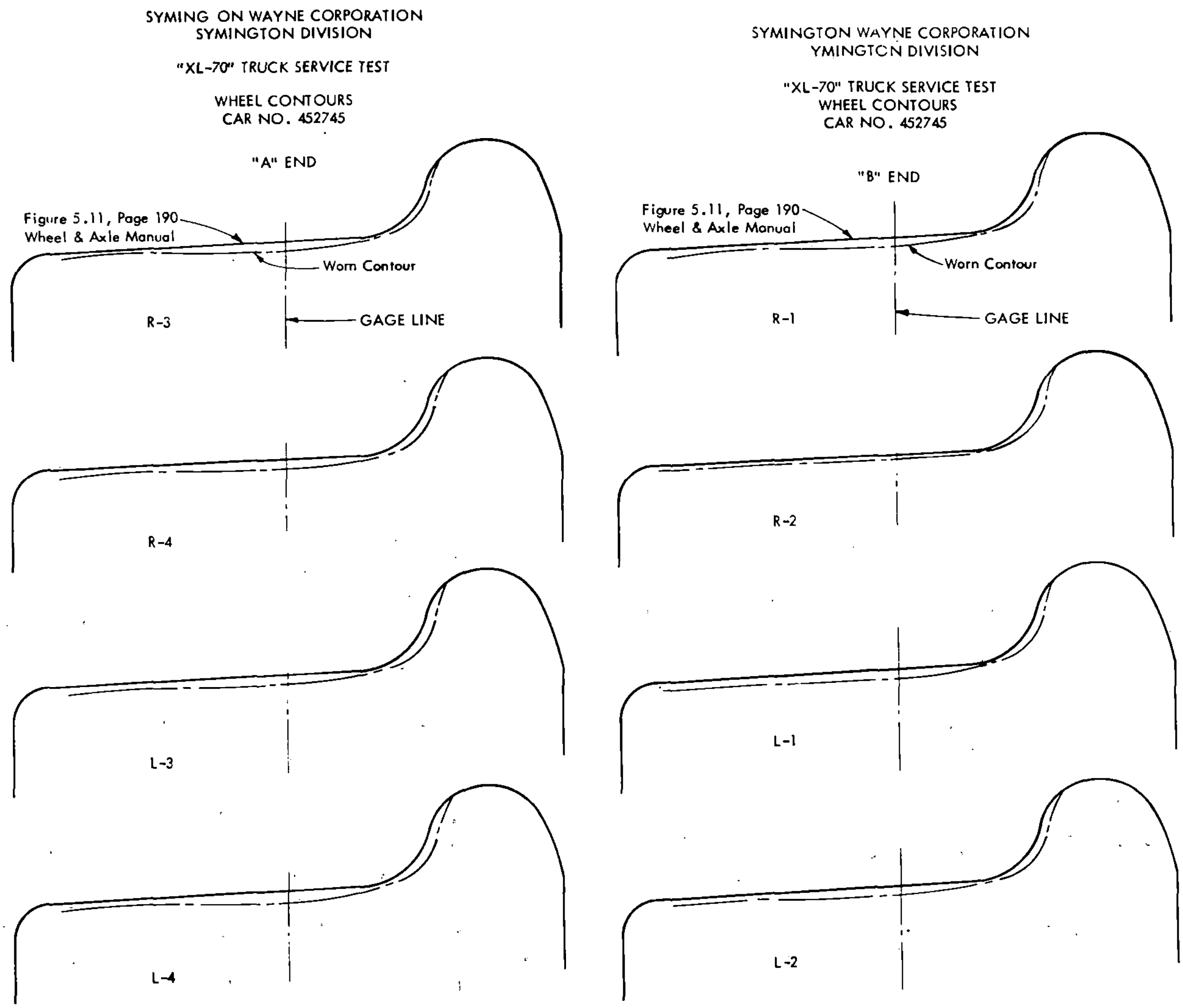


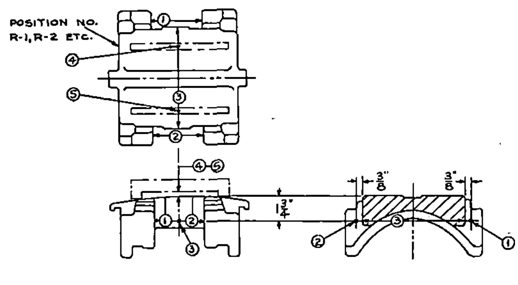
Fig. 9. Wheel Contours from the A-End and B-End Trucks.

an answer to the conventional truck shortcomings insofar as the roller bearing is concerned.

Mr. Warnock: The design of the XL-70 truck provides for long service life and low maintenance. We are not only talking of physical maintenance or replacement of truck parts but keeping the car that the truck is applied to in service without loss of time. We are therefore talking about maximum use of the car through maximum dependability of the truck components and through the consistent use of the truck design features reducing the lading shock and damage occurring with the present current conventional truck in high speed and high mileage service.

The vertical cushioning of any current truck is limited by the fact that the maximum sprung travel available is the approximately 3″ vertical travel allowed at the center of draft between light car coupler height and loaded car coupler height.

Therefore, for improvement you must design like we did, placing separate journal boxes mounting load springs at each axle end, providing cushioned vertical and lateral control before road shocks enter the car. This design concept reduces the upsprung weight of the truck and allows reduced friction control. The combination of the improved vertical and lateral control at this location prevents the build up of synchronized road shocks. The truck is designed to provide satisfactory service with roller bearings.



LOCATIONS 1 & 2	-	WIDTH BETWEEN GUIDE LUG STOPS	AVG. .01"
LOCATION 3	-	OVERALL WIDTH	AVG. .01"
LOCATIONS 4 & 5	-	MEASUREMENT TO DETERMINE WEAR ON CROWN	AVG. .01"

Fig. 10. A.A.R. Roller Bearing Adapter Dimensions.

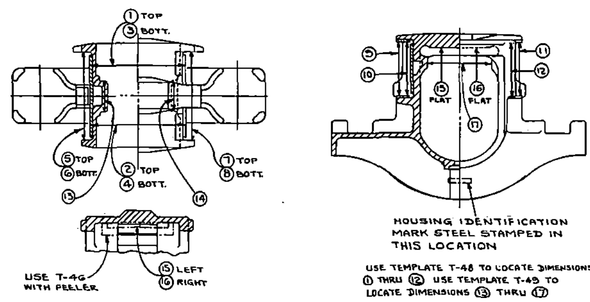
We know that wheel hunting and truck nosing conditions can be prevented by controlled lateral movement at the roller bearing location. We also know from experience that the current standard narrow pedestal jaw truck and roller bearing combination with limited lateral on our test cars gives violent truck nosing action with greased center plates when speeds exceed 60 miles per hour. I can see from the heads nodding in the audience that some of you have had the same experiences. From test run experience, we also know that the XL concept, with controlled lateral movement does not allow this synchronization to occur. Les, I believe this gives you the information you request.

Mr. McGregor: *I just wanted to get remarks in respect to roller bearing lateral oscillation which is a point of discussion in many of the general committee and other meetings. There was just one other aspect regarding this truck design. Are you contemplating developing and designing it for the 100-ton cars, and if so, do you think this might alleviate some of our truck problems regarding maintenance standard requirements, etc.*

Mr. Warnock: The 100-ton is designed on paper. The design is being evaluated. A 100-ton XL Truck for high speed service would be used on box cars or a similar type car. What is the future market for this type of service? In this respect this is the information we require, and you people will undoubtedly be asked this question in the future by our Sales.

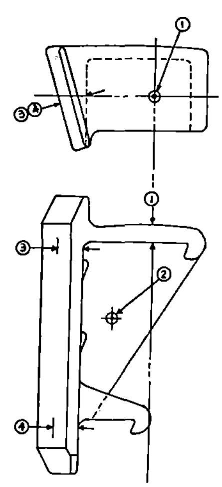
Mr. Koci: *L. F. Koci, Electro-Motive. I have three questions. On what basis did you establish the plus or minus 5/8" lateral of the journals, that is, did you try any other values?*

Mr. Warnock: The original XL 50-Ton Truck was designed with plain bearings, which provides approximately 3/8" lateral movement inboard and outboard between the axle and the journal box as allowed by clearance between the wedge stops, guide lugs, brass and wedge combinations. Tests and road runs of the 50-ton XL indicated that the addition of 5/8" travel inboard and outboard at each box between the box and the pedestal jaws provided the optimum results. This would be a total of 1" inboard and 1" outboard. Before designing the 70-ton XL we modified the plain bearing journal boxes on the 50-ton truck to accept A.A.R. package roller bearings and made several road tests without additional change. These tests indicated that with roller bearings the 5/8" lateral between the box and the frame gave a satisfactory control.



LOCATIONS 1, 2, 3, 4	SIDE LINERS	AVG. .01"
LOCATIONS 5, 6, 7, 8	WIDTH PEDESTAL FLANGES	AVG. .02"
LOCATIONS 9, 10, 11, 12	VERTICAL SHOE LINERS	AVG. .04"
LOCATIONS 13, 14	WIDTH OF GUIDES	AVG. 0"
LOCATIONS 15, 16	WEAR ON ROOF	AVG. .03"

Fig. 11. Journal Housing Dimensions.



LOCATION 1	
THICKNESS THRU TOP CONTACT WITH JNL. BOX LINER	WEAR 0"
LOCATION 2	
THICKNESS THRU SIDE CONTACT WITH JNL. BOX SIDE LINERS	WEAR 0"
LOCATIONS 3 & 4	
THICKNESS THRU SHOE FACE CONTACT WITH PEDESTAL LINER	WEAR 0"

Fig. 12. The Snubber Shoe Dimensions.

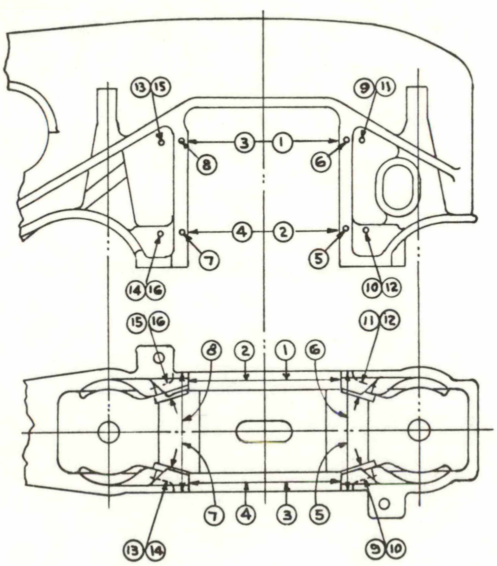
The XL-70 was designed with the ⅝" lateral and when road tested indicated with roller bearings that the lateral control was satisfactory. You must remember that the lateral resistance of the load springs and the amount of lateral friction control also entered in this performance.

This inspection reported on the PFE cars indicates that the present XL-70 arrangement controls lateral shocks, so that synchronization does not occur. There is very little evidence of contact between the journal housing flanges and the corresponding pedestal jaw bearing surface at the PFE inspection.

Mr. Koci: *My second question, again just so that I have it right. Were these measurements that you were getting after about 40,000 miles of service?*

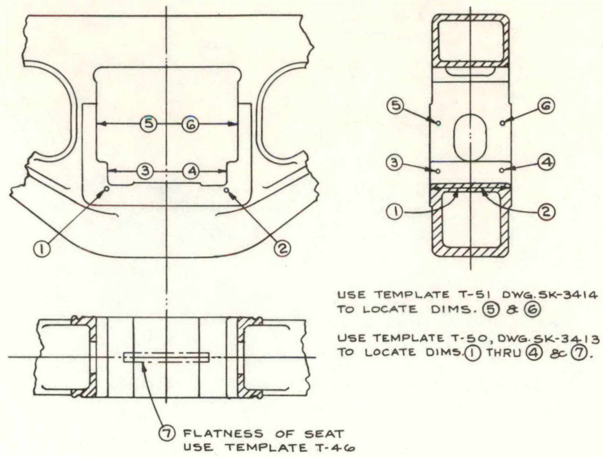
Mr. Warnock: That is correct. All components of the test trucks were measured as shown on the slides before and after the initial 40,000 miles service.

Mr. Koci: *My third question, you said the journal springs were equivalent to 3¾" effective travel. Is this free to solid or free to static?*



LOCATIONS 1, 2, 3, 4	
PEDESTAL OPENING WIDTH	AVG. WEAR .02"
LOCATIONS 5, 6, 7, 8	
PEDESTAL JAW WIDTH	AVG. WEAR .02"
LOCATIONS 9, 10, 11, 12, 13, 14, 15, 16	
THICKNESS THRU PEDESTAL JAW AND LINER	AVG. WEAR .01"

Fig. 13. Side Frame Pedestal Dimensions.



LOCATIONS 1 & 2	
WIDTH OF SEAT	AVG. WEAR .04"
LOCATIONS 3 & 4	
WIDTH BOLSTER STOPS	AVG. WEAR .04"
LOCATIONS 5 & 6	
UPPER BOLSTER OPENING WIDTH	AVG. WEAR 0"
LOCATION 7	
BOLSTER SEAT	AVG. WEAR .02"

Fig. 14. Side Frame Center Connection Dimensions.

Mr. Warnock: Actually, the spring has 3⅞" travel, but we designed the truck so that the journal box will go solid in the frame ⅛" before the spring goes solid. Therefore, this would be 3¾" effective travel of the spring group in the truck between free and solid.

Chairman Smith: Gentlemen, let me interrupt this session right here to say that it is time for intermission. We have allowed a final session for a general discussion, and this will include bringing up any paper that was discussed during the day. The last session this afternoon you can come back and ask any of the speakers further questions.

(APPLAUSE)

Chairman Smith: The next paper is entitled, "Cushioning Requirement Trends in New Car Design" and will be given by Mr. C. C. Leriche, Consulting Engineer — Design & Construction — Railway Freight Cars, Association of American Railroads. Mr. Leriche.

Mr. Leriche: Thank you, Mr. Smith. Gentlemen, as my subject deals with "Cushioning Requirement

Editor's Note: Two papers, one by Masaharu KUNIEDA and one by Keiji YOKOSE, which are directly pertinent to U.S. four wheel truck practice are published in full, following immediately. Copies of other papers, as listed below, may be obtained from Dresser Transportation Equipment Division upon request:

- Theoretical Study of the Side Thrust of Truck Wheels Running over Curves, Based Upon the Friction Theory – by Masaharu KUNIEDA
- Experiment of Hunting Derailment with a One-Fifth Model Wheelset – by Keiji YOKOSE
- On Optimum Value of Dry Friction in Bearing Spring for Two-Axle Cars – by Keiji YOKOSE
- Design of Wheel Tread for Two-Axle Railway Vehicle – by Keiji YOKOSE
- Effect of Non-Linear Characteristic on Hunting of Cars – Effect of Side Play Between Axle and Bearing Metal – by Keiji YOKOSE
- On the Tyre Profile of Freight Car Wheels – by Nobuo MATSUI

Calculation on Hunting of High Speed Railway Truck – Problems of Truck Design for SANYO SHIN KANSEN

by:
Keiji Yokose
Car Dynamics Laboratory

(As published in QUARTERLY REPORTS, Vol. 11, No. 2 (1970), Pgs. 108-113)

It is extremely important to prevent the hunting of railway vehicle which occurs at high speed, for securing stability of the car. In this report, there treated in detail for the hunting phenomenon of the railway vehicles, and for the calculation ground which makes the basic design of the truck for the SANYO SHIN KANSEN.

According to these results for the test truck described above, it is clarified that there are many factors containing the truck which affect the hunting velocity, for instance, the supporting stiffness of axles, the elastic stiffness between the truck frame and the side frame of truck, the frictional moment between the truck and the other various elements.

And it is clarified that there are many factors which prevent the truck hunting at the planned maximum, speed and the fundamental data on the truck design are presented.

1. Introduction

It has been experienced that a lateral oscillation occurs in case of the railway vehicle rolling on the rails at high speed. Therefore, the prevention of this lateral oscillation makes a contribution to both the running stability of the car and an increase of the transport capacity. Accordingly, this problem has been studied for a long time, for securing the solution of the hunting phenomenon and the preventional countermeasure. It will be regarded as good results to apply this study to the railway vehicles which run not only at high speed but also with good riding comfort such as the SHIN KANSEN.

The lateral oscillation, which is called "hunting", is not concerned with the rail condition, and it is classified as "self-exciting oscillation" that can take place even on an ideal straight track.

There are two kinds of hunting, that is to say, the geometrical hunting and the dynamical hunting. Namely, the former takes place by the effect of the inclination of the wheel treads, and the latter takes place through the medium of the creep force acting on the contact points between the wheel and the rail, but generally speaking, both are connected with each other.

The geometrical hunting had been solved for the first time by Klingel in 1883. The hunting study is becoming a worldwide study recently, and as a result, the hunting phenomenon and its preventional countermeasure have been clarified. But because of the complicated phenomenon there are many problems remaining unsolved.

In solving the above-mentioned hunting problem, main difficulty is that a car has not only many degrees of freedom of motion but also many nonlinearity parts, for instance, the side play existing various parts, the friction of rubbing surface, the variation of the contact condition which is accompanied by the relative wear between the wheel and the rail, and the slip which appears at the contact points between the wheel and the rails, etc. In addition to these factors, there is the irregularity of the rails which is transmitted to the car.

According to many studies which have been made so far, there are two kinds of the hunting of car. That is to say, one type, the body hunting (primary hunting) that takes place at relatively low speed, and the other type, the wheelset hunting or the truck hunting (secondary hunting) that occurs at relatively high speed.

The theoretical analysis of hunting many degrees of freedom can not be solved by any possibility by manual calculation, but accordingly as the development of electronic computer the recognition for the hunting phenomenon is becoming clarified. By this study the numerical optimum values concerning the car were determined, the

Reprinted from Technical Proceedings - 1973 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043.

values which held the satisfactory hunting characteristic of the car, and the calculation results provided the fundamental data of the car design[1].

The main points for the hunting of car are the preventing of the body hunting for the 2-axle car and that of the truck hunting for the high speed car as the SHIN KANSEN.

2. On the Slip Phenomenon between Wheel and Rail

The slip has already been clarified to exist at the contact point between wheel and rail. Considering the wheelset which is stationally placed on rails, it was clarified by Hertz that the contact area became elliptic due to the elastic deformation. In case of the wheelset rolling on the rails quietly as shown in Fig. 1, the forward part of the wheel is pressed and the backward part is stretched. On the other hand, the opposite phenomenon to the wheel occurs in the rail part and there happens a slight racing or a slip due to the shearing force. Because of this slip the wheel does not move forward as much as the peripheral length of wheel for one rotation.

It has already been clarified that there are two distinguished domains which are called "adhesion" and "slip" within the contact part of the elliptic zone. The adhesional zone decreases as the rolling speed of the wheel increases, and finally it is transformed to the slip completely. The ratio of slip speed to forward speed of wheelset is called "slip ratio" and the following relation holds between the tangential force acting at the contact surface and the slip ratio.

tangential force = – constant coefficient
× slip speed/forward speed ············(1)

The proportional constant is called the creep coefficient, the theoretical analysis and the experimental study have been made by Cain, Levi, Carter and Davies. Recently the study has been made also by Chartet, Johnson and Kalker. And in JNR Matsudaira and the author made a study using 1/10 and 1/5 scale model wheelsets. An example of the model experimental results is shown in Fig. 2.

According to Levi and Chartet, the following relation holds between the tangential force and the slip ratio.

$$\frac{1}{\tau^{\alpha}} = \frac{1}{\mu^{\alpha}} + \frac{1}{(\kappa\nu)^{\alpha}} \qquad \cdots\cdots\cdots(2)$$

where $r = T/N$, $\kappa = Eab/N$, ν is a slip ratio, E is a Young's modulus, a and b are respectively the half length of the major and the minor axis of the ellipse at the contact point between wheel and rail, μ is a static frictional coefficient and $\propto$ is an exponent.

From Eq. (2) when the slip ratio is very small, ν becomes nearly zero. That is:

The tangential force is proportional to the slip ratio, and as the slip gradually increases, that is to say, when ν is infinity the tangential force converges to a constant value. The model experimental

$$\tau = \kappa\nu \qquad \cdots\cdots\cdots(3)$$

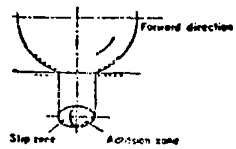


Fig. 1. Contact between wheel and rail

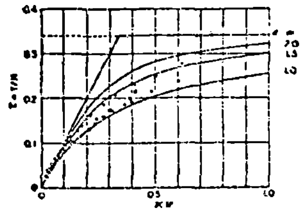


Fig. 2. Relation between slip ratio and tangential force

result coincides with the theoretical calculation which is given by Eq. (2).

The creep theory is extremely important to analyze the hunting. Carter's research was the first which recognized the importance of creep to analyze the hunting of the railway vehicle. Using this theory the differential equations of motion for the car are linearized. Generally the linear creep theory has been applied to the theoretical analysis of hunting.

There appears the same phenomemon for the lateral direction of the wheelset as the forward direction. Strictly speaking, both the creep co-efficients have little disparity in values, but in case of the hunting analysis we have carried out using the same value so far.

3. Hunting of Car

3.1 *Geometrical hunting*

The formula for the wave length of the geometrical hunting, in case of a conical wheel tread rolling on straight rails having corner edges, had been derived by Klingel for the first time, which is the basis of the hunting theory.

According to his theory, the differential equation of a wheelset motion is given by the following formula: $\frac{d^2y}{dx^2} + \frac{\gamma}{br}y = 0$ $\cdots\cdots\cdots(4)$

where x is a forward distance of wheelset, y is a lateral displacement, b is a half distance between the contact points of the wheel tread and the rails in lateral direction, γ is a slope of the wheel tread at the contact point between the wheel and the rail.

In solving Eq. (4) the wave length of hunting, S, is given by the following formula:

$$S = 2\pi\sqrt{br/\gamma} \qquad \cdots\cdots\cdots(5)$$

From the formula (5), as γ becomes small, the wave length of hunting becomes large. The value of γ for the standard wheel tread in the prototype car is one-twentieth, whereas that for the SHIN KANSEN is one-fortieth.

As the running speed of vehicle increases, and the tire and the rail becomes worn out relatively. On account of this reason, the slope of the tire tread at the contact point between wheel and rail becomes large.

Heumann and Matsudaira had derived the following formula for the effective slope, γ_e,

$$\gamma_e = \gamma/(1 - \rho_R/\rho_T) \qquad \cdots\cdots\cdots(6)$$

where ρ_R, ρ_r are radii of curvature of the rail and of the wheel respectively at the contact point between wheel and rail. Substituting (6) in place of γ of (5), the wavelength of the geometrical hunting under the condition of worn wheel and rail is calculated.

In the conical wheel tread, as $\rho r \to \infty$, γ_e of formula (6) becomes γ. Therefore, as the wheel tread becomes worn the value of γ_e which is in relation to $\rho_R / \rho r$ becomes large, and as the result the wave length of hunting becomes shorter.

3.2 *Truck hunting*

There are two kinds of hunting on bogie car; that is to say, the body hunting which occurs at low speed and the truck hunting which occurs at high speed. The body hunting can be eliminated by adding the reasonable damping to the body suspension. On the other hand, the truck hunting, once it occurs, becomes severe and never decreases, so it is very dangerous. The phenomenon has already been experienced through the model experiments and the tests of the SHIN KANSEN.[2]

It is extremely important to prevent the truck hunting of the high speed car, because the occurence of the truck hunting may become a main factor which affects not only on comfort to drive but also sometimes on the derailment and the gauge expansion.

To prevent the truck hunting, the following measures have been considered and accepted.

(1) Making the slope of wheel tread small.

(2) Giving the reasonable supporting stiffness to the longitudinal and lateral support of axles, respectively.

(3) Giving the moment of frictional resistance against the truck turning with spring in series.

In the truck of the SHIN KANSEN, it was determined to design the truck speed which never raised the truck hunting at the planned speed 250 km/h.

On the design of the SANYO SHIN KANSEN, it was required to determine the value of the various factors in the truck to prevent the hunting which appears at high speed.

The equations of motion for the truck were linearized by the describing function method and were solved. As the result, it was clarified that to prevent the hunting there existed the optimum values of the turning stiffness and of the frictional force against the truck turning. And many affecting factors in the truck to the hunting speed, and the fundamental data on truck design were presented.

4. Calculation of Truck Hunting for SANYO SHIN KANSEN[3]

4.1 *Theoretical calculation*

The test truck for the SANYO SHIN KANSEN is shown roughly in Fig. 3. Being omitted the rolling of the truck frame and the side frame, the lateral motion of the truck is expressed a system having 12 degrees of freedom. In order to derive the equations of motion, the following assumption and simplification for calculation are made.

(1) No motion of the side frame in longitudinal direction is assumed.

(2) The right and left side frames are assumed to oscillate relatively in phase.

According to the assumptions derived above, the truck motion can be expressed by the system having 8 degrees of freedom. That is, the truck frame and the side frame have 4 degrees of

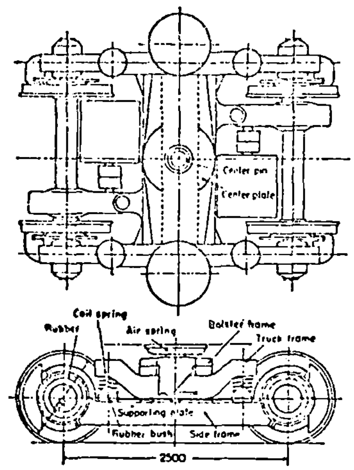


Fig. 3. Outline of the truck for SANYO SHIN KANSEN (above) plan view, (below) side view

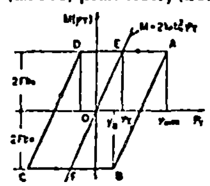


Fig. 4. Relation between turning angle ϕr and restoring moment $M(\phi r)$

freedom in all, in lateral motion and in the turning motion around the vertical axis, and the front axle and the rear one have 4 degrees of freedom in all, in the lateral motion and in the rotational motion around the vertical axis. In the system as the elastic force at the bolster anchor and the frictional moment by the centerplate are acting to the truck frame in the rotational motion around the vertical axis, the force of side bearers draws a hysteresis curve as shown in Fig. 4.[4] That is to say, when the turning angle ϕr is very small and the elastic force is less than the frictional force, the angular displacement of the truck is proportional to the restoring force of bolster anchors. And as ϕr increases to a certain value, the side bearers begin to slip, and as a result, the side bearer force draws a hysteresis curve. It is very difficult to solve the above-mentioned problem exactly. In this case, the system can be linearized by applying the describing function method of the automatic control theory.

Using this method, the following relation holds between the turning moment of the truck $M(\phi r)$ and the angular displacement ϕr.[5]

$$M(\phi r) = (A + jB), \quad j = \sqrt{-1} \qquad \cdots(7)$$

where A and B are determined by the describing function method, and these values are related in the frictional force of the centerplate, in the turning amplitude of truck and in the turning stiffness. They are constants in the steady state of the truck motion. When ϕr is very small, only the elastic force is the restoring force. That is to say, $A \neq 0$ and $B = 0$.

In order to derive the equations of motion of the truck, here it is assumed that the car body is fixed in space, the only force by the weight of car body acts to the truck as the vertical load, and again the linear creep law is assumed to apply the contact point between wheel and rail. Under these assumptions described above, the characteristic equation of the truck motion is derived as follows:

$$\sum_{i=0}^{16} (\bar{A}_i + j\bar{B}_i)\lambda^i = 0 \qquad \cdots\cdots\cdots(8)$$

where $\bar{A}_i$ and $\bar{B}_i$ are calculated by the Danilevsky's method with an electronic digital computer.

Solving Eq. (8), we obtain the roots of the characteristic equation as follows:

$$\lambda_i = \alpha_i + j\omega_i, \quad j = \sqrt{-1} \qquad \cdots\cdots\cdots(9)$$

where α_i represents the rate of divergence of vibration or the degree of instability, and ω_i is an angular frequency of vibration. And according to the sign of α_i the state of the truck motion can be clarified. Namely, $\alpha_i > 0$ shows unstable, $\alpha_i = 0$ shows a sinusoidal vibration, $\alpha_i < 0$ shows stable and when ∂_i is a real root the motion is not cyclic.

4.2 *Numerical calculation*

Fig. 5 shows a relationship between the roots of characteristic Eq. (8) and the running speed when the frictional moment is infinity ($A \neq 0$, $B = 0$). From this figure the state of the truck motion can be clarified. And Eq. (8), having degrees of 16, may have at most 16 roots, but Fig. 5 shows the three pair roots which have relation to the hunting.

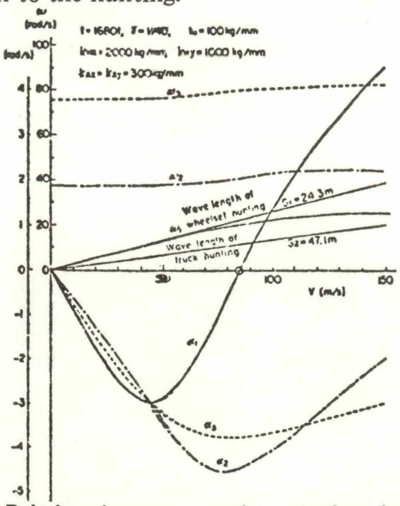


Fig. 5. Relation between running speed and roots of characteristic equation

When the truck runs in slow speed, all the real parts of the complex roots are negative, and hence the truck motion is stable. But as the speed increases, the stability of the truck motion gradually decreases, and when the speed passes over a critical point, hunting motion occurs. When α_i is zero, it corresponds to the critical speed of hunting. And above the critical speed the truck hunting diverges rapidly so that the truck hunting will never attain stability. In this figure, ω_1 is a hunting frequency of wheelset, and ω_2 and ω_3 are the lateral frequency of the truck frame and the yawing frequency respectively.

Being considered just the critical speed, Fig. 6 shows the effect of variation of the supporting stiffness (k_{Ax}, k_{Ay}) between the truck frame and the side frame, and Fig. 7 shows the relation between the supporting stiffness of the wheelset (k_{wx}, k_{wy}) and the critical speed of hunting. As these figures show, there are k_{Ax}, k_{Ay} and k_{wx}, k_{wy} which maximize the critical speed of the truck hunting.

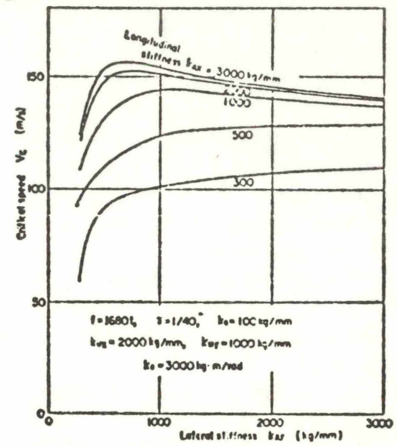


Fig. 6. Relation between supporting stiffness, which inserted between truck frame and side frame and critical speed

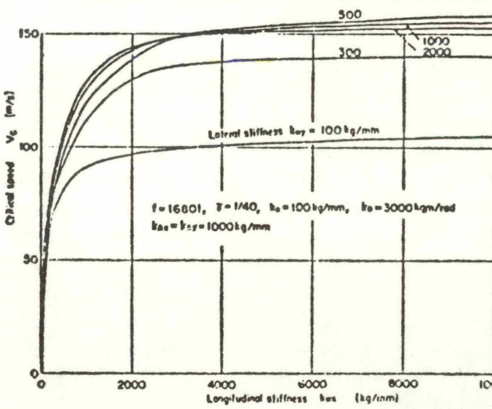


Fig. 7. Relation between supporting stiffness of axles and critical speed

Fig. 8 shows the calculation results when spring and friction are connected in series. In such a system, at the transitional point where the stability changes to the unstable zone, there exists the unstable limit cycle.

In case of the non-linear vibration, the stability of such system is related to the magnitude of its amplitude. Accordingly, when the amplitude is

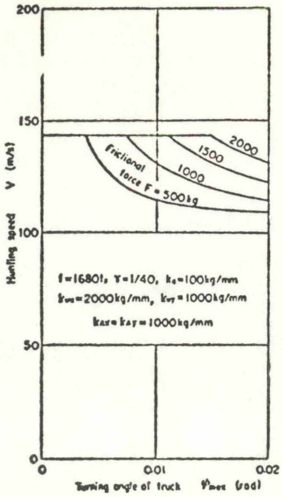


Fig. 8. Relating among turning angle of truck, frictional force and hunting speed

increased by a very small amount for the system which vibrates in the constant amplitude, if the system becomes restorable both in amplitude and frequency, there exists stable limit cycle, but if the system becomes divergent, there exists an unstable limit cycle.

In such a system as this truck, there exists an unstable limit cycle, so when the turning amplitude of the truck increases, for instance by the track irregularity or by the other impact force, the hunting speed of the truck may be decreased. Accordingly, under such conditions, we have to select the value of the various elements in the truck, the values which never raise the truck hunting at the planned maximum speed.

As the rotational frictional force of the truck becomes large, the speed of the unstable limit cycle becomes higher, so to increase the frictional force has an effect to prevent the truck hunting (in Fig. 8). However, taking into consideration of various conditions, such as side thrust of wheel to rail during the truck running through curved track, the frictional force can not be increased without any restriction.

On the test truck of the SANYO SHIN KANSEN being considered of the wheel flange wear and of the allowable limit of the rail gauge etc., it is recommended to have 0.01 radian as a possible maximum turning angle of the truck to the rail for safety.

According to the above calculation, as the basic data of car design for the SANYO SHIN KANSEN the following values were proposed.

$k_{wx} = 5{,}000 \sim 10{,}000$ kg/mm,
$k_{wi} = 1{,}000 \sim 2{,}000$ kg/mm,
$k_{Ax} = k_{Ay} = 1{,}500 \sim 2{,}000$ kg/mm,
$k_o = 500$ kg/mm, $F = 500$ kg

Where k_{wx} is the longitudinal supporting stiffness, k_{wy} is the lateral supporting stiffness, k_{Ax} and k_{Ay} are the longitudinal and the lateral supporting stiffness between the truck frame and the side frame, k_o is the turning stiffness for the truck, and F is the frictional force for turning the truck.

5. Conclusion

In this report, the writer's outline deals with the fundamental characteristic of a hunting car, and of the theoretical foundation on the basic design of the test truck for the SYNYO SHIN KANSEN, citing an example of numerical calculation.

According to these calculation results, various factors which affect the hunting speed of the truck were clarified. For example: the supporting stiffness of the wheelset; the spring constant between the side frame and the truck; and the frictional and elastic movements against truck turning. It was clear that reasonable values exist which never raise truck hunting at the planned maximum speed. As a result, basic data for car design are proposed.

In this report, we assume that the body is fixed in space; therefore, a more advanced and general study on the hunting, including car body, will have to be studied.

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3. K. Yokose: Report of RTRI, No. 683, (1969–5)
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Principles of Truck Design

Barwell, F. T., "Problems Of Support, Guidance And Propulsion Involved In High Speed Transport Systems," Bulletin of the International Railway Congress Association, Weissenbruch Co., Ltd., 40, Rue De L'Orphelinat, Brussels 7, Belgium (paper presented at the "High-Speeds" Symposium, Vienna, 1968).

This paper discusses the provision of means whereby high speeds (100 m/sec or above) may be economically attained, and road congestion thereby relieved. Economy demands the effective use of the existing rights of way, and thus necessitates higher degrees of superelevation than are achievable by conventional railway practice. It is suggested that the separation of support and guidance functions may be advantageous in the development of a new system.

TDOP: 02-001

British Railways, "Improving Riding Of Passenger Vehicles," by J. L. Koffman, Railway Gazette, Vol. 107, November 22, 1957, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 585-586.

This paper describes the five fundamental modes of vehicle body vibration of particular interest to bogie designers. These are: lateral oscillations, rolling and swaying oscillations, body nosing, fore and aft oscillations, and vertical oscillations. Each of these principles is briefly described and is related to coach and locomotive production bogies to illustrate how observance of these principles results in satisfactory performance.

RRIS: 037707

TDOP: 02-002

Byrne, R. , "Time To 'Spring' For A Better Truck," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N. Y., 14043, pp. 16-19.

A historic perspective of the freight car truck is presented indicating that modern truck designs will be required to meet the increasing demands of railroad service. There is a discussion of the work of AAR's Ad Hoc Committee on Truck Design and a listing of suspension system factors which are important in design considerations.

TDOP: 02-003

Cassidy, R. J. , "The Train With A Flexible Spine - For Faster Cheaper Rail Travel," Research Trends, Summer 1971, CALSPAN Corp. , Box 235, Buffalo, N. Y. 14221, pp. 42-48.

This paper describes a fundamental railroad design change, now under study. This change involves substituting a continuous flexible train bed for the conventional string of coupled car trucks. The flexible train's "backbone" would be a steel beam, two to four inches wide and twelve inches high. Calculations indicate that this concept would be far less affected by wheelset hunting than current designs, and could be used in conjunction with existing road beds.

TDOP: 02-004

Chesapeake and Ohio Railway, The Drag of Railway Cars of New Design, by H. Burlage, Jr., December 13, 1954, Terminal Tower, P. O. Box 6419, Cleveland, Ohio 44101.

The air resistance characteristics of two train configurations, one of modern, conventional design and the other aerodynamically designed to reduce drag, were determined from train models. The lower drag of the new design was due to the reduction in total surface area, to the important improvement in underbody design including the recessing of trucks and wheels, to the removal of the many appurtenances on the tops and sides of the cars, and the reduction of wake drag. Not only was the absolute drag reduced, but the tests showed that the rate of change of drag with velocity was considerably less for the new design than for the conventional train.

RRIS: 040128

TDOP: 02-005

Chesapeake and Ohio Railway, Workshop - Cum - Study Tour On Problems Of Dieselization Of Railways, by S. G. Guins, May 5, 1966, Terminal Tower, P. O. Box 6419, Cleveland, Ohio 44101.

This correspondence is based on a trip to Russia to attend a UN activity sponsored by the Railway Division of ECAFE. It briefly covers maintenance practices, engine-crew operations, oil analysis, engine development, truck tramming, and diesel-hydraulic locomotive developments.

RRIS: 040074

TDOP: 02-006

Cox, E. S., "Some Problems In Vehicle Riding," Institution of Locomotive Engineers Journal, paper No. 625, Vol. 51, No. 283, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London S. W. 1, England, pp. 574-659.

This is an integrating paper that seeks to survey the field. It emphasizes that sound suspension design only meets part of the problem, and refers to the many aspects of mechanical interaction and wear which are just as important. The practical application of the theoretical and developmental work which has been undertaken by a number of groups on British Railways is described, six particular riding problems are analyzed, and the steps taken to deal with them are outlined.

RRIS: 040154

TDOP: 02-007

Coxon, H. E. and L. D. McNaughton, "Bogie Design For Australian Conditions," Railway Engineering Journal, March 1973, Institution of Mehcanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England, pp. 16-31.

The authors discuss geographic and climatic conditions making the Australian truck design problem unique. They then touch on various suspension systems applied to passenger, freight, and locomotive service. The paper, being general in nature, mentions many facets of the problems indicated.

RRIS: 047913

TDOP: 02-008

Cripe, A. R., "High Speed Railway Transportation Today and Tomorrow," Society of Automotive Engineers, Inc., Two Pennsylvania Plaza, N. Y., N. Y. 10001, (paper No. 700815 presented at Automotive Engineering Congress, Detroit, Michigan, January 12-16, 1970).

This paper discusses the present generation of high-speed trains. A brief discussion of the designs of present and future high-speed trucks is given.

TDOP: 02-009

Dwyer, H. L., Jr., "American Freight Car Trucks," Technical Proceedings - 1973 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043, pp. 20-23.

This paper reviews AAR's 1973 involvement in research in three main studies: the RPI-AAR Cooperative Program on Truck Component Safety, AAR-RPI-FRA National Research Program on Train-Track Dynamics, and a program by Washington University of St. Louis in conjunction with AMCAR Division of ACF to provide improved analytical methods for use in structural design. Truck functions summarized are support, guidance, braking/traction interface, and isolation. How much improvement is economically justified is a recurring question that must be considered.

TDOP: 02-010

Engineering Exchange Forum, Technical Proceedings Of The Engineering Exchange Forum, September 22, 1966, Symington Wayne Corporation, 2 Main Street, Depew, New York.

Papers from a railroad forum which discuss car design trends, high speed track design, roll and wheel lift tests, and coupling requirements. Also included with the papers are comments and questions concerning the papers presented at the forum.

RRIS: 033132

TDOP: 02- 011

Gaertner, W. , "Toward A Wearless Truck," Eisenbahntechnische Rundschau, Vol. 14, No. 9, September 1965, Hestra-Verlag, Darmstadt, West Germany, pp. 406-413.

Apart from the wear at the wheel treads, bogie or truck wear is confined to the journal boxes and guides, the truck and bolster suspension and connections, and the truck pivots or centerplates. The use of modern rubber suspensions can now make the "wearless" truck a practical reality, as well as providing the desirable qualities of dampening the oscillatory motions vertically, longitudinally, and laterally. The author describes these developments and presents with illustrations locomotive and car trucks built with the various types of rubber suspensions.

RRIS: 037604

TDOP: 02- 012

Garin, P. V. , "Engineering Criteria For Future Freight Car Truck Design And Performance," Technical Proceedings - 1969 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043, pp. 25-30 (paper presented at 1969 Railroad Engineering Conference).

This article directs attention to freight car truck design as related to economics to see what can be done to improve performance and reduce maintenance and lading damage costs. Dynamic performance of the freight car truck is a complex family of motions. Track-truck-carbody interactions have not been evaluated. External effects include forces and vibrations related to the track structure, subgrade, ballast, tie plates, rail, and rail anchoring which produce a dynamic track profile. Car body dynamics and geometric properties under empty-load range affect truck performance. Article lists six questions or parameters pertaining to the economic performance balance of truck design.

TDOP: 02-013

Giesking, P. F., "Composite Car And Free Wheeling," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043, pp. 64-68, (paper presented at 1971 Railroad Engineering Conference).

This paper discusses two experimental car designs, one built by National Steel Car Corporation and the Department of Industry of the Government of Canada, and the other built by National Steel Corporation and various Canadian railway suppliers. The first car represents a composite design incorporating the physical characteristics of a covered hopper car with some box car design elements. The other experimental car is an ore car, which has been modified for free wheeling. This car has thus far accumulated 34,000 miles in service. One side effect experienced with this car was a 49 percent reduction in draw bar pull necessary to negotiate a 12 degree curve.

TDOP: 02-014

Hancock, R. M., "Vehicle Suspension And Bogie Design In Relation To Track Conditions," Institution of Locomotive Engineers Journal, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London S. W. 1, England, pp. 457-565.

This article expresses analytically the relationship of vehicle suspension to track conditions. The significance of coning lies in its property to transmit and magnify the effects of track misalignment to the vehicle body. This effect will become greater as tread wear develops. In the event of build-up of wheel movement, flange-climbing and bogie hunting oscillations may occur. The effects of conicity can be reduced in new designs by the positive guiding of axles by means of telescopic or other guides, together with roller bearing boxes and wheels having a minimum of coning, to avoid running against one rail. Frictional effects of laminated springs and suspension links transmit shock vertically and laterally from the track, and the elimination of such friction is desirable and beneficial. Suspension link length is important, and an increase in length will diminish the transmitted shock. A relationship has been established between track shape and vehicle response, which may be applied by a special application to an analog.

RRIS: 040195

TDOP: 02-015

International Union of Railways, Study Of The Optimum Damping Required By The Suspension Systems Of Wagons So As To Enable Their Running, Under Any Loading Conditions, At A Speed Of 80 KM/H On Tracks In An Average State Of Repair, by P. Moron, Question B56, July 1962, Office for Research and Experiments, Utrecht, Netherlands.

The running of trains in ordinary service at a speed of 80 km/h as of 1970 entails problems especially as regards certain series of existing wagons, which will not have been redeemed in the forthcoming years and which may give rise to difficulties when being run at higher speeds. In the present report of enquiry the measures to be taken in order to improve the riding stability of these latter vehicles are examined. These arrangements concern both the constructional modifications to be made and the adaptation of the damping systems. They should, however, be economic in order that they can rapidly be redeemed. The study shows, as regards the two-axled wagons, the importance of the ratio between wheelbase and length of the wagon body and, more exactly, between the wheelbase and the radius of gyration to the vertical axis of the centre of gravity. The increase of the wheelbase has shown itself to be one of the most efficacious measures in order to improve the stability of the wagons having insufficient riding qualities. In the case of bogie wagons, the study gives an account of the damping systems based on friction aiming at an improvement in the behaviour of the vehicles fitted with helical spring suspension systems. Such systems also make possible the visualization of bogies that can be run at speeds up to 120 km/h. The report finally supplies some general indications on the choice of the lines where the tests on vehicles could be undertaken within the frame of studies of a Specialists Committee. These tests should make it possible to guarantee the safety of the running of the modified wagons on all the lines where a speed of 80 km/h is authorized.

RRIS: 033203

TDOP: 02-016

Japanese National Railways, "Dynamics Of High Speed Rolling Stock," by T. Matsudaira, Railway Technical Research Institute Quarterly Report, April 1960, Kunitachi, Box 9, Tokyo, Japan, pp. 57-65.

The main object of this research group was to solve the problems concerning the safety of rolling stock and passenger ride comfort. The researches for the determination of allowable limits with respect to safety and the researches for the prevention of derailment as well as destruction of track, the running stability of trucks, and the elimination of rolling stock vibration and shock are dealt with here.

RRIS: 033726

TDOP: 02-017

Japanese National Railways, "Dynamics Of High Speed Rolling Stock," by T. Matsudaira, Railway Technical Research Institute Quarterly Report, September 1964, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 21-25.

Basic researches on running safety and riding quality were required for the design of vehicles for the new Tokaido line, and high-speed tests of the prototype vehicles on the test track section have been almost finished in the fiscal year 1962. In the fiscal year 1963, researches on some remaining problems for the final design of production-type vehicles, especially minute researches on hunting prevention and on the lateral load-deflection characteristics of air springs, have been made continuously. In March 1964, the running performance test of the first six production-type vehicles built was performed. The main results of these researches and test are described.

RRIS: 033731

TDOP: 02-018

Japanese National Railways, "Dynamics Of High-Speed Rolling Stock," by N. Matsui, Railway Technical Research Institute Quarterly Report, Vol. 7, No. 1, March 1966, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 45-97.

This article discusses the results of running tests conducted with defective tracks, a rescue diesel locomotive, and repeated speed up and operation tests at frequent intervals. In addition, the hunting of rolling stock and vibration of a vehicle on a defective track is detailed. Finally, the performance test of production vehicles is discussed on the whole line.

RRIS: 033724

TDOP: 02-019

Japanese National Railways, "Static Distribution of Wheel Load of Two-Axle Bogie Car," by S. Koyanagi, Railway Technical Research Institute Quarterly Report, Vol. 7, No. 3, September 1966, Kunitachi, Box 9, Tokyo, Japan, pp. 37-40.

Since the off-loading of wheels is one of the important factors of the derailment phenomenon, it is significant to analyze relations between off-loading of wheels and various car parameters. Analysis of this sort has been done before for a two-axle car; this report concerns a two-axle bogie car.

RRIS: 033076

TDOP: 02-020

Japanese National Railways, "Theoretical Study On The Distribution Of Wheel Load Of Two Axle Railway Car," by M. Kunieda, Railway Technical Research Institute Quarterly Report, Vol. 5, No. 3, September 1964, Kuntachi, Box 9, Tokyo, Japan, pp. 36-40.

This paper presents a means by which the distribution of wheel load on a two-axle railway car can be calculated. This paper addresses the static problems relating to unevenness in wheel loads.

TDOP: 02-021

Kayserling, U., "Developments By The Maschinenfabrik Augsburg-Nuernberg A. G. In Air Sprung Trucks For Rail Vehicles," Eisenbahntechnische Rundschau, Vol. 18, No. 4, April 1969, Hestra-Verlag, Darmstadt, West Germany, pp. 127-133 (German).

In accordance with increased train speeds on existing track structures, designs of truck suspension systems that would permit higher speeds on curves by tilting the coach inward to compensate for the centrifugal force have been completed and tested recently. An important factor in such designs has been the development of air-spring suspension systems for the trucks, which are described.

RRIS: 040144

TDOP: 02-022

Koffman, J. L., "Body Rolling As Influenced By Bogie Suspension--2," Railway Gazette, Vol. 113, September 30, 1960, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 393-396.

This paper points out that the effective length of the swing links should be considered in terms of the natural frequencies of lateral and nosing oscillations, as well as swaying. Frequently, the use of relatively long links is desirable to reduce the sensitivity of the suspension to lateral track irregularities and to achieve a low natural frequency of lateral and body nosing oscillations. Particular attention must be paid to lateral displacement, for this is determined by the effective link length and the lateral force. Another solution consists of preloading the bolster laterally, the centering springs' action increasing usually in direct proportion with the displacement, a feature found on leading bogies of steam locomotives and some modern electric locomotive bogies. The natural frequency of oscillation for a British Railways coach is determined mathematically. Articulated swing links used in Switzerland are briefly described.

RRIS: 037700

TDOP: 02-023

Koffman, J. L., "British Railways Carriage And Wagon Axle Design," Railway Gazette, Vol. 122, April 1, 1966, Temple Press Limited, 161-166 Fleet

Street, London EC4, England, pp. 281-283.

Information is presented relating to the effect of shapes and stresses on the fatigue properties of components. The journal load, wheel load, and flange force are tabulated for a 74-ton car with 4. 5-ton trucks. Fatigue strength of steel axles and bending moments for steel are shown. The ride quality and center of gravity for passenger trains are calculated.

RRIS: 040109

TDOP: 02-024

Koffman, J. L., "Carriage-Bogie Design," Railway Gazette, Vol. 88, March 12, 1948, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p. 297.

Factors involved in determining the riding characteristics of coaching stock are numerous. By means of suitable formulae, the path of a wheel set will be a sine curve. It is influenced by the profile of the rail head. The characteristics of springs are carefully considered as it is suggested that helical springs might be used to deal with both vertical and lateral forces or that these forces might be controlled by arranging for the bogie center to bear against large rubber pads disposed at an oblique angle so as always to be in shear.

RRIS: 039476

TDOP: 02-025

Koffman, J. L., "Carriage And Railcar Bogies: Their Design And Development--IV," Railway Gazette, Vol. 115, August 25, 1961, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 216-218.

Design factors considered in this part include brake ratios, axle fatigue, adhesion, and P. D. More effective braking will require ratios 1:1. 5 to 2. 2 of the tare weight, cut back to 0. 8 at lower speeds. A routine method of axle fatigue calculation indicating the influence of such design variables as fillet radii, type of vehicle, speeds, vehicle type (four-wheeler or bogie), and so on, relating to dynamic wheel load allowance and other factors is long overdue. Adhesion improvement through truck linkage to the body at low levels requires care that bogie pitching will not cause high stress peaks at the kingpin or cause intense shuttle of the body.

RRIS: 040083
TDOP: 02-026

Koffman, J. L., "The Effect Of Suspension Design On Rail Stresses," Railway Gazette, Vol. 110, March 27, 1959, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p. 361.

The matching of spring stiffness and damper characteristics is discussed as an aid to improving riding and reducing rail stresses. Dynamic wheel load due to spring deflection as a function of deflection distribution and damping factors for a 100-ton truck locomotive is illustrated. The approximate dependence of the total wheel-load versus speed, which might be encountered in service, is plotted. The effects of spring stiffness, mass ratio, and bolster damping factors on body displacement relative to the ground, on the deflection of bolster springs, on truck frame displacement relative to the ground, and on the deflection of axlebox springs, are shown. The theory that the stresses imposed on rails decrease inversely to wheel diameter is discredited.

RRIS: 037695
TDOP: 02-027

Koffman, J. L., "Limitations of The Three-Piece Bogie," Railway Gazette, Vol. 126, May 15, 1970, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 379-384.

This article cites the design limitations of the conventional, three-piece bogie with regard to torsionally rigid freight car bodies, and with regard to running over 60 mph with worn wheels. Truck dynamics are discussed with regard to:

- Coil spring rigidity
- Rotational resistance
- Truck twist
- Adhesion values
- Lateral oscillations

The article suggests several modifications to the existing design, and ends by suggesting that future designs be based on rigid underframes with primary suspensions between the frames and axleboxes.

TDOP: 02-028

Koffman, J. L., "Rational Approach to Design," Rail Engineering International, Vol. 3, No. 6, July 1973, Broadfields (Technical Publishers) Limited, Little Leighs, Chelmsford, Essex CM3 1 PF, England, pp. 248-254.

This paper points out the fact that the pattern of design activities is changing in the wake of engineering progress. Up to now, the art of design was mostly expounded and taught on the basis of available "excellent examples," and the designer learned his craft predominantly as a craftsman or artist. This method had its drawbacks since by the time the designer was ready to apply himself to the design of complete machines, years went by, and the "excellent examples" were often out of date. The ever increasing speed of development makes it necessary to consider design on the basis of more effective methods, and one of these is to be considered here. Design should be regarded as a series of logical steps.

RRIS: 048018

TDOP: 02-029

List, H. A., "An Evaluation of Recent Developments in Rail Car Truck Design," ASME, 71-RR-1, 345 E. 47th St., N. Y., N. Y. 10017, (paper presented at IEEE-ASME Joint Railroad Conference, New York, April 1971).

The author reviews and evaluates technical developments, and proposes use of a self-aligning articulated truck to improve riding and tracking qualities of freight cars and rapid transit vehicles.

RRIS: 050340
TDOP: 02-030

Love, A. and E. A. Sugden, "Design And Maintenance Aspects Of Freight Rolling Stock Relative To The Effects On The Track," Railway Division Journal, Vol. 2, No. 4, 1971, Institution of Mechanical Engineers, 1 Bird-cage Walk, London S. W. 1, England, pp. 467-482.

A modern railway has grown up in this country with new and more powerful forms of traction and improved signaling and track, but the majority of wagons in use still consist of small four-wheelers, the basic design of which has not changed for over fifty years. These wagons incorporate a "box-on-wheels" design that was robust and cheap, suitable for the low axle weight, low speeds, and low utilization of their day. They are, however, incapable of meeting today's conditions of high utilization and speeds without an unacceptable degree of maintenance and inspection. Until recently, very little was known of the behaviour and design parameters necessary for high-speed wagons. Because of this, in 1963, when it became obvious that improved wagon suspension was required, B. R. adopted the U. I. C. Double Link Suspension. This was a proven design which was, and still is, in wide use on the Continent, but here again problems arose when operating at the higher speeds and axle loads permitted in Great Britain. These problems concerned the rapid wear of the links and saddles due to the friction necessary for lateral damping, and spring failures caused by the torsion induced in the spring, which was superimposed on the normal vertical loading.

RRIS: 033864
TDOP: 02-031

Manzo, M., "Evolution Of Bogies For Fast Rolling Stock (Evoluzione Dei Carrelli Per Rotabili Motori Veloci)," Ingegneria Ferroviaria, February 1973, Collegio Ingegneri Ferroviari Italiani, Piazza Croce Rossa, Rome, Italy, pp. 137-150.

Competitiveness with other means of transport imposes on the railways, in addition to the need for an improvement of the services (such as speed, safety, and comfort), an appreciable reduction in expenditures for the maintenance of rolling stock and track. Rolling stock with satisfactory running qualities can be obtained only with bogies of particular design, making use of appropriate constructional solutions utilizing the latest techniques. For the coding of the constructional principles, means of measurement are available which permit an exact knowledge of the influence of various parameters, leading one, at the same time, towards fundamental choices for running stability. A brief indication is given of the lines followed in the design of the latest bogies for Italian Railroad rolling stock, bringing out the results in practical tests and the savings achieved in the field of construction and maintenance.

RRIS: 051442

TDOP: 02-032

McGregor, L.S., "Railroad View," (paper for Panel Symposium on Freight Car Truck Design), American Society of Mechanical Engineers Winter Annual Meeting 1962, 345 East 47th St., N.Y., N.Y. 10017.

A general discussion of freight car truck design is given. Present standard integral side frame trucks provide acceptable riding qualities and are operationally reliable up to speeds of 70 mph. Restrictions and costs require truck designers to conform to some limitations.

TDOP: 02-033

McLean, L.A., "A Re-Evaluation Of Freight Car Truck Performance Requirements," Technical Proceedings - 1973 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, New York 14043, pp. 38-47 (abstract of paper given at 1973 Railroad Engineering Conference).

This paper presents a general discussion and description of the conditions of truck hunting. Design improvements are suggested with respect to a hunting study performed by S. C. L. The discussion following the presentation is particularly illuminating in that it demonstrates a general lack of consensus concerning causes and cures of truck hunting.

TDOP: 02-034

Moron, P., "Basic Principles For The Design Of Bogies For Passenger Rolling Stock," French Railway Techniques, No. 4, 1970, Federation des Industriels Ferroviaires, 92 Rue Bonaparte, 75 Paris 6e, France, pp. 117-139.

This paper discusses the parameters to be used in the design of the bogie unit. The principles which are established concern good lateral control of car body, critical speed of bogie, and tread profile. New devices and materials may be necessary for the design of units for speeds in excess of 200 km/h. Good matching between the characteristics of bogie and attached body will also be necessary.

RRIS: 033375
TDOP: 02-035

Newland, D. E., "Steering A Flexible Railway Truck On Curved Track," ASME, 66-RR-5, 345 East 47th Street, N. Y., N. Y. 10017 (paper presented at the ASME-IEEE Joint Railroad Conference, April 1969).

This paper gives a linear analysis for the steady motion of a flexible two-axle railway truck on curved track. Most existing designs are too rigid to be self-steering but it is shown that a flexible truck with sufficiently soft primary suspension can negotiate main line curves without slipping or experiencing flange contact. Results for the maximum rolling displacement of the wheelsets and the minimum curving radius for no slipping are obtained expressed as a function of suspension stiffness. It is demonstrated that lateral loads due to super-elevation deficiency have a minimal effect on truck motion that is mainly determined by creep forces arising from the geometric inability to roll freely on curved track.

RRIS: 047430
TDOP: 02-036

Reed, G. E., "Freight Car Suspension Systems - A Car Builder's Appraisal," Technical Proceedings - 1970 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N. Y. 14043, pp. 38-40 (paper presented at 1970 Railroad Engineering Conference).

This article provides a car builder's appraisal of freight car suspension systems and is intended to stimulate design innovations. Problem areas include space required for the suspension system, center plate design and height, brakes, rock and roll, and car ride. Design innovation in terms of system and components is suggested as a real need today.

TDOP: 02-037

Robertson, J. T., "Third Report On Free-Wheeling Truck," Technical Proceedings - 1971 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N. Y. 14043, pp. 84-86 (paper presented at 1971 Railroad Engineering Conference).

This third report presumes the reader's acquaintance and understanding of earlier presentations covering a free-wheeling truck (See Giesking, P. F., "Total Car Design For Optimum Car Utilization," Volume II, Section 4, Track Train Dynamics, and Giesking, P. F., "Composite Car and Free-wheeling," TDOP: 02-014). Reference is made to a problem of rail corrugation at ends of curves. Mention is made of a six-wheel truck concept.

TDOP: 02-038

Sparrow, R. W., "Vehicle And Track Dynamics Research On British Railways," Technical Proceedings - 1973 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N. Y. 14043, pp. 14-19.

This paper deals with British Railways' current approach to dynamic analysis of vehicle suspensions. Analytical and design tools that aid in achieving lateral stability and a high critical hunting speed are outlined.

TDOP: 02-039

Sheppard, F. E., "Sub-Critical And Super-Critical Bogies," Railway Gazette, Vol. 118, February 1963, Temple Press Limited, 161-166 Fleet Street, London EC4, England, p. 220.

In this paper, subcritical and supercritical design aspects of bogies are discussed, and a diagram reveals pitching frequency versus speed and relative severity for various bogie designs. It was considered improbable that a bogie could be designed more subcritical than the Commonwealth, so that the problem was one of the feasibility of designing a supercritical bogie. Such a bogie must exhibit a natural frequency of 11.6 cycles/sec. for a maximum speed of 90 mph or 12.4 cycles/sec. for 100 mph. It was concluded that such a bogie would pose no design problems even with a pivot load of 9 tons and a maximum speed of 100 mph.

RRIS: 037264
TDOP: 02-040

Temple Press Limited, "The Barber Stabilised Freight-Wagon Bogie," Railway Gazette, Vol. 85, October 11, 1946, 161-166 Fleet Street, London EC4, England, p. 416.

Roller bearings in conjunction with a lateral control device are the chief features of this new American design. Separate axleboxes of special design incorporate a built-in friction device for damping out vertical movements and shocks. The new Barber bogie is considerably lighter than the conventional four-wheel, cast-steel, freight-wagon bogie. Wheel changing is exceptionally easy, as the stabilizer parts, springs, or axleboxes need not be removed when a wheel change is made.

RRIS: 039478

TDOP: 02-041

Temple Press Limited, "Bogie Design For High Speed", Railway Gazette, Vol. 82, April 6, 1945, 161-166 Fleet Street, London EC4, England, p. 337.

Opinion holds that the line of demarcation between low-speed and high-speed operation, from the coach design point of view, is in the region of 80 to 85 mph. Bogies that have been tested in freight wagons have exhibited good riding qualities at 80 mph or lower, but have shown themselves entirely unsuited for speeds above 85 mph; in some of the tests, the shocks recorded were doubled in intensity as speed increased from 80 to 90 mph. Up to 80 mph, the amplitudes of body swing were within reasonable limits, but at 90 mph, the body of the box wagon under test was becoming unstable. Experiments proved that, given equal conditions of springing and of maintenance, a six-wheel bogie gives slightly better riding in both vertical and horizontal planes than a four-wheel bogie, and has better braking qualities. But the gain is not considered to be worth the increase in weight, first cost, and cost of maintenance. In the design of passenger-car bogies, coil bolster springs perform the same duty as the swing hangers in the elliptical spring bogie. Another important requirement is that in wheels for high-speed equipment the treads shall be concentric within 0-10 in. Experiments showed that one of the principal factors in causing bogies to "hunt" at speed, is a short and sharp taper close to the throat of the main flange of the wheels, even if the extent of the taper be no more than 1/10 in. No bogie design tried by the Milwaukee has given good riding at high speed if the wheels have been in this condition, and the only way to restore such wheels to good riding qualities is by re-turning or grinding them.

RRIS: 037286
TDOP: 02-042

Temple Press Limited, "Bogie Development In Relation To Wagon Characteristics," Railway Gazette, Vol. 126, July 3, 1970, 161-166 Fleet Street, London EC4, England, pp. 505-507.

One of the first aftermaths of extensive modernization has been the realization that low freight train speeds are becoming a handicap to exploitation of the new types of motive power being introduced. Increases in speed with existing designs can bring about considerable increase in wear and consequently upset maintenance costs and programming. British Railways has led the way in Europe with 25-ton axleloads for bogie wagons, and gross loads of 100 tons are now widely accepted for speeds of up to 96 km/h. For speeds of 120 km/h, the axleloading on BR is currently restricted to 17-1/2 tons, but loading at this speed must be matched by good riding characteristics. Design work is going ahead with higher speeds in mind. Recent French, English, and U.S. bogies designed for high speeds are shown.

RRIS: 037761
TDOP: 02-043

Temple Press Limited, "The Design Of Railway Bogies," Railway Gazette, Vol. 115, November 24, 1961, 161-166 Fleet Street, London EC4, England, pp. 588-589.

In connection with bogie design, research and development has led to the establishment of ten particular design features which cover primary and secondary spring deflection; primary damping; secondary spring disposition and damping; swing-link design and suspension; bolster anchorage by traction bars; bogie frame stiffness; rotational damping; secondary spring anchorage; and check stop clearances. It has been clearly established by investigation and testing that the inclusion of the recommendations on these ten points into a new bogie design will give a consistently good ride that is relatively insensitive to tyre wear and to normal track irregularities.

RRIS: 037275

TDOP: 02-044

Temple Press Limited, "Six-Wheel Or Four-Wheel Bogies?," Railway Gazette, Vol. 82, March 13, 1945, 161-166 Fleet Street, London EC4, England, p. 361.

The principal factors in smooth riding are the method of suspension and the quality of maintenance, which can affect the running in considerably greater degree than the provision of one axle more or less in the bogie. A test with four-wheel bogies was done to show that the coach had suffered no deterioration in its riding qualities, and that the eight-wheel coach with bogies of 11-foot wheelbase rode no better than a similar vehicle carried on four-wheel bogies of the more usual 8-foot wheelbase. After prolonged experiments directed towards the improvement of riding, the four-wheel bogie, even with vehicles up to 80 feet in length and 10 feet in width, retained an unchallenged position.

RRIS: 039483

TDOP: 02-045

Temple Press Limited, "Some Aspects Of Bogie Design," Railway Gazette, Vol. 96, January 11, 1952, 161-166 Fleet Street, London EC4, England, p. 32.

Normal design has proved its practicability over a long period and most efforts have been directed towards improvement in detail rather than toward the production of something fundamentally different. Arduous conditions have led to efforts to eliminate or reduce wear so that a vehicle can run longer between less expensive overhauls. These and other problems were covered in this paper to the Institution of Mechanical Engineers.

RRIS: 039938

TDOP: 02-046

Temple Press Limited, "Vehicle Riding Convention," Railway Gazette, Vol. 121, November 19, 1965, 161-166 Fleet Street, London EC4, England, pp. 913-916.

The convention on interaction between vehicle and track convened by the Railway Engineering Group of the Institution of Mechanical Engineers consisted of four sessions at which 10 papers were read including the following: "An Appreciation of the Practical Problems--a Survey of the Problems and their Importance," "Some Observations on Linear Theory of Railway Vehicle Instability," "The Dynamics of Railway Vehicles on Straight Track: Fundamental Considerations on Lateral Stability," "Dynamics of Railway Vehicles on Curved Track," "Hunting Problem of High-Speed Railway Vehicles with Special Reference to Bogie Design for the new Tokaido Line," "Track Parameters Static and Dynamic," "The Influence of Track Twist on Vehicle Design," "The Static and Dynamic Parameters of Railway Coaches."

RRIS: 040097

TDOP: 02-047

Threlfell, W.G., "Operating Characteristics Of Locomotive And Car Trucks," 1973, Railway Fuel and Operating Officer's Association, 10414 South Wood Street, Chicago, Illinois 60643 (paper presented at 37th Annual Proceedings of the Railway Fuel and Operating Officer's Association).

In late 1971, problems were experienced by the CN on high curvature main-line subdivisions even though most of them had been recently upgraded. There were frequent changeouts of outer rails on curves due to flange wear, misalignment and loss of superelevation at the spiral or entrance to curves, high spikes and rail turnover (or partial turnover), and gauge widening due to the tie plates shifting and cutting into the ties, especially during the many months when the roadbed was frozen. The frequency of derailments increased simultaneously on certain subdivisions and the seriousness of the situation demanded immediate corrective action. The successful and quickly implemented program of action taken by CN is outlined in this paper.

RRIS: 054600
TDOP: 02-048

Wickens, A. H., "The Riding Of Railway Vehicles," Proceedings of Society of Environmental Engineers, 1963, 68a Wigmore Street, London W1, England, pp. 39-44.

Some of the basic facts are outlined for the definition and control of vibration and shock in railway vehicles and their loads. The action of conic wheelsets and the kinematic motion of wheelsets are clearly illustrated. The hunting principle is described. A laterally restrained wheelset connected to a high inertia vehicle body by perfectly elastic lateral springs is described, and the stability criterion is shown. These theories are then applied to the analysis of railway vehicle stability and response.

RRIS: 040198
TDOP: 02-049

Williams, D., "A New Method Of Dynamically Stabilizing Railway Bogies, Four-Wheel Wagons, And Road-Railers Against Undesirable Lateral Oscillations," Proceedings of Institution of Mechanical Engineers, Vol. 180, Part 3F, 1966, 1 Birdcage Walk, Westminster, London S. W. 1, England, pp. 125-139.

This paper describes a new method of dynamically stabilizing railway bogies, four-wheel wagons, and road-railers so as to be proof against undesirable lateral oscillations. It is generally accepted that the root cause of all such oscillations is the inherent instability of the ordinary axle-set, i.e. the integral unit of live axle and two wheels. By seeking to eliminate this instability, the new method tackles the problem at its source. The principle relied upon is that of the inertia-guided axle, i.e. an axle with a leading guiding arm that is integral with it in yaw, and that is connected at its forward end, by means of a velocity damper, to a floating mass whose lateral inertia provides the necessary guidance. Alternate guiding arms come into action wherever the direction of travel is reversed.

RRIS: 037705

TDOP: 02-050

Yokose, K., "Calculations On Hunting Of High Speed Railway Truck - Problems Of Truck Design For SANYO SHIN KANSEN," Technical Proceedings - 1973 Railroad Engineering Conference, Dresser Transportation Equipment Division, 2 Main Street, Depew, N.Y. 14043, pp. 30-34 (paper presented at 1973 Railroad Engineering Conference [see Reprints--Truck Design]).

This paper deals with the fundamental characteristics of a hunting truck, and explains the theoretical foundation of the test truck for SANYO SHIN KANSEN. An example of the numerical calculations is given. According to the results, factors such as: supporting stiffness of the wheelset, the spring constant between the side frame and the truck, and the frictional and elastic movements against truck turning are clarified with respect to truck hunting. Basic design data are proposed for the test truck which recommend modifications raising the critical hunting speed above the maximum system speed.

TDOP: 02-051

Freight Car Truck Design

Adams, D. W., "Truck Problems From Current Operating Demands," Technical Proceedings from 1964 Railroad Engineering Conference, September 24, 1964, Symington Wayne Corporation, 2 Main Street, Depew, New York 14043, pp. 30-34 (paper presented at 1964 Railroad Engineering Conference).

Increasing axle loads, vertical vibration causing damage to lading, wheel lift on autorack flat cars, long cars, and heavy hoppers, and fabrication and durability problems connected with recent truck designs are discussed. The economic desirability of adopting new truck designs is briefly mentioned.

RRIS: 039518
TDOP: 02-052

Chrysler Corporation of America, A Report To The Railroads Of America, Engineering Division, P. O. Box 857, Detroit, Michigan 48231, pp. 6-13.

In this commercial report, a controlled lateral and vertical motion freight car truck design intended to provide absorption of lateral impacts and permitting long-travel (AAR D-5) coil springs is described. Swing hangers, spring planks, and constant friction snubbers are included in the truck assembly. Alignment is accomplished without reliance on contact between bolster and side frames, assuring stability and tracking of wheels.

TDOP: 02-053

Cope, G. W., "Four and Six-Wheel Hydraulically Equalized Trucks," Technical Proceedings - 1965 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, New York 14043, pp. 26-30 (paper presented at 1965 Railroad Engineering Conference).

The article discusses basic principles of a hydropneumatic, six-wheel truck suspension system that utilizes coil springs and hydraulic cylinders to support the load above the bearings. The concept aims to produce a six-wheel truck that is smaller, lower, shorter, and lighter, requiring less maintenance. An accumulator is charged with gas on one side and with hydraulic fluid on the other. The motion of the truck has been sequenced to show wheel movements by hydraulic equalization. Hydraulic fluid from the cylinders flows through control valves to provide snubbing action.

TDOP: 02-054

Dwyer, H. I., "A New Approach To Unit Train Equipment," ASME, 66-RR-5, 345 East 47th Street, N. Y., N. Y. 10017 (presented at the Ninth Joint ASME-IEEE Railroad Conference, May 1966).

This paper discusses an economical background and technical approach to unit-train equipment. Use of lightweight structure, single-axle structures, air-operated unloading doors, and articulated car configuration in the Aeronca Model 65 Hopper Car are presented as examples. The use of computer analysis in evaluating car dynamics is discussed.

TDOP: 02-055

General Steel Industries, Inc., General Steel Industries Chevron Truck, Castings Division, 1417 State Street, Granite City, Illinois 62040.

The CHEVRON truck concept features a rubber chevron spring arrangement that replaces coil springs, snubbing devices, and column guides as used in conventional trucks for freight cars.

TDOP: 02-056

Hawthorne, J. W., "The Six-Wheel High Capacity Truck," Technical Proceedings - 1965 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, New York 14043, pp. 22-26 (paper presented at 1965 Railroad Engineering Conference).

This paper, which deals with the six-wheel, high-capacity truck, discusses larger cars, incentive loadings, and heavier wheel loads in relation to traffic considerations and track and structures. Wheel loading as related to multiple-wear wheels is mentioned.

TDOP: 02-057

Hori, K., A. Harada, S. Yoshi, and H. Miyamoto, "Sumitomo Freight Car Trucks And Manufacturing Equipment," Sumitomo Search, Sumitomo Electric Industries Limited, Osaka, Japan, May 1972, pp. 58-77.

This paper reports on Sumitomo freight car trucks and mentions the outline of their features and components. Inspections and manufacturing equipment are also covered.

RRIS: 046806
TDOP: 02-058

IPC Transport Press Limited, "Eurospeed Bogie For Freight," Railway Gazette International, Vol. 130, No. 4, April 1974, Dorset House, Stamford Street, London SE1 9LU, England, p. 149.

This three-piece cast steel bogie for speeds above 120 km/h has roller bearing axle-boxes mounted in rubber rings to provide a degree of primary suspension.

RRIS: 054937
TDOP: 02-059

IPC Transport Press, "Welded Aluminum Alloy Bogie," Railway Gazette International, October 1970, Dorset House, Stamford Street, London SE1, England, p. 755.

A prototype unit with a one-piece frame and axleboxes fabricated from aluminum alloys has been on trial in Switzerland. Use of the alloys in construction has resulted in a weight saving of about 20 percent over the equivalent steel bogie. Experience in service mounted under a tank wagon on the metregauge line where conditions are more severe (sharper curves and steeper gradients) than those encountered on standard-gauge main lines has resulted in no operating problems.

RRIS: 039487
TDOP: 02-060

Institution of Mechanical Engineers, "Axle Motion Bogie," The Railway Engineering Journal, November 1972, 1 Birdcage Walk, London SW1H 9JJ, England.

Summary is given on a "rigid" H-frame design together with a primary suspension design for axle loads up to 25,400 kg (60,000 lb) operating at speeds up to 100 km/h (60 mph). The truck has long-travel, soft helical springs controlled by load variable friction damping.

TDOP: 02-061

International Union of Railways, Standard Bogie For Wagons, Comm Rpt, ORE No. 26, January 1968, Office for Research and Experiments, Utrecht, Netherlands, pp. 9-12.

The assignment of the B 12 Committee was to define a bogie able to run at 120 km/h both on standard-gauge (1.435 m) and on broad-gauge (1.524 m) track, which will permit the automatic traction and shock coupler to be used on wagons and which, with the aid of some simple intermediate parts, could replace the bogie of the present standard flat wagon. Two solutions were proposed: (1) a bogie with supercritical running gear and a suspension system, (2) a bogie with subcritical running gear, helical springs, and vertical and horizontal friction-damping. Testing consisted of static strength tests, dynamic strength tests, running tests, and removing and mounting tests with the automatic traction and shock coupler. The results obtained with both types of bogie are good. Comparisons of construction and probable maintenance costs of the bogies have not revealed substantial differences. The bogie with leaf springs is 140 kg lighter than the similar unit with helical springs. Both are lighter than the present standard bogie. In view of the small differences in the quality of each type of bogie, either could be selected as standard.

RRIS: 037231
TDOP: 02-062

Mauzin, Andre, "New Basic Principles Of Body/Bogie Suspension Two Prototype Bogies Studied By The S. N. C. F.," French Railway Techniques Quarterly Review, No. 2, 1965, 20 Boulevard Diderot, Paris - 12°, pp. 86-93.

This report covers design information concerning two prototype trucks tested by S. N. C. F. It concludes that these trucks would optimize riding qualities at high speeds. Tests were performed up to 84 mph (140 km/h).

TDOP: 02-063

Murphy-Richter Publishing Company, "'Million Mile' Freight Car Suspension System," Progressive Railroading, Vol. 16, No. 2, March 1973, 9 South Clinton Street, Chicago, Illinois 60606.

A freight car suspension system has been introduced by American Steel Foundries. The T-11 system is designed expressly for 100-ton capacity freight cars. It combines an advanced snubbing system to absorb and dissipate energy and a design change involving the use of long-travel, load-carrying coil springs. The system is warranted for a million miles or 10 years of life. The T-11 system also includes the Simplex side bearing which uses controlled friction to reduce rock and roll motion at critical speeds and to control the high-speed swiveling of the truck. Recent tests indicate that the T-11 reduced the damaging effects of lateral shock to an unloaded freight car by as much as 75 percent at speeds above 50 mph. It also reduced the damaging effects of vertical shock inside a loaded car by as much as 60 percent. Comparisons are made with a conventional truck. Better riding qualities should result in longer life for truck components. The life expectancy is rated at 2.65 times that of a conventional truck.

RRIS: 044006

TDOP: 02-064

National Castings Division, National Swing Motion Truck, Commercial Engineering Bulletin No. 527, Midland-Ross Corporation. 2570 Woodhill Road, Cleveland, Ohio 44104.

This commercial report details a new freight truck design, the National Swing Motion Truck.

Design innovations include:

- Modification of frame pedestal jaw to permit rocking or swinging of side frame on adapters
- Elimination of standard side frame spring seat
- Incorporation of replaceable rocker seat and hardened rocker seat bearings to permit rocking or swinging of side frames in relation to the seats
- Use of a transom connecting the rocker seats of both side frames
- The use of bolster lateral stop brackets added to the bolster bottom
- Elimination of conventional bolster gibs
- Use of two-stage, long-travel load springs

TDOP: 02-065

ORE, "Description Of The New Bogie With Elastic Primary Suspension Between Axles And Frame," Rail International, November 1972, pp. 650-652, (also, ORE Report, B52/RP3, October 1971, p. 41).

This article describes a new truck design and the numerous tests that were carried out by S. N. C. F. with the truck on track in good condition. The purpose of these tests was to study the effects of elastic connections between the truck frame and wheel sets and the effects of the tire profiles on the lateral stability of the truck.

TDOP: 02-066

Railway Engineering Associates, Incorporated, "Articulated Rail Car Truck Development," (R&D program), January 1971 - July 1974, 38 West University Avenue, Bethlehem, Pennsylvania 18015.

Objectives and Scope.

Develop a dramatically improved freight car truck. Obtain background information for applying basic design to locomotives, rapid-transit cars, and passenger cars.

Approach and Methods.

Design, build, and test a 100-ton-capacity car set of trucks based on earlier work with 1/8-size scale models and continue work with mathematical models (computer simulation).

Progress and Results.

Low-speed testing over switching railroad trackage indicates that basic design and principles are sound. Plans being made for further testing over longer distances and at higher speeds.

RRIS: 050338

TDOP: 02-067

Railway Engineering Associates, Inc., The Articulated Truck, November 10, 1972, 38 W. University Avenue, Bethlehem, Pennsylvania 18015.

The article discusses an articulated truck concept to suit basic requirements for the freight car. Greater safety, reduced maintenance, and improved riding are claimed while using mostly standard AAR components. Elastomeric connections between parts reduced wear and vibration and improved tracking characteristics, even at high speeds. Performance under various modes of travel is discussed.

TDOP: 02-068

Temple Press Limited, "Bogie Type Container Wagons," Railway Gazette, Vol. 103, August 14, 1955, 161-166 Fleet Street, London EC4, England, p 218.

Among the recent developments in the U.S. for the conveying of granulated and powdered products by rail is a new type of container and platform bogie wagon. This special equipment was developed by the Shippers' Car Line Corporation. The bogie wagon, which is standard gauge, is 54 ft, 2 in. over buffers and has an overall width of 10 ft 1-1/4 in. The total container area is 2,072 cu ft and the loaded weight is 50.6 short tons. The wagon is capable of carrying 28 containers each of 74 cu ft capacity, and can be loaded or unloaded by fork lift truck. The placing of the containers on the wagon is simplified by means of inserts let into the flooring, and the containers are locked in position by means of side and top transverse bars.

RRIS: 039646
TDOP: 02-069

Temple Press Limited, "Cast-Steel Bogies For Goods Vehicles," Railway Gazette, Vol. 94, March 9, 1951, 161-166 Fleet Street, London EC4, England, p. 269.

Accurate bolster guides and locking devices on axlebox lids are features of a cast-steel spring plankless bogie with integral axleboxes and brake suspension brackets to the side frame. This design has been fitted to tank wagons with a 12-1/2-ton axle load and 10 in. x 5 in. journals.

RRIS: 039472
TDOP: 02-070

Temple Press Limited, "Cast Steel Freight Bogie With Friction Damper," Railway Gazette, Vol. 116, April 27, 1962, 161-166 Fleet Street, London EC4, England, p. 484.

The Gloucester cast-steel freight bogie is of the spring plankless type, which, in its latest form, incorporates a patent constant-friction damping device. This feature permits the use of long-travel coil springs by preventing spring breakage arising from low-frequency oscillations. This bogie is available to suit all gauges from 2 ft-6 in. to 5 ft-6 in. and for axle-loads up to 17-1/2 tons.

RRIS: 040089
TDOP: 02-071

Temple Press Limited, "DB Wagons For Large Indivisible Loads," Railway Gazette, Vol. 126, September 4, 1970, 161-166 Fleet Street, London EC4, England.

The German Federal Railway has produced plans for four new bogie wagons to carry heavy out-of-gauge or over-length loads throughout the system. They are all designed to carry heavy and lengthy indivisible loads, such as girders, transformers, and chemical plants, and have maximum payloads ranging from 56 to 186 tons. The pivots of all the bogies and connecting beams are without pins. A load mounted on the bearer beams can be transferred to DB heavy road vehicles without additional equipment being required. Wheelsets can be changed to enable the wagon to run on broad-gauge tracks.

RRIS: 037764

T DOP: 02-072

Temple Press Limited, "Developments In Wagon Bogie Design," Railway Gazette, Vol. 100, April 23, 1954, 161-166 Fleet Street, London EC4, England, pp. 466-467.

Cast-steel bogie frames and bolsters are being used for goods stock on many railways. Basically, there are two designs, one employing springs without friction damping, and the other employing springs that are friction controlled. The bogies can be of either the plank or plankless type, but the self-aligning spring plankless type gives better riding qualities with consequent reduction of wear on bogie guides and on wheel flanges and rails. There is also a reduction in costs and weight due to the elimination of the plank. Friction controlled bogies have long-travel bearing springs, although they can be designed for short-travel springs, with adequate reserve travel and constant control of spring action. Tests have shown friction-controlled bogies give improved riding of wagons at all speeds with consequent reduction in wear and maintenance. Unit brake beams are described.

RRIS: 037771

TDOP: 02-073

Temple Press Limited, "High Speed Wagon Bogies For Japanese National Railways," Railway Gazette, Vol. 121, November 19, 1965, 161-166 Fleet Street, London EC4, England, pp. 902-904.

Two types of bogies with air and steel springing were developed for 70 mph, fast freight services in Japan. The axleload has been set at 13.2 tons, and 860 mm diameter solid-rolled wheels have been used. Tapered roller bearings are used to take both vertical and lateral loads. The restraint between the axlebox and the bogie frame is 2,000 kg/mm longitudinally and 500 kg/m laterally. Either steel coil or air springing will be used in the bolsters. The air springs are of 550-mm diameter and are of the three-ply bellows type.

RRIS: 040095
TDOP: 02-074

Temple Press Limited, "Obtaining A Smooth-Running Bogie--2," Railway Gazette, Vol. 114, January 20, 1961, 161-166 Fleet Street, London EC4, England, pp. 74-77.

This article suggests an approach to improve the conventional type truck as used on multiple-unit and locomotive-hauled stock by considering its faults. The tendency of the bogie to snake can be greatly reduced by guiding both axles rigidly in the frame and so forcing them to remain parallel, i.e., within 0.04 in. by using roller bearings. The second method of attacking hunting is to prevent the whole bogie frame from snaking by deliberately introducing friction to discourage it from rotating slightly about its pivot. The sidebearer friction will rise in step with tire wear and snaking tendency if the sidebearers are greased only once when the tires are profiled.

RRIS: 040081
TDOP: 02-075

Temple Press Limited, "One-Piece Primary Suspension Bogie," Railway Gazette International, August 1971, 161-166 Fleet Street, London EC4, England, p. 325.

This article discusses the characteristics of a one-piece truck with welded rigid frames, designed for freight service. The truck is equipped with a two-stage primary suspension, load sensitive friction dampers, a hemispherical center pivot, and resilient side bearers. The main frame is fabricated from mild steel plate, which is of welded construction. These trucks are approved for operation at 60 mph under 100-ton cars.

RRIS: 037802
TDOP: 02-076

Temple Press Limited, "Rubber-Sprung Bogie," Railway Gazette, Vol. 116, 622309, 161-166 Fleet Street, London EC4, England, p. 289.

The Gloucester rubber-sprung freight bogie is a springless unit of three-piece form, but without sliding bolster-guides. Design features of this bogie are described in detail.

RRIS: 037272
TDOP: 02-077

Temple Press Limited, "Torsion Bar Bogies On The Rhaetian Railway," Railway Gazette, Vol. 102, February 11, 1955, 161-166 Fleet Street, London EC4, England, p. 158.

As a result of experiments with a novel type of bogie designed to provide improved riding at speeds up to 40 mph on their metre-gauge electrified system, the Rhaetian Railway is equipping four types of carriages with a bogie built by SIG. The results of road tests with the SIG bogie are shown, along with the road tests of the bogies originally on the carriages. These latter bogies were built before 1930. Transverse oscillations have been reduced considerably when compared to that experienced with the original bogies. Longitudinal movement is little influenced by the type of bogie, and this was slightly reduced. Tare weight of the cars is approximately one ton less than the original weight.

RRIS: 039635
TDOP: 02-078

Temple Press Limited, "Two-Axle Patent Bogie For Freight Vehicles," Railway Gazette, Vol. 122, July 1, 1966, 161-166 Fleet Street, London EC4, England, p. 543.

A principal feature of a newly developed bogie is that each wheel can follow rail inequalities closely, irrespective of the position of the others. Thus, constant loading between wheel and rail is enhanced and the guiding effect of the wheel flange remains steady. A balanced system has been evolved in which the only direct connection of the brake system and rigging with the bogie frame is through the brake-hanger pins and brackets. The free space around bogies of this type makes a wagon fitted with them well suited to the fitting of automatic centre couplers, a point that is likely to become of some importance in Western Europe.

RRIS: 037839

TDOP: 02-079

Warnock, E. J., "Progressive Development Of A High Speed Truck Design," Technical Proceedings - 1966 Railroad Engineering Conference, Symington Wayne Corporation, 2 Main Street, Depew, New York 14043, pp. 11-18 (paper presented at 1966 Railroad Engineering Conference [see Reprints--Truck Design]).

This article describes the development of the XL freight car truck concept, which provides for high-speed roller bearing trucks. This design allows the bolster to float laterally on spring groups to compensate for reduced lateral movement with roller bearings. Description and illustrations present this development in detail. Reduced wear on parts are expected.

TDOP: 02-080

Warnock, E. J., "A Truck Design For Today's Car And Operating Conditions," Technical Proceedings - 1964 Railroad Engineering Conference, September 24, 1964, Symington Wayne Corporation, 2 Main Street, Depew, New York 14043, pp. 34-40 (paper presented at 1966 Railroad Engineering Conference).

Early four-wheel trucks with coil spring suspensions are briefly discussed. The XL-70, 70-ton-capacity truck is illustrated and described in detail. The XL-70 can use the AAR-type hanger brake device, but can also be adapted for other types of brake devices. The journal box assembly and a sectional view of the frame and bolster assembly are shown.

RRIS: 039519
TDOP: 02-081

Webb, C. E., "Freight Car Innovation And Design," Progressive Railroading, Vol. 17, No. 2, February 1974, Murphy-Richter Publishing Company, 9 South Clinton Street, Chicago, Illinois 60606, pp. 1-5.

This article describes the Southern Railway's efforts to obtain better freight cars. Car and truck design improvements are discussed. Truck hunting research is discussed, and so is the use of computers in car design and in fleet analysis.

RRIS: 053866
TDOP: 02-082

Weber, H. B., "Single-Axle Trucks And Articulated Connections For Multiple-Unit, High-Capacity Railway Cars," ASME, 66-RR-7, 345 East 47th Street, N. Y., N. Y. 10017 (paper presented at the 9th Joint ASME-IEEE Railroad Conference, May 1966).

This paper discusses the development and design of single-axle trucks as tested on an articulated hopper car. A summary of test results is given.

RRIS: 046783
TDOP: 02-083

Wilcock, H., "Freightliner Vehicles-Design And Development In the Light Of Service Experience," Railway Engineering Journal, March 1972, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England, pp. 19-29.

The design and development of freightliner vehicles are discussed in light of service experience in England's domestic container transport system. This paper gives details of the original concept and design, with information relative to the design and maintenance problems encountered and the solution produced in overcoming these.

TDOP: 02-084

Passenger Car Truck Design

Burdick, W. E. and K. L. Jackson, "Development And Test Of The General 70-E Truck For BARTD," ASME, 66-RR-4, 345 East 47th Street, N. Y., N. Y. 10017, (paper presented at ASME-IEEE Railroad Conference, May 1966) pp. 1-11.

Basic design principles considered in developing the General 70-E truck for BARTD are discussed in combination with test results obtained.

RRIS: 046785

TDOP: 02-085

Chesapeake and Ohio Railway, Roll Stabilizer For New Passenger Cars, by K. A. Browne and S. G. Guins, Research Report No. 16, October 1951, Terminal Tower, P. O. Box 6419, Cleveland, Ohio 44101.

The object of this study was to design a torque bar type of roll stabilizer for application to General Steel Castings' trucks to meet roll limitations of certain railroads. The resulting design almost meets the roll restraint that is produced by the stiff springs recommended by B&O, and should noticeably improve the ride of long-travel springs.

RRIS: 040181

TDOP: 02-086

Cripe, A. R., "The United Aircraft High Speed Train Suspension Design," ASME, 67-RR-2, 345 East 47th Street, N. Y., N. Y. 10017 (paper presented at the Joint ASME-IEEE Railroad Conference, April 1967).

A suspension system for a turbine powered high-speed train is described. The system banks the car body proportionally to lateral force. Intermediate axles are guided to eliminate truck hunting.

TDOP: 02-087

Dean, A.G., "The Pioneer III Truck - Technical Aspects," ASME, 64-RR-2, 345 East 47th Street, N.Y., N.Y. 10017, (paper presented at ASME-IEEE Railroad Conference, April 1964) pp. 1-8.

The lightweight Pioneer III truck for passenger service is discussed by the Chief Engineer of the Railway Division of the Budd Company. The weight of an 85-ft coach with trucks was reduced by half, the principal weight saving being in the trucks. No sacrifice in ride quality was observed. Many details of the design were considered radical, but a favorable service history has been experienced.

TDOP: 02-088

Federal Railroad Administration, "Improve Metroliner Trucks" (FRA Contract), LTV Aerospace Corporation, Ground Transportation Division, P. O. Box 5907, Dallas, Texas 75222.

The objective of this contract is to design, fabricate, test, integrate, and railcar test improved metroliner trucks.

RRIS: 038849
TDOP: 02-089

Ishizawa, M., N. Matsui, and S. Otsuka, "Bogie Of The New Tokaido Line Railcar And The Results Of The Use Thereof," Bulletin of the International Railway Congress Association, (abstract of paper presented at IRC-UIC-"High Speeds" Symposium, Vienna, 1968) pp. 1-11.

This paper deals with the introduction of a passenger car truck construction that constitutes the essential part of Japan's new Tokaido Line rail car. Results of the use of the truck are discussed as well as the problems of safety and vibration in a high-speed rail car.

TDOP: 02-090

Japanese National Railways, "High Speed Rolling Stock I, Aerodynamic Problems," by T. Hara, Railway Technical Research Institute Quarterly Report, August 1964, Japanese National Railways, Kunitachi, Box 9, Tokyo, Japan, pp. 9-19.

Article discusses unique problems of high-speed trains. Aerodynamic problems such as testing of sealed-nonsealed train units and means to measure aerodynamic drag are discussed. Structural analyses of side frames, load tests, strength of bodies, and components are also included. Testing of power transmission, effects of wheel flat, bearings, and life guard are further investigated.

RRIS: 033229
TDOP: 02-091

Koffman, J. L., "Carriage And Rail Car Bogies: Their Design and Development--VI," Railway Gazette, October 20, 1961, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 446-449.

Article discusses the interrelationship between design of a bogie and the coach under which it will be installed. Differences in coach length, width, centers of gravity, wheel diameters, and rail condition are considered. The factors of sound intensity in the coach and car bouncing and pitching are related to ride and comfort index.

RRIS: 033225
TDOP: 02-092

Koffman, J. L., "Design For Comfort," Institution of Locomotive Engineers Journal, Vol. 57, No. 319, Part 5, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London SW1, England, pp. 428-508.

The work done during the last ten years to improve passenger comfort has resulted in marked advances as far as riding, noise levels, heating, and ventilation are concerned. Increasing speeds combined with road and air competition make it essential to ensure the development of still more effective coaches. The coaches should be lighter and they must be strong and energy-absorbing in accidents. Riding qualities have become less of a problem and running-gear maintenance requirements will be reduced, particularly by the use of rational tire profiles, but effective braking from high speeds will demand considerable attention. Heating and noise insulation will also demand further attention.

RRIS: 040499

TDOP: 02-093

Koffman, J. L., "Primary Suspensions For Coach Bogies," Railway Gazette, Vol. 124, February 2, 1968, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 108-111.

British Railways adopted the Schlieren type of cylindrical axle guides with their B4 and B5 bogies, now used in large numbers under coaches running extensively at 100 mph. The use of friction dampers to reduce bogie pitching oscillations at speeds above 75 mph is described. The design of the integral friction damper for the primary suspension of high-speed Russian coaches is illustrated and described. The Clouth roll spring used on Hamburg Underground and the German Railway interurban coaches is also described. Test results for Clouth springs on a British Railways experimental bogie are reported.

RRIS: 037466

TDOP: 02-094

Koffman, J. L., "Test Truck Of British Railways With Rubber Suspension," Glasers Annalen ZEV, Vol. 96, No. 9, September 1972, Siemens (Georg) Verlagsbuchhandlung, Luetzowstrasse 6, 1 Berlin 30, West Germany, pp. 285-288.

This paper covers the design of a truck based on the use of rubber rolling ring primary suspension units for the first stage and air springs for the second stage (bolster) springs, which ensure riding qualities at speeds of up to 145 km/h.

RRIS: 046803

TDOP: 02-095

Lejeune, M., "Improving The Suspension And Stability Of Running Of Bogies," Railway Gazette, Vol. 95, December 14, 1951, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 660-661.

New coaches have been equipped with an improved Pennsylvania bogie having reduced play about 1 mm in new stock designed to take nearly 50 percent of the total load on the bogie. Coaches fitted with these new bogies have an excellent degree of stability in running up to speeds of 100 mph and over. Double elliptical springs in the bogie were eliminated and were replaced by something flexible and able to damp all stock gradually. The design enabled the total vertical accelerative movements, measured inside the vehicle, to be reduced by some 50 percent, and the shocks caused by passing over the rail joints to be noticeably lessened. These bogies have a control device or system of longitudinal and lateral stay-bars, which dispenses with horn plates on the axle boxes or guides at the ends of the swing bolsters.

RRIS: 037788

TDOP: 02-096

Lich, R. L., "The General 70, A New Family Of Trucks For Rapid Transit, Commuter And Main Line Equipment," ASME, 63-WA-331, 345 East 47th Street, N. Y., N. Y. 10017.

This article describes the development of an improved passenger car truck concept meeting the equipment requirements of future rapid transit and commuter systems. The new arrangement is designed to provide an optimum combination of low cost, light weight, good riding qualities, overall stability, and flexibility. Numerous photographs of the trucks are included.

RRIS: 040134
TDOP: 02-097

Manzo, M., "Trial Runs At Speeds Exceeding 200 km/h Of New Rolling Stock Designed For The Italian State Railways," Rail International May 1965, International Railway Congress Association, 17-21 Rue de Louvrain, 1000 Brussels, Belgium, pp. 323-345.

Report on the performance and capabilities of an Italian high-speed rail car. The car was designed for speeds of 160 km/h, though tests included speeds of 200 km/h. Discussion of pantograph used for current collection is included. A detailed report on the bogie design and structural considerations is also included. The test results of the components of the pantograph and bogie assembly suggest that the units are capable of exceeding maximum speed of 225 km/h.

RRIS: 033382
TDOP: 02-098

Mauzin, A., "Results Of Trials With A New Type Of Bogie Designed By The S. N. C. F.," French Railway Techniques, No. 3, 1966, 20 Boulevard Diderot, Paris - 120, pp. 145-148.

The article considers the transverse stability of a prototype Y 207 passenger car truck operating at speeds from 140-145 km/h. Wheel treads were worn the equivalent of 350,000 kilometers to generate shaking of the truck and vehicle. Also considered is the effect of longitudinal play in the axle boxes on transverse stability.

TDOP: 02-099

Nelson, J. A. and M. J. Hapeman, "A New Transit Propulsion Unit Suspension Proved On Northeast Corridor High Speed Test Cars," ASME, 66-RR-3, 345 East 47th Street, N. Y., N. Y. 10017 (paper presented at Joint ASME-IEEE Railroad Conference, 1969).

This paper describes a new concept in the suspension of transit propulsion units. The new suspension system corrects deficiencies in the previous standard design while retaining the principle of floating the motor in rubber to isolate it from rail shocks. The new propulsion unit arrangement for lightweight, inboard-journal trucks has allowed car operating speeds to increase from 75 mph to 150 mph. The paper includes both diagrams and an analysis of the system.

RRIS: 033237

TDOP: 02-100

Parsons-Brinkerhoff-Tudor-Bechtel, Assuring The Stability Of The BARTD Lightweight Rapid Transit Vehicle, by W. A. Bugge, April 1974, Project Report for the San Francisco Bay Area Rapid Transit District, 800 Madison Street, Oakland, California 94607.

This report concludes that a vehicle-track system designed to a 5 ft 6 in. gauge would provide the necessary degree of stability for the BART lightweight cars. The lateral stability investigation, research procedures employed, findings, and recommendations are included.

TDOP: 02-101

Robert, J., "Improving The Running Qualities Of The Coaches To Be Included In High Speed Luxury Trains. Modern Bogies. Possible Technical Evolution," Bulletin of the International Railway Congress Association, High Speed Symposium, 1968, Vienna.

This article discusses possible technical evolutions of traction and rolling stock designs of coaches for high-speed luxury trains of Europe (1968). Schedule regularity and safety are requirements but speed and comfort are attractions which must be included in technical design. Research work covers elastic connections between axles and truck frames, wheel profiles to obtain stability at speeds of 250 km/h, vertical suspension for modern light flexible coaches, and reduction of unsprung weight which limits maximum speeds. Compensation for insufficiency of superelevation on curves and the relation of the lack of superelevation to braking of fast, heavy trains must be considered.

TDOP: 02-102

Temple Press Limited, "Air Resistance Of Fast Passenger Trains," Railway Gazette, Vol. 123, August 15, 1967, 161-166 Fleet Street, London EC4, England, pp. 710-711.

The article discusses factors to be considered in the design of fast passenger trains based upon experiments with model rolling stock. Factors discussed are body shape, window shape, corrugations, space between coaches, flush lining of roof and sides, truck design, and coach ends.

RRIS: 037814
TDOP: 02-103

Temple Press Limited, "Lightweight Passenger Stock Development In U. S. A.," Railway Gazette Vol. 107, August 2, 1957, 161-166 Fleet Street, London EC4, England, pp. 135-136.

A prototype stainless-steel passenger car, the Pioneer III, which meets all the strength and safety requirements of the Association of American Railroads, with a net weight of 595 lb per passenger, has been designed by the Budd Company, Philadelphia. A feature is the lightweight four-wheel bogie with a single air spring on each side, tubular axles, and outside-mounted disc brakes. The basic design is suitable for both suburban and main-line stock. The principal parts of the bogie are the side frames, bolster, and two air springs. There are no equalizer beams, springs, swing hangers, spring planks, or transom. Interior partitions are of micarta-faced plywood, which is resistant to wear. Use is also made of laminated plastics for doors, vestibule ceiling panels, exterior door inner panels, air ducts, enclosure around the air conditioning, and seating.

RRIS: 037637
TDOP: 02-104
(Also see 02-088)

Temple Press Limited, "Standard German Passenger Bogie" Railway Gazette, Vol. 113, December 2, 1960, 161-166 Fleet Street, London EC4, England.

The Minden-Dentz standard passenger bogie is described in detail. Development objectives are stated, and the methods for achieving these objectives are given. Modifications to the basic design are mentioned, as well as indications of experimental work to incorporate the latest forms of rubber and pneumatic springing.

RRIS: 039685
TDOP: 02-105

Temple Press Limited, "Tokaido Line Standards, Test Length and Trials," Railway Gazette Vol. 118, No. 11, March 15, 1963, 161-166 Fleet Street, London EC4, England, pp. 209-301.

Various characteristics of the new Tokaido Line are presented, including track, tunnel, and bridge construction, bogie design, and passenger and track inspection vehicles. Testing of rolling stock at 124 mph prior to the opening of the line revealed wheel side thrust to be only 3-4 tons and lateral and vertical vibration accelerations of 0.2 g and 0.3 g, respectively.

RRIS: 037262
TDOP: 02-106

Terrase, R. and R. Joly, "Running Of Tilting Railway Vehicles Over Curved Track--General Aspects Recent Tests With The S. N. C. F. Tilting Coach," French Railway Techniques, No. 3, 1970, Federation des Industriels Ferroviaires, 92 Rue Bonaparte, 75 Paris 6e, France, pp. 89-103.

This article discusses the problems of passenger comfort as related to high-speed trains on a canted, curved track. The French experiments using a test vehicle to determine means to correct for this cant are detailed including the specifications of the vehicle. The ways in which solutions were found to the problem of cant were tested. Recommendations concerning types of equipment to solve the problem are also discussed.

RRIS: 033849
TDOP: 02-107

Wilcock, H., "High Speed Mark III Coach," Railway Engineering Journal, July 1974, Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London SW1, England, pp. 26-31.

This article deals with the development of British Rail's Mark III coach, which is represented as a major advance in many aspects of coach and truck design. In 1974, prototype trains were undergoing trials at speeds up to 200 km/h (125 mph), and completed trains are expected to be in service in 1976.

TDOP: 02-108

Locomotive Truck Design

Batchelor, G. H., "Bogie Locomotive Riding Problems," Railway Gazette, Vol. 120, May 1, 1964, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 353-356.

The performance of vehicle suspension can be adversely affected by lateral plane design, which appears to be of minor importance. The low-order restoring forces due to swinglinks, which are about one ton per inch per bolster, are discussed as to their relationship to ride quality. A Bo-Bo electric locomotive with 20-inch vertical swinglinks is used as an example. The link ends were mounted on rubber and rubber snubbers were required to prevent impact between the bolster and the truck frame. Lateral pull tests, which were conducted to determine the source of the poor ride quality, are described for this system. It is shown that the effective length of the 20-inch swinglinks is only 6.8 inches.

RRIS: 040197

TDOP: 02-109

Brittell, C. W., "Motive Power Life Cycle Costing," Railway Systems and Management Association, February 1969, 163 East Walton Street, Chicago, Illinois 60611, pp. 49-53.

The design changes to trucks and locomotive components needed to upgrade a four-motor diesel locomotive to 3,000 horsepower are described. More horsepower per traction motor increases the tendency of the locomotive to lose adhesion. ALCO has a new high-adhesion, four-wheel truck designed to reduce weight transfer to a minimum. Cash flow analysis for making decisions on replacement of locomotives is illustrated.

RRIS: 039523
TDOP: 02-110

Croft, E. H., "Bogie Design For Electric Locomotives--1," Railway Gazette, Vol. 105, August 1956, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 264-265.

Methods are given for limiting transverse forces on the track, and stresses on the bogie and body. The design concepts are discussed from the aspects of both wheel diameter and bogie oscillations. The use of rubber as a cushioning medium is also described.

RRIS: 039961
TDOP: 02-111

Croft, E. H., "Bogie Design For Electric Locomotives--2," Railway Gazette, Vol. 105, September 28, 1956, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 384-385.

Suspension, adhesion, and riding requirements of general service locomotives are discussed. Attention is given to three-point suspension, weight transfer reduction, Bo-Bo characteristics, bogie behavior on curves, and the need for three-dimensional rigidity in the frame types.

RRIS: 039963
TDOP: 02-112

Ell, S. O. "Some Design Problems Of Diesel Locomotives," Institution of Locomotive Engineers Journal, Paper No. 685, Vol. 56, No. 6, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London SW1, England, pp. 543-571.

This paper focuses on the problems of diesel-hydraulic vibration systems, the power transmission to axles via a geared system, and the ride problems of the D. 800 and D. 1000 locomotive classes. Illustrations reveal shaft failures, comparisons of original and tuned vibration systems, crankshaft torsional vibrations, vertical movements of locomotive bogies, and tire profiles of new and worn wheels.

RRIS: 040486
TDOP: 02-113

Guillemard, "Details Of An Initial Experimental Gas Turbine Railcar Set," French Railway Techniques, No. 2, 1967, Federation des Industriels Ferroviaires, 92 rue Bonaparte, 75 Paris 6e, France, pp. 67-74.

For the prototype gas turbine set, a 330-kW, diesel-powered railcar set was selected for starting. The original trucks were satisfactory for operation to 150 k/h, but were replaced with Y-214 trucks for greater speed capability. The air intake and exhaust systems and the soundproofing necessary for the installation of the 297 kg turbine motor are described. The transmission system is described and illustrated. The fire protection system, heating system, air condition system, and the towing gear are briefly discussed.

RRIS: 040017
TDOP: 02-114

Lampe, C. and N. Gossl, "The 2,000 HP Diesel Locomotive Of The German Federal Railways," Railway Technical Review, October 1955, Carl Rohrig Verlag, Darmstadt, West Germany, pp. 2-16.

This document discusses the advantages of the design of the 2,000 hp diesel locomotive as used by the German Federal Railways. The power transmission unit, cooling system, and the engine are described and illustrated. Truck design is also noted. The maintenance requirements are given.

RRIS: 037811
TDOP: 02-115

Marta, H. A., K. D. Mels, and G. S. Itami, "Design Features and Performance Characteristics of the High Traction, Three-Axle Truck," ASME, 72-RT-3, 345 East 47th Street, N. Y., N. Y. 10017 (see Reprints--Truck Design).

This article describes a locomotive truck design providing improved adhesion performance. The improved adhesion was made possible by the ability of the truck to minimize the reduction of axle load on rail during operation. Results of predicted performance characteristics, and of laboratory and field tests are presented and compared to the more conventional SD type flexicoil truck.

TDOP: 02-116

McArd, G. W., "Unit Bearing Pressures For Locomotive Details," Railway Gazette, Vol. 95, November 14, 1941, Temple Press Limited, 161-166 Fleet Street, London EC4, England, pp. 497-498.

Data are compiled and reported for unit bearing pressures for locomotives giving satisfactory service. The pressure or load curve and the bending moment diagram are shown for the outside cylinder slidebars of some express passenger engines built for the Central Argentine Railway. British and U.S. design standards for axlebox bearing pressures for passenger, freight, and shunting engines are given. The pressure curve is given for one revolution of the driving wheel of an engine having two outside cylinders with 19 in. diameter by 26 in. stroke, coupled wheels with a 6 foot, 2.5 in. diameter, and a boiler pressure of 225 lb/sq in.

RRIS: 037260
TDOP: 02-117

Schmuecker, H., "Switch Locomotive V 90 Of The German Federated Railways," Eisenbahntechnische Rundschau, Vol. 14, No. 3, Hestra-Verlag, Darmstadt, West Germany, pp. 91-92.

Only a portion of this article describes the locomotive. One portion deals with the latest design of cardan shafts having ball joints made of plastic that eliminate the need for periodic lubrication servicing. The other describes the truck design, which includes springs made of rubber "sandwiched" with steel plates that are used at the journal boxes and over the coil springs between the truck and the locomotive frame. The truck is of a simple design, with cast members welded together, and with the traction transfer points below the axle center line. The composite design has effectively reduced lateral and vertical accelerations.

RRIS: 037611
TDOP: 02-118

Temple Press Limited, "Bogie Designs For Russian Diesel Locomotives," Railway Gazette, Vol. 110, February 27, 1959, 161-166 Fleet Street, London EC4, England, pp. 242-244.

Road tests of the Russian TE. 3 Co-Co diesel-electric locomotives, which contain primary, half-elliptic spring systems only, concluded that bogie design problems limit speeds to 60-75 mph. A modified TE. 3, designated TE. 7, was built for speeds up to 88 mph. Because of lateral forces between the bogie frame and the leading axles, ride quality was considered unsatisfactory above 60 mph. A new, yet untested design for a six-wheel bogie with Alsthom-type axlebox suspension and helical spring bolster supports is described and illustrated.

RRIS: 037708
TDOP: 02-119

Temple Press Limited, "Design of Bogies For Electric Locomotives," Railway Gazette, 161-166 Fleet Street, London EC4, England, pp. 601-602.

An improved bogie developed by the Swiss Locomotive and Machine Works of Winterthur incorporates transverse coupling together with a centering device to prevent hunting and friction dampers on the axlebox guides. Experimental results showed that if two bogies are coupled by means of a transverse bar that there is a reduction in (a) the guiding pressure on the leading axles of both bogies, and (b) the angle of incidence of the leading axles. The two bogies are connected by a spring coupling at the end of a triangular yoke on each bogie. The coupling permits a degree of side play adjustable to any value between 20 and 500 mm after which further relative movement is controlled by two helical springs, which may be adjusted to allow a maximum movement of 24 mm.

RRIS: 037936
TDOP: 02-120

Temple Press Limited, "English Electric 2, 700 hp Diesel-Electrics for B. R.," Railway Gazette, Vol. 123, August 18, 1967, 161-166 Fleet Street, London EC4, England, pp. 625-628.

This article discusses the first of a fleet of 50 Co-Co Type 4 locomotives with charge-air cooled 16-cylinder engines which will be put in service by British Railways. The D400 class will be the first diesel-electric locomotives to incorporate automatic control of tractive effort, slow-speed running, and integrated braking. This latter provides the correct dynamic brake application for the degree of train brake applied. The cast steel bogies and the traction motors have the provision for the fitting of weight transfer compensation equipment, if required. The maximum axle-load is 19-1/2 tons, and the bogie wheelbase is 13 ft. 6 in.

RRIS: 037825
TDOP: 02-121

Temple Press Limited, "Italian Diesels of 2,000 HP," Railway Gazette, Vol. 123, No. 7, July 21, 1967, 161-166 Fleet Street, London EC4, England, pp. 529-537.

Italian diesels of 2,000 hp of advanced four-axle design are 33 percent greater in output than the 75 GR. D343 locomotives being built. Many parts are standard and interchangeable. In particular, the gear-drive systems for the monomotor bogies and the traction motors are interchangeable. Driving cabs and contents are the same, as are such details as air compressors, filters, radiator elements, and so forth. The systems and components of each engine are described and illustrated.

RRIS: 037823

TDOP: 02-122

Temple Press Limited, "Powered Tipper Wagons For Iron Ore Trains," Railway Gazette, 161-166 Fleet Street, London EC4, England, pp. 654-658.

Paper describes how by motoring the axles of two cars next to the locomotive, long 65-percent gradients can be tackled with heavy trains. The solution of the problem in East Germany was found by constructing special cars with powered trucks similar to those on the locomotive, which, in effect, created a 12-axle power unit. This combination could also haul eight cars but, in addition, the two motored cars were also carrying ore so net train load was increased to 750 tons for no increase in gross train weight.

RRIS: 037176

TDOP: 02-123

Temple Press Limited, "Reducing Flange-Wear On Bogie Locomotives," Railway Gazette, Vol. 111, December 18, 1959, 161-166 Fleet Street, London EC4, England, p 559.

The phenomenon of relatively rapid flange-wear on sharp curves with the bogies of Bo-Bo and Co-Co locomotives must be accepted pending the results of further research into the reduction of bogie weight and unsprung weight, the height of the bogie centre of gravity, different relative speeds of the axles and motors, weight transfer, etc. Flange or rail lubrication using a lubricant matched in viscosity to suit the ambient temperature lengthens the flange life from six to nine times, this factor varying according to the tire steel used. A change from ties of "D" steel to those of 70-ton tensile steel alone can increase flange life by as much as three times. A combination of lubrication, hard tires, and the fitting of a centralizer linking the movement of the two bodies, which is capable of drastically reducing flange forces on curves, will improve flange life by 12 to 15 times.

RRIS: 039311
TDOP: 02-124

Temple Press Limited, "Universal Bogie Designs For Hungarian Built Locomotives," Railway Gazette, Vol. 123, February 3, 1967, 161-166 Fleet Street, London EC4, England, pp. 112-113.

Multi-gauge, twin-axle truck designs are described for locomotive power from 600 to 2,000 hp and axleloads up to 20 tons. An important aspect was to design with three principal objects: to keep down the unsprung weight, increase the wheelbase, and provide suitable spring characteristics and adequate damping. Riding properties of locomotives equipped with this basic type of bogie have been found to be very good. Effectiveness of the nonlinear primary suspension has fulfilled expectations and, even without damping at this stage, no tendency to resonance conditions has been encountered.

RRIS: 037829
TDOP: 02-125

Voith Getriebe, K. G., "4,000 HP Locomotive Units For U. S. A.," Diesel Railway Traction, August 1961, Heidenheim, West Germany, pp. 295-325.

European design locomotives forming the most powerful single locomotive units in the world were delivered to the United States for heavy drag-freight service. The six large diesel-hydraulic locomotives are powered by two Maybach engines of 2,000 bhp each, U. I. C. rating, driving through a special Voith L. 830 transmission. To each set, a fluid coupling has been added to deal with resistance braking, which is a leading feature of these locomotives.

RRIS: 037807

TDOP: 02-126

Wakefield, F.H.G., "Twenty Years' Experience With Diesel Railcars," Railway Division Journal, Vol. 2, Part 1, 1971, Institution of Mechanical Engineers, 1 Birdcage Walk, London SW1, England, pp. 45-83.

The following topics are discussed: (1.) Bodies--A short discussion on the relative merits of light alloy and steel construction is given together with some notes on the writer's experience with various materials used in the passenger areas and a discussion on the problems of heating this type of vehicle. (2.) Bogies--Some notes on the problems which have been associated with the brake gear are given, followed by a discussion of the riding problems experienced with these vehicles, and then some notes on the problems encountered with axle design. (3.) Power Equipment--A short discussion is included on the relative merits (based on experience) of gear boxes and torque converters. Some notes on the various problems that have been associated with the final drive units are given together with a discussion of performance.

RRIS: 040529

TDOP: 02-127

Watts, P.H., "Weight Transfer Compensation In Four-Axle Direct Current Locomotives," Institution of Locomotive Engineers Journal, Vol. 59, No. 328, Part 2, Institution of Locomotive Engineers, Locomotive House, 30 Buckingham Gate, London SW1, England, pp. 143-153.

The intention in providing a compensation mechanism is to adjust the individual axleloads, their motor outputs, or both, so that the adhesion level required at each axle approaches more closely the general level of adhesion required by the locomotive as a whole. Electrical compensation methods use the latter approach of modifying the motor outputs, and the mechanical methods affect the relative axle loadings. If the compensated and uncompensated locomotives are compared while producing equal tractive efforts, it can be seen that the uncompensated locomotive requires a considerably higher level of adhesion if the leading axle is not to slip.

RRIS: 040538

TDOP: 02-128