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TRUCK DESIGN OPTIMIZATION PROJECT PHASE II

INTRODUCTORY REPORT

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DEPARTMENT OF TRANSPORTATION • UNITED STATES OF AMERICA •

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16. Abstract

The Truck Design Optimization Project (TDOP) Phase II is being conducted to establish the performance and cost-effectiveness of premium freight car trucks with reference to the standard three-piece truck. This report sets up a framework for quantitative characterization of truck performance and outlines a method for collecting economic data. Means of achieving these objectives by road-testing, mathematical modeling, and review of maintenance and operational data are described. An important goal of the project is to supply the railroad industry with a basis for performance specifications for freight car trucks.

This report is the first of a series that will be published under the major title Truck Design Optimization Project, Phase II as the multi-year program develops.

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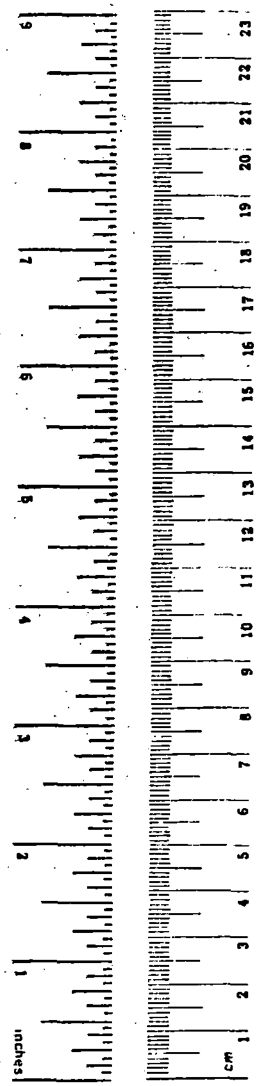
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

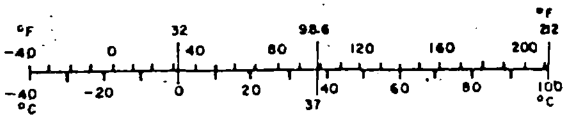
Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
	MASS (weight)			
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
	VOLUME			
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
	TEMPERATURE (exact)			
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



EXECUTIVE SUMMARY

Increasing demands on freight car performance during the past several decades have revealed the shortcomings of the standard three-piece freight car truck. The evidence comprises increased truck maintenance, damage to track, and more frequent derailments.

Truck manufacturers are responding to the need for better truck performance by introducing modifications of the standard truck as well as novel configurations designed to reduce problems under specific operating conditions. Justification for the higher cost of improved trucks is difficult to establish for two reasons:

- Too little quantitative information is available on the characteristics of the standard truck, as well as that of the various new designs, to make possible a comparison in engineering terms.
- While data on truck maintenance cost and freight car utilization are available in the files of railroads and operators, these data have not been systematically related to truck performance.

The objectives of the Truck Design Optimization Project (TDOP) Phase II conducted by the Federal Railrod Administration are:

- To define the performance of both standard and premium trucks in quantitative terms, represented by performance indices.
- To establish a plan for collecting economic data on the cost of acquiring, operating and maintaining the standard three-piece truck.
- To establish a quantitative basis for evaluating the economic benefits to be derived from improved freight car trucks.
- To supply a basis for a performance specification for freight car trucks.

The means by which these objectives will be achieved are:

- Road testing of several representative car body types on a number of premium trucks.
- Mathematical modeling of freight cars and trucks and comparison of model test results with a view towards extending knowledge of truck behavior to configurations not tested.
- Determination of wear of premium trucks in unit train service over an extended period of time.
- Collection of economic data on truck maintenance and operation, and correlation of such data with information on truck performance.

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1.1 INCREASING DEMANDS ON RAIL TRANSPORTATION SYSTEMS

The demands made on the country's rail transportation systems have steadily risen over the past few decades. The gross rail weight of freight cars has shown a continuing increase, which results in higher wheel loads and accelerated deterioration of the track, while higher centers of gravity increase the risk of derailment. In accommodating themselves to shippers' requirements, the railroads have introduced a number of new freight car configurations which, by virtue of extreme inertial or structural properties, such as low torsional stiffness, or high center of gravity, give rise to problems of operation and maintenance more severe than was the case for cars of more conventional proportions. In addition, maintenance and repair problems are aggravated by the high utilization rate of unit trains.

1.2 PERFORMANCE SPECIFICATION FOR FREIGHT CAR TRUCKS

The freight car truck performs a number of essential functions. The truck 1) attenuates the magnitude of impulsive and periodic forces, arising from track imperfections, which are transmitted to the carbody and lading, 2) damps motions of the carbody excited by rail irregularities, 3) maintains adequate vertical wheel loads to guard against derailment in the presence of lateral forces, 4) transmits braking forces, and 5) guides the car around curves. In performing these functions, the components of the truck suffer various forms of deterioration, such as wear and fatigue failure.

While we can determine the cost of repair and replacement of worn and failed components, of impaired operation, and of damage to lading, there is a lack of specific, detailed information that would make it possible to evaluate quantitatively the performance of a freight car truck against its cost of acquisition and maintenance.

Recent advances in freight car truck design have made the task of evaluating truck performance even more complex. Design changes range from add-on devices, such as constant-contact side bearings, to new truck configurations that are based on the better understanding of vehicle dynamics gained from analysis, laboratory experiments and road tests.

In the area related to economics, a large body of data has been collected in the form of maintenance and repair data, car movement files and track conditions. However, except for a few valuable pilot studies, this information has remained substantially uncorrelated with the design features of specific trucks. It is a principal objective of this report to set up a framework within which relationships between the performance and cost of freight car trucks can be established.

1.3 "TRUCK PERFORMANCE" DEFINED

A railroad or operator needs a means for evaluating the cost effectiveness of a freight car truck with respect to its operating conditions. These conditions may differ considerably from one railroad to another. For example, a railroad operating primarily in mountainous territory at relatively low speeds may be concerned with reducing wear of wheels and rails in curves. On the other hand, a railroad operating in flat terrain at high speeds will require that its trucks have a high critical speed of hunting. The design features of a truck that fulfills both these requirements are, to some extent, incompatible. Another example of such comparative incompatibility is the rigid truck with primary suspension. The low unsprung mass of this design reduces dynamic rail loads, and the rigidity of the frame improves curving behavior; however, the ability to maintain adequate vertical load distribution under cross-level irregularities is impaired.

A useful characterization of truck performance thus requires the identification of specific performance regimes which may be defined as sets of conditions associated with predominant features that distinguish one regime from another. This is illustrated by the examples of curving, hunting and equalization mentioned above. Besides being distinct and non-overlapping, the set of performance regimes should be inclusive, i.e., identify all aspects of truck behavior. This subject is discussed in greater detail in Section 3.

Performance criteria express qualitatively the aspects of truck behavior considered desirable in the various performance regimes. Criteria may range from the most general, such as safety from derailment or low wear rates, to the specific, such as lateral stability or curve negotiability.

In order to make possible the quantitative evaluation of truck performance, both absolute and comparative, each performance regime must be associated with a performance index, by which is meant a measurable quantity typical of that regime. Examples are the critical speed of hunting, lateral wheel load in curves, and minimum vertical wheel load, which are characteristic of the lateral stability, curve negotiation and equalization regimes, respectively.

A truck performance specification defines a range of performance indices for each performance regime that a truck must meet under specified operating conditions, such as speed, track quality and degree of curves, with due regard to state of wear or other deterioration associated with age or ton mileage. Specification of performance in quantitative, operational terms will, on the one hand, give latitude to design innovation and on the other, will facilitate correlation with economic factors.

The establishment, by testing and analysis, of performance indices for both conventional and improved freight car trucks is the objective of the Truck Design Optimization Project (TDOP).

1.4 TRUCK DESIGN OPTIMIZATION PROJECT (TDOP)

The first formal attempts to specify truck performance were made by the Committee on Freight Car Trucks of the Association of American Railroads (AAR) in the 1960's. In the context of Section 1.3, the results of that earlier effort would be equivalent to a set of performance criteria of trucks. P.V. Garin and R. Byrne of the Southern Pacific Transportation Company took a leading part in this activity.

1.4.1 TDOP Phase I

Beginning in 1972, the Federal Railroad Administration (FRA) sponsored a broad-based effort to quantify freight car truck performance. Southern Pacific Transportation Company was the contractor for this first phase of the Truck Design Optimization Project. A number of conventional three-piece trucks built by American Steel Foundries (Ride Control) and the Standard Car Truck Company (Barber S-2) were tested under five different car bodies, both 70 and 100-ton. Additional tests were performed with trucks modified by the addition of special hardware intended to improve operation under specific conditions. The data from Phase I constitute the main basis for characterizing the performance of the standard or Type I freight car truck. As will be noted later on, it will be necessary to supplement these data by additional tests.

1.4.2 TDOP Phase II

Wyle Laboratories is the contractor for TDOP Phase II, with the Union Pacific Railroad as the principal subcontractor. The scope of the project includes the testing and analysis of premium or Type II trucks. For the purposes of this project, the main restriction placed on a Type II truck is that it preserve coupler height, but the method of mounting the wheelsets on the frame, and of supporting the carbody are not specified. This allows the project to evaluate a number of truck designs, both domestic and foreign, that differ to varying degrees from the conventional three-piece truck defined as Type I.

The widening of the performance range due to the inclusion of premium trucks underscores the need for establishing the quantitative measures of Type I truck performance discussed in Section 1.3. Thus, the first task to be performed in TDOP Phase II and the planned approach constitutes the subject of this report. The task is summarized in the Statement of Work issued by the FRA:

> "The Contractor shall define performance indices which can be correlated to the economics of railroad operation. Such performance indices should include ride quality, curve negotiability, safe operating speed, track forces, and truck component loadings as related to wear maintenance, efficiency, and operating conditions. The Contractor shall then define these performance indices in terms of the economic relationships to railroad operating costs and profits. The Contractor shall make use of the data collected in Phase I TDOP as well as any appropriate data collected under the programs itemized in Attachment 2, which lists government-sponsored research efforts. The Contractor shall review this information for the purpose of determining the adequacy of available test data for quantifying these performance indices for a Type I freight car truck, for quantifying the relationship

between the performance indices and the economic data for the complete characterization of the Type I truck, for deriving performance and test specifications, and for yielding data applicable to model development and validation. Consideration should be given to such items as: Is there adequate data to a) characterize Type I truck performance under the most representative conditions of wear; b) define performance variation in terms of suspension system modifications including side bearing and stabilizer influence; c) evaluate ride quality characteristics with modified and unmodified truck suspensions under varying car body designs; d) correlate performance and wear versus operating conditions; e) provide input to and validation criteria for analysis tools; and f) generate total economically based performance specifications for this class of truck."

In order to meet the objectives, it is necessary to establish a frame of reference and a set of performance criteria that would be both inclusive and flexible enough to accommodate all aspects of truck performance, and also responsive to the needs of those responsible for the acquisition, operation and maintenance of freight car trucks.

1.5 REPORT ORGANIZATION

Section 2 of this report describes the characteristics of the freight car truck, and the environment in which it must operate. It outlines the methods by which its performance is investigated; these include road testing, mathematical modeling, and analysis of maintenance and repair records.

Section 3 introduces a comprehensive scheme for establishing the characteristics of the truck in terms of performance and deterioration, and points out the need for translating conventional descriptions of these aspects into a terminology more amenable to engineering and economic analysis.

Section 4 outlines the approach to quantification of truck performance in several distinct operational regimes. The goal is the establishment of performance indices by means of which trucks of different characteristics can be objectively rated and specified to suit specific operational requirements.

Section 5 gives an overview of the parallel engineering and economic studies aimed at establishing the relationships between performance and cost of the three piece truck. A plan for collecting additional economic data is outlined. Finally, Section 6 presents a summary of work to be performed in TDOP Phase II.

SECTION 2 - THE FREIGHT CAR TRUCK: ITS DESIGN AND EVALUATION

Apart from track, the freight car truck has long been recognized as the most critical component of the rail vehicle system. It supports the carbody and its lading, guides it along the track, transmits braking forces, and isolates the car to some extent from dynamic excitation caused by track irregularities. In performing these functions, the truck is subjected to wear and impact due to the highly concentrated forces at the contact points between its components.

The design of the standard three-piece freight car truck, last improved by the addition of friction snubbing in the "thirties", must be considered as an engineering, manufacturing and economic achievement. It is relatively inexpensive to produce in large quantities by casting. Its loose construction, i.e., the bolster-side frame and adapter-pedestal connections, make it tolerant of vertical track irregularities (a property generally termed "equalization" which refers to the load distribution over the four wheels when their contact points are not in a plane). The truck's relatively few standardized components can be readily stocked at repair facilities, and maintaining or repairing it does not require a highly skilled labor force.

The efficiency of the standard truck design may be appreciated from the fact that the weight of a carset (i.e., a pair of trucks) is only 10% or less of the gross vehicle weight. This is remarkable in view of the restrictions placed on the truck envelope by the limiting dimensions of height, width and rail clearance within which both the suspension components and the load-carrying structure must be accommodated. In an engineering sense, the three-piece truck may be regarded as optimized.

2.1 THE TRUCK ENVIRONMENT

The performance of the standard truck becomes even more remarkable when one considers that it must operate under a wide range of conditions. The period during which these conditions change range from years to fractions of a second.

Deterioration of truck components, such as wear of wheels, snubbers, gibs and adapters, takes place over several years and profoundly modifies truck performance, usually for the worse. Changes in the foundation modulus of the track are in part caused by seasonal and diurnal variations of temperature and humidity; these changes in turn affect truck behavior, as may be evidenced by the number of derailments on frozen ground, or on rail forced out of gauge limits by thermal expansion, and by the increase in critical speed on wet rail.

A major change in operating conditions is represented by the total gross rail weight, which may vary by a factor of two or more (depending on the lading) and occurs over periods measured in days or weeks. This variation in weight greatly affects the dynamic behavior of some cars, particularly with respect to hunting stability and harmonic roll.

The most important change in operating conditions to which the truck must accommodate itself is the variability in the track, which varies during time intervals measured in seconds to minutes, although some characteristics due to terrain may vary over hours. Changes encountered by the truck include transitions from tangent to curved track, from continuously welded to jointed rail, crossings, turnouts and bridges. Another variation in operating condition is caused by the difference in the standards to which a railroad maintains the track.

Adaptation of the truck to all of these changes in condition is, of course, not practical. Long term changes (e.g., deterioration) are routinely corrected by maintenance. At the other end of the time scale, changes in track properties must simply be handled by the ability of the truck to mitigate their effect on the car and lading. Maintaining gross rail weight is feasible only in unusual cases, for example, when a railroad runs unit trains loaded with ore in one direction, and with coal on the return trip. It is not surprising that hunting is less of a problem for cars that always run fully loaded, than for those that must travel empty at high speeds.

In recommending suspension parameters, truck manufacturers take into account, to a limited extent, the predominant service condition of the freight cars to be carried by their trucks. However, the range of available suspension parameters of three-piece trucks is limited to spring travel and friction snubber column load. This range is small compared to the operational variables discussed above.

2.2 TRUCK COMPONENT DETERIORATION

An essential requirement of the freight car truck is that it must perform in the environment described above in a state of increasing deterioration, which is brought about by its interaction with that environment. In the standard three-piece truck, the only precision components are the roller bearings, the journals of the axles on which they are mounted, and the surfaces of the adapters that mate them with the truck. All other components, such as the center bowl, friction snubbers, gibs, and the adapters themselves, transmit loads by metal-to-metal contacts through surfaces capable of moving relative to each other - -sliding - or normal to each other, through clearances. The resulting deterioration of these highly loaded connecting points or interfaces represents a large cost to the industry because repairs or replacements must be made to ensure safe operation and prevent delays. In addition, the changes in clearances as the connecting surfaces wear or deform changes the kinematics of the truck as a mechanism.

These changes have a profound effect on the dynamic response of the truck and the railcar body to the conditions of the track described above. For example, wear of the friction snubber shoe decreases the resistance to harmonic roll and thus results in increased probability of derailment when the vehicle is crossing staggered rail joints at certain critical speeds. Snubber wear also decreases the yaw stiffness of the bolster-side frame connection; the resulting increase in the degree of parallelogramming increases the angle of attack of the leading wheelset during curve negotiation and thus leads to increased wear of both wheels and rails. Comparable changes in truck performance can be associated with almost every other group of components.

Wheels, brakes and rails interact to produce complex wear patterns that affect performance long before components are changed or repaired to ensure safety of operation. As with the other examples given above, the cost of impaired effectiveness due to deterioration is not easy to establish since the mechanism of damage cannot be viewed in action and must be inferred by analysis of such damage.

2.3 DIFFICULTIES OF TRUCK IMPROVEMENTS

The standard three-piece truck is highly serviceable, but not perfect. Attempts to improve its performance and durability have a long history, with both railroads and suppliers contributing novel components as well as entirely new design concepts. The acceptance of these improvements is impeded not only by their unavoidably higher initial cost, but by the difficulty of justifying this cost in terms of improved performance, which is defined as a combination of higher safe operating speed, greater load-carrying capacity, improved ride quality, reduced damage to track structures, and lower maintenance cost.

Improvement requires detailed knowledge of what exists; thus, characterization must precede, and form the basis for, optimization. Three main sources of information on truck performance are road tests, mathematical models and maintenance records. Data from these sources are apt to be limited by the selection of variables, distorted by the method of acquisition, and sometimes deprived of potential usefulness by inappropriate interpretation. The problems associated with these information sources are briefly outlined below.

2.3.1 Road Tests

Since railroads tend to be pragmatic, their preferred procedure for characterizing truck performance from the operational and economic standpoints has been road tests. It seems safe to state that there is no time when at least one road test is not being conducted somewhere on the North American continent. A road test represents the real environment in all its complexity, rather than simulates it partially. This tends to lend credibility to the results which may be enhanced by direct observation of the test specimen.

The test approach to information gathering is useful when the variables can be unambiguously identified. For example, the stress distribution in a bolster under known conditions of speed and track properties, when combined with laboratory data, may be used with some confidence to predict fatigue life in bolsters manufactured to the same standards of quality control with respect to materials and workmanship. On the other hand, when some of the important variables are imprecisely known, prediction of vehicle behavior for conditions other than those tested becomes less reliable. An example is the influence of moist rail on the critical speed of hunting which illustrates the effect of a large change of conditions on the magnitude of the creep coefficient which is not accurately known in the first place.

The difficulty of predicting long-term results from road tests has been illustrated by a number of case histories. Modifications of a vehicle shown by road tests to solve a particular problem have at times given rise to other problems during prolonged service, caused by unforeseen interactions. The possibility of multiple causes contributing to an observed failure or malfunction greatly increases the difficulties of interpretation.

Lack of suitable transducers, and the difficulties involved in their development and application, have emerged in recent years as factors limiting the information that can be extracted from road tests. The variables most difficult to measure are generally the same as those causing the greatest problems in the mathematical modeling of the freight car truck. These are the areas between components in sliding contact which exhibit non-linear interactions. They include the bolster-side frame connection made through the friction snubbers, the contact areas between the pedestal and the adapter, the bolster gibs and side frame columns, and the wheel and rail. Since these sets of components constitute the major wear points of the truck (apart from the brake shoes), their changes in shape modify the kinematics of the three-piece truck and thus affect the dynamics of the vehicle to a large extent.

The problem of measuring the interaction between truck components at their contact areas has received considerable attention in TDOP Phase II. A transducer system for measuring snubbing friction forces and column loads (including differential column loads due to center plate friction moments during curve entry) is about to be road tested. Consideration is being given to the measurement of vertical, lateral and longitudinal wheel loads, including the lateral shift of the vertical load on the adapter crown due to side frame roll. It is also considered important to locate the transverse position of the wheel with respect to the rail, to make possible the correlation of wheel contour with curving behavior.

In summary, care will be taken that the road tests planned for TDOP Phase II will be properly conducted, adequately instrumented, and rationally interpreted.

2.3.2 Mathematical Models

Mathematical modeling is a useful and often indispensable tool for interpreting and predicting the behavior of dynamic systems. However, applications of modeling techniques to freight car trucks have had mixed success, and opinions concerning its usefulness for solving the railroads' problems are divided.

The foundation for the systematic application of mathematical modeling to some aspects of rail vehicle dynamics was laid in the 1960's by Alan Wickens and his group at the British Rail Research Establishment in Derby. Wickens' linearized analyses of hunting were supported by the results of both roller rig and track tests, although discrepancies between predicted and measured friction forces were noted.

2.3.2.1 Models of Freight Car Trucks. The mathematical modeling of the three-piece freight car truck represents a more complex task than the linearized versions of the passenger cars analyzed by Wickens. The non-linearities already referred to allow simplifying linearizations, with their efficient analysis in the frequency domain, to be made only under the assumption of small motions. However, the main interest of the railroad industry centers on those aspects of safety related to extreme behavior of freight cars, and on the damaging effects of both track- and self-excited force inputs of large magnitude, when system parameters change rapidly and often discontinuously. Typical examples are a freight car undergoing harmonic roll of large amplitude, or the wheel flanges of a hunting truck striking the rails. Mathematical models that deal with details of such behavior require analysis in the time domain, which is more costly and time consuming. The modeling is made more difficult by the scarcity of reliable measurements for the parameters that characterize the non-linear connections between truck components.

2.3.2.2 Validation. A mathematical model is said to be validated when it is shown to be able to reproduce or predict a physical event, sequence of events, or other representation of reality according to a set of criteria. These validation criteria may define an acceptable match between the measured and calculated values of a set of parameters, or the probability of an event's happening or not happening, or any other verifiable quantity or circumstance. However these criteria are defined, they must be related to some significant feature of the system being modeled in order to be meaningful in a practical engineering context. In the end, the selection of validation criteria is based on professional judgment, and a consensus of informed opinion is required to decide on the validity of the model.

The validation criteria may thus be said to be equal in importance not only to the construction of the mathematical model but also to the kind of measurements, and their accuracy, with which the modeling results are to be compared. It follows that the activities of model construction, identification of validation criteria, and selection of test conditions and measurement channels must be closely coordinated.

Besides providing valuable insight, a well-validated model can reduce the amount of testing required. However, the confidence that may be placed in a model's ability to predict conditions other than those shown to agree with test results is commensurate with the extent to which it has been validated.

2.3.2.3 Mathematical Models in TDOP Phase II. Mathematical models will be utilized in TDOP Phase II to analyze and predict the behavior of trucks and freight cars. Validation of existing models, and others that may be required, is an essential part of the work being carried out.

Validation of models representing freight cars on three-piece trucks will initially be based on test data acquired in Phase I. An approach to modeling, given in the Appendix, is illustrated by the analysis of the dynamics of a freight car under normal operating conditions through an evaluation of test data and their comparison with the results of some simple models. The example demonstrates a number of the points made above, including limitations arising from unavoidable imperfections in the data, which circumscribe the limits of validity but do not preclude some useful conclusions and insights.

2.3.3 Evaluation of Wear and Damage

Establishing the cost-effectiveness of the freight car truck requires not only the analysis of performance under given operating conditions, but also an assessment of the deterioration of truck components, the car body, and the lading, as a result of these operating conditions.

One impediment to determining the cost effectiveness of a specific freight car truck is the form in which records of repair and maintenance are kept. It is difficult to establish the real causes of a particular case of deterioration in a truck component. One of the most important results to be achieved in TDOP Phase II is the establishment of correlations between the performance of the truck, and the cost of realizing that performance. The collection of additional economic data will be discussed in a separate report. A framework for correlating data on truck performance and deterioration is presented in Section 3 of this report.

2.4 TRUCK DESIGN IMPROVEMENTS

While many details on truck performance and deterioration remain to be dealt with quantitatively, there exists a basis of knowledge and data sufficient to suggest major improvements in design. These improvements are of two main kinds:

- Modification of the three-piece truck by addition of special hardware
- Alternate truck configurations.

2.4.1 Modified Three-Piece Trucks

A number of special components have been developed to improve the roll dynamics of the carbody. Among these are constant contact side bearings, center plate extension pads (C-PEPS) and adjustable pneumatic side bearings (air springs). Modifications of the suspension, intended to improve ride quality and dampen harmonic roll, include augmented snubbing, through higher column loads, by additional dampers inserted in the spring nest, or by hydraulic dampers.

Several approaches have been tried to improve lateral stability. Apart from larger center plates, which increase resistance to truck swivel, such modifications include hydraulic dampers between the carbody and truck bolster, or between the truck bolster and side frames, and structural ties between the side frames to reduce parallelogramming. The rigidity or resistance of these devices required to maintain the truck in tram against the moments applied by creep forces appears in general to have been underestimated.

Several of the modifications listed above have been in service for some time, and a number were tested in TDOP Phase I. Evaluation of these test data will be performed in Phase II.

2.4.2 Alternate Truck Designs

Almost without exception, all Type II trucks are made up of two side frames connected at their centers by a member analogous to the bolster of the conventional Type I truck. What distinguishes alternate truck designs from each other, and from the conventional truck, are the nature of the side frame/bolster connection, the suspension, and in a few cases, the method of supporting the carbody. These differences exert a significant effect on truck performance, particularly with respect to curve negotiation, lateral stability and load equalization.

For purposes of classification, premium or Type II trucks may be broadly divided into "rigid" and "radial" configurations. Differences between individual members of each class are discussed below with reference to the standard or Type I three-piece truck.

2.4.2.1 Rigid Trucks. The connection between the side frame and bolster allows relative displacement in all six degrees of freedom. The bolster moves vertically to change the spring loading under varying vertical loads, and its capability for lateral displacement, limited by the bolster gibs, isolates the carbody to some extent from truck motions caused by lateral rail irregularities, or from self-excited motions. Longitudinal motion of the side frames occurs during braking and thus isolates the center plate to some extent from shock loading.

Since the springs are placed close to the axis of rotation in pitch of the side frame (i.e., a horizontal axis through the bolster), they cannot exert a large moment and thus offer little resistance to the rotation of individual side frames caused by cross level rail irregularities. This response is called equalization or tracking ability and is important in maintaining vertical wheel loads adequate to prevent derailment under simultaneous lateral forces.

However, this advantage is obtained at the cost of impaired characteristics in other respects. The side frames are free to roll (about longitudinal axes) which requires that the bearing adapters be crowned to prevent rocking. Crown wear due to sliding restricts the roll freedom of the side frame.

The least desirable characteristic of the loose bolster-side frame connection is that it allows relative rotation in yaw (about a vertical axis). This reduces the kinematic wavelength of the truck and lowers the critical speed at which body hunting begins. A second effect of yaw rotation is the parallelogramming of the truck in curves, which increases the angle of attack of the leading wheels with respect to the rails and thus promotes wear of both.

A different compromise is represented by the truck with a completely rigid bolster-side frame connection, such as the French Y-25 or the British Gloucester GPS 25. The rigid connection, of course, requires a primary suspension (i.e. springs at the axle bearings) which has the advantage of reducing the unsprung mass. However, while the stiffness in tram brings the advantages mentioned above with regard to hunting and curve negotiation, larger deflections of the primary suspension springs are required for wheel load equalization than are compatible with other important features, such as desired natural frequencies of the carbody and available space. Thus the capability for load equalization of European trucks is considered less adequate for American service than that of the standard truck.

This problem of equalization is solved in the Symington XL-70 truck by a side frame-bolster connection which keeps the truck in tram but allows the side frames to pitch. The primary suspension springs thus need not be designed for the additional travel required for load equalization.

A number of American trucks with secondary suspensions have also been designed with a view towards combining stiffness in tram with the freedom of the side frames to rock (pitch) so as to improve equalization. The Rockwell truck achieves this by a pair of connecting rods, parallel to the bolster, which are mounted in spherical bearings.

The ACF fabricated truck makes use of a transom plate connecting the side frames, which is rigid in bending but flexible in torsion and keeps the side frames in tram while providing freedom in pitch. There is no bolster since the weight of the car body is carried directly on the side frames.

Yet another solution of the tram stiffness-load equalization problem is represented by the National Swing Motion Truck. The side frames are connected by a rigid transom which is mounted on rockers. This arrangement holds the truck in tram, but permits them not only to rock for load equalization, but to roll as well. In combination with the crowned pedestal roofs, this roll freedom permits the transom, and with it the spring nest and bolster, to translate or swing laterally. The transom and side frames thus act as a pendulum, the gravitational stiffness of which is in series with the lateral stiffness of the suspension, and decreases the coupling between the truck and car body. This feature, combined with the stiffness in tram, makes for good lateral stability.

2.4.2.2 Flexible and Radial Trucks. A completely different approach to better truck performance is based on the improvement in curve negotiability by reduction of the lateral flange forces that occur in both rigid and standard trucks. Embodiments of this design approach leave the good equalization characteristics of the standard truck unimpaired but must include provisions for enhancing lateral stability which is not an inherent feature of this type of truck. The following brief outline of curving kinematics may help to explain the main features of the radial trucks.

A single, unrestrained, coned or profiled wheelset can theoretically negotiate a constant curve without relying on creep forces. The wheelset displaces radially outward so that the diameter of the rolling line of the outer wheel is greater than that of the inner wheel. One revolution of the axle thus advances the outer wheel farther along the track than the inner wheel.

When the difference in advance is exactly equal to the difference in length between the outer and inner rails between two radii through the center of the curve, the wheelset performs pure rolling motions, without sliding or creeping.

When the wheelset is prevented from assuming a radial position in a curve, as it would be in a rigid or standard three-piece truck, the direction of rolling of the wheels is not aligned with their direction of motion tangential to the curve. The wheels thus slip radially, and the resulting lateral creep forces, outward at the leading axle and inward at the trailing axle, constitute a moment that resists the rotation of the truck as it rounds the curve. This moment is balanced by longitudinal creep forces, directed forward at the outer wheels, and backward at the inner wheels, which are due to the fact that the truck is displaced radially outward beyond the rolling equilibrium position of the single wheelset described above. Thus, the wheels also slip in the longitudinal direction.

The condition described above, which is called flange-free curving, prevails only as long as sufficient flange clearance remains. As the radius of the curve decreases, contact occurs between the flange of the outer leading wheel and the rail. From that point on, the truck is guided around the curve by the flange force which must overcome the friction forces of the slipping wheels. (This simplified description has omitted the consideration of gravitational stiffness which may be significant in profiled wheels.) The lateral forces can reach considerable magnitude even when the curve is traversed at equilibrium speed, and this condition is thought to contribute significantly to wheel and rail wear. It is aggravated in the loose three-piece truck because the angle of attack increases as the truck is forced out of tram by the friction forces.

The regime of flange-free curving can be extended to curves of smaller radius by providing some longitudinal flexibility at the axial boxes, so as to allow the wheelsets to approach a radial position. This approach has been tried with some success in Germany where the lateral forces on the outer wheels of a so-called flexible truck were about 40% of that at the leading outer wheel of a rigid truck. However, a compromise is required between good curving ability and lateral stability: a truck with too soft a wheelset yaw restraint has a low critical speed.

Another disadvantage of the flexible truck is that the capability for radial alignment of the axles is largely lost when the curve is traversed at other than equilibrium speed.

In order to extend the range of flange-free curving to curves of smaller radius, and to non-equilibrium speeds, a number of truck designs have been proposed, in the last 40 years, in which the axles or side frames are constrained by special linkages, into near-radial alignment in curves. Recent versions of this concept are the British Scales truck, the List truck (built as the Dresser DR-1), and the South African Scheffel or Cross Anchor truck, developed by Standard Car Truck Company.

All of these trucks require suitable elastic longitudinal restraints at the axle boxes, in the form of elastomeric springs, to assure lateral stability. It should be noted that the bolster-side frame connection and the secondary suspension of the American radial trucks are conventional and thus preserve the advantages of equalization, as well as standard components such as snubbers.

Since radial trucks depend on creep forces for flange-free curving they require wheels of relatively high effective conicity. This requirement conflicts with the low effective conicity desirable for high critical speed. Therefore, some wheel profiles used with radial trucks have a low taper at the rolling line, a throat with a large radius, and a drop-off taper. The later two properties produce a large difference in rolling radii when the wheelset is displaced from the rolling line (where both wheels run on equal radii) and produce the longitudinal creep forces that guide the truck around the curve. The effect of wear on the change in wheel contour, which may affect the curving performance, remains to be investigated.

2.4.3 Assessment of Type II Trucks

A number of the Type II or premium trucks described above will be road tested and analyzed in TDOP Phase II to establish quantitative information on their performance in the principal regimes mentioned in Section 1, and more fully discussed in Section 4. Since most of these trucks have had relatively little service, they will be operated in unit trains for one to two years, and their wear characteristics determined by periodic measurement.

All of the premium trucks have an initial cost higher than that of the standard three-piece truck. While this cost may well be justified in terms of savings (through better performance and reduced maintenance), it cannot be assessed without first establishing the relationships between cost and performance of standard trucks. The standard three-piece truck has yet to be quantified in sufficient detail to permit an economic evaluation.

The basic data for such a study were acquired in Phase I of TDOP. Phase II will evaluate these data, supplement them, if necessary, by additional tests, and develop cost and performance baselines against which Type II trucks can be compared.

In order to meet this objective, it is necessary to establish a frame of reference and a set of performance indices that cover all essential engineering and economic aspects of the freight car truck.

SECTION 3 - CHARACTERIZATION OF TRUCK PERFORMANCE

3.1 TERMINOLOGY

One of the problems of dealing effectively with the truck is that it is seen and dealt with in widely differing ways, and from different points of view, by those engaged in its design, analysis, mathematical modeling, construction, operation, maintenance, and financing. Each group tends to emphasize certain features or characteristics of the truck at the expense of others, and arrives at conclusions and recommendations not necessarily in agreement with those of another group that uses a different frame of reference. In order to characterize the performance of the freight car truck, as a necessary prerequisite to quantification, it is necessary to settle on a terminology that is both useful to the operating and maintenance of the truck, and technically discriminating so as to be able to identify problems and point to solutions at the engineering level.

There is some discrepancy between the railroads' and operators' approach to the truck, and that of the theoretician and constructor of mathematical models. A railroad knows what truck-car combinations give rise to operational or maintenance problems; in looking for long-term effects it enjoys the advantage of large sample size and is thus able to identify components, or combinations of components, directly associated with unsatisfactory operation. The drawback of this pragmatic approach is that it tends to address a problem too specifically. A component may be modified when it has caused trouble, but the solution of that problem leads to another problem because the modification has changed the performance of the entire system in ways that were not predicted, but may not have been unpredictable. One often hears this sequence of events described as the "band-aid" approach.

The mathematical modeler faces the opposite problem. In order to deal with a workable system, he must simplify, abstract and generalize. When the model does not agree sufficiently well with reality, he is driven to including greater amounts of detail. Beyond a certain amount of "refining" or "tuning", there is the risk of losing sight of the main objective of the model, which is to solve a particular problem, or to gain insight into an important phenomenon. Thus, while the railroad's and the modeler's truck start from the same object, their characterizations are sometimes too far apart for fruitful communication.

The objective of this section is to bring together the various points of view from which the freight car truck may be approached, and to clarify the relations between them.

3.2 THE TRUCK "UNIVERSE"

Figure 1 shows the frames of reference used to deal with the freight car truck. The diagram is divided into two main domains, performance and deterioration, with cost indicated as a factor in both. The interface between these two domains is formed by the truck components.

3.2.1 Operational Regimes

Performance may be defined either in operational or in functional terms. Operational descriptions include a combination of problem areas, phenomena, regimes, and conditions. Some of these are indicated in Table 1. For example, hunting may be considered a phenomenon, a dynamic regime, and a problem area. Other major performance regimes listed are curve negotiation, harmonic roll, ride quality, derailment, load equalization, braking response, and possibly others. Under each major heading are listed some of the characteristics associated with the particular regime. Such a classification of truck performance features is useful for demonstrating the range and variety of requirements, and there is probably little doubt as to the characteristics of the ideal truck in any of these performance regimes. For example, all railroads would like a high critical speed and low sensitivity to wheel profiles with respect to hunting, and also a minimum flange force during curve negotiation. Unfortunately, the requirements for high performance in several regimes are frequently incompatible.

The weight given to some aspect of performance over another depends to some extent on a railroad's particular operating conditions, as well as on economic factors. The process of trading-off to reach an acceptable level of performance is called optimization in the context of this report. Because of the complexity of the freight car truck and its environment, optimization should not be considered in the same light as the exact discipline of the same name.

3.2.2 Engineering Characteristics

Optimization requires that the truck be considered in functional terms, that is, in terms of the engineering characteristics of its components and their interaction. This approach makes available a large range of techniques and experience developed and

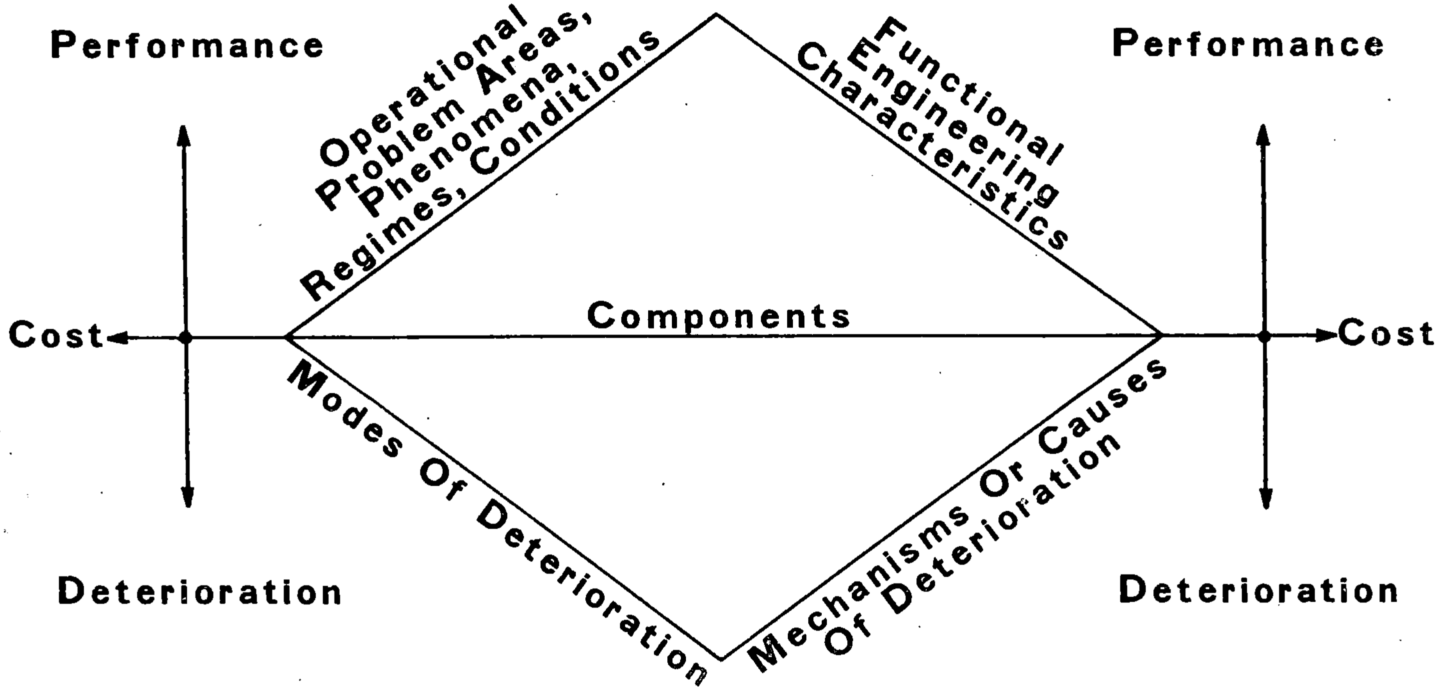

Figure 1. Freightcar Truck "Universe"

Table 1. Truck Operational Characteristics

<u>HUNTING</u>
- Critical Speed
- Lateral Acceleration
- Sensitivity To
  - Wheel & Rail Contour
  - Gauge
  - Carbody Properties

<u>CURVE NEGOTIATION</u>
- Axle Alignment
- Lateral Wheel Loads
- Wheel Climbing
- Wheel Unloading
- Resistances

<u>HARMONIC ROLL</u>
- Critical Speed
- Maximum Amplitude
- Centerplate Liftoff
- Buildup And Decay Rates
- Wheel Lift
- Snubbing

<u>RIDE QUALITY</u>
- Shock
- Vibration
- Isolation/Transmission
- Damage Potential

<u>DERAILMENT</u>
- Wheel Climbing
- Loss of Guidance (Wheel Lift)
- Change in Gauge
  - Pre-Existing
  - Forced

<u>LOAD EQUALIZATION</u>
- Resistances

<u>BRAKING RESPONSE</u>
- Crabbing

validated for other mechanical and dynamic systems. Major engineering characteristics are listed on Table 2. These include the kinematics both of the truck and of the wheel on the rail, stiffnesses and damping characteristics, clearances and other significant nonlinearities, load paths, statics of the braking system, and finally the significant characteristics of the car body, such as its dimensions, including the truck spacing, its mass, its mass distribution, (i.e., moments of inertia and location of center of gravity) and in some cases, its flexibility in bending and torsion.

The analysis of these functional characteristics of the truck and car body is an essential part of the mathematical modeling process to be discussed later in this report. The engineering characteristics shown in this list are, of course, abstractions that present significant features of the actual truck components on the centerline of the truck universe (see Table 3). Some of these components require a detailed breakdown to reflect their importance in various performance regimes, and to make possible the required engineering analysis.

3.2.3 Deterioration

To reflect actual conditions, a realistic engineering analysis cannot be based on the assumption that all of the components remain unchanged over a period of time and perform their functions according to the equations describing them when new. Almost all components of the freight car truck are subject to various forms of deterioration. The deterioration in performance may be high in proportion to the amount of metal removed or damaged. For example, a small change in wheel tread contour may greatly lower the critical speed as may a widening of the clearances between the bearing adapter and its seat in the pedestal.

Deterioration is accepted as a reality inherent in freight car operation and is allowed for in maintenance schedules and repair facilities, as well as in replacement budgets. Frequencies and rates of failure above the expected are cause for concern, especially when they occur in a particular type or class of components.

In spite of steadily improving methods of mathematical and physical analysis, circumstances of accelerated and catastrophic deterioration remain difficult to establish.

Table 2. Truck Functional Characteristics

KINEMATICS
- Truck Frame
- Wheel/Rail

STIFFNESS - TRUCK
- Warp
- Torsion
- Roll
- Nonlinearities

STIFFNESS - SUSPENSION
- Vertical
- Lateral
- Yaw (Swivel)
- Roll

DAMPING
- Vertical
- Lateral
- Yaw
- Nonlinearities

BRAKE MECHANISM - STATICS
- Braking Load
- Eccentricities

CLEARANCES
- Adapters
  - Longitudinal
  - Lateral
- Gibs
  - Lateral
  - Longitudinal
- Side Bearings

LOAD PATHS
- Vertical
- Lateral
- Primary
- Secondary

CAR BODY
- Dimensions
- Truck Spacing
- Mass
- Mass Distribution
- Flexibilities

AXLES

WHEELS

- Tread
- Throat
- Flange
- Plate
- Hub

BEARINGS

ADAPTERS

SIDE FRAMES

- Pedestals
- Columns

RAIL

- Profile
- Gauge
- (Foundation)

Table 3. Truck Components

<u>BOLSTER</u> Beam Center Bowl Gibs
<u>SUSPENSION</u> Springs Friction Snubbers Side Bearings
<u>BRAKE SYSTEM</u> Shoes Brake Beam Linkages
<u>SPECIAL COMPONENTS</u> Links & Ties Hydraulic Shock Absorbers Elastomers

This may in part be due to limitations in the method of reporting and accounting which are limited to few categories. For example, some reporting systems make provisions for only three categories of deterioration: broken, worn or missing. There are good reasons for setting up condemning limits for worn wheels, such as "thin flange" and "high flange"; however, these regulations do not cover subtle differences in wear patterns which may have a profound effect on lateral stability, nor are they able to identify differences in causes, and sometimes not even in the mechanism of wear. For example, the change in contour in the throat of a flange that may appear to be caused by wear may in fact be due to metal flow caused by severe and repeated impact associated with hunting.

The lower half of the freight car truck universe diagram (Figure 1) illustrates the need for such clarification. It distinguishes between modes or evidence of deterioration on the left, and mechanisms or causes of deterioration on the right. Some of the major modes of deterioration are:

- abrasion
- plastic deformation
- gross loss of material
- change in properties
- metal flow
- surface/subsurface cracks
- brittle fracture
- dislocation

The more important mechanisms or causes are shown in Table 4.

The identification of failure modes and mechanisms in more precise technical terms may well facilitate the solution of such problems, again by relating them to the existing technology. That is the main advantage of, for example, characterizing center plate wear as being associated with boundary lubricated sliding friction.

Table 4. Mechanisms or Causes of Truck Deterioration

Overload/Overstress
Fatigue
Impact
Thermal Stress
Sliding Friction - Unlubricated
Sliding Friction - Boundary Lubricated
Contact Stress
- Shear
- Fretting
- Stress Corrosion
Cutting
Manufacturing Flaw
Misfit

3.3 TRUCK "UNIVERSE" APPLIED TO HUNTING

Although deterioration shows up in individual components it must be treated as a system problem, just like performance, for the most effective solution. In fact, the aspects of performance and deterioration are connected and interact. We may illustrate this interaction by applying the diagram of the freight car truck universe to a specific phenomenon, that of hunting, shown as a single entry in the operational category (see Figure 2).

Functionally, this is connected with a large number of engineering characteristics, shown on the right, which include the kinematics of both the suspension and the wheel and rail, the stiffness of the truck and suspension, clearances in all the mechanisms, and the characteristics of the car body. The components of the truck directly involved are the wheel tread and throat as well as some properties of the rail, including profile and gauge, and surface condition.

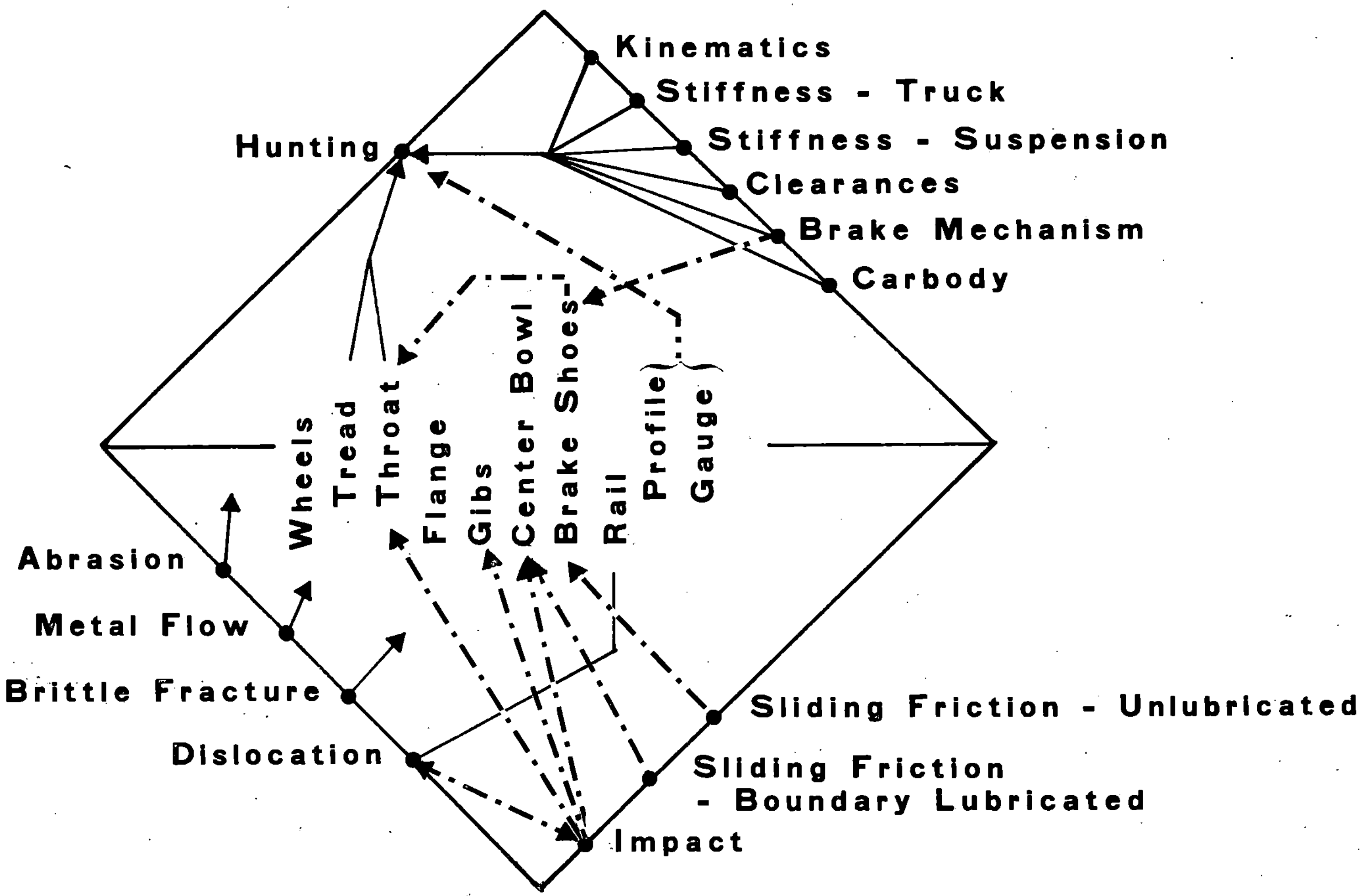

Figure 2. Freightcar Truck "Universe" Applied to Hunting

The deterioration side of the diagram shows some of the possible modes and mechanism of damage that may be involved, such as metal flow and fracture in wheels, dislocation of the rail due to lateral impact, impact at the gibs and the rim of the center bowl, and center plate wear due to boundary lubricated sliding friction. Another possible interaction begins with the configuration of the brake rigging which influences the transverse location of the brake shoes with respect to the wheel and is believed to affect the development of the worn contour. This in turn can lead to large differences in critical speed, an effect that has been verified in several road tests and in the analyses based on test results.

The interaction between performance and deterioration for the case of hunting are shown in Figure 3. Hunting in freight cars occurs when one of the lateral natural frequencies of the car body on its suspension comes close to the kinematic frequency of at least one of the two trucks. Therefore, different cars with the same truck, or the same car with different lading, will not necessarily become unstable at the same critical speed. When two identical cars with the same lading have widely different critical speeds, under the same track conditions, this difference in performance can be shown to be due to differences in the contours of the worn wheel treads which in turn are believed to be produced by the characteristics of the brake rigging, and by differences in the brake shoes themselves.

The contributions of these various factors to the development of stable and unstable wheel profiles are not well known, and consideration is being given to the construction of a test rig that will allow a more positive identification of the importance of the various factors involved in the wear process. Thus, the left side of Figure 3 represents a combination of known facts, such as the dynamics of the spring mass system represented by the car, and the importance of the effective conicity of the wheel profile, with conjectures concerning changes in the system due to localized deterioration, that is, the removal of small amounts of metal from the circumference in the wheel.

By contrast the right side of the diagram portrays the observable damage and the resulting cost of repair or replacement. Such damage is caused by impact between the wheel and rail, which may result in damage to both, as well as to other truck

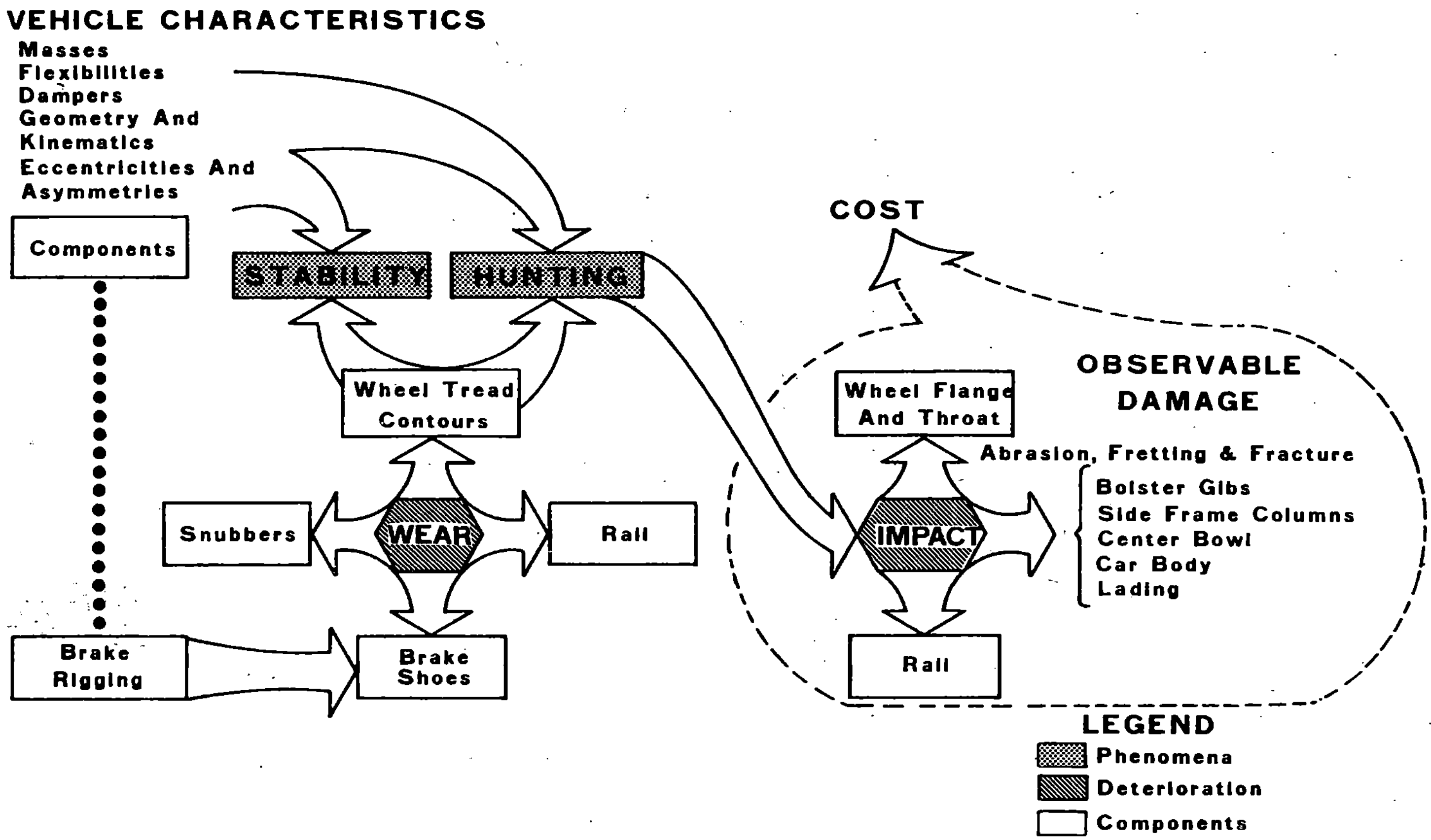

Figure 3. Hunting Environment Simplified

components through which the impact forces are propagated, such as the bolster gibs, the side frame columns and the center bowl. In severe cases, the car structure may be fatigued and the lading damaged. In addition, if there is relative motion between truck components during impacts caused by hunting there will obviously be abrasion and fretting, even fractures may occur.

Similar relationships between performance and deterioration aspects may be established for other major performance regimes of the freight car truck. The consideration of these relationships is an essential basis for the establishment of performance guidelines.

SECTION 4 - QUANTIFICATION OF TRUCK PERFORMANCE

4.1 SELECTION OF PRINCIPAL PERFORMANCE REGIMES

The preceding section has presented the truck environment in a number of reference frames and has pointed out some of the relationships between performance and deterioration. In order to define truck performance in quantitative terms, it is necessary, as a first step, to select a number of operating conditions or regimes that fulfill the criteria mentioned in Section 1. Individually these regimes should be associated with distinctly different operating conditions, and collectively, they should permit overall evaluation of truck performance.

The following regimes have been selected for quantification of truck performance:

- Hunting
- Steady State Curve Negotiation
- Load Equalization
- Ride Quality

They represent distinctly different and complementary aspects of truck performance.

4.1.1 Hunting

Hunting is a self-excited lateral and yaw oscillation of the truck and carbody that occurs above a certain speed (the "critical speed"). The range of the critical speed is determined by a number of factors which include the contours of the wheel tread and rail, the surface condition of the rail, the design features of the truck, the characteristics of the suspension system, and the mass and mass distribution of the carbody. With all of these parameters being equal, the contour of the wheel tread has been found to have an overriding influence on lateral stability.

In dynamic terms, hunting represents an interplay between creep forces at the wheel-rail interface, elastic restoring forces that tend to center and stabilize the truck, inertial forces that act to increase the excursions of hunting, and suspension damping that attenuates them. The effect of track irregularities is believed to be secondary to the self-excited cause of hunting.

In economic terms, the large relative motions of components in a hunting truck cause accelerated wear; impacts between wheel flange and rail cause gauge widening and pose the danger of wheel fracture, and severe lading damage is known to be caused by the lateral vibration of the carbody.

4.1.2 Steady-State Curve Negotiation

In steady-state curve negotiation, horizontal forces between the wheels and rails act to rotate the truck about the center of the curve, even though there is no relative rotation between the truck and carbody. For standard freight car trucks, the lateral force that turns the truck in the curve is usually the flange force at the outer leading wheel, and is likely to contribute to the resistance of the truck to forward motion. It is believed that this flange force is responsible for much of the wear that leads to condemnation of wheels for "thin flange". As mentioned in Section 2, one of the objectives of truck designs has been the improvement of truck kinematics to extend the region of flange-free curving to curves of smaller radius.

4.1.3 Equalization

Equalization ("trackability") refers to the ability of the truck to maintain an adequate load on all four wheels under a range of track conditions, and the dynamics of the vehicle resulting from transient or periodic changes on these conditions. A combination low vertical wheel load with a simultaneous lateral load can lead to derailment. Wheel lift is an extreme case of wheel unloading.

All trucks have some provision for accommodating vertical rail irregularities with a wave length of the order of magnitude of the axle spacing. This adjustment is intended to retain a safe vertical load on the four wheel-rail contact points when they no longer lie in a plane. As discussed in Section 2, the standard truck adjusts itself to such vertical rail irregularities by independent rotating of the side frames, while rigid trucks rely on the displacement allowed by the secondary suspension, or on torsionally flexible side frame connections. For all trucks, there is some limit of track twist beyond which vertical wheel loads may fall below a safe minimum, and the probability of derailment is increased.

Vertical rail irregularities with a wavelength of the order of the truck spacing also lead to wheel unloading by exciting some of the natural modes of vibration of the carbody on its suspension. Resonant vibrations are set up when the frequency of excitation, given by the speed divided by the wavelength, is close to one of the body mode natural frequencies.

The mode with the lowest natural frequency in freight cars is generally lower center roll in which lateral displacement of the center of gravity and roll about that center occur roughly in phase. Alternating vertical forces, most usually due to misaligned rail joints, gives rise to car body oscillations called harmonic roll or rock-and-roll, in which the carbody pivots about centers successively farther offset from the geometric center of the truck: first, the edge of the center plate, and finally the side bearings.

The lateral shift of the center of gravity compresses the springs on the one side until they are solid and in extreme cases the opposite wheels lift off the rail. Derailment is inevitable if any kinetic energy of roll rotation remains when the center of gravity of the body is vertically above the side bearing.

Harmonic roll is a low-speed phenomenon, occurring at speeds below 20 mph. The critical speed is determined by rail joint spacing, mass and mass distribution of the car body (including height of center of gravity) and the characteristics of the suspension system, i.e., the spring rates. The main resistance to harmonic roll comes from friction snubbing. The effect of windage (i.e., the air resistance of the car body) on roll damping appears not to have been investigated.

At higher speeds, and thus higher frequencies, an important dynamic cause of wheel unloading is the phenomenon called bounce, although it should more properly be called pitch. When the carbody oscillates about a transverse horizontal axis, with the ends of the carbody rising and dropping out of phase; a vertical bounce motion may be superimposed. The natural frequency is determined by the mass moment of inertia in pitch of the carbody and by the suspension spring stiffness.

In both the harmonic roll and bounce regimes, derailment is highly likely if the lifting wheelset is subjected either to lateral loads, or if there is a relative lateral displacement between wheel and rail. This may be due to a curved track, in the case of harmonic roll, and to a lateral velocity of the wheelset near the bouncing end of a car.

4.1.4 Ride Quality

Ride quality denotes a standard of performance rather than a performance regime. It is generally taken to refer to the acceleration environment in the carbody and thus reflects the capability of the truck to attenuate both periodic and transient distributions, i.e., vibration and shock arising from track irregularities. This characteristic of the truck to act as a mechanical filter is also termed transmissibility. Unlike ride quality, which requires a definition of disturbances introduced by the track, it represents a characteristic of the truck alone and is thus more suitable as a performance criterion. This point illustrates the case required in selecting suitable performance criteria for each regime, a subject that is discussed in greater detail below.

4.2 ESTABLISHMENT OF PERFORMANCE INDICES

4.2.1 Definitions

Section 1.3 has defined the relationships between performance regimes, criteria and indices. To repeat briefly, criteria denote features characteristic of a specific regime; they may be expressed in positive or negative terms (for example, "high lateral stability" or "freedom from hunting"). A number of more or less general performance criteria may be identified for each regime, but not all are equally suitable to characterize truck behavior in a quantitative way. Such characterization requires the selection of a measurable physical quantity that can be unambiguously associated with performance in a specific regime. The measure of performance serves as the performance index.

It may be evident that the conditions under which a truck is evaluated in each regime must be carefully specified. For example, track characteristics are of the greatest importance in the harmonic roll regime, and in specifying ride quality, while it is relatively unimportant with regard to hunting, except perhaps for gauge. A second factor that must be specified is the state of wear of various truck components. The overriding effect of the worn wheel contour on the critical speed of hunting has been well established. In addition, worn friction shoes reduce the level of energy dissipation during harmonic roll, besides permitting greater truck warp in curves and thus increasing flange forces.

The process of selecting suitable performance criteria for establishing performance indices is discussed below for each of the five major performance regimes.

4.2.2 Hunting

The importance of picking suitable criteria in establishing a performance index is illustrated in Table 5, with respect to hunting.

Table 5. Hunting

CRITERIA	INDICES
Safe Operation in Desired Speed Range	
High Lateral Stability	
High Critical Speed	Critical Speed
Low Sensitivity to Unfavorably Worn Wheel Profiles	
Low Lateral Accelerations Near Critical Speed	Magnitude of Lateral Accelerations

The first two criteria listed on the left are obviously important from the operational point of view but do not lend themselves easily to quantitative expression. The third criterion, high critical speed, can obviously be directly translated into a number which has long served as a performance index for lateral stability. The next criterion, low sensitivity to unfavorably worn wheel profiles, represents a desirable feature in a rail vehicle, but again would be difficult to express quantitatively. The final criterion, low lateral accelerations near the critical speed, may be the basis for a useful second performance index. It allows for differences in the severity of hunting which has been observed in different truck configurations.

4.2.3 Steady State Curve Negotiation

The second performance regime, curve negotiation, shown in Table 6, again illustrates the care required in passing from an operationally defined performance criterion to a performance index based on engineering factors.

Table 6. Curve Negotiation

CRITERIA	INDICES
Low Wheel & Rail Wear	
Safety From Derailment	
Low Lateral Wheel Load	Lateral Force on Leading Outer Wheel per 1000 lb Axle Load, per Degree of Curve, Balance Speed

Operational considerations would lead to such criteria as safety from derailment and low wheel and rail wear. These also are difficult to quantify. In order to isolate basic differences in curve negotiation performance between different truck designs, we must eliminate a number of extraneous factors such as the effect of unbalanced centrifugal force not compensated by superelevation, and transient effects that occur during curve entry, and focus on the basic kinematic characteristics of a given truck that determine its orientation in a curve of constant radius, under the influence of creep and gravitational forces alone.

This suggests the lateral force on the outer leading wheel of the truck during steady state curving as a likely candidate for a performance index. As is well known, and has been confirmed by both road tests and mathematical modeling, this force is strongly determined by the ability of the axles to align themselves with the radius of the curve. By contrast, any property of the truck tending to increase the angle of attack, such as the parallelogramming of the standard three-piece truck, increases the lateral wheel load directed toward the inside of the curve. Thus, a suggested performance index for curve negotiation could be the lateral force on the leading outer wheel, per thousand pounds of axle load, per degree of curve, at balance speed.

The imperfections of this index are associated with differences in wheel profile which would thus have to be specified in detail in the comparison between the curve negotiation capability of different trucks, as well as in the creep coefficients that determine the angle of attack of the leading outer wheel in flange-free curving. Nevertheless, the most important characteristics of the truck in this regime are determined by its kinematics, as embodied in the relationship of the rigid components and the properties of the elastic connections between them.

4.2.4 <u>Equalization</u>

As pointed out in Section 4.1.3, the conditions that affect reduction in vertical wheel load may be classified according to the wave length of the vertical rail irregularity that leads to such reduction.

4.2.4.1 <u>Track Twist</u>. The performance index for equalization with respect to track irregularities of short wave length, such as rail joints and track twist, is the easiest to define, at least in a static sense:

Let W_H = sum of forces on the three most heavily loaded wheels

Let W_L = force on most lightly loaded wheel

= angle of twist of track within axle spacing of truck, degrees

WUI = wheel unloading index

then

$$WUI = \left[\frac{W_H/3 - W_L}{W_H/3}\right] \div \theta = \left[1 - \frac{W_L}{W_H/3}\right] \div \theta \text{ , degree}^{-1}$$

It may be seen that this index may vary from zero for a perfectly equalized truck (since $W_H/3 = W_L$) to $1/\theta$ for a truck with one wheel completely unloaded, or unity for unit twist in degrees. This simple, provisional performance index for equalization must be qualified as follows:

a. Because of frictional resistance of the snubber, a truck on perfectly level track may not have all wheels equally loaded.

b. A series of wheel unloading indices, expressed as a curve, will have to be used with trucks having nonlinear elements, such as multi-rate springs or limits to the relative displacement of components involved in equalization.

c. The index does not take into account dynamic conditions of rapidly varying track twist, such as might occur during spiral entry or exit. The inertia of a wheel may transiently relieve the vertical force on a rapidly dropping inner rail, or the wheel on the rapidly rising outer rail may bounce and thus relieve the vertical load. However, the wheel unloading index could be cast in dynamic terms by introducing the rate of twist into the denominator, either directly or as a function of rolling velocity and axle spacing.

It may be noted that the suggested performance index for wheel unloading due to truck twist is not sufficient to measure derailment potential since it is not referred to such indices as L/V ratio, or the duration of lateral wheel impact on the rail. Derailment associated with wheel unloading is a complex dynamic process which it may be impossible to define in terms of a truck performance specification. On the other hand, the proposed wheel unloading index describes a characteristic of the truck itself that could easily be specified, and which it may be possible eventually to relate to the derailment potential on the basis of test data and analyses.

4.2.4.2 Harmonic Roll. The performance index for harmonic roll, associated with vertical roll irregularities of long wave length, illustrates the difficulty of quantifying a complex dynamic regime. From an operational point of view, one obviously wants to prevent hazardous conditions, and damage to truck components, such as would occur during center plate lift-off and wheel lift. In the first case the suspension springs are compressed solidly and are thus subject to fretting damage, and during wheel lift there is always the danger of derailment when the track has even a slight curvature.

Table 7. Harmonic Roll

CRITERIA	INDICES
Prevention of Center Plate Lift-off	
Prevention of Wheel Lift	
Low Maximum Roll Amplitude Under Given Excitation	Maximum Roll Amplitude Under Given Excitation Amplitude and Frequency
Rapid Decay From Maximum Amplitude	Rate of Energy Dissipation in Snubbing System: lb-ft/sec or per cycle
Amplitude	

When the springs are solidly compressed during harmonic roll, the motion of the car body is determined almost entirely by the kinetic energy of roll rotation existing in the car body. With a given excitation due to resonance with rail joint spacing, that kinetic energy is largely determined by the energy that has been dissipated in the snubbing system prior to that point. This circumstance suggests that a performance index for harmonic roll should in some way be associated with the capability of the suspension system to dissipate energy, for example, by snubbing friction. Parameters available for variation are level of friction force, distance through which this force acts (i.e., spring travel) and rate of increase of friction level with spring compression. The effectiveness of this energy dissipation system could be measured in a number of ways, such as maximum amplitude after initial excursion. The mass and mass distribution, that is the car weight and the height of the center of gravity, will have to be specified in the establishment of this performance index.

4.2.5 Ride Quality

The aspect of performance shown in Table 8, Ride Quality, does not fit neatly into the proposed scheme. Ride quality should be "good," according to some scale. As usually expressed, ride quality is already a performance index, and the choice here is not between different criteria to be expressed as performance indices, but rather between various ways of quantifying that performance index, some of which are indicated in the table.

Table 8. Ride Quality

Function of Speed, Track Quality and Vehicle Suspension
Referred to a Specific Location on Carbody
Identified as Acceleration Response
Expressed Statistically
Mean and Standard Deviation
Exceedances or Probability Distribution
Expressed as a Function of Frequency
Transmissibility
Power Spectral Density

It is evident that many qualifications are necessary to ensure fair comparison between different truck designs, such as the speed, the quality of a track, and the characteristics of vehicle suspension as well as its mass and mass distribution. Furthermore, in a usable performance index, a standard for track quality must be agreed upon, and the magnitude of accelerations (expressed either statistically or as a function of frequency) must be referenced to the same location on the car body, for example, a point directly above the center plate. These restrictions will be taken into account both in the instrumentation of vehicles during Phase II testing, and in the evaluation of data from these tests.

SECTION 5 - CORRELATION OF ENGINEERING AND ECONOMIC FACTORS

5.1 INTRODUCTION

The preceding sections have dealt almost exclusively with the engineering and operational aspects of the freight car truck. In order to fulfill one of the major objectives of the program, the engineering aspects must be correlated with the economics of truck operation and maintenance.

The difficulties of achieving such correlation should not be underestimated. They arise in part from the fact that mechanisms of deterioration are not easily observed in process, but must be inferred from an inspection of the observed damage. Such interpretations are not always unambiguous. A second important reason is the incompleteness of the available data, in the areas of both performance and economics relating to the truck. It is, of course, one of the objectives of TDOP to identify missing data, and to analyze and utilize data that exist in various files and records. A method for locating and collecting the required data is outlined in Section 5.3.

5.2 SUMMARY OF PHASE II PROGRESSION

The progress towards the goal of correlating performance and economics is shown schematically in Figure 4. The "Truck Universe" which was discussed in Section 3 is shown at the top of the figure. The universe is divided into two main sections, performance and deterioration, with cost as an element common to both. The left side of Figure 4 deals with the engineering aspects of performance and deterioration, while the right side comprises the economic aspects related to deterioration and cost. The first band illustrates procedures for utilizing various data sources, which, on the engineering side, consist mainly of Phase I performance data supplemented by mathematical modeling studies. On the economic side, data sources consist of various types of files and records maintained by railroads.

At the next level, provisional performance baselines are established by major regimes in the engineering area, and for attrition and cost with respect to economics. In the latter case, we have tried to show that the attrition and cost baselines may be raised by specific operating conditions that require identification.

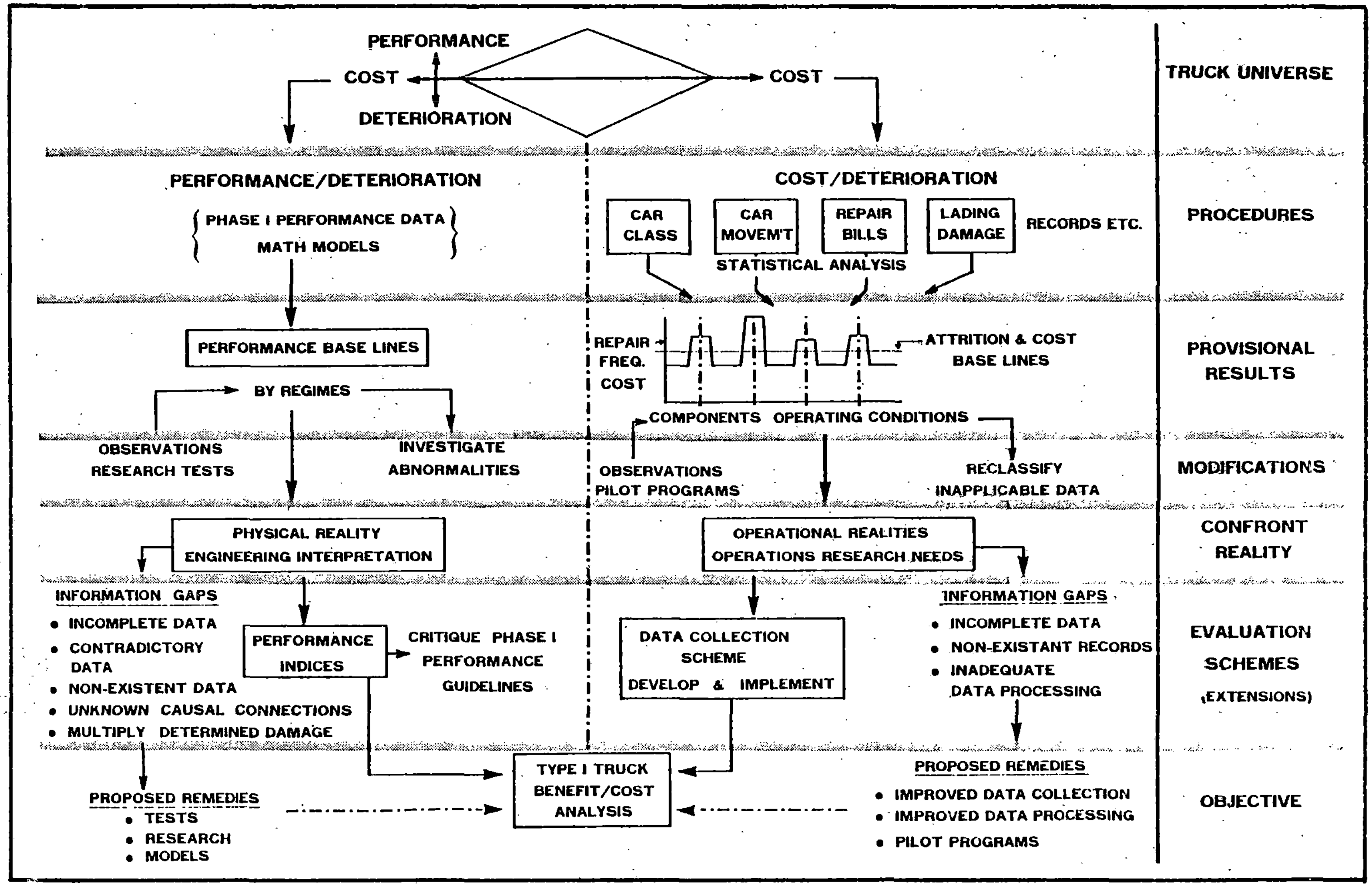

Figure 4. TDOP Phase II Progression

This simplified approach requires modifications necessary to accommodate conclusions of other ongoing work, such as research, testing, pilot programs, and observations by knowledgeable individuals. One must expect to find data that cannot be fitted into simplified schemes set up to organize a mass of data into some intelligible form.

As shown in the next band, it is necessary to confront reality and explain or remove discrepancies with the analytical schemes. In the engineering area, information gaps will inevitably show up during attempts at interpreting data. The data may be incomplete, contradictory or non-existent; causes of observed events may be unknown, or more than one cause may be assigned to an observed event or incident of damage. In order to fill such information gaps, it may be necessary to perform additional tests and research, and to develop special models to explain conditions not reproducible by existing models. Examples of such information gaps are the proposed wear test program, the measurement of snubbing friction under operating conditions, and the proposed measurement of accelerated wheel and brake wear on a truck rather than a dynamometer. Similar information gaps are likely to be found in the economic area, in the form of incomplete data, non-existent records, and inadequate data processing. Possible remedies are shown in the figure.

At this stage, the principal results of our study are the establishment of performance indices on the engineering side, accompanied by a critique of the performance guidelines developed under Phase I, and the development and implementation of a truck data collection scheme.

The two branches of the study, possibly supplemented by additional activities intended to fill information gaps, will come together in the final objective, the benefit/cost analysis of the Type I truck.

The main purpose of Figure 4 is to show the various activities being carried on and planned for the project, and to illustrate the connections between them.

The interaction between the engineering and economic studies may occur in a variety of ways. For example, an observation of accelerated wear or other deterioration may suggest the presence of a particular mechanism or details of a dynamic regime that could be confirmed by testing or mathematical modeling. On the other hand, a review of test data, combined with model analysis, can lead to prediction of types or rates of deterioration of particular components. An example of the latter approach is presented in the Appendix.

It may be noted that the procedures outlined in the preceding are followed in general outline by railroads and suppliers in their efforts to improve equipment or evaluate new design features or concepts. The responsibility of TDOP Phase II is somewhat wider: as mentioned in Section 1, it will attempt to provide a quantitative basis for performance specifications for both standard and premium trucks. These specifications must encompass the complete range of operating conditions and must be applicable to a wide variety of truck configurations. Since there are some functional similarities between competing truck designs, a formal approach to the definition of performance is considered mandatory.

5.3 ECONOMICS METHODOLOGY FOR EVALUATING IMPROVED TRUCK PERFORMANCE

It is recommended that an incremental cost/benefit analysis be conducted to evaluate proposed investments in improved trucks. Railroads follow such procedures in arriving at their profit-making decisions with regard to rolling stock investment. However, the data required to establish the operating cost of existing conventional trucks and for comparing their cost effectiveness with improved trucks, have never been available in usable form. It has now been established that the basic data can be obtained from the Union Pacific Railroad (UP) as a result of their data collection efforts over the last 18 months.

There are two broad categories of economic data required:

a. The cost data required are the capital investments or purchase prices of the conventional and improved trucks, adjusted for any credits and debits such as investment tax credits or additional working capital requirements. The difference is the net incremental cost.

b. The benefits data required are derived from the actual operating cost of the conventional truck and the estimated operating cost of the improved truck.

Their difference is the estimated gross incremental benefit which is adjusted for the income tax shield arising from depreciation allowances and discounted to present value using a railroad's cost rate to acquire the investment funds (i.e., their cost of capital). That calculation produces the estimated net incremental benefit.

To measure the cost effectiveness of the improved truck, the estimated net incremental benefit is divided by the actual net incremental cost over a range of expectations. Values greater than one provide benefits greater than cost, a profitable investment; less than one, an unprofitable investment; equal to one, an "indifferent" investment. (The timing of the costs and benefits will be based on an assumed implementation schedule for any improvements.)

The proposed data collection plan will discuss the data required to establish operating costs for the standard three-piece truck, and the data's availability at UP. These data are to be used in special studies in the following areas: truck maintenance, truck life cycle, and track life. Some areas can be more effectively studied through engineering work at this time (e.g., fuel consumption). In several areas it is difficult to separate the truck-caused costs from other causes (e.g., derailment and lading damage). In these areas, we are requesting typical data for further study. In any event, the results of any of our special studies will ultimately be used in estimating operating costs for the standard truck and to evaluate the effectiveness of an improved truck.

5.4 DATA AVAILABLE FOR COLLECTION

It has been determined that the actual data collected in TDOP Phase I were too limited for determining the cost effectiveness of trucks. The data are in non-machine readable hardcopy form and consist of two-year old information on off-line component repairs for new box and flat cars. The data were collected in Phase I for the purpose of demonstrating the feasibility of implementing a cost acquisition system for trucks which did not exist at the time. The Phase I data are too limited because they cover only one category of costs, do not include mileage and other operating conditions, are too restrictive with respect to car age, and are not representative of the national experience of different railroads in various service.

Through Wyle's subcontract with the Union Pacific, it has been established that the following data is available. Light maintenance, car movements, track data (including posted speed) and accidents were found available in machine readable form. Heavy repairs, lading damage and track delays are in semi-automatic form; UP plans to automate it within the TDOP contract period. Fuel consumption and actual speed are not being collected for the fleet and may require special data collection handling. Details of the planned economic data collection effort will be described in a future report.

SECTION 6 - FUTURE PLANS

The information required for an understanding of the dynamics and deterioration of the standard three-piece truck will be derived from a number of sources. Broadly speaking, the engineering information will come from test data, and their interpretation. The collection of economic data, mentioned briefly in Section 5, will be discussed in a separate report. This concluding section outlines the procedures to be followed to acquire and interpret quantitative information on the various performance aspects of the truck that were discussed qualitatively in the earlier sections of the report.

6.1 PHASE I TEST DATA

The road tests conducted during Phase I involved five different car bodies in three conditions of lading, five different trucks supplied by two manufacturers, five different wheel profiles, and five different kinds of track. Testing all possible combinations of these variables would have constituted a much larger effort than was contemplated for that program, while omitting any one variable completely might have failed to provide important clues. With some exceptions to be noted later, the instrumentation of the trucks and car bodies appears to have been adequate.

The combinations of variables tested in Phase I is shown in the matrices of Figures 5 through 9, one for each type of track. Each matrix contains the combination of car body and state of lading with the truck and wheel profile. Actual combinations tested are indicated by a solid circle. A cross entry in the matrix shows an inapplicable combination (such as a 100-ton car body on a 70-ton truck). A blank space indicates a combination not tested.

The analysis given in the Appendix of the dynamics of the 100-ton boxcar shows that a considerable amount of information on the dynamics and probable wear of the vehicle may be extracted from the test data. Similar, more extensive analyses will be performed on data relating to other test conditions.

Comparison of the performance of different trucks under the same car body will require considerably more analysis since many combinations of trucks and car bodies

CAR BODY	TRUCK	ASF 70 TON RIDE CONTROL					ASF 100 TON RIDE CONTROL					BARBER 70 TON					BARBER 100 TON					ASF 70 TON LOW LEVEL				
	LOAD COND.	WHEELS *																								
		A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E
REFRIGERATOR	EMPTY	●	●	●		●	X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED	●	●	●		●	X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
70 TON BOXCAR	EMPTY						X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED						X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
100 TON BOXCAR	EMPTY	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
89 FT FLATCAR	EMPTY	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	●				
	HALF FULL	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X					
	LOADED	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	●				
100 TON HOPPERCAR	EMPTY	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X

* A - NEW 1/20 B - NEW 1/40 C - CYLINDRICAL D - HALF WORN E - WORN

● TEST DATA AVAILABLE

☐ NO TEST CONDUCTED

☒ NON-APPLICABLE TEST CONDITION

Figure 5. Phase I Test Matrix: Curved Track

CAR BODY	TRUCK	ASF 70 TON RIDE CONTROL					ASF 100 TON RIDE CONTROL					BARBER 70 TON					BARBER 100 TON					ASF 70 TON LOW LEVEL				
	LOAD COND.	WHEELS *																								
		A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E
REFRIGERATOR	EMPTY	●	●	●	●	●	X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED	●	●	●		●	X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
70 TON BOXCAR	EMPTY						X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED						X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
100 TON BOXCAR	EMPTY	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
89 FT FLATCAR	EMPTY	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	●				●
	HALF FULL	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X					
	LOADED	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	●				
100 TON HOPPERCAR	EMPTY	X	X	X	X	X	●					X	X	X	X	X						X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X	●					X	X	X	X	X						X	X	X	X	X

* A - NEW 1/20 B - NEW 1/40 C - CYLINDRICAL D - HALF WORN E - WORN

● TEST DATA AVAILABLE
☐ NO TEST CONDUCTED
☒ NON-APPLICABLE TEST CONDITION

Figure 6. Phase I Test Matrix: High Speed Jointed Track

CAR BODY	TRUCK	ASF 70 TON RIDE CONTROL					ASF 100 TON RIDE CONTROL					BARBER 70 TON					BARBER 100 TON					ASF 70 TON LOW LEVEL				
	LOAD COND.	WHEELS *																								
		A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E
REFRIGERATOR	EMPTY	●	●	●	●	●	X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL	●					X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED	●	●	●		●	X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
70 TON BOXCAR	EMPTY						X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED						X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
100 TON BOXCAR	EMPTY	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X						X	X	X	X	X	●					X	X	X	X	X
89 FT FLATCAR	EMPTY	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	●				●
	HALF FULL	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X					
	LOADED	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	●				
100 TON HOPPERCAR	EMPTY	X	X	X	X	X	●					X	X	X	X	X						X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X	●					X	X	X	X	X						X	X	X	X	X

* A - NEW 1/20 B - NEW 1/40 C - CYLINDRICAL D - HALF WORN E - WORN

● TEST DATA AVAILABLE
(blank) NO TEST CONDUCTED
X NON-APPLICABLE TEST CONDITION

Figure 7. Phase I Test Matrix: High Speed CWR Track

CAR BODY	TRUCK	ASF 70 TON RIDE CONTROL					ASF 100 TON RIDE CONTROL					BARBER 70 TON					BARBER 100 TON					ASF 70 TON LOW LEVEL				
	LOAD COND.	WHEELS* A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E
REFRIGERATOR	EMPTY	●	●	●	●	●	☒	☒	☒	☒	☒	●					☒	☒	☒	☒	☒	☒	☒	☒	☒	☒
	HALF FULL	●					☒	☒	☒	☒	☒						☒	☒	☒	☒	☒	☒	☒	☒	☒	☒
	LOADED	●	●	●		●	☒	☒	☒	☒	☒	●					☒	☒	☒	☒	☒	☒	☒	☒	☒	☒
70 TON BOXCAR	EMPTY						☒	☒	☒	☒	☒	●					☒	☒	☒	☒	☒	☒	☒	☒	☒	☒
	HALF FULL						☒	☒	☒	☒	☒						☒	☒	☒	☒	☒	☒	☒	☒	☒	☒
	LOADED						☒	☒	☒	☒	☒	●					☒	☒	☒	☒	☒	☒	☒	☒	☒	☒
100 TON BOXCAR	EMPTY	☒	☒	☒	☒	☒						☒	☒	☒	☒	☒	●		●			☒	☒	☒	☒	☒
	HALF FULL	☒	☒	☒	☒	☒						☒	☒	☒	☒	☒						☒	☒	☒	☒	☒
	LOADED	☒	☒	☒	☒	☒						☒	☒	☒	☒	☒	●		●			☒	☒	☒	☒	☒
89 FT FLATCAR	EMPTY	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	●				
	HALF FULL	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒					
	LOADED	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	●				
100 TON HOPPERCAR	EMPTY	☒	☒	☒	☒	☒	●					☒	☒	☒	☒	☒						☒	☒	☒	☒	☒
	HALF FULL	☒	☒	☒	☒	☒						☒	☒	☒	☒	☒						☒	☒	☒	☒	☒
	LOADED	☒	☒	☒	☒	☒	●					☒	☒	☒	☒	☒						☒	☒	☒	☒	☒

* A - NEW 1/20 B - NEW 1/40 C - CYLINDRICAL D - HALF WORN E - WORN

● TEST DATA AVAILABLE
☐ NO TEST CONDUCTED
☒ NON-APPLICABLE TEST CONDITION

Figure 8. Phase I Test Matrix: Medium Speed Jointed Track

CAR BODY	TRUCK	ASF 70 TON RIDE CONTROL					ASF 100 TON RIDE CONTROL					BARBER 70 TON					BARBER 100 TON					ASF 70 TON LOW LEVEL				
	LOAD COND.	WHEELS *																								
		A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E	A	B	C	D	E
REFRIGERATOR	EMPTY		●	●			X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED		●	●			X	X	X	X	X	●					X	X	X	X	X	X	X	X	X	X
70 TON BOXCAR	EMPTY						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	HALF FULL						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
	LOADED						X	X	X	X	X						X	X	X	X	X	X	X	X	X	X
100 TON BOXCAR	EMPTY	X	X	X	X	X						X	X	X	X	X			●			X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X						X	X	X	X	X			●			X	X	X	X	X
89 FT FLATCAR	EMPTY	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X					
	HALF FULL	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X					
	LOADED	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X					
100 TON HOPPERCAR	EMPTY	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	HALF FULL	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X
	LOADED	X	X	X	X	X						X	X	X	X	X						X	X	X	X	X

* A - NEW 1/20 B - NEW 1/40 C - CYLINDRICAL D - HALF WORN E - WORN

● TEST DATA AVAILABLE
☐ NO TEST CONDUCTED
☒ NON-APPLICABLE TEST CONDITION

Figure 9. Phase I Test Matrix: Shimmed Track

were not tested in Phase I. Such comparison will rely heavily on validated mathematical models. In particular, it would have been desirable to test the same car body on low-speed jointed track, on trucks with both load-dependent and load-independent friction snubbing.

During Phase I, few tests were run on worn wheels on high-speed tangent track, so that little information on hunting behavior is contained in the test data. However, this scarcity of data on lateral stability is not considered detrimental since a number of very good hunting tests have been performed in the last few years, which have investigated the effect of a number of parameters on the critical speed. These tests include those conducted by Southern Pacific, and by American Steel Foundries and the Burlington Northern Railroad on the tracks of the Missouri Pacific.

Thus, adequate data appear to be available to characterize the Type I truck with respect to ride quality and hunting. Performance data relating to the remaining regimes are less complete and will require additional testing and analyses, to be discussed below. Also, tests will be required to measure snubbing friction with instrumentation developed during Phase I, but not completed in time to be incorporated in the test plan.

6.2 FRICTION SNUBBER FORCE MEASUREMENT SYSTEM

During the planning of instrumentation early in Phase I, it was realized that the measurement of snubbing friction would be essential to the complete understanding of truck behavior. Instrumentation of the test trucks included measurement of accelerations and relative displacement of truck components, relative displacement between car body and truck, and car body accelerations.

For a time, consideration was given to extracting the value of snubbing forces from displacement and acceleration data by means of computer analysis. This approach was judged to be complex and unlikely to produce satisfactory results. In November 1974, Wyle Laboratories proposed the development of a transducer for measuring both vertical and lateral snubber friction forces, as well as normal column loads. This project was funded through Southern Pacific under TDOP Phase I and completed under the auspices of the Federal Railroad Administration after the termination of Phase I.

The transducers were designed, built, installed and tested in an ASF Ride Control and a Barber S-2 truck, both of 70-ton capacity. An account of the device will be given in an FRA report to be published in the near future.

A road test of the Friction Snubber Measurement System is planned for the Fall of 1978. The two trucks will be provided with transducers to measure relative displacements between side frames and bolster, as well as between car body and truck bolsters. This should make it possible to determine the predicted increase in column load, and therefore friction force, during relative rotation between side frame and bolster, and also the amount of energy dissipation accompanying car body roll. It should also be possible to determine center plate friction by measurement of differential column loads which indicate the moment applied by the side frames to the truck bolster during curve entry and exit.

6.3 CURVE NEGOTIATION

The one performance regime for which Phase I data are inadequate is curve negotiation. This is because lateral wheel loads, which guide the truck around the curve, were not measured. The lateral forces between the bearing adapters and pedestals, which were measured by strain-gauged pins, represent axle loads produced by such external conditions as unbalanced superelevation, forces due to angled couplers, and transients due to axle or side frame accelerations. They do not measure lateral wheel loads arising from tread creep and flange friction forces which are characteristic of curve negotiation, and constitute major causes of wheel and rail wear.

It is planned to repeat the curving tests of Phase I with improved instrumentation designed to measure true lateral wheel loads. Two schemes are under consideration: Measurement of axle bending moments, according to the method developed by Peterson et al. and further perfected by American Steel Foundries; and the instrumented wheel plate method worked out by Illinois Institute of Technology. Wyle is presently working on modifications of the axle bending technique with a view towards increasing its accuracy.

An additional desired improvement relates to the measurement of vertical wheel loads which are required to make possible separation of axle bending moments due to both vertical and lateral wheel loads. Existing vertical wheel load transducers in the form of instrumented bearing adapters are not considered satisfactory.

The alternative method of strain-gauging the side frame has been perfected by American Steel Foundries, but requires a separate scheme for almost every truck design and is not suitable for field calibration.

Furthermore, it appears desirable to measure not only the total vertical wheel load, but the lateral position of its line of action which indicates the degree to which the two roller bearing races share the load unequally. Such unsymmetrical loading, due to side frame roll, may accelerate bearing wear. Finally, longitudinal forces are transmitted between the pedestal and the bearing adapter, particularly during curve negotiation and braking. Detailed knowledge of the load distribution on the adapter would not only be of value in understanding wear patterns at the pedestal and bearing, but provide additional insight into truck dynamics which is known to be affected by longitudinal clearances in this area.

A transducer is being designed to measure forces on four segments of the interface between the roller bearing and the adapter. This will make possible identification of unsymmetrical vertical bearing loads as well as longitudinal force components. In combination with the improved method of measuring lateral wheel loads, such a transducer should make possible a more complete picture of interaction between rail, wheel and truck than has been achievable before.

6.4 EQUALIZATION ("TRACKABILITY")

As already mentioned, the use of a suitable transducer will make it possible to measure quasi-static vertical wheel loads from which the Wheel Unloading Index can be calculated when data on track twist are available. Further, the availability of means of measuring lateral wheel loads can supply data on dynamic unloading of wheels as well as L/V ratios from which derailment potentials can be calculated.

A common type of derailment, due to harmonic roll,* occurs on curved track. When one wheel lifts off, the guiding capability of the axle is lost, so that it continues in a path tangent to the curve. When the wheel drops back on the rail, the relative position of the wheel tread and rail head has been shifted laterally by the curvature of the rail, so that the wheel contacts the rail on the flange or may miss the rail altogether. This condition depends on the angle of roll and the curvature of the track.

Two approaches are contemplated to study harmonic roll. The first considers energy dissipation by the snubbers; this study will be greatly aided by the results from tests of the Friction Snubber Force Measurement System described in Section 6.2. Other contributions will come from mathematical models that have succeeded in closely reproducing the roll dynamics of a vehicle on shimmed track and are judged to be usable, with proper care, to extrapolate to amplitudes greater than those permitted in the tests. It will be possible to reproduce roll conditions approaching derailment with the car body restrained against complete overturning in the Rail Dynamics Laboratory at the Transportation Test Center (TTC). Additional test results are expected from the powered roll table built and operated by Southern Railways.

The second approach considers the nature of the track irregularities in greater detail than has been customary. The shimmed track used in harmonic roll tests represents a useful idealization for the study of vehicle dynamics. However, vertical irregularities in real track are not uniform, and the sequence of irregularities of different magnitudes may have an important effect on the amplitude of roll. For example, it is conceivable that an ensemble of rail offsets of given height could produce a greater amplitude of roll when they occur in ascending rather than descending order of magnitude. Nor do present methods of testing and analysis take into account

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*Less severe forms of harmonic roll may occur at higher speeds when the upper center roll mode is excited, or at a low speed on tangent track when the kinematic frequency of the truck coincides with the car body's natural frequency of lower center roll. The latter is thus, strictly speaking, a case of hunting. Both variants of harmonic roll were encountered during Phase I testing.

permanent depressions in the rail beyond the joint, caused by the impact of the wheel returning after lift-off.

In order to provide a more realistic basis for studies of harmonic roll, Phase II of TDOP will utilize the results of a national track survey. These data will make it possible, via mathematical modeling, to determine the effect of random sequences of vertical rail irregularities on the angle of roll. Since for a given segment of track, the range of irregularities can be specified, but the sequence may vary from one subsegment to the next, the derailment potential due to harmonic roll of a given railcar will be associated with a probability distribution that reflects the sequential distribution of vertical irregularities along the segment of track under consideration.

6.5 WEAR DATA COLLECTION

The Wear Data Collection Program will collect wear data on several Type I and Type II freight car trucks beginning in November 1978. The objectives of the program are to collect sample wear data, establish wear trends, evaluate wear measurement methods, develop a schedule for measurement occurrence, and provide data for economic models.

A limited Wear Data Collection Program using unit coal trains operating in actual revenue service over several railroad systems will be initiated. The program's measurement techniques have been developed through evaluation of existing railroad and industry measurement techniques as well as those being used on the Facility for Accelerated Service Testing (FAST) program at the TTC. Initial verification sampling at close mileage intervals will be made to ensure that baseline data on early wear are preserved. The wear data will be entered into the TDOP data base and will be compared by engineering and economic analysts with wear data from other data sources (e.g. industry, AAR, and the FAST program).

Wear data will be collected on six types of freight car trucks. Two trucks of each type will be installed on 100-ton hopper cars and will run in two unit coal trains, the Carbondale/Sunnyside Unit West (CSUW) and the Kaiser Unit West (KUW) operating on a 1600-mile round trip between Utah and California. Three test cars will be run in the CSUW train and three test cars in the KUW train. Except for these six 100-ton test cars, the rest of the unit train consists of 125-ton hopper cars. However, these 125-ton cars are normally filled to only the 100-ton level.

The unit trains will travel on Union Pacific, Santa Fe, and Denver and Rio Grande tracks. The unit coal trains will allow Wyle to collect data on both Type I and Type II trucks from service that is typical for the railroads.

Both of the trains pass through Las Vegas, Nevada and are easily accessible to Union Pacific's Repair-in-Place (RIP) track. Most of the service will be on Union Pacific's main line class 4 track, with curves of at least 10 degrees. The 1600-mile round trip is completed in four days with the trains leaving at staggered, two-day intervals. The maximum speed of the trains will be about 50 miles per hour with an expected accumulation of 140,000 miles per year.

The six trucks to be tested are the American Steel Foundries (ASF) Ride Control truck, the Dresser DR-1 Steering Assembly truck, the Modified Barber S-2 truck, the Barber S-2 Center Plate Extension Pad (C-PEP) truck, the Standard Car Radial Barber-Scheffel truck, and the National Casting Swing Motion truck.

The ASF Ride Control and the Modified Barber S-2 will provide representative data on the Type I trucks. The Barber C-PEP represents a modified normal 100-ton truck, and is considered a Type II truck. The DR-1, Barber-Scheffel, and Swing Motion trucks will provide wear information for Type II trucks. These last four trucks meet the critical elements of the proposed selection criteria for Type II trucks (i.e., feasibility of technical design, length of service experience, and availability).

6.6 SUMMARY AND OUTLOOK

The performance rating of freight car trucks by means of performance indices is intended to enable a potential user to select the truck configuration best suited to the needs of his particular operation. While it is not too difficult to establish a few distinct regimes that suffice to characterize truck performance, not all technical details are available yet to define performance unambigiously even for the standard three-piece, Type I truck. As was shown, a large amount of information can be extracted from Phase I test data; at the same time, the need for additional data and more finely discriminating measurements has become apparent. Thus, the effort of specifying Type I truck performance, as a base against which the performance of premium or Type II trucks will be compared, will continue for much of the duration of the project.

The economics of truck operation are closely related to the deterioration of truck and car components, and possibly to lading damage, as affected by operating conditions such as speed, load, track type and quality, and state of wear. The identification of mechanisms and rates of wear, as a function of the variables mentioned above, as well as of truck configuration, is one of the most important activities of Phase II. It will be complemented by the collection and study of economic data. An example of an attempt to correlate some parts of the performance and deterioration segments of the "Truck Universe" is given in the Appendix; there are obviously many similar relationships that can, and will, be investigated.

A major challenge of this task is seen to be the need for general conclusions, expressed in some simple quantitative way (such as performance indices) so as to be useful to operating personnel without time-consuming effort, but at the same time adequately founded on a large base of technical detail. The amount of detailed information is increasing at a great rate, as a result of new truck and car designs, and of tests and analyses being conducted by members of the industrial and academic communities. It will require a continuing effort to integrate this information, as well as that being generated by TDOP, into the sort of general framework within which truck performance can be established for the benefit of the railroad industry.

The eventual objective of TDOP is to provide a technical and economic basis for the establishment of truck performance specifications that will enable a railroad to obtain the particular features suited to its needs.

APPENDIX A

DYNAMICS AND WEAR OF 100-TON BOXCAR FROM PHASE I DATA

1 INTRODUCTION

The body of this report has presented the characteristics and performance of the freight car truck, and the methods proposed to study and define it, in the most general terms. Reference frames, such as the Truck Universe, precisely defined performance indices, and the Phase II progression chart are useful for dealing with a mass of detailed observations and measurements which may not fall into place and reveal their meaning without some structured approach. As pointed out by Charles Darwin, "to be a good observer, one must be an active theorizer".

As already mentioned, there are bound to be discrepancies between our simplified schemes and theories, and the real world in which extraneous and significant details occur together and are difficult to separate. Thus, one cannot claim to understand a set of test data until one has explained all features and correlated them with each other on the basis of applicable physical principles.

To illustrate this approach, we present the analysis of a limited set of Phase I test data relating to the general area of ride quality. The data are shown to make physical sense in terms of a few simple mathematical models and permit an estimate of snubber wear that appears to be in the range of observed values.

2 100-TON BOX CAR: COMPARISON OF MODEL AND TEST DATA

2.1 Undamped Natural Frequencies and Normal Modes

Figures A-1 through A-4 show the vertical and lateral accelerations at various points in the car body and truck of the 100-ton box car on Barber trucks, on both jointed rail and continuous welded rail (CWR), for a range of speeds. In the case of jointed track, one of the most striking features is the pronounced peak in the both vertical and lateral accelerations at 50 mph. A slight hump at that speed is also noticeable in the record of lateral accelerations on CWR in one of the two test zones into which the tested section of track was divided. Such a peak could be due to resonance of the car body on its suspension in one of the rigid body modes.

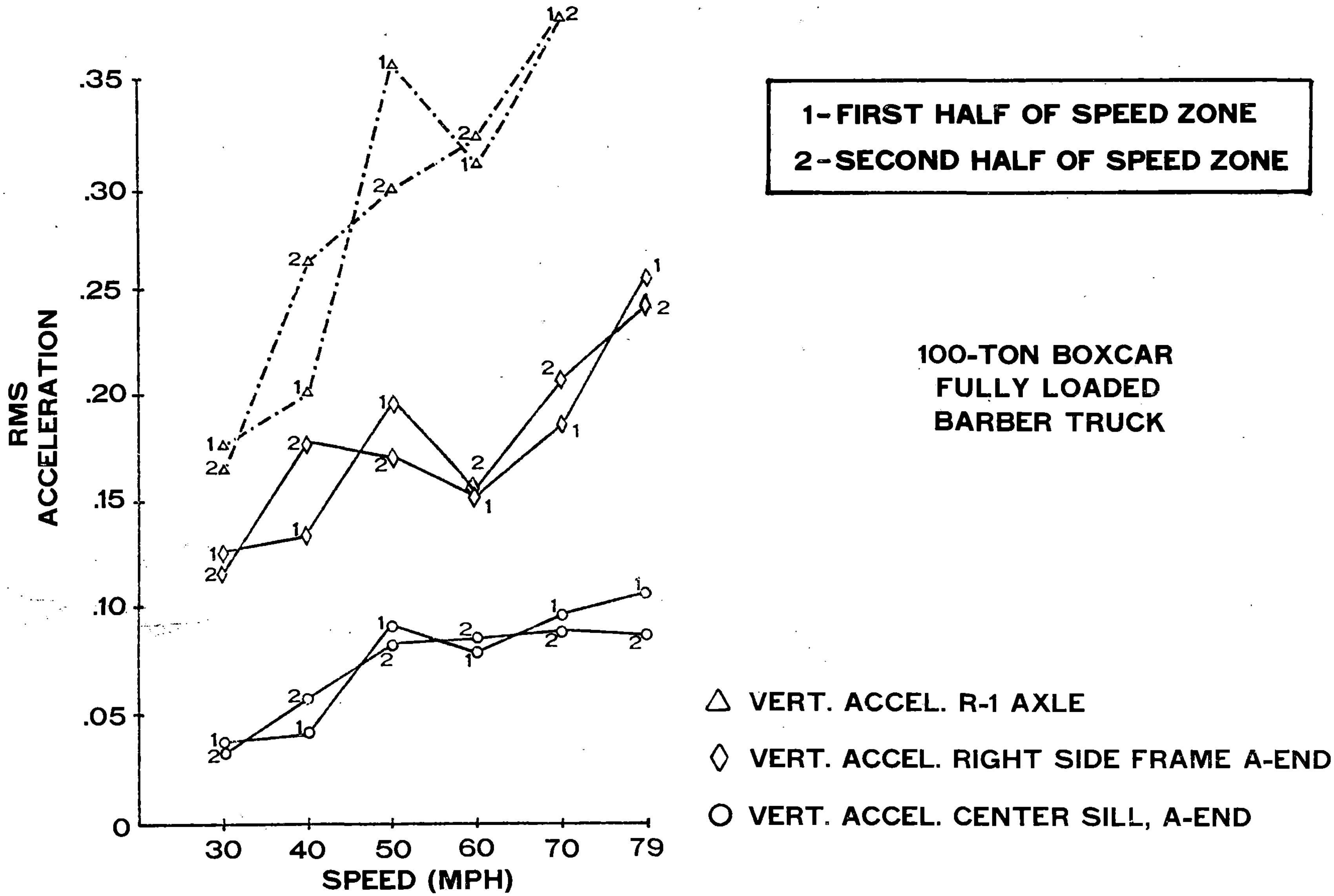


Figure A-1. Vertical Acceleration on Jointed Track

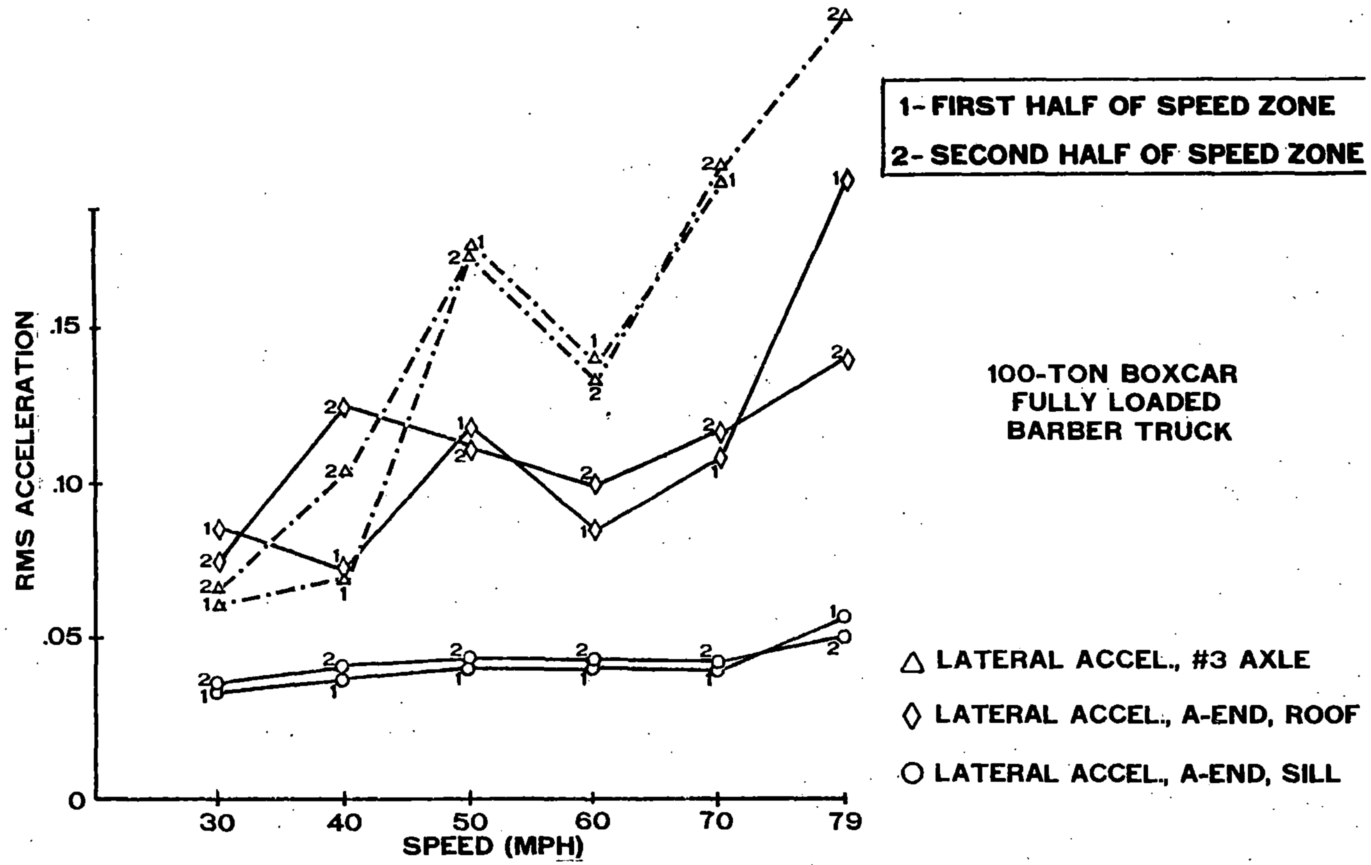


Figure A-2. Lateral Acceleration on Jointed Track

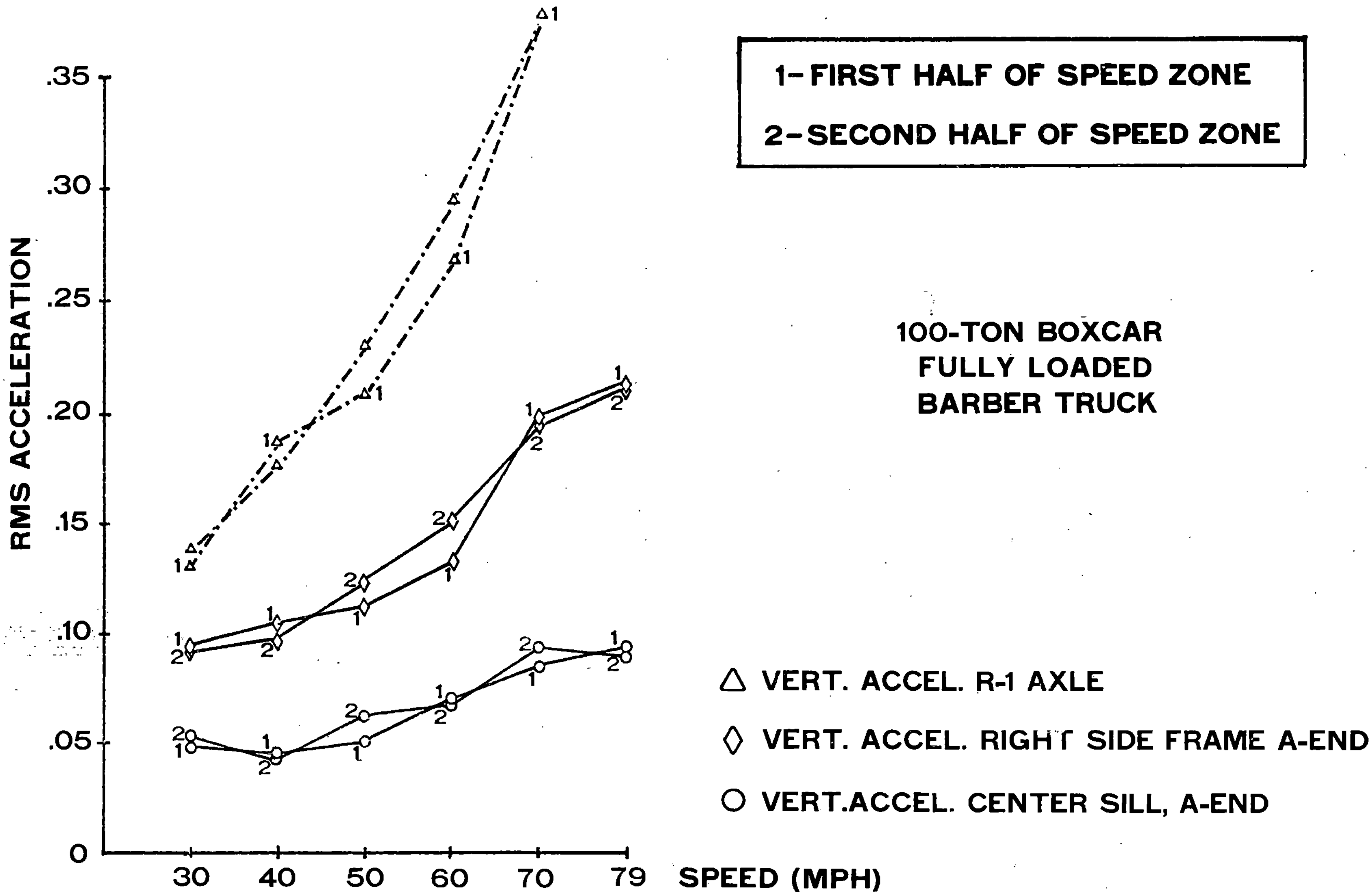


Figure A-3. Vertical Acceleration on CWR Track

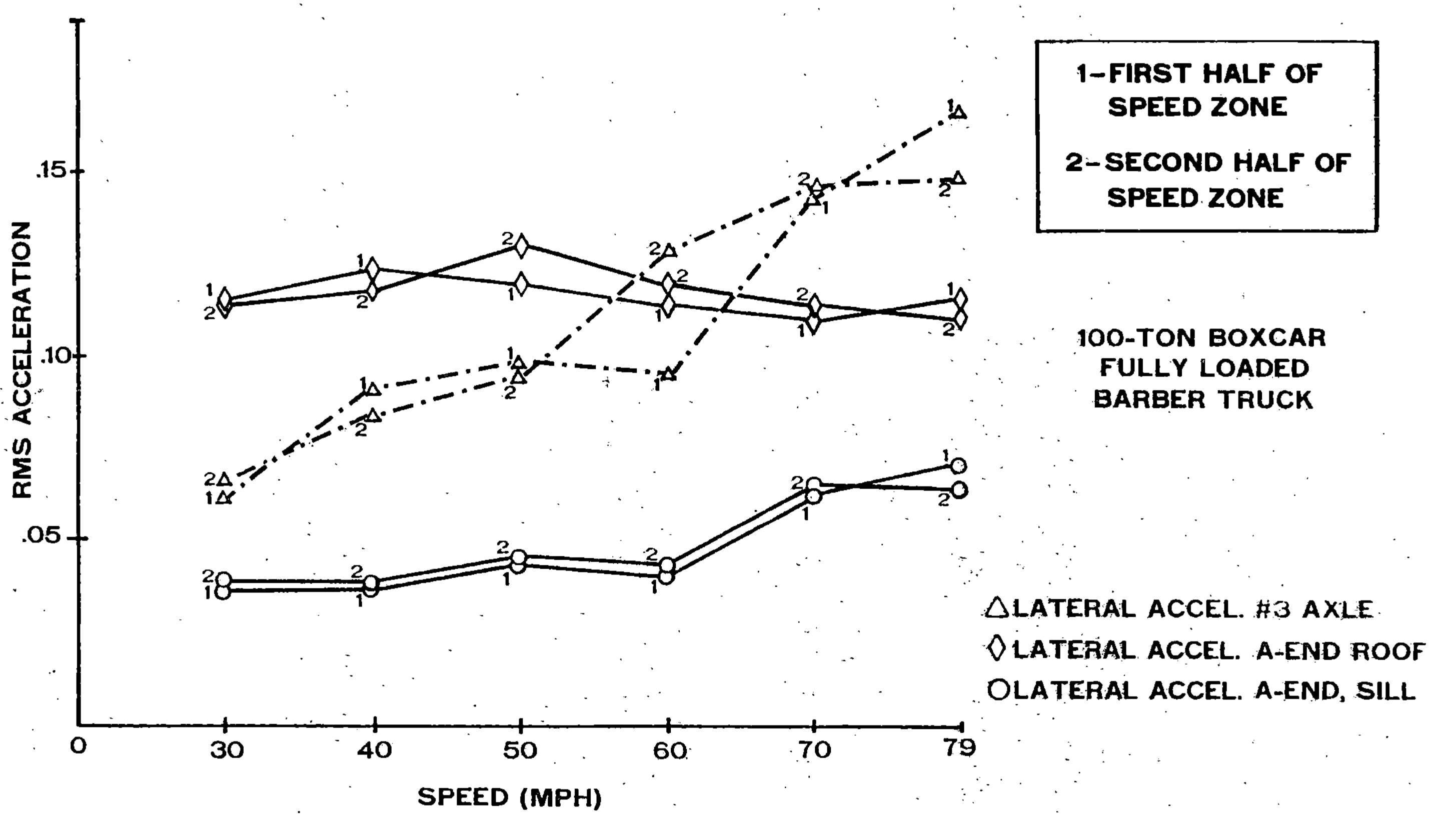


Figure A-4. Lateral Acceleration on CWR Track

This hypothesis was explored by means of the simple math model shown in Figure A-5, which has the car body supported vertically and laterally on suspension springs, without damping. The known mass of the loaded and empty car, less trucks, and the known height of the centers of gravity, were used to compute the mass moments of inertia in roll, pitch and yaw. Longitudinal motion being disregarded, there are in addition two linear degrees of freedom, lateral and vertical. The values for the lateral spring rates were taken from the Sumitomo report prepared for Southern Pacific, and the vertical spring rates are those specified on standard drawings.

Results for the loaded car are shown in Table A-1. Under each of the natural frequencies are listed the normal modes of vibration, which are relative values with the largest displacement being set equal to 1. It will be noted that, in this idealized representation, all modes in which the springs are arranged with symmetrical offsets from the center of gravity are uncoupled, that is, only one mode of vibration occurs. The only modes for which this does not apply are those identified as upper center and lower center roll which are in fact combinations of roll rotations and lateral displacements. The frequency of upper center roll is about 1.89 Hz. Table A-2 gives the same information for the empty car.

Figure A-6 represents an attempt to correlate test and model data. It is clear that the frequency of upper center roll, of 1.89 Hz is so close to the frequency of the excitation of 39-foot rail joints at 50 mph, 1.88 Hz, that the hypothesis of resonance appears to be well supported. However, correspondence between the calculated and measured mode shape is not nearly as good, as shown on the figure. Using the RMS lateral accelerations at the roof and center sill, we can locate the virtual center of upper center roll, but the ratio of roll to lateral displacements is off by a wide margin. However, an error of measurement of 1/100 of a g in the lateral acceleration of the center sill would reduce this error to 10 per cent. This suggests the limits to which attempts to correlate test and model data can be forced. In this case, the correspondence in frequency is considered more important than the discrepancy in the mode shapes. In other words, the measurements cannot locate the node of the vibration mode very accurately. It should be noted that close identification of the lower natural frequencies is generally accepted as evidence for the validity of a model. In any case, friction or other forms of damping produce coupling between modes so that the uncoupled modes resulting from the model represent only useful idealizations.

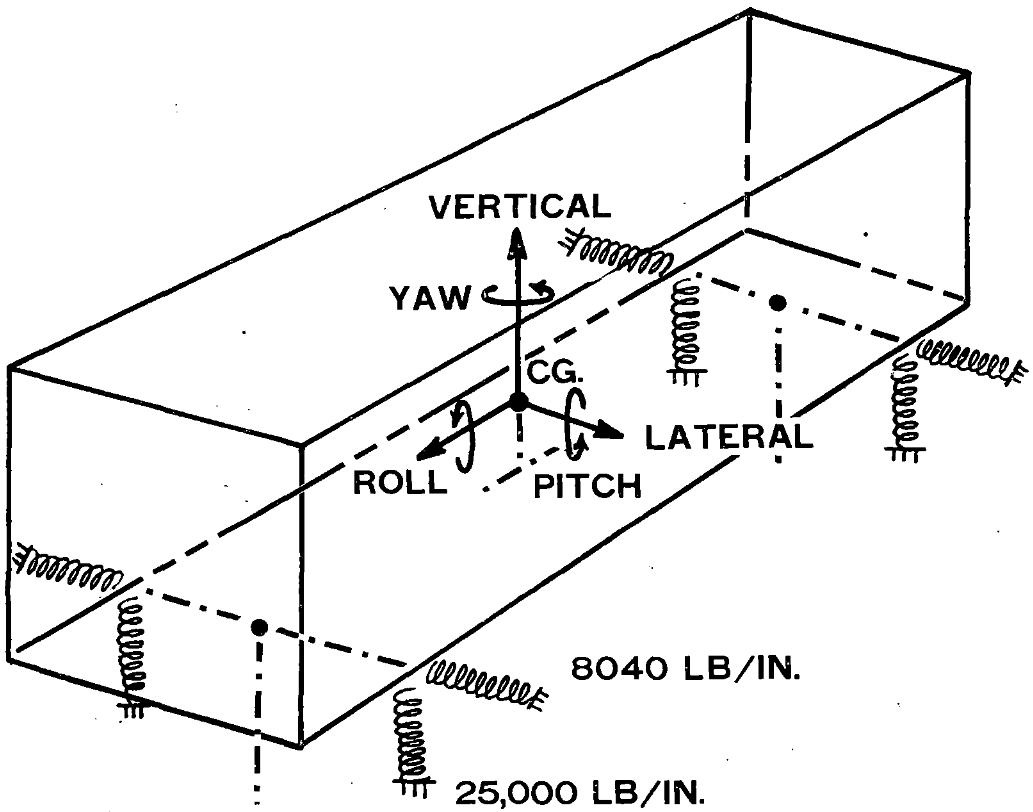


Figure A-5. Math Model of 100-ton Boxcar

Table A-1. Normal Modes and Natural Frequencies of Loaded 100-ton Boxcar

INPUT DATA

W	$.25130 \times 10^6$	- Weight	
XJ1	$.22668 \times 10^7$	- Roll	Mass Moments of Inertia
XJ2	$.30556 \times 10^8$	- Pitch	Mass Moments of Inertia
XJ3	$.29851 \times 10^8$	- Yaw	Mass Moments of Inertia
SKV	25,000	- Vertical Spring Rate	Per Side Frame
SK.	8,000	- Lateral Spring Rate	Per Side Frame
XL	555	- Truck Centers	
B1	39	- Spring Nest Moment Arm	
H	58	- C.G. Height Above Floor	

NORMAL MODES AND NATURAL FREQUENCIES (Hz)

	1	2	3	4	5
Frequency	1.89	1.97	0.773	2.53	1.45
Lateral 1	1.0	0.	1.0	0.	0.
Vertical 2	0.	1.0	0.	0.	0.
Roll 3	.031829	0.	-.00902	0.	0.
Pitch 4	0.	0.	0.	1.0	0.
Yaw 5	0.	0.	0.	0.	1.0

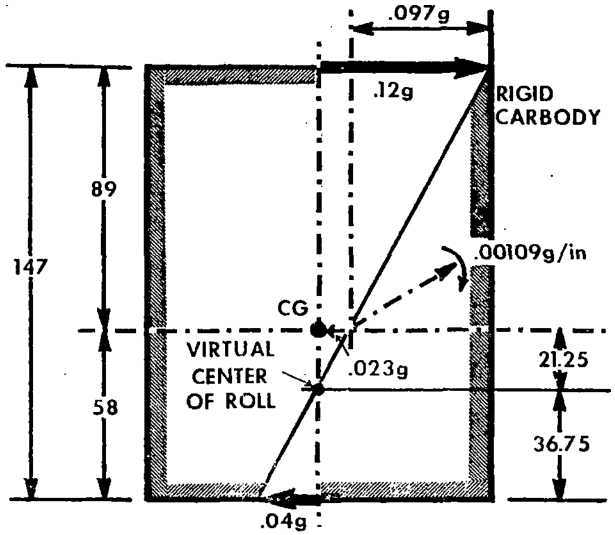
Table A-2. Normal Modes and Natural Frequencies of Empty 100-ton Boxcar

INPUT DATA

W	$.7300 \times 10^5$
XJ1	$.79825 \times 10^6$
XJ2	$.10751 \times 10^8$
XJ3	$.10614 \times 10^8$
SKV	25,000
SKL	8,040
XL	555
R1	39
H	26

NORMAL MODES AND NATURAL FREQUENCIES (Hz)

	1	2	3	4	5
Hz:	2.59	3.65	1.75	4.25	2.45
1	1.0	0.	1.0	0.	0.
2	0.	1.0	0.	0.	0.
3	0.02184	0.	-0.01089	0.	0.
4	0.	0.	0.	1.0	0.
5	0.	0.	0.	0.	1.0



ACCELERATIONS MEASURED AT ROOF AND FLOOR LOCATE VIRTUAL CENTER OF ROLL.

MOTION IS COMPOSED OF A LATERAL ACCELERATION OF .023g AT CG, PLUS A ROTATION OF (.12-.023)/89 = .00109g/in ABOUT CG.

NORMALIZED ON THE LARGEST COMPONENT , THE ROLL MODE IS .00109/.023 = .047.

THE MODEL RESULT IS .032. HOWEVER, AN ERROR OF .01g IN THE MEASURED ACCELERATION WOULD GIVE A RESULT OF .035, AN ERROR OF ABOUT 10%.

PREDICTED NATURAL FREQUENCY OF UPPER CENTER ROLL IS 1.89 Hz. FREQUENCY OF EXCITATION BY 39 FT RAIL JOINTS AT 50 MPH IS (50 x 5280)/(3600 × 39) = 1.88 Hz. PSD'S OF BOTH VERTICAL AND LATERAL ACCELERATIONS AT ROOF OF CAR SHOW SHARP PEAKS AT 1.8 - 1.9 Hz. THE MODEL EXPLAINS THESE RESULTS.

Figure A-6. Comparison of Test and Model Data for 100-ton Loaded Boxcar at 50 mph

Additional confirmation of resonance is provided by the PSD of lateral acceleration at the roof of the car which shows a pronounced peak at about 1.9 Hz (Figure A-7). A peak of equal magnitude at twice that frequency may be due to either or both of two causes: Since the truck center spacing of the 100-ton box car, 46 ft 3 in., is larger than the 39-ft rail spacing, an excitation at twice the rail joint frequency will occur. On the other hand, since the excitation is not sinusoidal, a second harmonic of considerable amplitude is likely to be present. Since the amplitudes of other harmonics are much lower, the first explanation is probably more correct.

It should be noted that resonant oscillations in upper center roll may also be excited on continuous welded rail because of departures from perfect alignment and cross level. In fact, the RMS amplitudes at the roof and center sill have practically the same values for both jointed and continuous rail.

2.2 Estimate of Snubber Wear

If we accept the measured dynamic behavior of the 100-ton box car as typical we may use the data for a rough prediction of wear between components of the truck. The steps involved in such a calculation which takes into account only the vertical relative displacement between the bolster and side frame, due to upper center roll are shown below.

Assume Roll Mode as 0.001g/in at 1.89 Hz

Angle of Roll = $(0.001)(386)/\left[2\pi \times 1.89\right]^2 = 0.00274$ Radians

Half-amplitude of Linear Displacement at Side Frame = (39) (0.00274) = 0.10675 in.

Rubbing Distance per Cycle = (4) (0.10675) = 0.427 in./cycle

Rubbing Distance Over 1,000,000 Miles at 50 mph:

$$d = \frac{1{,}000{,}000}{50} \times 3600 \times 1.89 \times 0.427 = 5.81062 \times 10^7 \text{ in.}$$

$$\text{Depth of Wear} = \frac{\text{K x Bearing Pressure x Sliding Distance}}{\text{3 x Hardness}}$$

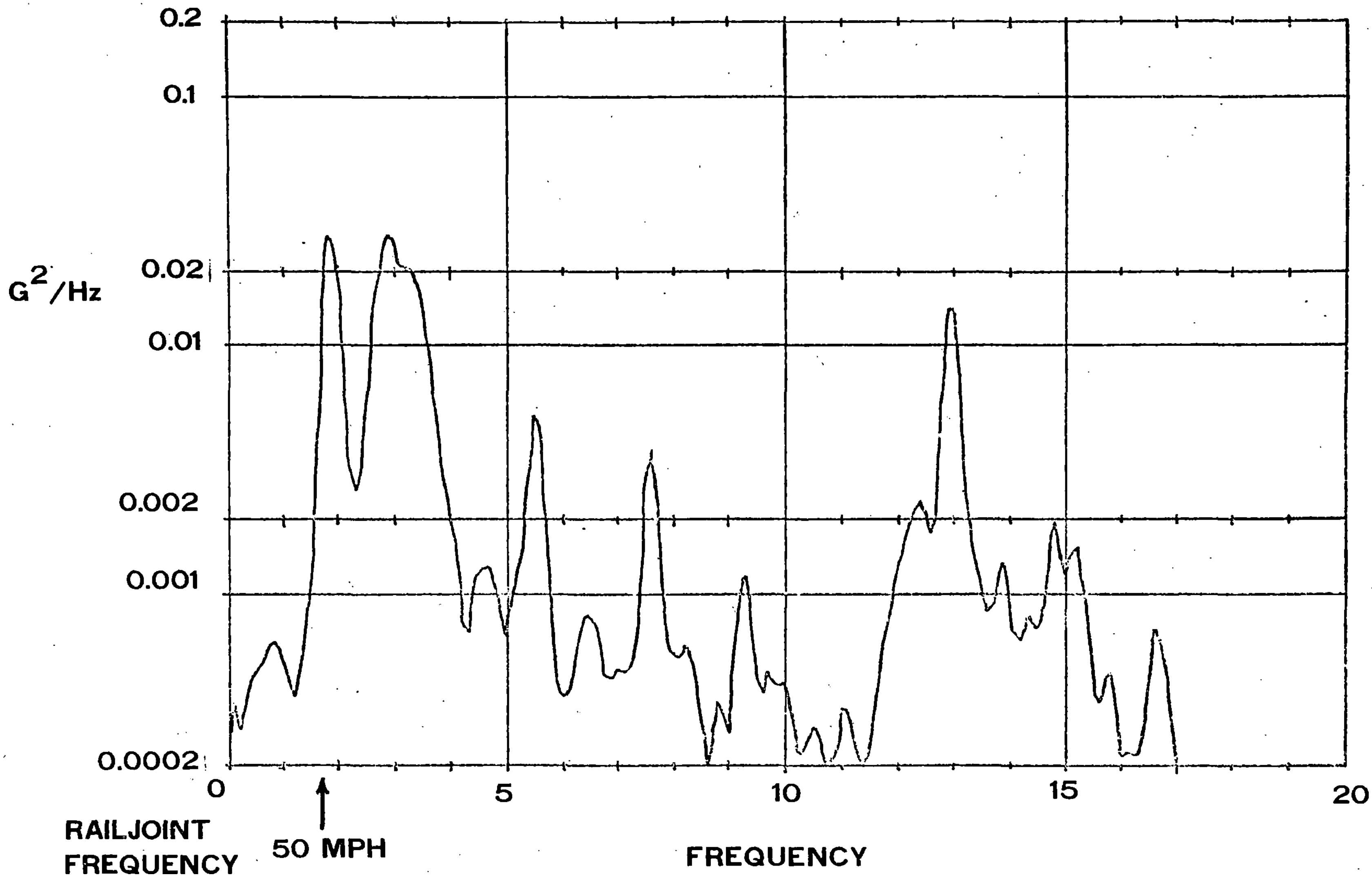


Figure A-7. **PSD of Lateral Acceleration at Roof, A-End, of Fully Loaded, 100-ton Boxcar with Barber Trucks on Jointed Track**

Where K = Archard Wear Coefficient $\approx 1 \times 10^{-4}$

Assume Bearing Pressure = 100 psi

Brinell Hardness = 300 (kg/mm^2) = 426,700 psi

CONCLUSION

Depth of Wear = 0.454 in. (Due to This Motion Alone)

With the roll amplitude derived at as a compromise between the test and model data, the total rubbing distance during a travel of one million miles at 50 mph is slightly under sixty million inches. The standard method of calculating the depth of adhesive wear utilizes the total sliding distance, the unit bearing pressure, the Brinell hardness of the softer of the two materials in contact, and a dimensionless coefficient, the Archard wear coefficient, which ranges between 5×10^{-5} and 2×10^{-4} for poorly lubricated or unlubricated surfaces. Assuming a mean value of 1×10^{-4}, a unit bearing pressure of 100 psi, and a Brinell hardness of 300 (which corresponds to 426,700 psi) we arrive at a "combined" depth of wear of slightly under one-half inch due to the vertical component of upper center roll alone. It should be emphasized that this kind of calculation is presented merely to illustrate an approach towards predicting wear, and that contributions of other relative motions will have to be considered, and compared to actually measured data, to establish baseline values that may be used with some confidence.

2.3 Response of Truck Components

Additional wear must be expected from movements of truck components relative to each other and to the car body. For example, Figure A-8 shows the PSD of the vertical acceleration at one roller bearing adapter for a 50 mph run on jointed track. Within the frequency range considered, all harmonics of the fundamental frequency of about 1.83 Hz are present, indicating that the run was made at 48.66 mph. The harmonics would make it possible to work out the shape of the periodic pulse applied to the wheel when passing over a rail joint.

It may be noted that the peaks at 7-1/2 and about 15 Hz are higher than the rest, and that there are two peaks close together near 15 Hz. The explanation for these results is provided by Figure A-9 which shows accelerations for the same roller bearing adapter, at the same speed, but on continuous welded rail. Only the peaks at 7-1/2 and

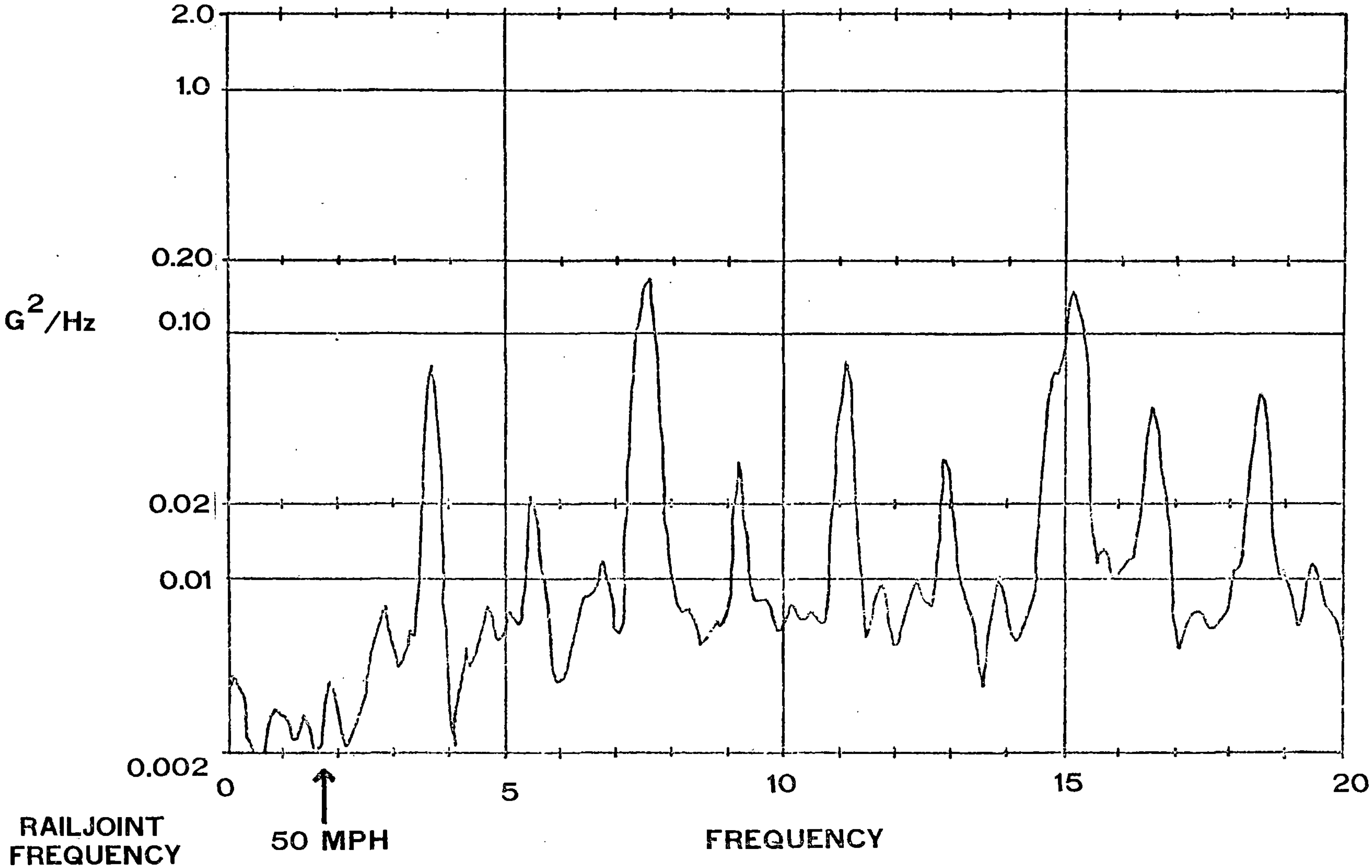


Figure A-8. PSD of Vertical Acceleration of R-1 Axle, Fully Loaded, 100-ton Boxcar with Barber Trucks on Jointed Track

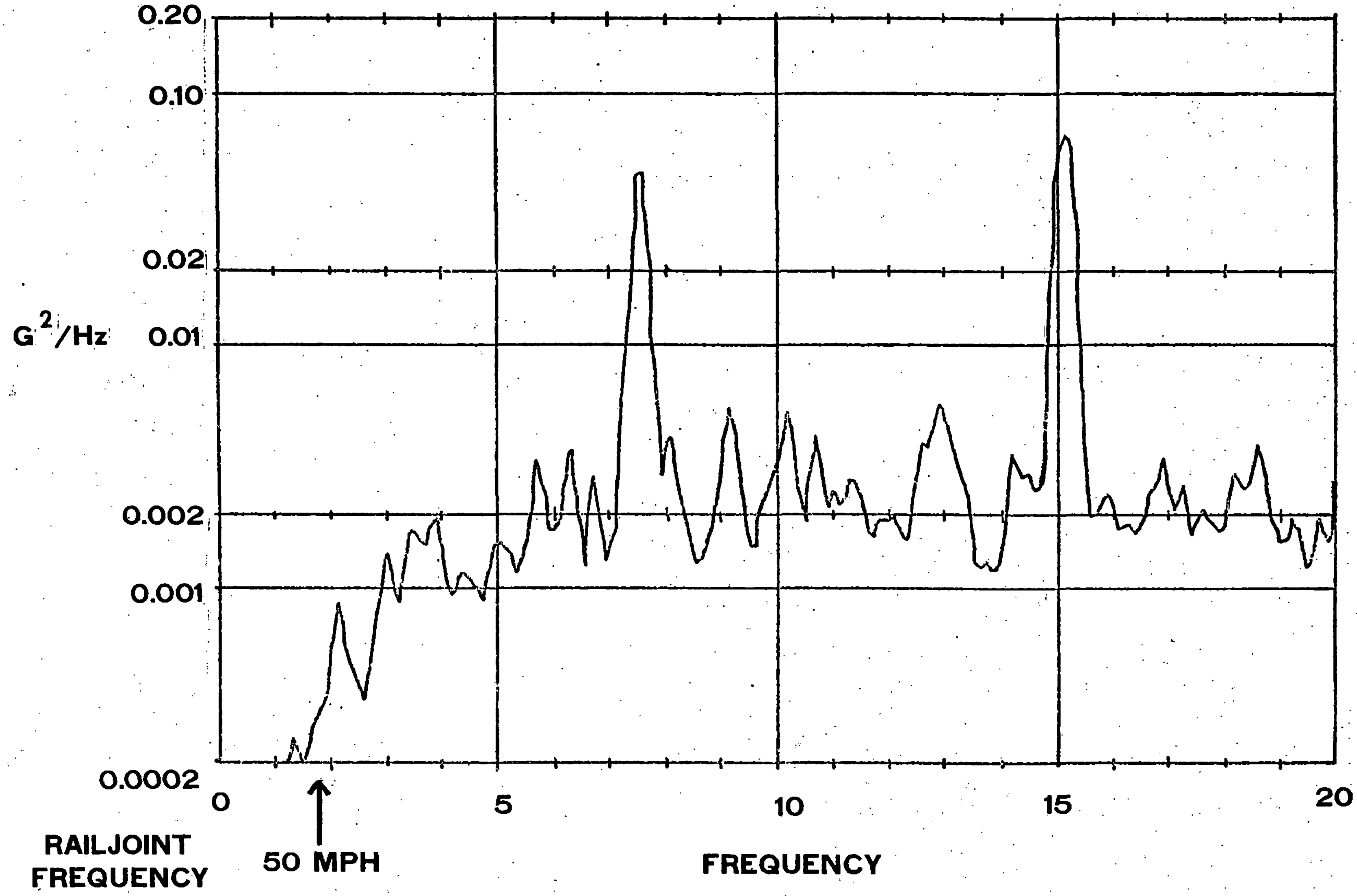


Figure A-9. PSD of Vertical Acceleration of R-1 Axle, Fully Loaded, 100-ton Boxcar with Barber Trucks on CWR Track

15 Hz are significant. This suggests that the wheel may have been unbalanced, since a 36-inch diameter wheel at 48.66 mph will have a basic frequency of 7.57 Hz. Here again, the excitation contains higher harmonics, possibly because the stiffness of the track provides greater resistance to downward acceleration than the suspension system does to upward acceleration.

Additional information that may be useful for studies of ride quality and wear are shown in the next two figures. Figure A-10 is the PSD of vertical acceleration at one side frame that apparently did not have an unbalanced wheel. The distribution of the harmonics is somewhat different from that for the axle, with the double rail joint frequency predominating at the lower end of the spectrum. The high peak at 13 Hz may be accounted for by the fact that the seventh harmonic of the rail joint impulse is very close to the frequency of excitation of the side frame due to the double impulse on two axles spaced 5 ft, 10 in. apart.

Figure A-11 shows that the lateral accelerations on the axle are of the same order of magnitude as the vertical accelerations on the side frame. The simultaneous occurrence of vertical and lateral motions, in the presence of a normal steady state load, describes conditions conducive to wear of bearing adapters and pedestal roofs which it may be possible to quantify by further analysis of the test data, perhaps supplemented by additional tests.

2.4 Wear Due to Lateral Self-Excitation

The preceding discussion may have suggested that jointed rail is the main cause of wear in truck components. According to our interpretation of the test results, it appears that considerable movement of the vehicle takes place also on continuous welded rail. One reason for this is obvious: if the rails are not perfectly plane and parallel, they will excite some of the natural modes of vibration of the vehicle. The impressions in the roadbed, made by rail joints of former jointed track replaced by CWR, have also been shown to persist even after repeated track repair. All that can be said is that the large impulses caused by rail joints, which give rise to the many harmonics in the PSD, will be absent on CWR.

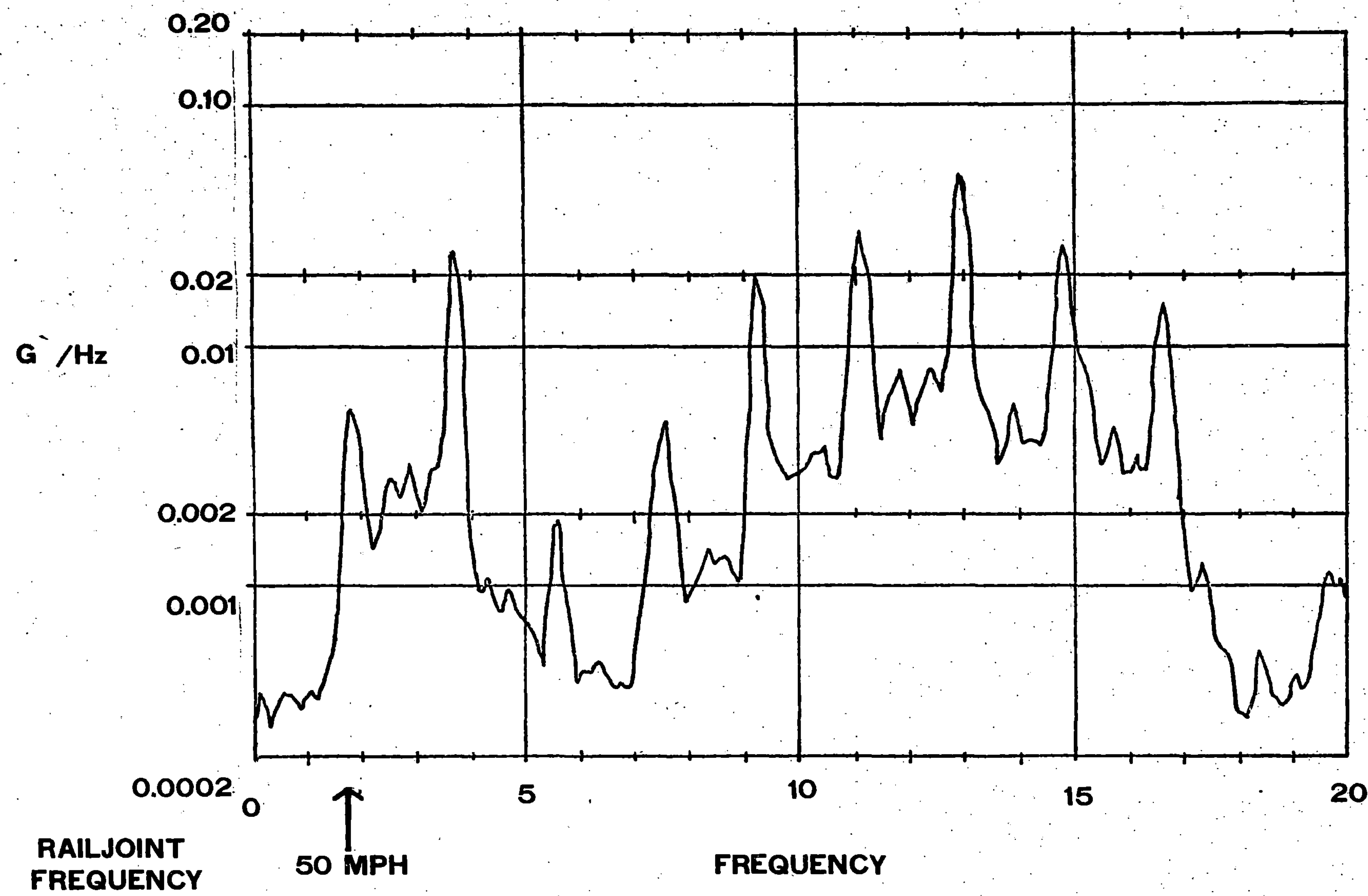


Figure A-10. PSD of Vertical Acceleration of Right Side Frame, A-End, Fully Loaded, 100-ton Boxcar with Barber Trucks on Jointed Track

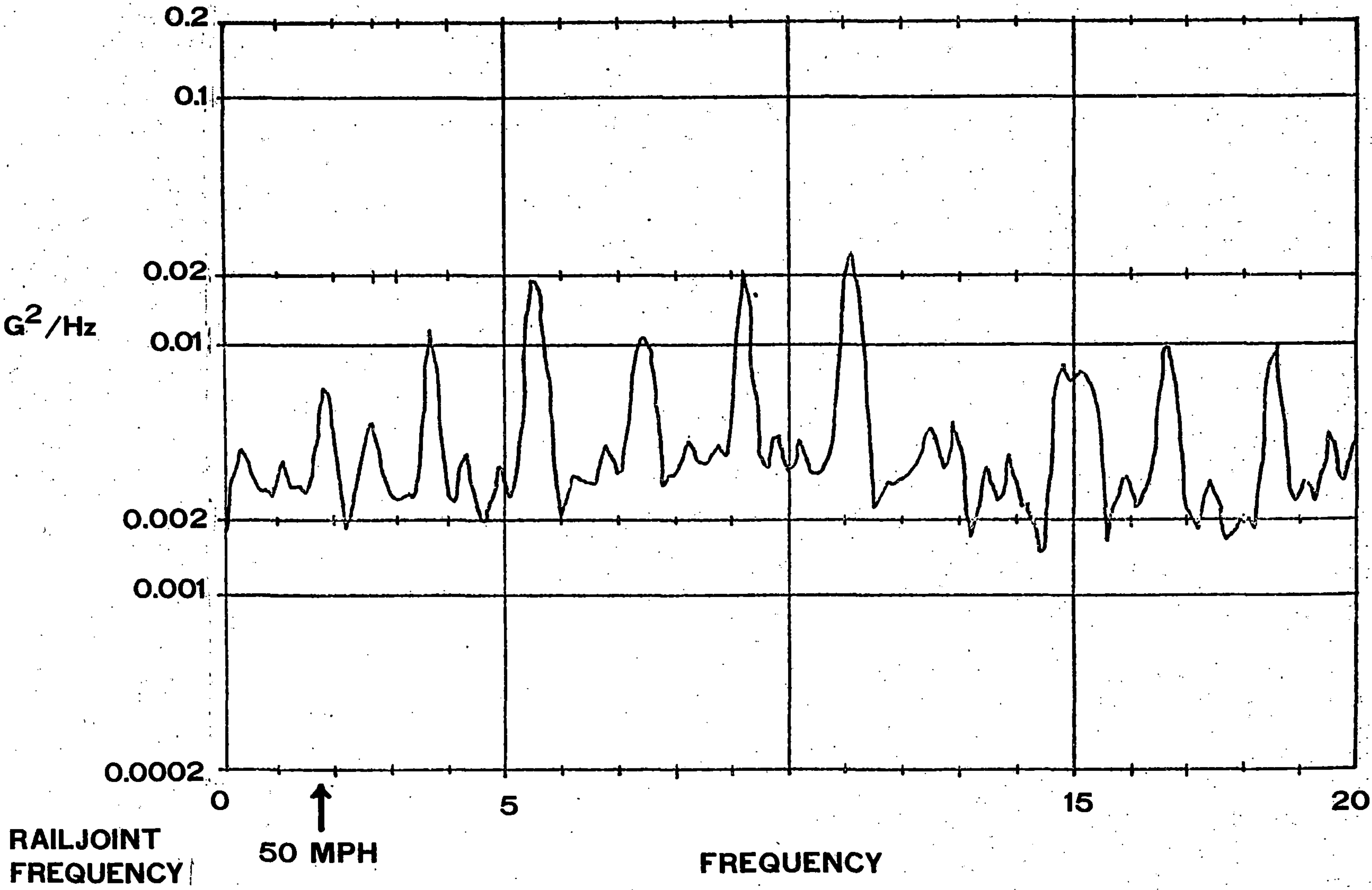


Figure A-11. PSD of Lateral Acceleration on No. 3 Axle Fully Loaded, 100-ton Boxcar with Barber Trucks on Jointed Track

However, car oscillations may occur even on perfect track, owing to excitation caused by creep forces. The dynamics of this process is analogous to hunting, although the motions are much smaller.

A wheelset performing the lightly damped sinusoidal motion is sometimes called "lively". The technical term for this motion is "kinematic hunting," which implies that inertial forces are negligible. The wavelength of kinematic hunting of a wheelset is very constant for a given state of wear of the wheel contour. It can be recognized in many of the test data from Phase I. For a profiled wheel, the wavelength is:

$$\lambda = 2\pi\sqrt{\frac{aR}{\alpha}}$$

where a = half gauge, in.

R = wheel radius, in.

α = wheel taper (or effective conicity)

For a 36 in. wheel with a 1:20 taper, and a distance of 60 in. between contact patches, the wavelength is:

$$\lambda = \frac{2\pi}{12}\sqrt{\frac{(30)(18)}{.05}} \quad \text{ft}$$

$$\lambda = 54.4 \text{ ft}$$

The frequency of excitation at 50 mph is:

$$f = \frac{V}{\lambda} = \frac{(50)(5280)}{(3600)(54.4)}$$

$$f = 1.35 \text{ Hz}$$

The kinematic wavelength of a two-axle truck depends on the stiffness of the connection between axles, which in the case of the standard three-piece truck is determined by the rotational stiffness of the sideframe/bolster connection. Figure A-12 shows the relationship between kinematic wavelength and truck stiffness, with wheel taper as a parameter. It is evident that, for standard roller bearing trucks, the kine-

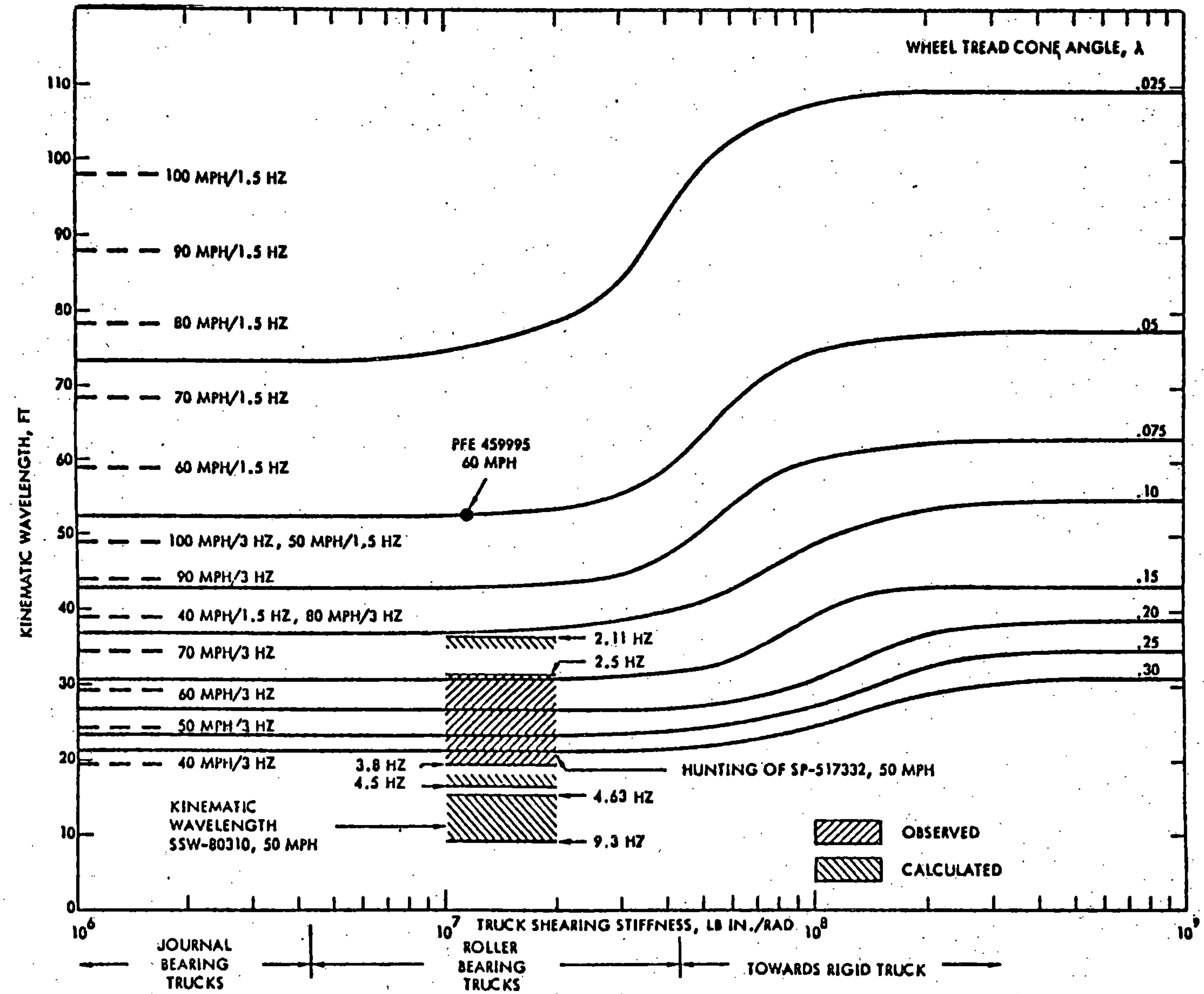


Figure A-12. Kinematic Wavelength of Truck as a Function of Shearing Stiffness and Wheel Taper matic wavelength differs negligibly from that of a single wheelset except at the upper end of the range, where it approaches, but does not quite reach, the longer wavelength of the completely rigid truck.

The relationship between wheel profile and kinematic wavelength was demonstrated by hunting tests in Phase I, as well as other tests run by Southern Pacific. This suggests a search of the test results for the frequency of kinematic hunting.

The PSD of lateral accelerations at the roof of the empty 100-ton boxcar at 50 mph, on CWR, shown in Figure A-13, has its highest peak at about 1.25 Hz. In view of the slight uncertainty in speed one may confidently conclude that this mode of vibration is forced by the wheelset, particularly since the empty carbody has no resonance as low as 1.25 Hz.

Since the lateral acceleration at the center sill is negligible, the car body appears to be rolling about a center near its floor.

We may use these data to compute the approximate amount of wear between the side frame and bolster, due to the vertical component of this roll, in the same way as was done for the loaded car on jointed rail. From the PSD, the acceleration at the top of the roof is about .0707 g. For a roof height of 145 in. above the floor, at 1.25 Hz, the vertical displacement at the side frame (half amplitude) is:

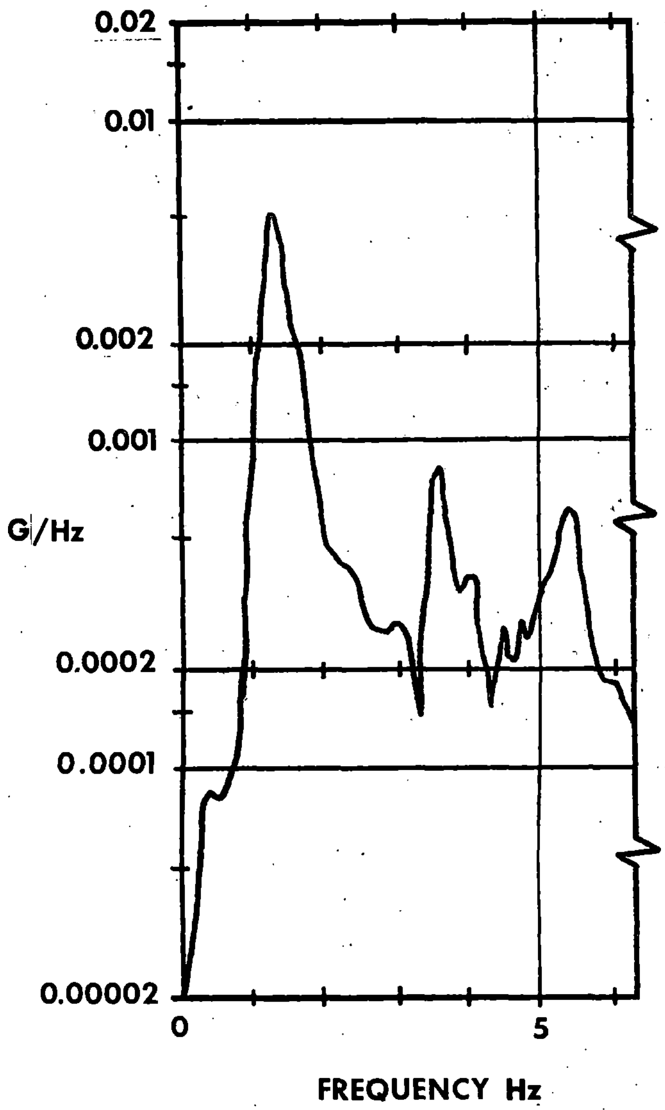
$$X = \frac{(.0707)\,(386)\,(39)}{(2\,\pi \times 1.25)^2\,(145)} = 0.119 \text{ in.}$$

The total rubbing distance over one million miles at 50 mph and 1.25 Hz, is

$$d = \frac{1{,}000{,}000}{50} \times 3600 \times 1.25 \times 4 \times 0.119$$

$$d = 4.2844 \times 10^7 \text{ in.}$$

Using the same data for unit bearing pressure and material hardness, we arrive at a depth of wear of 0.335 in. This is about three quarters of that computed for the wear of the loaded box car due to upper center roll.



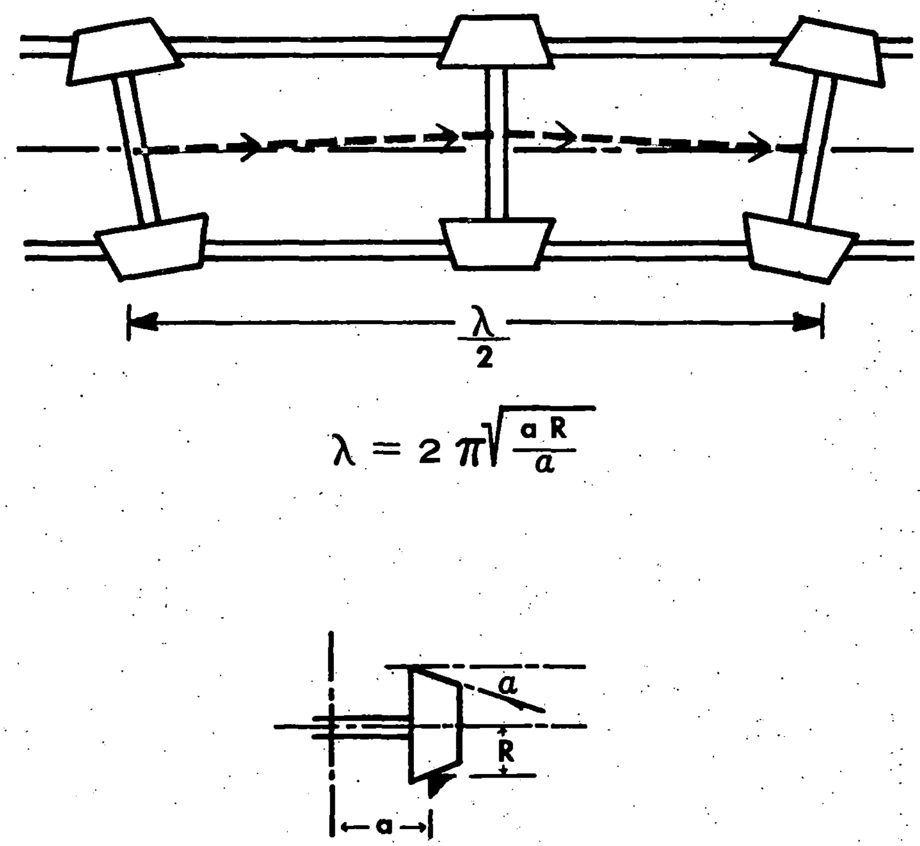


Figure A-13. PSD of Lateral Acceleration at Roof, A-End Empty 100-ton Boxcar with Barber Trucks on CWR Track

3 SUMMARY

The preceding discussion has attempted to illustrate one approach to the interpretation of the test data upon which the characterization of the Type I truck performance will be based.

It is clear that our models will never perfectly reproduce the behavior of the real system for a variety of reasons. Among these are the simplifying assumption made in the modeling, and imperfections in the data that may be due to limitations in measuring techniques.

However, inaccuracies in the performance of a model do not necessarily negate its validity. As mentioned earlier, the validity of a model is, in the last analysis, a matter of technical judgment, as is the confidence with which even incomplete results may be utilized to evaluate the real system, to suggest refinements in the model that would reduce the observed discrepancies, and to identify improvements in measurement techniques that would explain the discrepancies.

APPENDIX B - GLOSSARY

ADHESION

The absence of gross slippage of the wheel on the rail in the presence of tangential forces at the interface.

ANGLE OF ATTACK

Horizontal angle between the vertical plane of the wheel and the tangent to the rail at the point of contact.

ANOVA

Analysis of variance, a body of tests of hypotheses, methods of estimation, etc. using statistics which are linear combinations of sums of squares of linear functions of the observed values.

AXLE

The steel shaft on which the car wheels are mounted. The axle not only holds the wheels to gage, but also transmits the load from the journal boxes to the wheels.

"B" END OF CAR

The end on which the hand brake is located.

BALANCE SPEED

The speed with which a vehicle traverses a superelevated curve of constant radius when the centrifugal force exactly balances the horizontal component of the weight due to inclination.

BILLING REPAIR CARD

The card which, under AAR Interchange Rules, is furnished to the car owner when repair work is done on a foreign car.

BOLSTER (BODY)

The transverse members on the underframe of a car which transmit the load carried by the longitudinal sills through the center plate to the trucks.

BOLSTER SPRING

The main suspension spring of a car directly supporting the truck bolster and the weight of the car body.

BOLSTER (TRUCK)

A beam placed across the frame of a truck to receive, through the center plate, the weight of the carbody and transfer it to the truck frame and wheels through the spring sets in the side frames.

BOUNCE

Vertical oscillation of the center of gravity of the sprung mass (car body, truck bolster, etc.).

BRAKE BEAM

The immediate supporting structure for the two brake heads and brake shoes acting upon any given pair of wheels.

BRAKE SHOE

Friction unit contacting the wheels.

BUFF

A term used to describe compressive coupler forces.

CENTER PLATE

One of a pair of plates which fit one into the other and which support the car body on the trucks, allowing them to turn freely under the car.

CENTER SILL

The central longitudinal member of the car underframe which forms the backbone of the underframe and transmits most of the buffing shocks from one end of the car to the other.

CHAINING

A computer programming technique where linkages are provided connecting data records having the same key and associated with a master record. In the case discussed here, the master record is a file containing a list of all the access keys, and the linkage may be thought of as being the disk address of the record before and the record after each record in the chain.

CLIMB

The process of the wheel flange contacting and climbing the rail frequently onto the railhead.

COUPLER

The device connecting one rail vehicle to another.

CURVES

In the United States it is customary to express track curvature in degrees noted by the deflection from the tangent measured at stations 100 feet apart. The number of degrees of central angle subtended by a chord of 100 feet is the "degree curve." One degree of curvature is equal to a radius of 5,750 feet.

CREEP

The capability of two bodies to displace in their plane of contact without slipping. It is made possible by shear deformation of both bodies in the region of the interface, which can support tractive forces.

CRITICAL DAMPING

Amount of damping at which no oscillatory vibration occurs after a spring-mass has been released from a nonequilibrium position.

CRITICAL HUNTING SPEED

Minimum speed at which violent truck shimmy occurs.

CRITICAL SPEED

Excitation forces applied to the vehicle are related to forward speed. A critical speed is one at which a car-truck dynamic resonance occurs.

CUSHIONING DEVICE

Part of the coupler used to absorb shocks.

DAMPING

Means to absorb vibration energy in the system.

DAMPING COEFFICIENT

Number describing the energy absorbing property of a physical system.

DATA BASE

A collection of interrelated data stored together to serve one or more applications; data is stored so that they are independent of programs using the data, a common and controlled approach to maintenance and retrieval of data.

DEFECT CARD

Issued by the railroads to acknowledge their responsibility for damage done to a car for which they are liable in accordance with the rules as set forth by the Association of American Railroads.

DEGREE OF FREEDOM

A single parameter or coordinate of choice with physical objects, it implies the ability to displace along or about one coordinate axis.

DRAFT

A term used to describe tensile coupler forces.

DRAFT GEAR

The name of that unit which forms the connection between the coupler rigging and the center sill. The purpose of this unit is to receive the shocks incidental to train movements and coupling of cars, and so cushion the force of impact that the maximum unit stress is brought within the capacity of the car structure.

DRAG RATING

The amount of drawbar pull available from a particular locomotive consist calculated at the minimum continuous speed.

DRAWBAR FORCES

Longitudinal forces at the couplers between cars and/or locomotives that may be either tensile (draft) or compressive (buff), depending on the operation of the train at the time.

DYNAMIC BRAKING

An electrical means used to convert some of the power developed by the momentum of a moving locomotive into an effective retarding force.

ELEVATION

The higher position of one of the two rails.

FISHTAILING

Special case of yaw describing the motion of the rear-end lateral motion of the vehicle about the front truck as a center.

FLAT SPOT

Loss of roundness of the tread of a railroad wheel, caused by wheel-sliding.

FLEXIBLE TRUCK

A truck in which the axles are allowed to displace relative to the frame in the lateral and yaw degrees of freedom, against elastic constraints, usually in the form of elastomers.

FORCE-DISPLACEMENT CHARACTERISTICS

Graph describing the force necessary to reach/maintain a specific amount of displacement.

FORCED FREQUENCY

Frequency imposed on the system by superimposed forces (rail joints, etc).

FOREIGN CAR

Any car not belonging to the particular railway on which it is running.

FRICTION BLOCK

A casting attached to the truck bolster as a guide and to take the wear between the bolster and transom. Commonly called Bolster Guide.

FRICTION PLATE

A removable plate to prevent wear on the main body of a component.

GAGE (OF THE TRACK)

The distance between the rails measured from the inside head of each rail at a right angle 5/8 inches below the top of the rail. The standard for this dimension on North American Railways is 4 ft. $8\frac{1}{2}$ in.

GRADE

Part of roadbed with changing elevation.

GROSS WEIGHT

The total weight of a car, including the lading.

HARMONIC ROLL

Periodic angular displacement of the vehicle body about its longitudinal axis, due to vertical track inputs close to the natural frequency of the carbody on its suspension referred to as Rock and Roll.

HI-CUBE CAR

A box car of approximately 85-ft length and 10,000 cu ft capacity designed for hauling automobile body stampings and other low density freight.

HIGH SLIDE GONDOLA CAR

A gondola car, with sides and ends over 36 in. high, for carrying coal or minerals.

HOPPER CAR

A car, open or covered, with the floor sloping from the ends and sides to one or more hoppers, which will discharge its entire load by gravity through the hopper doors.

HOT BOX

An overheated journal (commonly known as a hot box) is caused by excessive friction between bearing and journal, lack of lubricant or foreign matter.

HUMP (CREST)

The act of switching and classifying trains with gravity being used as the prime mover usually accomplished with the use of a small hill.

HUNTING

Dynamic instability of sets of wheels or entire trucks consisting of a lateral translation along the axle and rotational vibration about a vertical axis.

HYSTERESIS

Multivalued section of physical device characteristics.

INTERCHANGE

The transfer of cars from one road to another.

INTERPOLATION

Derivation of function values at arguments not listed in the look-up table.

ITERATION

Method of computation through refinements. The iteration is called convergent if the sequence of computed values is convergent. The first element of that sequence is called initialization.

JACKING PAD

Bosses with flat surfaces incorporated on the under frame surface of a locomotive or car body bolster or frame to provide places to apply jacks for lifting the vehicle.

JACKKNIFING

A condition involving two coupled rail vehicles in which there is excessive center sill misalignment and coupler angularity. Jackknifing is caused by high buff forces in the train.

JOURNAL BEARING

A combination of rollers and races or a block of metal, usually brass or bronze, in contact with a journal, on which the load rests. In car construction the term when unqualified means a car axle journal bearing.

JUNCTION BOX

A metallic receptacle in which several lines of electrical conductors are joined.

KINEMATICS

The branch of mechanics that deals with motion without consideration of inertial forces.

KINEMATIC WAVELENGTH

The wavelength of the sinusoidal motion of a wheelset or truck along the track when inertial forces are negligible.

KINETIC ENERGY

A moving body possesses an amount of energy equal to the energy needed to bring it to rest. The energy of the moving body is called Kinetic Energy. It is defined as half the product of the mass (mass movement of inertia) and the square of the velocity (angular velocity).

KINETIC FRICTION

Friction of motion, such as that between the brake shoe and wheel (when the wheel is turning) or as between a wheel and rail (during sliding or slipping). Kinetic friction is always less than static friction.

LATERAL VIBRATION

Pure side to side movement in the horizontal plane.

LIFE CYCLE

The expected life of truck components over the life of a freight car.

LOOK-UP TABLE

Values of a function at several arguments arranged in a way that a computer can access it.

LOW LEVEL

Referring to the height of the Trailer on Flat Car (TOFC) or auto rack flat car deck above the top of the rail. Low level cars have a deck height of 31½" as opposed to 41½" height for "standard level" cars.

LOWER CENTER ROLL

Rotation of the car body about a virtual longitudinal axis below its center of gravity.

L/V RATIO

Defined as the ratio of the lateral force to the vertical force of a car or locomotive wheel on a rail. It is an important indicator of wheel climb, shifting of the track structure, rail turnover and/or derailments.

NATURAL FREQUENCY

The frequency at which the system tends to vibrate when released after being displaced from neutral position.

NEST SPRING

A helical spring with one or more coils of springs inside it.

NET INCREMENTAL BENEFIT

Present value of future operating costs of Type I Trucks subtracted from the future operating costs of a Type II Truck.

NET INCREMENTAL COST

Purchase price of a Type II Truck minus investment tax credits subtracted from the same number for a Type I Truck.

NOSING

Special case of yaw usually describing a motion of a locomotive which applies lateral forces alternately on the right and left rails of the track.

PARALLELOGRAMMING

Relative longitudinal displacement of truck side frames which causes the truck to go in and out of tram.

PERFORMANCE CRITERIA

The aspects of truck behavior considered desirable in various performance regimes. Criteria may range from the most general, such as safety from derailment or low wear rates, to the specific, such as lateral stability or a curve negotiability.

PERFORMANCE INDEX

A measurable physical quantity characteristic of performance in a particular regime. An example of a performance index for hunting would be the critical speed, and for curve negotiation, the lateral load on the outer leading wheel of the truck. Each performance index must be qualified by a statement of conditions for which it applies, and which may affect its magnitude to varying degrees.

PERFORMANCE REGIME

The characteristic way in which a railcar or truck responds to a combination of track and operating conditions (such as speed). Inherent in this definition is a comparison with stable vehicle behavior on "ideal" tangent track. Performance regimes selected for truck characterization should be sufficiently distinct to permit ranking of truck performance on non-overlapping scales. The five primary regimes chosen are hunting, steady state curve negotiation, harmonic roll, ride quality and derailment.

PITCH

Angular motion in the vertical plane about the axle perpendicular to the direction of the track.

POWER SPECTRAL DENSITY

This represents the distribution of energy in a vibrating system under defined test conditions over the frequency spectrum specified.

PRESENT VALUE

A method of accounting for the effect of time in economics. A dollar given to you today is worth more than a dollar given to you next year.

RADIAL TRUCK

A truck in which the axles of the wheelsets are made to assume an approximately radial orientation in a curve. A radial truck is a special form of flexible truck in which the wheelsets or side frames are connected by special devices that determine the curving kinematics.

RAIL OVERTURNING

While this is not a car or truck motion term, it should be applied only when there is conclusive evidence of excessively high lateral forces generated by the rail vehicle involved.

REGRESSION

A statistical procedure for fitting the coefficients of an equation, usually based on minimizing the sum of the squares of the errors between the observed and predicted values of the equation.

RESONANCE

The condition at which forcing frequency is equal to natural frequency; this usually results in violent motion.

ROCK AND ROLL

A slang term for the excessive lateral rocking of cars and locomotives, usually at low speeds and associated with jointed rail. The speed range at which this cyclic phenomenon occurs is between 10 and 25 mph, with the exact speed determined by such factors as the wheel base, height of the center of gravity of each individual car or engine, the spring dampening associated with each vehicle's suspension system, and the relative difference in elevation between successive joints in jointed rail territory. In extreme cases, actual wheel lift can occur which can result in derailments (see also Harmonic Roll).

ROLL

Rotation of the car body about a longitudinal axis through the center of gravity.

ROLLABILITY

The relative resistance of the truck to longitudinal motion.

ROLLER BEARING

The general term applied to a group of journal bearings which depend upon the rolling action of a set of rollers, in order to reduce rotational friction. The different types are distinguished by the shapes of the rollers and by their arrangement in the bearing. Three types of rollers are in common use at present for car journals - cylindrical, tapered, and spherical.

ROLLER BEARING ADAPTER

A specially shaped piece of steel which assures proper seating of a roller bearing in the pedestal type side frame.

RUN-IN

Describes the relative movement of the cars in the train to a state of compression.

RUN-OUT

Describes the relative movement of the cars in the train to a state of tension.

SAG OR DIP

A rapid decrease in grade followed by an increase in grade sufficient to result in abnormal slack adjustment.

SHEARING STRESS

The action or force causing two contacting parts or layers to slide upon each other, moving apart in opposite directions parallel to the plane of their contact.

SHIMMY

A synonym for hunting.

SIDE BEARING

Bearings attached to the bolsters of a car body, or truck, on each side of the center plate to prevent excessive rocking. The upper, or body side bearing, and the lower, or truck side bearing, are sometimes merely large flat surfaces. Other types of side bearings employ rollers, springs and friction elements to maintain constant contact and control relative movement between body and truck.

SIDE FRAME

The frame which forms the side of a car body or a truck. It includes the columns braces, plate, belt rail, etc., for the car body; and the side member of a truck frame.

SILL (CAR)

The main longitudinal members of a car underframe which are connected transversely by the end sills, body bolsters, and cross ties. Sills are divided into side sills, intermediate sills and center sills.

SNUBBERS

Damping devices which are used to attenuate oscillations of a car or truck. They may be similar to hydraulic shock absorbers. Friction devices are commonly used in rail vehicles.

SPRING GROUP

A helical car spring assembly formed of a number of separate springs, single, or nested and united by a common pair of spring plates.

STRINGLINING

A term used to describe the tendency of cars to pull off the inside of curves, trying to approach a straight line when the train is in draft.

SUPERELEVATION

The vertical distance between the heights of inner and outer edges of railroad rails.

SWING BOLSTER

A truck bolster (so called in distinction from a rigid bolster) which is suspended by hangers or links so that it can swing laterally in relation to the truck. The object of providing this swinging motion to the bolster is to prevent, as much as possible, lateral blows and shocks from being communicated to the car body, and, vice versa, to prevent the momentum of the car body from acting with its full force on the truck frame and wheel flanges.

SWING HANGER

Bars or links, attached at their upper ends to the frame of a swing motion truck, and carrying the spring plank at their lower ends. Also called bolster hanger.

SWING MOTION

A term applied to an arrangement of hangers or transom for the springs and truck bolster which enables a car body to swing laterally on the truck.

SWIVELING

Angular oscillation about an axis; a symmetry, usually applied to truck action when the bolster oscillates around the center-pin.

TARE WEIGHT

The weight of an empty car.

THROAT (CAR WHEEL)

The curved transition between the wheel tread and flange.

TRACK-TRAIN DYNAMICS

A term used to describe the dynamic motion and the resulting dynamic forces that result from the interaction of the vehicles coupled into a train interacting with the track, under given climatic conditions, train handling, train makeup, grades, curvature and operating policies.

TRACK-TRAIN ENVIRONMENT

All the conditions which effect the track and/or the train, such as grades, curvature, locomotive and car characteristics, train handling, etc.

TRACTIVE EFFORT

The force exerted by a locomotive on the track for the movement of a train, measured in pounds.

TRAIN BLOCKING

The organization of cars within a train which minimizes the dynamic instability of the train.

TRAM

This term applies to the diagonal measurement of axle bearing locations. When, in a four-wheel truck, the two diagonal measurements are equal, the truck is said to be in tram.

TRANSMISSIBILITY RATIOS

These criteria represent the ratio of maximum car body accelerations to input accelerations that can be accepted for a specified duration to the basic dynamic index of the track profile used in the qualifying tests.

TREAD

The exterior cylindrical surface of a car wheel next to the flange which comes in contact with the rail.

TRUCK CENTERS

The center point of a truck. The distance between truck centers is that distance as measured from one truck center to the other truck center on a single car.

TRUCK CLASSIFICATIONS

TYPE I: GENERAL PURPOSE DESIGN (STANDARD THREE-PIECE)

This design is interchangeable with existing trucks so as to preserve the present truck coupler height, supports the car body on center plates, utilizes air brakes which are compatible with existing systems, accepts, standard wheel sets and journal bearings, and whose components meet applicable Association of American Railroads (AAR) requirements.

TYPE II: SPECIAL PURPOSE DESIGN (PREMIUM)

This design utilizes current wheel set and journal bearing assemblies, is compatible with existing air brake systems, and preserves car coupler height. The Type II truck may employ mechanisms other than center plate and side bearings for support and stabilization of the car body.

TRUCK WHEEL BASE

The horizontal distance between the centers of the first and last axles of a truck.

UNDULATING GRADE

A track profile with grade changes so often that an average train passing over the track has some cars on three or more alternating ascending and descending grades. The train slack is always tending to adjust as cars on descending grades tend to roll faster than those on ascending grades.

UNIT TRAIN

A train transporting a single commodity from one source (shipper) to one destination (consignee) in accordance with an applicable tariff and with assigned cars.

UPPER CENTER ROLL

Rotation of the car body about a longitudinal axis above its center of gravity.

VERTICAL VIBRATION

Pure up and down motion often described as bounce.

WALKING

This term describes the vertical equalization and flexibility of a truck. Trucks are required to negotiate rough track conditions which demand that each wheel follow the rail head with minimum tendency to unload. Proper equalization is implied and sufficient mechanical freedom to permit independent rise and fall is necessary. When a truck meets these requirements, it is said to "walk" freely on rough track without derailing or unloading of any of its wheels.

WEARPLATES

Renewable, wear-resistant, hardened steel plates which may be applied to center-plates, side bearing pads, draft gear housings, etc.

WHEEL

The flanged rolling element which carried the weight and provides guidance for rail vehicles. It also serves as the brake drum for tread-braked equipment. Major classifications are "forged" (wrought) and "cast" steel wheels.

WHEEL CLIMB

This term applies to the condition where the lateral (axial) force between the wheel flange and rail head is great enough so that the resulting friction force causes the wheel flange to climb up on the rail.

WHEEL FLANGE

The projecting edge or rim on the periphery of a car wheel for keeping it on the rail.

WHEEL LIFT

This term applies to the lifting of a lightly loaded wheel due to high vertical force on the opposite bearing and the resulting moment. Such forces are encountered when rail vehicles are operated at speeds too great for the existing super-elevation on a curve, from very slow speed operation on a high super-elevation curve, from high draft (or buff) forces on a curve, or from harmonious rocking of a car on rough track.

WHEEL PLATE

The part of a disc car wheel which connects the rim and the hub. It occupies the place and fulfills the same purpose as the spokes do in an open or spoke wheel.

WHEEL SLIDING

The situation where the wheel is rotating slower than longitudinal movement would dictate, and adhesion is lost.

WHEEL SLIPPING

The situation where the wheel rotates faster than longitudinal movement would dictate, and adhesion is lost.

WHEEL TREAD

The exterior cylindrical surface of a wheel which bears on the rails.

WHEEL UNLOADING

Reduction of vertical wheel reaction on the rail.

YAW

Angular motion in the horizontal plane about a vertical axis.

ERRATA

Page	
iii/iv	Executive Summary, third paragraph, change "Federal Railrod Administration" to "Federal Railroad Administration".
40	Paragraph 4.2.4.1, sixth line should read: "Θ = angle of twist of track within axle spacing of truck, degrees".
B-6	Change "HIGH SLIDE CONDOLA" to "HIGH SIDE CONDOLA".

EXECUTIVE SUMMARY

Increasing demands on freight car performance during the past several decades have revealed the shortcomings of the standard three-piece freight car truck. The evidence comprises increased truck maintenance, damage to track, and more frequent derailments.

Truck manufacturers are responding to the need for better truck performance by introducing modifications of the standard truck as well as novel configurations designed to reduce problems under specific operating conditions. Justification for the higher cost of improved trucks is difficult to establish for two reasons:

- Too little quantitative information is available on the characteristics of the standard truck, as well as that of the various new designs, to make possible a comparison in engineering terms.
- While data on truck maintenance cost and freight car utilization are available in the files of railroads and operators, these data have not been systematically related to truck performance.

The objectives of the Truck Design Optimization Project (TDOP) Phase II conducted by the Federal Railroad Administration are:

- To define the performance of both standard and premium trucks in quantitative terms, represented by performance indices.
- To establish a plan for collecting economic data on the cost of acquiring, operating and maintaining the standard three-piece truck.
- To establish a quantitative basis for evaluating the economic benefits to be derived from improved freight car trucks.
- To supply a basis for a performance specification for freight car trucks.

The means by which these objectives will be achieved are:

- Road testing of several representative car body types on a number of premium trucks.
- Mathematical modeling of freight cars and trucks and comparison of model test results with a view towards extending knowledge of truck behavior to configurations not tested.
- Determination of wear of premium trucks in unit train service over an extended period of time.
- Collection of economic data on truck maintenance and operation, and correlation of such data with information on truck performance.

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