

DESIGN AND ANALYSIS OF A TRACK COMPLIANCE MEASUREMENT SYSTEM



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PHASE II FINAL REPORT**

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16. Abstract <p>This report covers the design of a vehicle for measuring the static and dynamic compliance of railroad track. The static compliance system uses the rail deflections due to different axle loads to calculate the track compliance. The static compliance measurements are taken continuously as the vehicle travels a section of track. The dynamic compliance system utilizes an electrohydraulic excitation system capable of cyclically applying in-service wheel loads simulating a passing train and superimposing either pulse, random, or sinusoidal dynamic excitation on the wheel load to dynamically excite the track structure. The excitation and resulting response of the system are analyzed using a digital Fast Fourier Transform program. From these data, the dynamic characteristics of the track structure can be determined.</p> <p>The information in this report is intended for use by research personnel who have an interest in railroad track performance as related to vehicle/track interaction and track maintenance, and in the measurement of track deflections and dynamic characteristics for developing track analysis models and evaluating track structure condition.</p>				13. Type of Report and Period Covered PHASE II FINAL REPORT JANUARY 1974-JUNE 1978	
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PREFACE

This report presents the Phase II results from a 3 Phase Program having the objective of designing and building equipment to measure the static stiffness and dynamic compliance of railroad track. The report was prepared by Battelle's Columbus Laboratories (BCL) under Contract DOT-FR-30051 from the Office of Research and Development of the Federal Railroad Administration (FRA).

Messrs. Thomas P. Woll and William B. O'Sullivan have been successive contracting officer's technical representatives for this contract, to date. The cooperation and suggestions provided by Messrs. Woll and O'Sullivan are gratefully acknowledged. The authors are also grateful for the cooperation of many persons in the railroad community who contributed their time and knowledge to help identify the requirements of a track measurement system and techniques which might be used to meet those requirements, and to the Kalmbach Publishing Company for permission to reproduce data from Our GM Scrapbook.

EXECUTIVE SUMMARY

In recent years the need for increased train speeds, train loads, and reduced track maintenance costs has resulted in the initiation of several research programs to study track structures, vehicle dynamic performance, and how the track and vehicles interact. One result obtained from many of these programs is the recognition that the available data -- and means for obtaining data -- on the properties of track structures are inadequate.

To obtain track data required in other related programs and to evaluate existing track, the Federal Railroad Administration (FRA) initiated a program at Battelle's Columbus Laboratories (BCL) to develop a system capable of measuring track structural parameters. A major task in the program has been to determine what parameters this system should measure and how the measurements should be made. This task included analysis, experimental studies, a review of the literature, and discussions with other investigators and railroad personnel.

Some of the conclusions reached from these studies were that:

- By applying a vertical load to track and measuring the resulting deflection it is possible to identify track which will deteriorate rapidly under heavy loads.
- An important use for a track structure measurement system would be to measure the lateral load carrying capacity of the track, however techniques for making this type of measurement have not yet been developed.
- Data obtained in measuring track structures is highly dependent upon the measurement technique. To obtain valid data it is important to use or simulate actual operating conditions.
- Several analytical and experimental studies, completed, in progress, or projected, require data on track structure. The requests are diverse enough so that any equipment developed to obtain data for these types of studies should be very versatile.
- There is no equipment currently available for use by the FRA or railroads which is suitable for making the required measurements of track structural parameters.

Although there is a consensus that vertical track stiffness is indicative of track load carrying capacity and the need for track maintenance work, there are no known applications of this technology in planning track maintenance or setting load limits -- primarily because there is no equipment available which will provide the necessary track stiffness data. To develop this technology a system is needed which will measure vertical track stiffness over long sections of track at normal operating speeds. The data produced by the system would be used initially by the railroads and/or by the FRA to correlate stiffness data with maintenance data in order to develop criteria for recognizing load limits and planning track maintenance, and later to actually develop maintenance and load limit data.

It is expected that development of the technology for evaluating track condition and planning maintenance could proceed in a manner similar to that which occurred in the development of the FRA rail geometry measuring system. Following construction of the original system, data were supplied to railroads and their comments on its usage were used in developing improved data acquisition techniques in order to increase the usefulness of the data.

In order to provide the required track data, it was concluded that two basic types of equipment are required -- one which will measure the vertical deflection of the track over long distances at normal operating speeds by applying a known vertical load, and a second which will measure a large number of track parameters at a limited number of locations with the vehicle stationary but with vehicle motion simulated.

To provide these capabilities it is recommended that a system be constructed using a surplus locomotive for the basic vehicle structure. To make the vertical load-deflection (stiffness) measurements from the moving vehicle the trucks would be modified by removing the center axle and by adding air springs to produce unequal axle loading. Removal of the center axle is necessary to provide an axle spacing large enough to obtain acceptable measurement accuracy on both continuous and jointed rail. Cylindrical instead of coned wheels may be used to insure stability and accurate measurements. In this system the track deflection at a point on the track is measured when wheels with different loads pass over that point. The difference in deflection resulting from the difference in wheel loads indicates the track stiffness at that point. In practice the measurements are made on

a continuous basis so that a continuous measurement of track stiffness is obtained as the vehicle moves along the track.

The basic system recommended for dynamic measurements made with the vehicle stationary consists of hydraulic actuators attached between the vehicle frame and rails. This system would be similar to the system shown in Figure 1, which was used in the experimental evaluation part of this program. A 35,000 pound per rail force capability would be provided, the system would be designed for rapid set up and retraction, and two lateral actuators would be used to simultaneously force both rails. This recommended system would not have the capability of developing data with the vehicle moving, but the control system would be designed so that vehicle motion could be simulated.

The motion simulation would consist of periodically loading and unloading the rail with the sequence that occurs when a train passes so that settling and other time-history effects are duplicated. The control system would also be versatile enough so that the actuator could be operated in or out of phase, with static or dynamic loads, or with different combinations of the above loadings.

Acceleration transducers could be moved to different points on the rail to obtain different types of data. The data processing system would analyze data from a single set of transducers to determine a specific track structure parameter during one data acquisition period, but by moving transducers and switching signals, all combinations of lateral and vertical motion needed for direct and cross compliances could be calculated. The data processor would be configured so that additional data acquisition channels could be added at a later time to allow simultaneous acquisition of all data.

It is estimated that the design, construction, and check-out of the recommended system would cost about \$720,000 and that it can be completed in a time period of 24 months. This system was selected based on a cost and performance trade-off study which included both simpler and more elaborate systems. Some features of the more elaborate system were (1) the capability of applying dynamic vertical forces to the track with the hydraulic exciters, and obtaining dynamic response data while the vehicle is moving, and (2) the capability of applying constant lateral loads to the track and measuring the resultant deflection while the vehicle is moving.

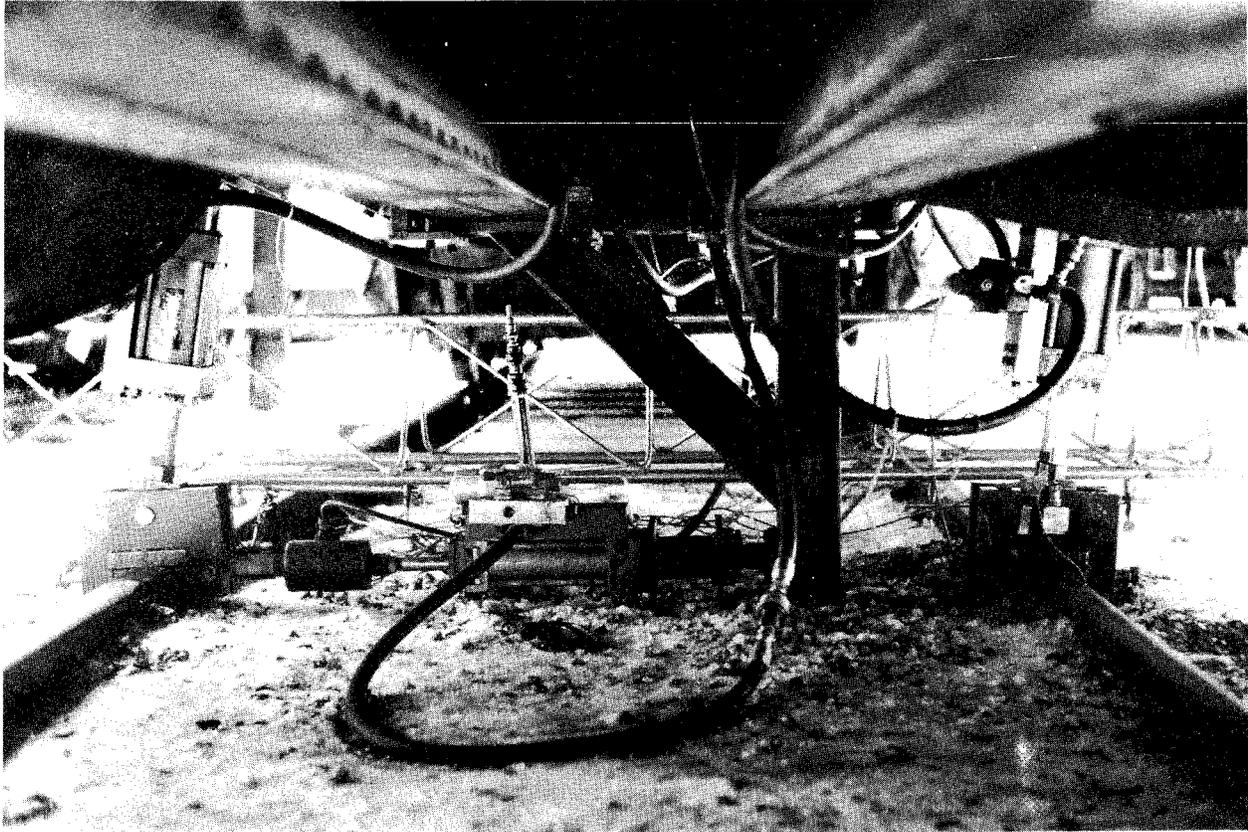


FIGURE 1. DYNAMIC COMPLIANCE MEASURING SYSTEM USED IN EXPERIMENTAL STUDY

The cost would be in the range of \$1,000,000 to \$2,000,000 for some of these more elaborate systems. The cost may not be justified at the present time because much of the same data can be obtained using the simpler recommended system and a stationary vehicle. Furthermore, an evaluation of user needs shows that current knowledge about the relationships between track stiffness and track strength is inadequate to justify the need for continuous measurements of vertical dynamic compliance and lateral stiffness.

A simpler system was not recommended because any simpler system would not be versatile enough to produce the dynamic data required at selected locations and also have the capability of producing continuous vertical track stiffness data over long sections of track.

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INTRODUCTION

During the past several years the need for greater train speeds, train loads, and reduced track maintenance costs has resulted in several research programs to study track structures, vehicle dynamic performance, and the interaction of track and vehicles. Arising from many of these programs is the recognition that the available data -- and means for obtaining data -- on the dynamic properties of track structures are inadequate. Without adequate data on the static and dynamic properties of the track structure, analyses of vehicle dynamic performance may result in vehicle designs which are unstable or overdesigned, analyses of track structures cannot be verified with experimental data, rail vehicle test and simulation equipment cannot be used with confidence to accurately simulate realistic railroad operating conditions, and track repair and maintenance cannot be most efficiently planned and implemented.

In order to develop the information needed for vehicle, track structure, and simulation equipment studies, the Federal Railroad Administration (FRA) initiated a program at Battelle's Columbus Laboratories (BCL) to develop a track dynamic compliance measuring system. This program has been divided into three phases. The Phase I work included review of measurement techniques which have been used previously, compilation of data on typical track dynamic characteristics, determination of track compliance measurement requirements, and identification of concepts for measuring track compliance.

Following the Phase I work, just described above, a supplementary track measurement task was initiated to obtain preliminary experimental compliance data that can be used as input data for Phase II. The data obtained from this measurement program included track damping properties, track compliance, natural frequencies, and variation of these parameters as a function of position along the track and track conditions. These experimental data were needed because very little information applicable to the Phase II feasibility and design study was found in the literature.

This report covers the Phase II work which included conceptual design studies and detailed evaluation of system concepts that can be developed to meet the identified measurement objectives. The Phase II work may then be followed by Phase III, a detailed design, construction, and testing phase in which a complete measurement system is constructed and evaluated.

CONCLUSIONS

As a result of the literature reviewed, discussions with investigators and the analyses completed during the program, it was concluded that:

- An indication of the vertical load carrying capacity of track and the need for track maintenance work can be obtained from vertical load versus deflection (stiffness) measurements.
- There is a major need for a technique which can be used to determine the lateral load carrying capacity of track and it appears probable to us that with further research and development that a technique using lateral load versus deflection measurements can be developed to meet this need.
- Track structural parameters are highly dependent on the data acquisition technique and to obtain data representative of specific service conditions requires that those conditions be reproduced or accurately simulated.
- At the present time, there is no equipment available for use by the FRA or railroads that is suitable for measuring required track structural parameters.

Although there is an initiative consensus that vertical track stiffness is indicative of track load carrying capacity and the need for track maintenance work, there are no known applications of this technology in planning track maintenance or setting load limits -- primarily because there is no equipment with a proven capability to make the measurements available which will provide the necessary track stiffness data. To develop this technology a system is needed which will measure vertical track stiffness over long sections of track at normal operating speeds. The data produced by the system would be used initially by the railroads and/or by the

FRA to correlate stiffness data with maintenance data in order to develop criteria for setting load limits and planning track maintenance, and later to actually develop maintenance and load limit data.

It is expected that development of the technology for evaluating track condition and planning maintenance would proceed in a manner similar to that which occurred in the development of the FRA rail geometry measuring system. Following construction of the original system, data would be supplied to the railroads and their comments on its usage would be used in developing improved data acquisition techniques in order to improve the usefulness of the data.

The system would also be used to identify sections of track where anomalies exist so those areas can be studied in more detail.

The system should also be capable of making many different types of measurements at specific locations along the track. This latter capability is required to aid in the development of improved track models, track failure criteria, and techniques for evaluation of track condition.

After sufficient work has been completed to demonstrate that track conditions can be evaluated with proven measurement techniques, either a separate measurement system which would rapidly and effectively make these measurements should be built, or the original system might be modified to convert it from a general purpose research and development system to predominantly a production test system. The modifications required for this conversion would be determined during the initial measurement program. These changes might consist of replacing transducers and electrohydraulic actuators that clamp to the rails with systems that load the rail and measure its motions through lightweight wheels.

Several configurations for loading the rails through wheels were studied and found to be feasible, but it was concluded that additional studies on track structures are needed before construction of a complex compliance measuring system could be justified. Table 1 shows some of the combinations of parameters that can be measured with different systems and the approximate cost of these systems.

TABLE 1. SYSTEM COMPARISONS

Measurement System	Approximate Cost	Track Parameter Measured				
		Vertical Static Track Stiffness	Lateral Static Track Stiffness	Direct Dynamic Compliance Under Reference Wheels	Cross Dynamic Compliance Between Reference Wheels and Arbitrary Point on Track	Cross Dynamic Compliance Between Truck Wheels
Recommended system vehicle frame with modified trucks and four hydraulic exciters from frame to rails	\$ 720,000	M-S*	S	S	S	N
Vehicle frame with modified trucks	480,000	M	N	N	N	N
Vehicle frame with four hydraulic exciters	690,000	S*	S	S	S	N
Vehicle frame with modified trucks, six hydraulic exciters and four auxiliary loading wheels	1,500,000	M-S*	S	M-S	M-S	M-S
Two vehicles, each with four hydraulic actuators and two auxiliary loading wheels	2,000,000	M-S	M-S	M-S	M-S	N

Note: S = measured with vehicle stationary

M = measured with vehicle moving.

N = not measured.

S* = requires set up with reference beam for stationary measurement.

RECOMMENDATIONS

It is recommended that a compliance measurement system be constructed on a rail vehicle which can:

- Measure the static stiffness or compliance of the track in the vertical direction while the vehicle is moving at normal traffic speed along the track
- Measure both the static and dynamic compliance of the track in the vertical and lateral directions on one or both rails with the vehicle stationary, but with vehicle motion simulated
- Be easily modified to make additional static or dynamic measurements while the vehicle is moving along the track at normal speeds.

To meet these objectives, the proposed measurement vehicle would contain two separate measurement systems. One system would measure static vertical track stiffness on a continuous basis as the vehicle moves down a section of track. The second system would consist of several electrohydraulic actuators and motion transducers which would measure both vertical and lateral dynamic track compliance at a single point on one or both rails of a section of track. Transducers which can easily be moved and attached to the rails would be provided so that different cross-coupling coefficients can be measured. It is recommended that initially these measurements be made from a nonmoving vehicle with an actuator control system designed to simulate the input force from a moving train. Provisions should also be made for modifying the system so as to obtain selected measurements with the vehicle moving.

The recommended static-stiffness measurement system uses the axle displacements of two unequally loaded trucks to calculate the track static compliance or stiffness. The displacements would be measured relative to the car body and then corrected to remove the effects of car-body motion. This system has the advantage of cancelling out any profile errors by using the measurements from two separate trucks to calculate the track static compliance.

The force needed for the dynamic compliance measurements would be produced by two vertical and two lateral hydraulic cylinders mounted under the center of the vehicle and attached to the rail head. The forces would be measured by transducers between the hydraulic actuators and rail, and motions would be measured by portable accelerometers which could be attached at different locations along the rail to obtain direct and cross-compliance data. In addition, a long reference beam and displacement transducers should be provided to obtain static or low-frequency compliance data as an absolute calibration.

The recommended system has been designed so that future expansions may be made without major retrofits. Using a building-block approach, it will be possible to expand the proposed system to enable dynamic measurements to be made on up to four load points with the vehicle moving.

To provide this capability, the recommended system would use an E8 or E9 EMD locomotive car body, underframe, and trucks for the basic vehicle framework. These units meet the general requirements for availability, structural strength, weight, and geometric factors. No extensive structural modifications would be required for the car body, although a large amount of body rebuilding would be required to provide the necessary instrumentation and working quarters. The trucks would require modifications to remove both the traction motors and the center axles to convert the trucks from three-axle to two-axle trucks, and to add loading devices to produce unequal axle loads and transducers to measure actual dynamic wheel loads.

The recommended vehicle would provide minimal living quarters. A small refrigerator, microwave oven, and table or convertible bunks would be included to provide for daytime needs and occasional overnight occupation.

TECHNICAL DISCUSSION

The nomenclature and definitions of terms used to describe the static and dynamic characteristics of railroad track can be quite confusing to persons having a varied background. Therefore, only a few descriptive terms have been selected for use in this report, and these have been used according to the definitions which follow.

Track modulus is a term commonly used in railroad engineering to describe the average elastic support of the foundation under the rail that is provided by the combination of discretely spaced ties supported by a roadbed composed of ballast on top of a subgrade. This is strictly a static parameter, with units of lb/in. per inch of rail length. It is not intended to include any dynamic effects such as frequency-dependent damping or mass. Since the measurements considered in this report are all related to the rail head and include the rail as a part of the track assembly, track modulus is not of direct interest.

Track stiffness is also used to describe the static, rather than dynamic, characteristics of track. Track stiffness as used in this report refers to the track load-deflection ratio (lb/in.) for a point load applied to the rail head, and this includes the stiffness contributed by both the rail and the foundation.

Track static compliance refers to the inverse of Track Stiffness (in./lb).

Track dynamic compliance is based on the definition that is commonly used in vibration analysis -- the complex ratio of displacement to force (includes amplitude and phase) representing the frequency-dependent transfer function for steady-state sinusoidal excitation. Dynamic compliance over a selected frequency range defines the dynamic characteristics of structural behavior, such as resonant frequencies, antiresonant frequencies, and energy dissipation (damping). Unless otherwise indicated, the term compliance, or track compliance, refers to forces and displacements measured at the rail head. Therefore, as the frequency of interest approaches zero, the track compliance is directly related to the inverse of the track stiffness. Also, the inverse of the dynamic track compliance at low frequencies is sometimes identified as the dynamic track stiffness to differentiate the results from a static and dynamic measurement.

It is important to realize that there are several other dynamic quantities found by the ratios of applied force and response variables (displacement, velocity, or acceleration) that differ only by being inverse ratios or are proportional to the exciting frequency. These include mobility, velocity/force; apparent mass, force/acceleration; apparent stiffness or dynamic stiffness, force/displacement; and impedance, force/velocity. Since all of these gave equally adequate information about track dynamics for sinusoidal excitation, it is sufficient to be able to measure any one in detail in order to determine track dynamic compliance. Therefore, a discussion of techniques for measuring dynamic displacements and force along with the term dynamic compliance is presented in this report, thereby employing an understanding of this relationship.

System Usage and Requirements

The initial phase of this program included a review of requirements for track compliance data^{(1)*}. As a part of that 1973 study, several persons working or concerned with track structures were questioned about their need for these data. Some of these people and some additional personnel were contacted again to obtain more recent information and opinions.

The major change in opinions that has occurred during the interval since the start of the program is a much greater interest in track lateral strength characteristics and a decreased interest in developing data for use in simulation studies. The reason for the increased emphasis on lateral strength characteristics is the recent significant increase in derailments due to rail rollover, misalignment, and gage widening. The railroad industry currently has a major program under way to study these problems and will require equipment to experimentally measure track lateral force versus deflection relationships in order to develop criteria and inspection techniques for measuring and predicting these phenomena. Development of data for simulation studies is currently of reduced interest because

*References are listed on page 49.

development plans for the wheel-rail simulator at the Transportation Test Center (TTC) near Pueblo, Colorado have been modified and there are no current plans for simulating track parameters with the equipment being built.

Railroad personnel have throughout the period of the program expressed an interest in using track stiffness or compliance data in efforts to minimize maintenance costs.

In general, recommended usages for a track compliance measuring system have fallen into two main categories. One of these is for research and development studies and the other is for evaluating track condition.

Research and development studies requiring use of a system capable of obtaining either static or dynamic track force-deflection measurements include:

- Track buckling
- Track load capacity in both the lateral and vertical directions
- The dynamic interaction between rail vehicles and track in both the lateral and vertical directions, including derailment causes
- Track life as a function of ballast and subgrade condition.

An example of the requirements for track compliance data in R&D work is in the Track Strength Characterization Program sponsored by the Association of American Railroads (AAR). In this program it will be necessary to measure a wide range of track parameters and to develop a compliance of stiffness measurement technique that can be used to predict track failure without damaging the track.

A second example is a Department of Transportation-Transportation Systems Center (DOT-TSC) program at The Analytic Sciences Corporation (TASC). In this program TASC is conducting vehicle parametric studies with the primary object to establish acceptably safe behavior limits due to dynamic vehicle response to track irregularities. In this study they require track compliance data in order to accurately model the vehicle-track interaction forces and motions.

A third example is the Facility for Accelerated Service Testing (FAST) program at the TTC where a rail vehicle has been modified by adding hydraulic actuators and transducers similar to the recommended system. This equipment is used in conjunction with other TTC track studies to calibrate track instrumentation.

There are also many studies under way to study and develop improved track structures or to improve existing track through stabilization techniques. Ultimately, it will be necessary to measure the stiffness or compliance of these structures in order to identify areas for improvement and to assess the performance of these improved structures.

For evaluation of track-structure condition, the functions which might possibly be performed using static stiffness or dynamic compliance data techniques are:

- Schedule overall track maintenance and rebuilding programs
- Evaluate effectiveness of track maintenance and rebuilding programs
- Identify locations for spot maintenance and subgrade stabilization and refurbishment
- Determine seasonal load-carrying capacity of existing track to avoid excessive rail failures or track deterioration from overweight loads
- Detect potential track buckling and rail rollover problems
- Measure vehicle dynamic excitation properties
- Detect defective ties or fasteners.

In discussing use of a compliance measuring system for evaluation usage, personnel from several roads indicated that they believe this type of information would be useful in planning track maintenance.

One railroad contacted has already initiated a feasibility study, similar to this study, of ways to measure track stiffness with the ultimate goal of using stiffness data in planning their track maintenance programs. Another railroad with many 100 ton cars and much light rail has experienced interest in measuring track stiffness, using the track stiffness data to calibrate rail stresses and thereby determine allowable load limits for their heavily-loaded, light-weight track.

Other railroad personnel have also indicated that they believe that it should be possible to use track stiffness data as a management tool in planning expenditures for track maintenance, for setting load limits, determine when heavy loads can be moved on their lines, and for identifying sections of track where localized maintenance or ballast stabilization is required. The concept of detecting defective ties or fasteners was also discussed. This capability would be considered desirable, but the

feasibility of accomplishing this was considered questionable. Research and development and field demonstrations would be required to establish this capability.

In addition to the above, two general areas of R&D and condition evaluation usage, a third potential future application, is to produce data for use in advanced rail-vehicle simulation facilities. On the basis of existing plans there is no known current need for these data in the U.S.; however, it is probable that at some future date these data will be needed to accurately simulate the interaction between track and vehicle. These types of data will probably ultimately be required because the techniques of using recorded rail profile errors as signal inputs will probably not accurately produce the input forces and motions that would be obtained with different types of vehicle suspensions and with different simulated speeds. In other words, the rail-geometry profile errors are a function of track compliance, vehicle speed, and truck design, and changes in any of these parameters will affect the wheel forces and motions developed at the wheel-rail interface. All of these parameters will ultimately have to be simulated to obtain satisfactory test results when a wide range of vehicle parameters and speeds are studied.

Although there is a general consensus that vertical track stiffness measurements can be used to identify poor quality track and thereby effectively plan maintenance and set load limits there are no known applications of this technology at this time -- primarily because there has not been any equipment available which is suitable for obtaining track stiffness data over significant lengths of track. To fully develop this technology it will be necessary to (1) measure track stiffness on selected track, (2) obtain maintenance and traffic data from the railroads for that track, and (3) develop a correlation between stiffness measurements, loads carried, and maintenance costs. The result of this type of study should be the development of criteria which can be used by the railroads to identify maintenance costs as a function of loads and traffic volume based on measured vertical track stiffness.

Although it is probably feasible to determine the lateral load carrying capacity of track with lateral stiffness or compliance measurements, little work has been completed to clearly define how lateral compliance measurements should be made to determine a safe lateral track load without damaging the track.

Because little is currently known about how lateral track strength should be measured, a dedicated system for lateral track strength measurements cannot be justified at the present time. Instead general purpose force generating and deflection measuring equipment is required to perform lateral track strength studies. The AAR, for their program, is recommending a phased series of studies, starting with a system capable of generating vertical, longitudinal, and lateral forces, measuring the resultant deflections at a stationary point on the track, and ultimately, at the completion of the program, ending with a vehicle capable of measuring lateral track strength while moving at normal traffic speeds.

Because several different configurations of this equipment may be required to make many different types of measurements for the initial series of studies in a typical R&D program, it is not considered practical to initially build this equipment to acquire the data while the vehicle is moving. Instead, the equipment used for initial studies in R&D programs should be sets of hydraulic exciters and transducers which can be attached between the track and vehicle to make the desired compliance measurements. After sufficient work has been completed to define a system that will produce the data necessary to evaluate specific track conditions, it should then be possible to modify the vehicle to obtain that data while moving at normal speeds. Even though a moving R&D type measurement system cannot be justified initially, a system which can be rapidly transported, set up, operated, retracted, and moved off the track is required to avoid excessive traffic delays. Railroad personnel have indicated that for work on main line track during normal working hours, it would usually be necessary to set up the system, acquire the desired data, and move off the test site in 1 hour or less.

The importance of minimizing interference with normal rail traffic depends on the ultimate use for the system. If the system is used frequently and/or for long time periods, this would be an important factor. For very occasional work on main line track it was indicated that it should be possible to schedule night or weekend tests for periods in excess of an hour.

In the work completed earlier in this study it was found⁽¹⁾ that track compliance is a function of loading history, and that either the data must be obtained from a moving vehicle or a moving vehicle must be simulated by the measurement system to provide compliance data representative of that encountered by moving rail vehicles. This required simulation consists primarily of cyclically loading and unloading the track in a manner similar to that which occurs when a train passes over a section of track.

For vehicle and track dynamic studies, the measurements considered to be of primary interest are the driving-point dynamic compliance at wheel loading points, and the transfer dynamic compliance between wheel loading points. It is desirable to be able to measure these dynamic compliances in both the vertical and lateral directions in the speed range from zero up to the highest train speeds that might be encountered in practice.

The frequencies of primary interest are 0 Hz and 10 to 100 Hz. There is a secondary interest in the frequency range 100 to 500 Hz, and a slight interest in higher frequencies up to about 1000 Hz. The compliance (or stiffness) at the 0 Hz frequency defines the static load-deflection characteristics of the track structure. This is important because it is the parameter most often used by both track-structure and vehicle-suspension designers, and probably provides the most information about the condition of the track structure. Other track-structure parameters will almost always be related to track compliance. Also, measurement of the 0 Hz, or static, compliance will usually provide a guide to the values of compliance that might be expected at other frequencies.

Measurements in the frequency range of 10 to 100 Hz are important because data in the literature indicate that with conventional track structures, the track structure's natural frequency with a typical vehicle unsprung mass is about 30 Hz. With new track structures and/or vehicle suspension systems, the natural frequencies will probably be increased, possibly to the 50 to 60 Hz range. The dynamic track compliance usually changes significantly at the track natural frequency, and the rapid changes in compliance can have a significant effect on measurements of track structure and vehicle stresses and motions.

There will be higher order resonant frequencies above 100 Hz where specific elements of the track and vehicle system may interact, but because the frequency is much higher than the fundamental natural frequency, the complete ballast, tie, and subgrade system will not respond. It is, of course, desirable to know how all parts of the system interact; however, because of the limited number of elements that interact at higher natural frequencies, these modes can usually be adequately studied in the laboratory.

Design Objectives

There are a number of specific design objectives which must be met in order to construct an accurate static compliance measuring system. The critical specifications for this measurement system apply to the vehicle configuration. A vehicle is needed which has an overall vehicle weight of between 160,000 to 240,000 pounds gross vehicle weight. This vehicle weight is determined by the fact that maximum allowable wheel loads are about 35,000 pounds and up to four vertical hydraulic exciters may be used under the vehicle. The vehicle must be heavy enough to produce the required static forces but not so heavy that its wheel loading is excessive. In order for the proposed measurement system to work correctly, each axle of the test vehicle must have a specific load applied. The trucks of the vehicle must have two axles. The axle spacing on the trucks must be as large as possible, preferably over 11 feet. The trucks must be constructed such that some type of modification could be made to one of the trucks so the axles may be loaded unequally.

If the vehicle is also to be used to measure dynamic track compliance, there are several additional requirements that it must satisfy. The vehicle must have a large amount of clearance under its center so that hydraulic loading cylinders may be mounted. The frame of the vehicle must be stiff to minimize car-body bending under dynamic forcing. The distance between the two center axles should be at least 25 feet.

The recommended specifications for the forcing equipment used in measuring the dynamic compliance must also be followed. Results from the measurement program showed the necessity for being able to load the track structure to the maximum wheel loads encountered in actual service. The maximum vertical wheel load that can be applied will be 35,000 pounds. This load can be applied to both rails simultaneously or to each rail individually so each rail is free to move independently. For the lateral direction, the maximum loading capacity will also be 35,000 pounds. This yields a lateral force/vertical force ratio (L/V) of 1. A L/V ratio of 1 has been measured in actual service. (2)

Also shown by the measurement program was the need for the load to be cycled, from zero to maximum load, then back to zero as shown in Figure 2 at a rate simulating a passing train. The rate at which the load is cycled simulates the train speed. The maximum simulated train speed is limited by the flow rate of the hydraulic system. In order to keep the size of the hydraulic system and power supply to a reasonable size, the maximum simulation speed will be limited to about 50 mph.

Superimposed on the cyclic load will be a dynamic excitation consisting of random, pulse, or sinusoidal excitation. The excitation frequency of the system will be up to about 100 cps or higher in both the vertical and lateral directions. The general arrangement of the hydraulic loading cylinders is shown in Figure 3.

At present, the effect of a second axle within the load-affected zone on the compliance measured by a given axle is unknown. So that this effect could be investigated at a later date, the vehicle should be designed to allow for adding a second pair of loading cylinders, giving the basic configuration shown in Figure 4. This second set of cylinders would also allow for measurement of cross compliance between the two loading points on one rail.

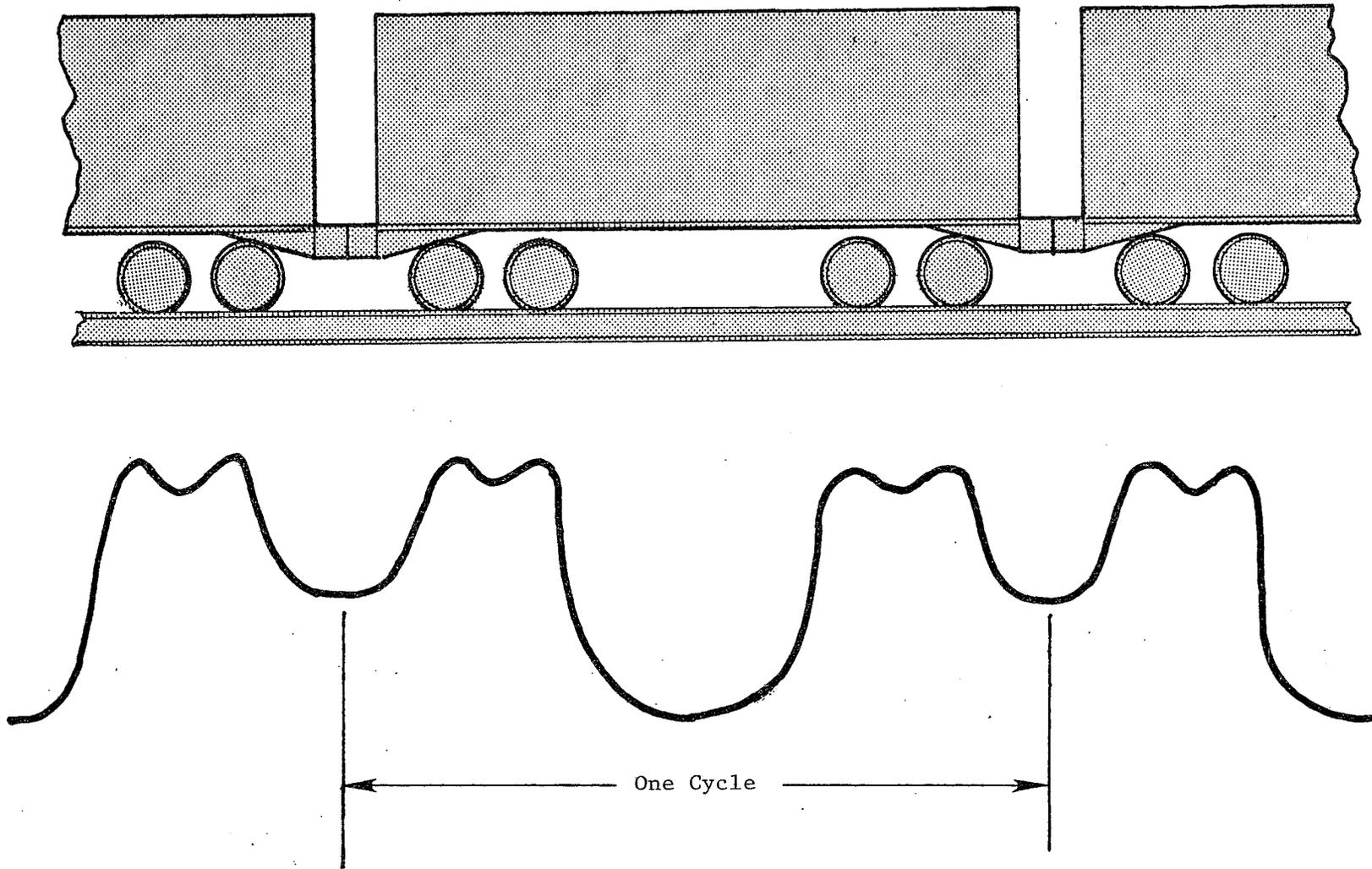


FIGURE 2. CYCLIC LOADING OF THE TRACK STRUCTURE DUE TO A PASSING TRAIN

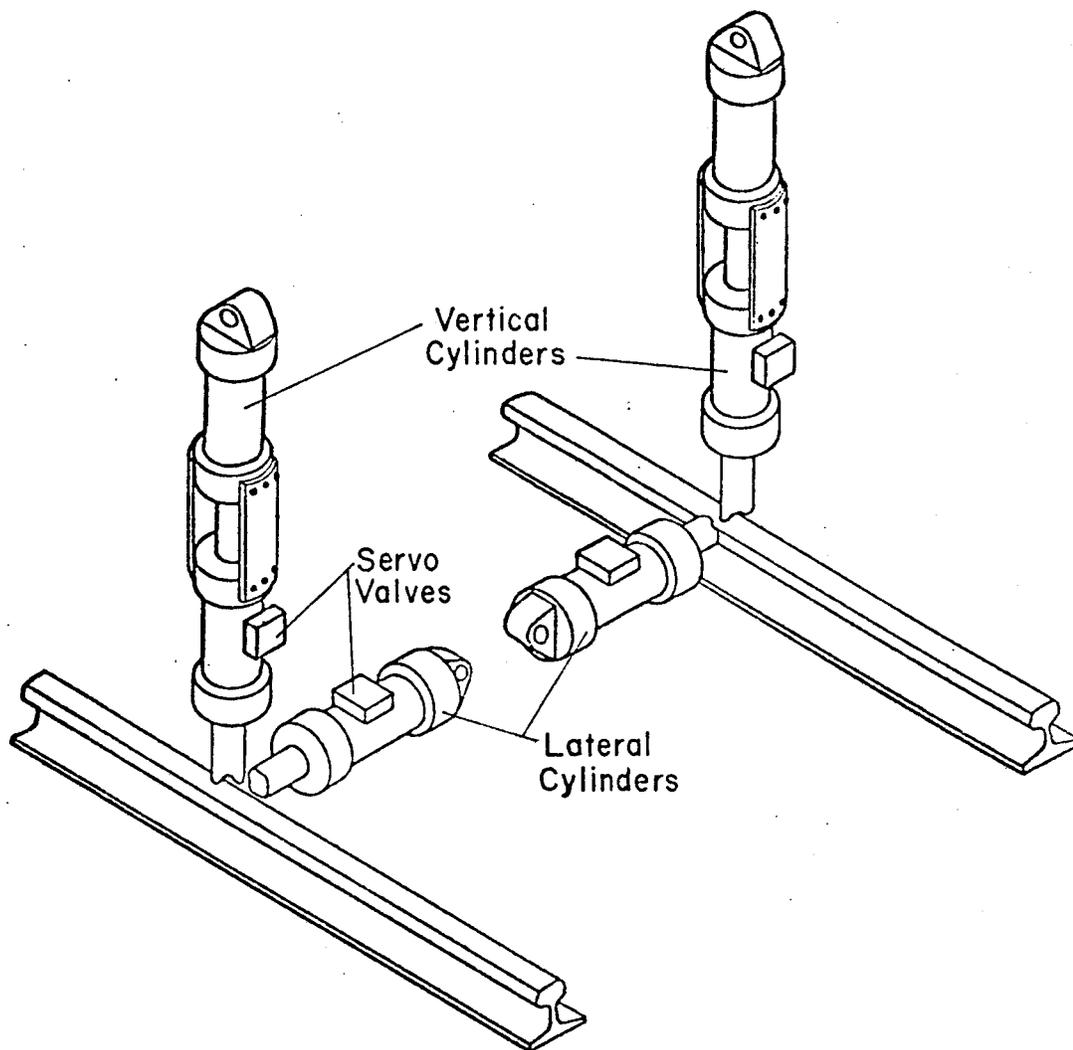


FIGURE 3. SINGLE LOADING SET (TWO MODULES)
SIMULATING A SINGLE AXLE

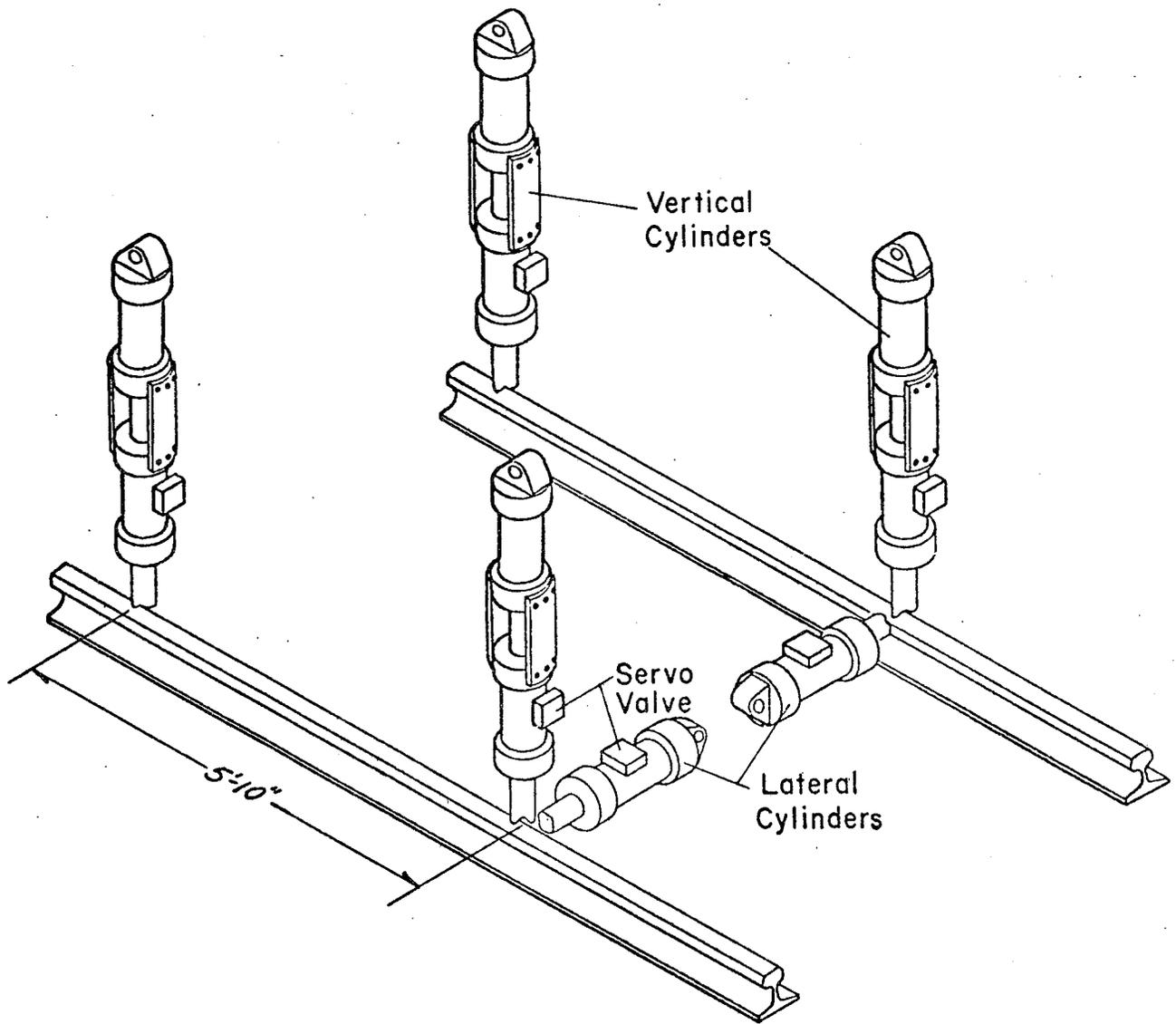


FIGURE 4. DUAL LOADING SET (FOUR MODULES) SIMULATING TWO AXLES OR ONE TRUCK

In order for the loading system mounted in the center of the car to be in alignment with the rails on a curve, the system must shift laterally with respect to the car. For a 10-degree curve and 43-foot truck spacing, the lateral shift is 6 inches. The system should be designed to shift at least +6 inches laterally.

All equipment must be constructed within AAR Clearance Plate C2 shown in Figure 5.⁽²⁾ The car must have standard AAR couplers and air brakes and meet the required compressive strength to be capable of transport in railroad freight or passenger trains.

Recommended System Design

Several types of vehicles and measurement systems were evaluated. A single system was chosen and this is described in this section. The alternative systems are described and illustrated in Appendixes C and D, respectively.

Static Measurement System

The recommended method for measuring track static stiffness or compliance is a combination of the best features of several of the methods detailed in Appendix C. The recommended method uses the loading configuration shown in Figure 6. The wheel displacements are measured relative to the car body and are measured for each truck over the same point on the track. The car-body pitch angle is measured with an inertial reference system having a sufficiently long time constant that the change in car-body angle occurring during the time required for the two sets of trucks to pass over a point on the track is accurately measured. For the system shown in Figure 6, the deflection of the track under the measurement wheels is given by:

EQUIPMENT DIAGRAM FOR LIMITED INTERCHANGE SERVICE *

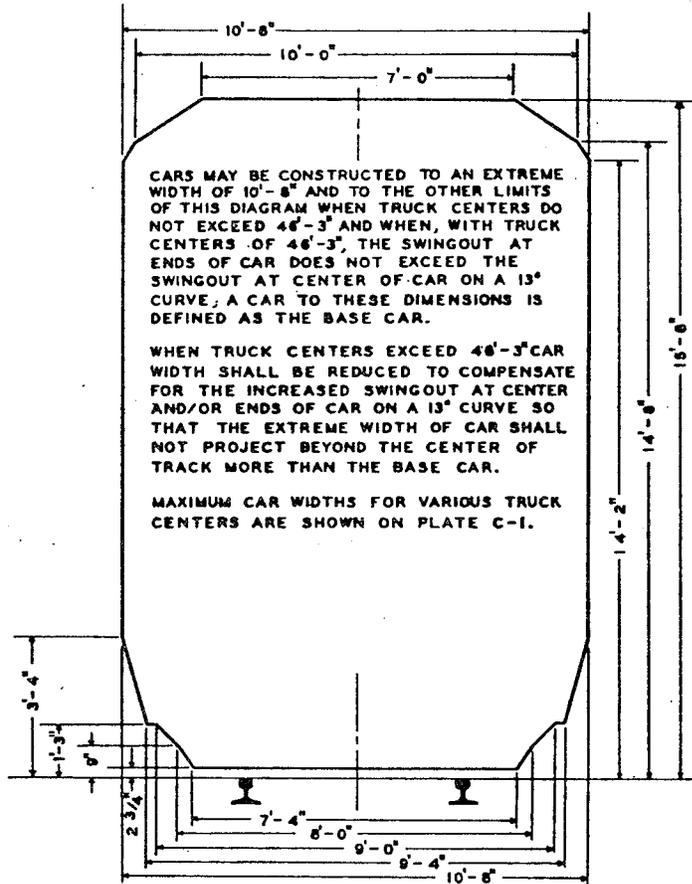


FIGURE 5. AAR CLEARANCE PLATE C

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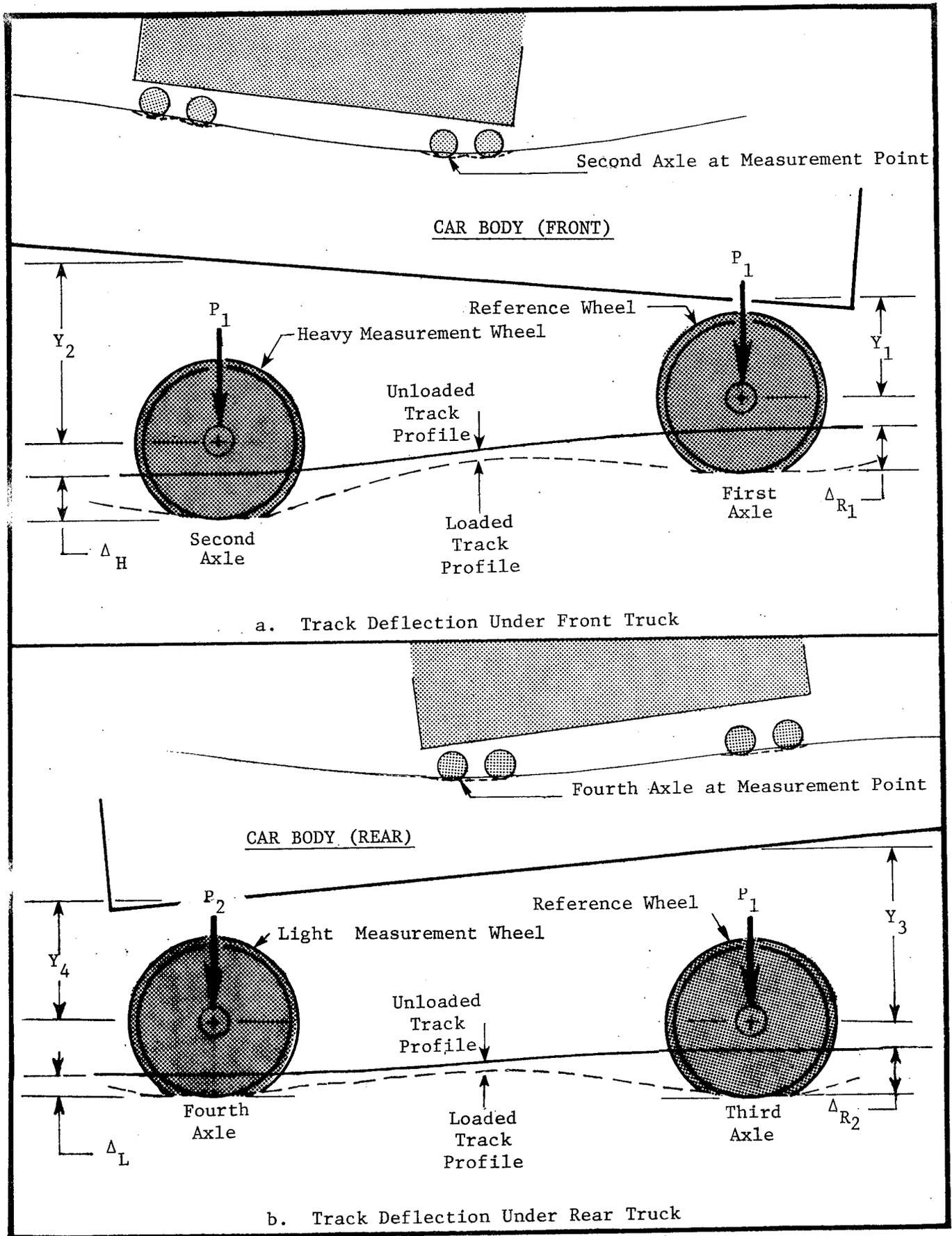


FIGURE 6. RECOMMENDED STATIC-STIFFNESS MEASURING SYSTEM

$$\Delta_H = Y_2 - Y_1 - \phi_1 L + \Delta_{R1} \quad (1)$$

and

$$\Delta_L = Y_4 - Y_3 - \phi_2 L + \Delta_{R2} \quad (2)$$

where

Δ_H = track deflection under the heavily loaded measurement wheels (second axle) when they are at the measurement point

Δ_L = track deflection under the lightly loaded measurement wheels (fourth axle) when they are at the measurement point

ϕ_1 = car-body pitch angle at the time the second axle is over the measurement point

ϕ_2 = car-body pitch angle at the time the fourth axle is over the measurement point

Δ_{R1} = track deflection under the first axle when the second axle is over the measurement point

Δ_{R2} = track deflection under the third axle when the fourth axle is over the measurement point

Y_1, Y_2, Y_3, Y_4 = displacement between axles and the car body as shown in Figure 6

L = spacing between axles.

If the loading on the second and fourth axles is identical, Δ_{R1} will be approximately equal to Δ_{R2} ; assuming $\Delta_{R1} = \Delta_{R2}$ and subtracting Equation (2) from Equation (1) gives:

$$\Delta_H - \Delta_L = Y_2 - Y_1 + Y_3 - Y_4 - L(\phi_1 - \phi_2) \quad (3)$$

where $\Delta_H - \Delta_L$ is the difference in track deflection due to the difference in wheel loading.

Using influence coefficients, the following expression may be obtained:

$$C_{11} - C_{12} = \frac{\Delta_H - \Delta_L}{P_1 - P_2} \quad (4)$$

where

C_{11} = compliance value obtained at an axle due to forces transmitted to track through that axle

C_{12} = compliance value obtained due to forces transmitted to track through an adjacent axle

P_1, P_2 = loads as shown in Figure 6.

C_{11} in the above equation is the static compliance of the track. To obtain accurate track static compliance measurements with the system shown in Figure 6, it is necessary that either C_{12} be small relative to C_{11} , or compensation must be made to account for C_{12} . Figure 7 shows the ratio of C_{12}/C_{11} versus truck length for $1/\beta = 50$ inches.

The parameter $1/\beta$ is a measure of the track stiffness, which is a function of the rail weight and track modulus. Typical values for $1/\beta$ are between 30 and 70 inches.

Figure 7 shows that for the typical track stiffnesses considered, C_{12} will be less than 10 percent of C_{11} for axle spacings greater than about 8 feet. For the 14-foot recommended axle spacing, C_{12} is calculated to be less than 4 percent of C_{11} both at joints and on continuous rail for all values of $1/\beta$ between 30 and 70. Because of the low influence of one axle on the adjacent axle with a 14-foot axle spacing, Equation (4) then reduces to:

$$C \approx \frac{\Delta_H - \Delta_L}{P_1 - P_2}, \quad (5)$$

where C is the static compliance, and a correction factor is not required.

A feature of the recommended system is its ability to cancel track-geometry profile errors. With the system shown in Figure 6, the track profile error is measured by both trucks of the vehicle and by subtracting the measurements made by the first truck from the measurements made by the second truck, the deflections produced by rail profile errors are canceled. The resulting measurement obtained is the deflection of the track resulting from the difference between the loads P_1 and P_2 .

A potential source of serious error with this measurement system is the error introduced by car-body pitch, since the measurements are made relative to the car body. To eliminate this error source the car-body pitch angle is measured and the car-body pitch angle added to the deflection signal as shown in Equation (3). At a speed of 20 mph, the time required for a car with a 43-foot center plate spacing to pass over a point on the track is about 1.5 seconds. Measurement of changes in pitch angle that occur during a 1.5 second period is well within the state-of-the-art using either a gyro or with accelerometers at each end of the vehicle. To measure pitch angle with accelerometers, the acceleration of each end of the vehicle is measured, the signals are combined,

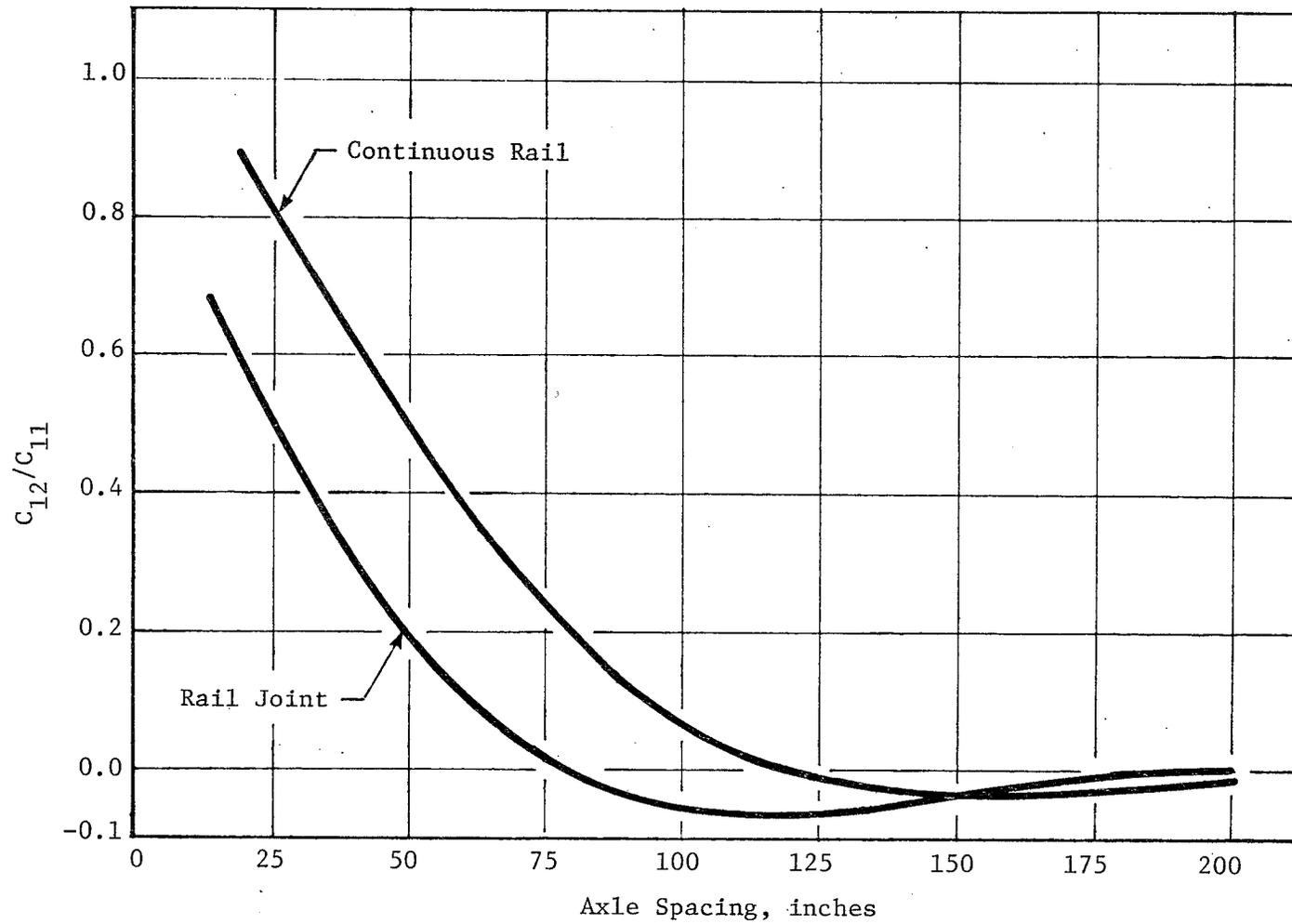


FIGURE 7. C_{12}/C_{11} VERSUS AXLE SPACING FOR $1/\beta = 50$ INCHES

and the resulting signal is double integrated. Due to the ratio of overall car-body length to truck length, the accuracy of the correction factor is better than the accuracy of the double-integrated signal by at least 4 to 1.

Other error sources are wheel eccentricity and dynamic changes in wheel loading that occur during high-speed operation on track with large geometry errors. To minimize the effect of wheel eccentricity, a phase-locked loop filtering technique will be used where a sinusoidal signal at a frequency equal to the wheel rotation speed is added to the measured signals to cancel the signal produced by wheel eccentricity errors. In other words, a signal at the same frequency, and with the same magnitude and phase as the error signal caused by wheel eccentricity is substrated from the total displacement signal to eliminate errors due to wheel runout. To minimize the errors produced by dynamic force changes, forces transmitted to each axle will be measured; these will then be used to compensate for dynamic force variations.

Calibration of the system will consist of a simple static displacement calibration of the transducers that measure wheel position, and a dynamic calibration, probably with an eccentric crank mechanism, of the accelerometers or gyro that measures the body pitch motions. Determining the absolute zero of the system will easily be accomplished by adjusting the wheel loads so that they are equal. This will be accomplished by turning off the air to the air springs that develop the unequal axle loads.

Using this method with wheel displacement accuracies of $\pm .005$ inch and with $P_1 - P_2 = 20,000$ pounds, the compliance values measured are expected to be within ± 10 percent of actual values for nominal track with a stiffness of 200,000 pound/inch, both on continuous welded rail and at joints.

Dynamic Compliance Measuring System

The recommended system uses the car as a reaction beam and mass, and a relatively lightweight fixture applies vertical and lateral loads to the rail head. The system is designed for rapid lowering of the shaker system to the rail head, gathering data, and retraction of the system for

moving. The recommended system is of modular design. There are two similar modules that will fit in the main frame to give two loading points. In this manner, the system can, if desired, be expanded to four loading points at a later date. Figure 8 shows the design for this system. The figures show one module (one loading system, vertical and lateral). This module would also be used for the other loading points with the possible exception that there is no need for lateral cylinders in a second loading set (see Figure 4). In order to obtain wheel loads of 35,000 pounds and to keep flow rates reasonable, two loading cylinders are used instead of one large-bore cylinder. Figure 4 shows the cylinders stacked on top of each other with each cylinder supplying one-half of the 35,000 pounds. The required flow rate using this configuration is one-half that of a single large-bore cylinder. Mass cancellation is not necessary in this design because the mass of the loading fixture that is between the load cell and rail head is negligible compared with the system mass. The load cell and accelerometer will be mounted on the fixture that is against the rail head.

When moving short distances, the fixture pulls up 4 inches and is within clearance Plate C. For long-distance traveling, the system is pulled up into the car 14 inches so each fixture is in the shadow of an axle to minimize damage in shipping.

To keep the dynamic forcing cylinders normal to the track when the vehicle is in a curve it is necessary to shift the forcing modules laterally. The forcing modules are mounted on a carriage which may shift laterally. The position of the carriage is controlled by means of a single hydraulic cylinder and 4-way valve. The carriage is centered over the rails manually by the operator before each set of test data is taken.

Car Type

Car Modification

An E8 or E9 EMD locomotive satisfies the design requirements for both the static and dynamic compliance measuring system. These units are from 20 to 30 years old. A large number of these models were built and consequently there are a large number of the units being scrapped as they are replaced by the railroads. A general outline of a complete unit as

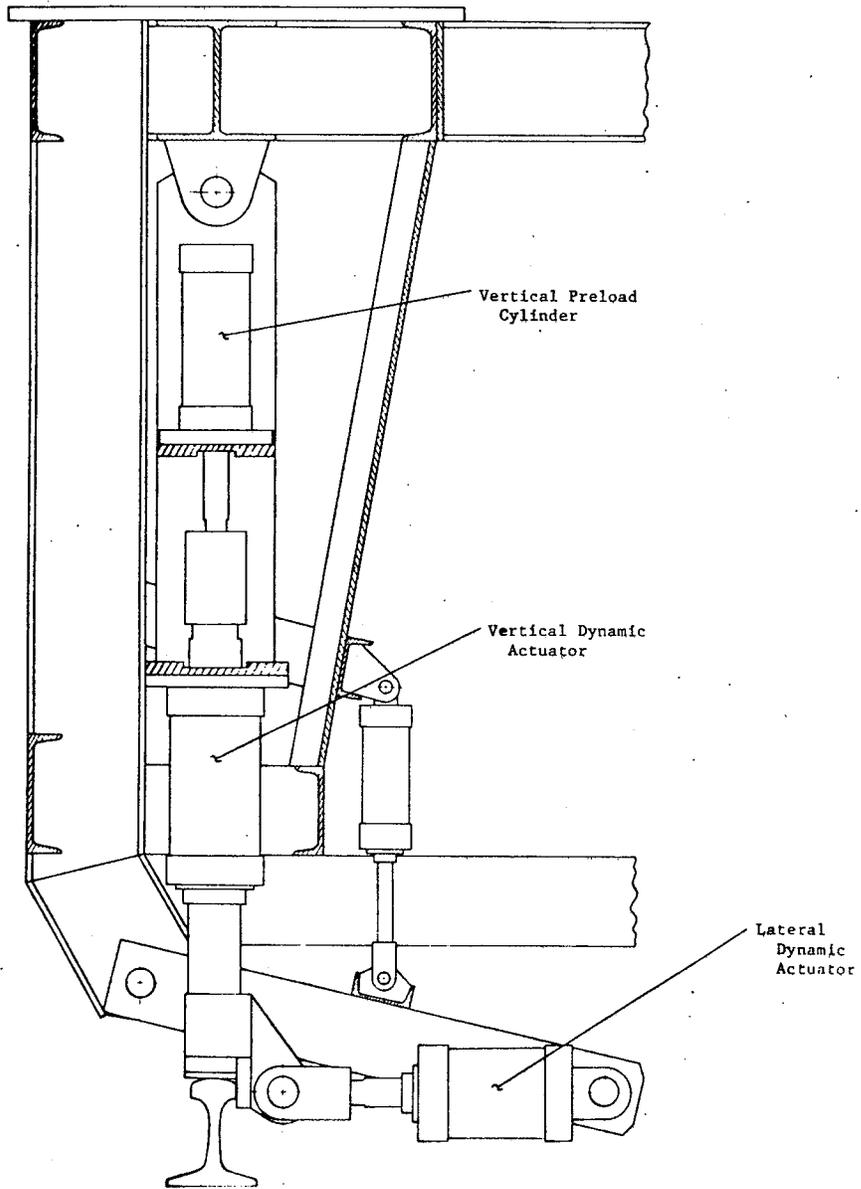


FIGURE 8. DYNAMIC EXCITATION MODULE

shown in Our GM Scrapbook⁽³⁾ is reproduced in Figure 9. The gross weight of this unit is approximately 334,000 pounds. For the proposed use, most of the mechanical equipment would be removed. The weight of the stripped vehicle is approximately 150,000 pounds. The weight is ideally suited to the requirements for the proposed system. After modification, the vehicle weight should be within the desired 160,000 to 240,000-pound range.

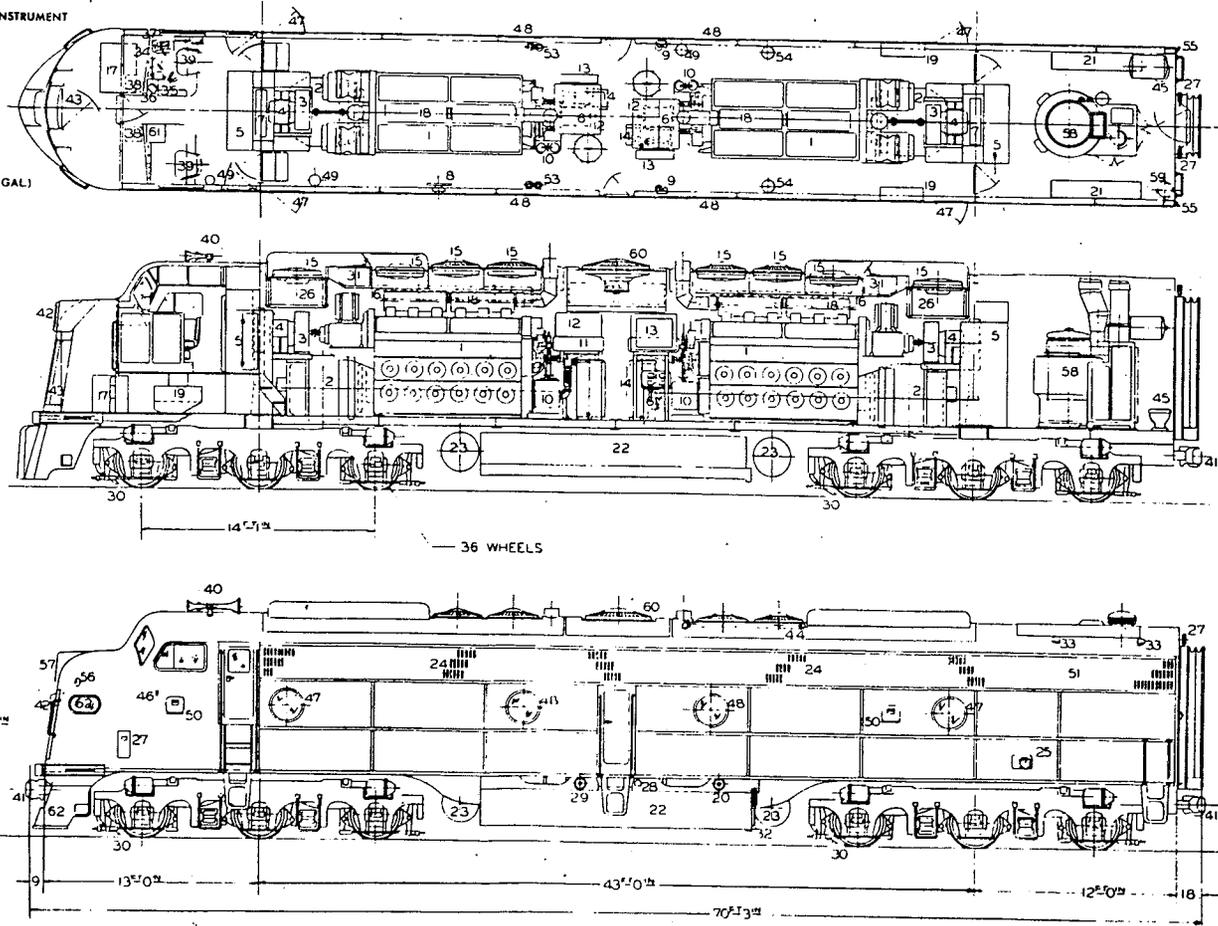
The trucks on this locomotive are 3-axle trucks with a load-equalizing suspension. With this equalizing suspension it would be a relatively simple modification to remove the center axles from the trucks and convert into 2-axle trucks with 14 feet between axle centers. The distance between the two center axles is 29 feet. This is again well suited to specifications. After the traction motors are removed from the trucks, there is ample room for mounting the necessary hardware for applying the unequal axle loads needed with the static-compliance measuring system. A sketch of the proposed system for providing unequal axle loads is shown in Figure 10.

The underframe of this type of locomotive is made up of two large I-beams running longitudinally the length of the locomotive. The car body is made up of a 4-inch WF beam truss framework system covered by flat plates. The basic structure is very stiff and should not require reinforcing to limit car-body bending. The car-body bending and suspension deflection must be minimized to keep the size of the hydraulic excitation system for the dynamic compliance measurements reasonable and to minimize vibration in the vehicle when applying dynamic forces to the track.

A major factor in the design of the hydraulic system is the maximum stroke of the vertical cylinders at the rates necessary to simulate a passing train. The majority of the stroke is being used in the deflection of the rail due to the applied load, the deflection of the car body in bending, and the deflection of the car suspension system due to removal of some of the car weight from the suspension system. Additional car-body suspension damping may be necessary to limit car-body resonant vibration motion during dynamic forcing. This damping would be added by using hydraulic cylinders between the axles and vehicle frame. Solenoid valves would be used to add this extra damping only while performing tests.

E9

- 1 ENGINE—EMD MODEL 12-567C
- 2 MAIN GENERATOR AND ALTERNATOR —EMD MODEL
- 3 GENERATOR BLOWER
- 4 AUXILIARY GENERATOR
- 5 CONTROL CABINET
- 6 AIR COMPRESSOR
- 7 TRACTION MOTOR BLOWER
- 8 HAND BRAKE
- 9 FUEL TANK VENT WITH FLAME ARRESTOR
- 10 LUBE OIL FILLER
- 11 LUBE OIL COOLER
- 12 ENGINE WATER TANK
- 13 ENGINE CONTROL AND INSTRUMENT PANEL
- 14 LOAD REGULATOR
- 15 36" FAN AND MOTOR
- 16 RADIATOR
- 17 AIR-BRAKE RACK
- 18 EXHAUST MANIFOLD
- 19 SAND BOX
- 20 FUEL FILLER
- 21 BATTERIES
- 22 FUEL TANK (1200 GAL.) AND WATER TANK (1350 GAL.)
- 23 MAIN AIR RESERVOIR
- 24 AIR INTAKE—SHUTTERS AND GRILLE
- 25 BATTERY CHARGING RECEPTACLE
- 26 ENGINE ROOM VENTILATOR
- 27 LIFTING LUGS
- 28 EMERGENCY FUEL CUTOFF
- 29 WATER FILLER
- 30 SANDING NOZZLES
- 31 A.C. CONTACTOR CABINET
- 32 FUEL TANK SIGHT GLASS
- 33 BATTERY BOX VENT
- 34 INSTRUMENT PANEL
- 35 CONTROLLER
- 36 SPEEDOMETER RECORDER
- 37 AIR-BRAKE STAND
- 38 CAB HEATER
- 39 SEAT
- 40 HORN
- 41 COUPLER
- 42 HEADLIGHT
- 43 DOOR—PLAIN
- 44 AIR COMPRESSOR AFTERCOOLER
- 45 TOILET
- 46 BLUE FLAG BRACKET
- 47 HINGED SASH
- 48 FIXED SASH
- 49 FIRE EXTINGUISHER
- 50 SAND BOX FILLER
- 51 VENTILATING SHUTTERS
- 52 NUMBER BOX
- 53 WATER TANK VENT
- 54 M. R. PIPELINE FILTER
- 55 MARKER LIGHT BRACKETS
- 56 CLASSIFICATION LIGHT
- 57 SIGNAL LIGHT
- 58 STEAM GENERATOR
- 59 LIQUID WATER TREATMENT
- 60 DYNAMIC BRAKE HATCH OR WATER TANK — 600 GAL.
- 61 WATER COOLER
- 62 EXPOSED COUPLER PILOT
- *MODIFICATIONS



29

141 142

FIGURE 9. E8 AND E9 LOCOMOTIVE DIMENSIONS
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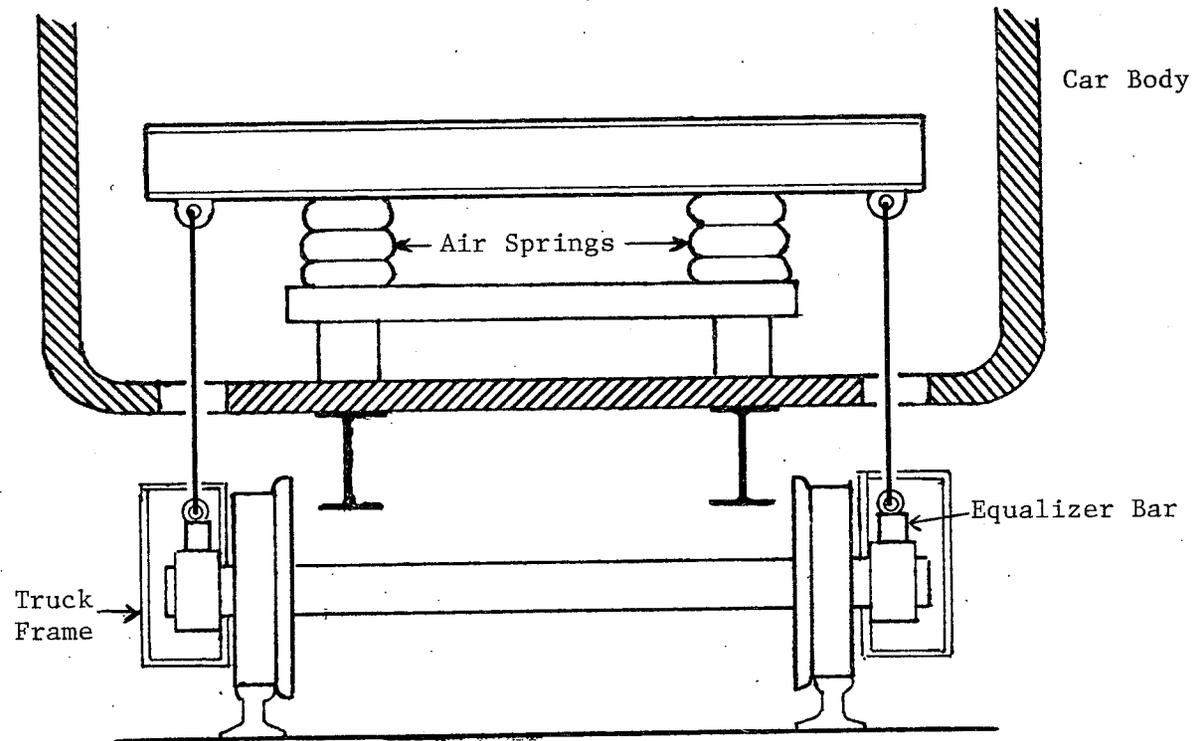


FIGURE 10. PROPOSED SYSTEM FOR PROVIDING UNEQUAL LOADS ON A TRUCK BY TRANSMITTING AXLE LOAD TO CAR BODY

Mechanical Equipment Room

A portion of the measurement vehicle will be partitioned off into a mechanical equipment room. In this room there will be the hydraulic system, a motor generator set, and the heating and cooling equipment. The hydraulic system power is supplied by a 60-gpm diesel-driven pump. This is based on up to four loading points being driven simultaneously. The motor generator set will also be diesel driven, but by a separate engine.

There will be an automatic fire-protection system in this compartment, with a fire-detector system that can shut down the system, extinguish the fire, and not harm the equipment.

Instrumentation Room and Living Quarters

The instrumentation room and the living quarters have essentially the same requirements for ventilation, air conditioning, lighting, and isolation from noise and vibration. Because of these similarities, the two spaces will share a common enclosure, constructed on an isolated base. A single air handler and plenum will circulate conditioned air through both spaces. A rough estimate of the air-conditioning requirement is 80,000 Btu/hr. Expressed in tons per day, this is equivalent to 6-2/3 tons. An air-conditioning system with 7-1/2-ton capacity should be adequate to handle maximum cooling loads. Space heating can be accomplished by circulating engine cooling water through a finned heat exchanger located in the air handler of the air-conditioning system.

Vibration isolation will be provided by constructing the enclosure on a concrete slab isolated from the car body by appropriate damped spring isolators.

The living quarters will be equipped with a microwave oven, a small refrigerator, a sink, an electric pyrolytic toilet, and a table or convertible bunks for occasional overnight habitation.

Control Systems

In the proposed system it is planned that electrohydraulic systems will be used to control all of the vertical and lateral actuators which apply the excitation forces to the rail. The functions that have to be performed with these systems are as follows:

- Start up the system and bring the actuators under position control without generating excessive forces or motions
- With the actuators under manual position control, extend the actuator until the loading pad or wheel is preloaded against the rail
- Automatically switch over from position to force control at predetermined force levels without generating excessive dynamic forces
- Control actuator forces for the duration of the programmed test
- Switch from force to position control at the conclusion of the test without generating excessive dynamic forces
- Retract the actuators to the store position under manual position control
- Shut the system down without generating large forces or motions

In addition to the above operational functions, the system must be equipped with safety interlocks to prevent operation when the carriage is not properly positioned, and with limit switches and comparator circuits to automatically shut the system down in case of a malfunction of the control system. Other manual safety devices required on the system are manually adjustable relief valves to limit actuator forces that can be developed to force levels just slightly higher than required for testing.

The technique to be used to switch from position to force control is one where the difference between the force being developed by the actuator while under position control is compared with the command force level set on the force-control circuit. When the actual force is approximately equal to the set force, comparator circuits automatically switch the control from the position-control circuits to the force-control circuits.

In switching from force control to position control, the command force level would be set to a reasonable level and the displacement command would be increased to a value greater than that actually measured. The displacement command would then be slowly decreased and when the actual displacement was approximately equal to the command displacement, comparator circuits would automatically switch the system from force control to displacement control.

Calculations and experience with the prototype system used in the measurement part of the program⁽¹⁾ has shown that high, fundamental-system, natural frequencies can be obtained and thus high forces can be developed to frequencies beyond 100 cps without using complex feedback control circuits.

Transducers and Signal Conditioning

The transducers and signal conditioners for this system are described in the following paragraphs. The discussion includes requirements for precision and accuracy, environmental considerations, and factors requiring consideration in integrating the transducers into the mechanical system.

The data acquisition and analysis system has been configured so that it can be efficiently expanded to accommodate a larger number of data channels at such time as the capability of the laboratory car is expanded. The data processing system, as specified for the nonmoving, two-loading module measurement car, has been structured to process data at a rate commensurate with the capability of the hydraulic system of the actuators. If increased analysis rates are required as a result of increased hydraulic system capacity, the data processing rate can be increased by addition of system modules, with a minimum obsolescence of existing equipment. The upgrading capability is explained in greater detail in the section on Data Processing.

Dynamic Compliance Transducer

General Requirements. Dynamic measurements of force, displacement, and acceleration in the vertical and lateral directions are required to evaluate the dynamic characteristics of the track structure. The signals from these transducers will also be used as feedback to the servo-control amplifiers in the excitation system.

The basic requirements for the transducers are shown in Table 2. These are based on the ultimate capabilities of the measurement car. The number of transducers will vary, depending on the complexity of the measuring system. Recommended transducers and some of the more important specifications are listed in Table 3.

Many of the transducers will be located in an area where they will be subjected to rather severe environmental factors. In addition to extremes of temperature and humidity, they will encounter dust and flying objects such as ballast and scrap metal. The vibration environment of the transducers will be severe. These environmental factors have been considered in specifying and mounting the transducers.

Force Transducers, Vertical. Vertical force is applied to the rail by a loading pad. Provision for measuring the combination of static and dynamic load consists of an array of three load cells situated between the loading pad and the cylinder end. This configuration has been chosen because it withstands large bending moments and it avoids introduction of spurious load data from bending moments which could be generated in a single rigid coaxial load cell by application of side loads. Because of the possibility that more than one-third of the vertical load could be applied through a single load cell of the array, each load cell has the capability to carry one-half the rated force capacity of the system. The load signals from the three load cells are summed at a point following the signal conditioners.

TABLE 2. TRANSDUCER REQUIREMENTS

Parameter	Range	Resolution	Frequency Response
Vertical Force	+35,000 pounds -10,000 pounds	400 pounds or 1 percent	1000 Hz
Lateral Force	±35,000 pounds	400 pounds or 1 percent	1000 Hz
Vertical Displacement	±5 inches	0.005 inch	250 Hz
Lateral Displacement	±2 inches	0.005 inch	250 Hz
Vertical Acceleration	0.1 to 100 G	0.1 G min.	250 Hz
Lateral Acceleration	0.1 to 100 G	0.1 G min.	250 Hz
Vertical Position ^(a,b)	(a)		
Lateral Position, Front ^(b,c)	±6 inches		10 Hz

(a) Vertical displacement serves this function.

(b) Position transducers are used only when deploying or stowing the shaker system.

(c) Wheeled system only.

TABLE 3. TRANSDUCERS SELECTED FOR TRACK DYNAMIC COMPLIANCE MEASUREMENT SYSTEM

Parameter	Device	Model	Manufacturer	Range	Nonlinearity	Nonrepeatability	Hysteresis	Excitation	Output	Temperature Effect	Quantity
Vertical Force	Load cell	FFL(18/±12) V-(3/±2)D	Strainert	18,000 lb	±45 lb	9 lb	±27 lb	15 VAC or DC	3 mV/V	0.45 lb/F	6
Lateral Force	Load cell	SD5CP 40 K	Strainert	40,000 lb	100 lb	60 lb	100 lb	12 VAC or DC	2 mV/V	1.2 lb/F	2
Vertical Displacement	LVDT	5000 MP	V	±5 inches	.005 inch at 50 percent	--	--	6 VAC 2.5 kHz	1.2 mV/.001 inch	--	2
Lateral Displacement	LVDT	2000 MP	V	±2 inches	.005 inch at 50 percent	--	--	5 VAC 2.5 kHz	1.8 mV/.001 inch	--	2
Vertical Acceleration	Accelerometer	111-100	SETRA	±100 G	±1 G	--	±1 G	6 VDC, 22 ma	15 mV/G	2 G/100 F	2
Lateral Acceleration	Accelerometer	111-100	SETRA	±100 G	±1 G	--	±1 G	6 VDC, 22 ma	15 mV/G	2 G/100 F	2

Force Transducers, Lateral. The lateral force is measured by means of an instrumented clevis pin between the end of the lateral force hydraulic cylinder and the load application pad. The commercially built instrumented clevis pin has shear sensing strain gages installed inside an axial hole to measure shear generated in the pin by the applied load. A stainless steel (17-4 PH) pin has been selected because of potentially severe environmental conditions.

Displacement Transducers, Vertical. The transducer which provides the vertical displacement data will provide the servo-feedback signal for positioning the actuators vertically. A linear variable displacement transducer (LVDT) has been chosen for this purpose. In order to operate the transducer in its region of greatest linearity, and near its electrical and mechanical null in the measurement mode, a unit with +5-inch range has been chosen. The plan is that the transducer will operate near the maximum stroke position when the measurement head is in the stowed position, and approximately 1 inch past the null position when measurements are being made. The linearity will be well within 0.15 percent of the full-scale output at this point, or less than .005 inch equivalent output. The model LVDT chosen is designed for operation in "hostile" industrial environments.

Displacement Transducer, Lateral. The lateral displacement transducer does not have the large stroke range requirement of the vertical unit. The transducer chosen for this application is from the same series as the vertical unit, but has a smaller overall stroke range.

Acceleration, Vertical. The requirements for the accelerometer used to measure vertical acceleration are difficult to meet if one considers the potential vibration environment in relation to the resolution required. As a result, it is necessary to use an accelerometer having either a large, over range capability, or one with a large range and relatively fine resolution. The preferred approach is to use an instrument with a wide range in order to obtain a high resonant frequency. One such instrument is the Setra Model 111 accelerometer. The 100-G instrument has ability to withstand a static acceleration of 500 G without damage. The full-range output is 1.5 volts, meaning that the output at the minimum required acceleration increment (0.1 G) is 0.15 millivolt. In order to obtain satisfactory results it may be necessary to provide amplification of the acceleration signal at the transducer so that the effects of electrical noise will not interfere with the signal.

Acceleration, Lateral. The accelerometer requirements for lateral measurements are the same as those for vertical measurements.

Signal Conditioning

Force Channels

All of the force transducers are based on strain gages as the transducing element. Signal conditioning for strain gages provides the following functions:

- Excitation of the strain-gage bridge
- Balancing of residual offset voltage from the bridge
- Amplification of the bridge output (signal)
- Shunt calibration to verify channel function.

Excitation of the strain-gage bridge may be ac or dc, depending on the type of amplification circuits employed.

The primary requirements for the amplifier section of the signal conditioner are:

- Low drift (thermal and power supply dependent)
- Linear gain characteristics
- Adequate frequency response
- Protection against amplifier damage if transducers are damaged or disconnected.

Shunt calibration of strain-gage transducers permits an in-place end-to-end check of cables, amplifiers, and recording equipment. Calibration is accomplished by connecting a shunt resistance across one arm of the transducer bridge. The effect is the same as that produced by a physical condition which would produce a negative strain in the gage which constitutes that arm. The result is a step change in system output which is determined by the gage resistance value, excitation voltage, amplifier gain, and shunt resistance value. Some signal conditioners connect the calibration shunt to the signal conditioner end of the transducer cable. This technique is satisfactory if the cables are relatively short and cable resistance is much smaller than the shunt resistance or the gage resistance. The calibration error which can be introduced by a long cable is insignificant as long as the strain-gage bridge is completed close to the gages, as is the case for all transducers described previously. Some signal conditioners provide for shunting at the gage by the use of a relay located at the gage location, or for connection of the shunt by means of a pair of wires separate from the wires carrying the excitation and bridge output. In either case, an additional pair of wires is required.

All of the strain-gage-based transducers specified for the track compliance instrumentation contain completed bridges so calibration shunts may be connected either at the transducer or in the signal conditioner. A desirable feature of the calibration circuit is a momentary rather than a positive detent switch for connecting the shunt. Should the calibration shunt be left in the circuit during measurements, the shunted gage would be temporarily desensitized and the output signal would be offset by the amount of the calibration step.

The gain requirements for the signal conditioner are established by the transducer sensitivity and the signal-level requirements of the data recording or acquisition equipment.

The force transducers selected for the system will have full-scale output on the order of 2 millivolts per volt of excitation. Assuming an excitation of 15 volts, the full-scale output will be 25 millivolts. The input sensitivity of the analog-to-digital conversion units will be in the range 100 millivolts to 10 volts, depending on which system is chosen; however, to maintain a favorable signal-to-noise ratio at the input to the antialiasing filters, it is recommended that the output of the signal conditioners be in the range 2.5 to 10 volts, full scale. The gain requirements of the signal conditioners is therefore in the range of 50 to 250.

The signal conditioner chosen for the strain gage load cells is the Ectron Model 418-WP. This unit meets the system requirements for gain range, linearity, noise, and other characteristics.

The Ectron unit provides regulated excitation to each transducer and provides ± 45 millivolts (referred to input) for transducer balance and offset.* Gain ranges can be controlled by the computer through relays which establish gain tap connections on the amplifiers. No provision is made within the signal conditioner modules for shunt calibration, this function being handled externally in circuits included in the rack in which the modules are mounted. For computer-controlled calibration and system checkout, relays, either solid state or electromechanical, are incorporated in the calibration system.

Displacement Channels

The linear variable differential transformer requires a signal conditioner to perform the following functions:

- Excitation of the primary with an alternating current voltage at a frequency approximately ten times the required frequency response
- Nulling of quadrature voltage components not related to the measured parameter
- Extraction of the signal from the output signal of the transducer secondary
- Amplification of the signal for subsequent use.

* 40 millivolts is equivalent to approximately 90 percent of full-scale output of the force transducers.

The signal-conditioning system chosen for the LVDT transducer is a modular system consisting of a rack with power supply and plug-in modules which contain signal-conditioning circuits. The signal conditioning for the basic system uses a 10-kHz excitation, which permits frequency response of 1 kHz.

Accelerometer Channels

The signal conditioners for the SETRA accelerometers will provide regulated power to the transducers, remote-controlled calibration, and control over gain and zero of the transducers. The plug-in rack provides power and signal connections to up to 12 signal conditioning modules.

Data Acquisition and Processing

The systems for data analysis and processing (DAP) are discussed together because in some systems both functions are under the control of a single unit — the Central Processing Unit, or CPU. In some cases, the entire system is available as a package unit from a single supplier. Because it is felt that a research vehicle should have considerable flexibility in its data processing system, as well as in the data acquisition functions, it was decided to recommend a system based on a minicomputer and using software routines for acquisition and analysis of data.

The basic DAP system will perform the following tasks:

(1) Acquire and store dynamic data from the following transducers:

- Vertical force, right front
- Vertical force, left front
- Vertical displacement, right front
- Vertical displacement, left front.

Alternatively, the data from lateral transducers will be acquired and stored.

(2) Generate forcing functions and input them, upon command, to the servo-control input to drive the excitation system.

- (3) Transfer selected blocks of data to the computer for processing.
- (4) Perform Fast Fourier Transform (FFT) analysis on blocks of data.
- (5) Store transformed data for further analysis.
- (6) Perform cross-spectrum analysis of transformed data.
- (7) Copy results of analysis to display and hard copy peripheral equipment.
- (8) Operate, in all modes, under keyboard or terminal control.
- (9) Permit keyboard selection of excitation parameters and locations.

Fourier Analyzer, Packaged Systems

For the purpose of cost estimating, two off-the-shelf packaged systems have been considered: the General Radio-Time/Data TDA53L Time Series Analysis System, and the Hewlett Packard 5451B Fourier Analyzer System. The Time Data system is designed around a Digital Equipment No. PDP-11/35 computer and the Hewlett Packard system around a HP-2100 computer. Both systems lend themselves to expansion through the addition of data acquisition hardware and to reduction in processing time through the incorporation of hardware FFT processors. Both systems have sufficient computer capacity to permit implementation of standard engineering programs in BASIC or FORTRAN language. In addition, each system has a special time series analysis language which facilitates flexibility in acquisition and processing of data.

In their original configuration, this equipment is suitable for measuring the basic desired parameters. However, additional optional features may be desired in the future to increase the speed and flexibility of the system. Some of these features are:

- Input channel expansion to at least 4 channels
- Magnetic tape recorder (digital)
- Plotter or hard copy unit
- High-speed tape punch and reader
- Relay register
- Core memory to support the additional hardware and provide scratch pad data storage.

Both of the systems considered have provision for adding more input channels by using sample and hold systems and multiplexed analog-to-digital conversion.

Recommended Data Acquisition System

A block diagram of the basic data acquisition system for the static compliance measuring system is shown in Figure 11. For the system shown, signals from accelerometers at the front and rear of the vehicle are double integrated to produce a signal proportional to the vehicle body pitching motion. The car-body pitch signal might also be produced with a gyro. This signal is summed with the displacement signals from each truck to generate signals proportional to the pitch angle of each truck. The pitch angle signal from the first truck is delayed by a computer, shift register, or other digital delay devices and the delay is controlled by a pulse generator on one of the vehicle wheels. The delay is made equal to the time required for the vehicle to travel a distance equal to the truck spacing.

The delayed signal from the front truck is combined with the signal from the second truck to produce a signal proportional to the difference in wheel displacements measured at a specific point on the track. This signal would be filtered by a filter phase locked to wheel position to remove wheel runout "noise" and probably also by conventional low pass filters to remove high-frequency noise. Wheel forces would also be measured and signals from the wheel force transducers would be used to compensate for the dynamic wheel loads generated at high operating speeds.

The resulting static compliance data along with position data would be recorded on conventional paper strip charts and on magnetic tape.

The basic recommended data acquisition system for the dynamic compliance measurements is shown in Figure 12. The heart of the system is a Fourier Analyzer of the type discussed previously. Commercial transducers and signal conditioners would be used to measure forces and motions. With this basic system, transducers would be moved and/or signals switched to measure different parameters; however, a set of transducers, signal conditioners, and servo-controllers would be used on each actuator so that both vertical and lateral loads could be applied to both rails simultaneously.

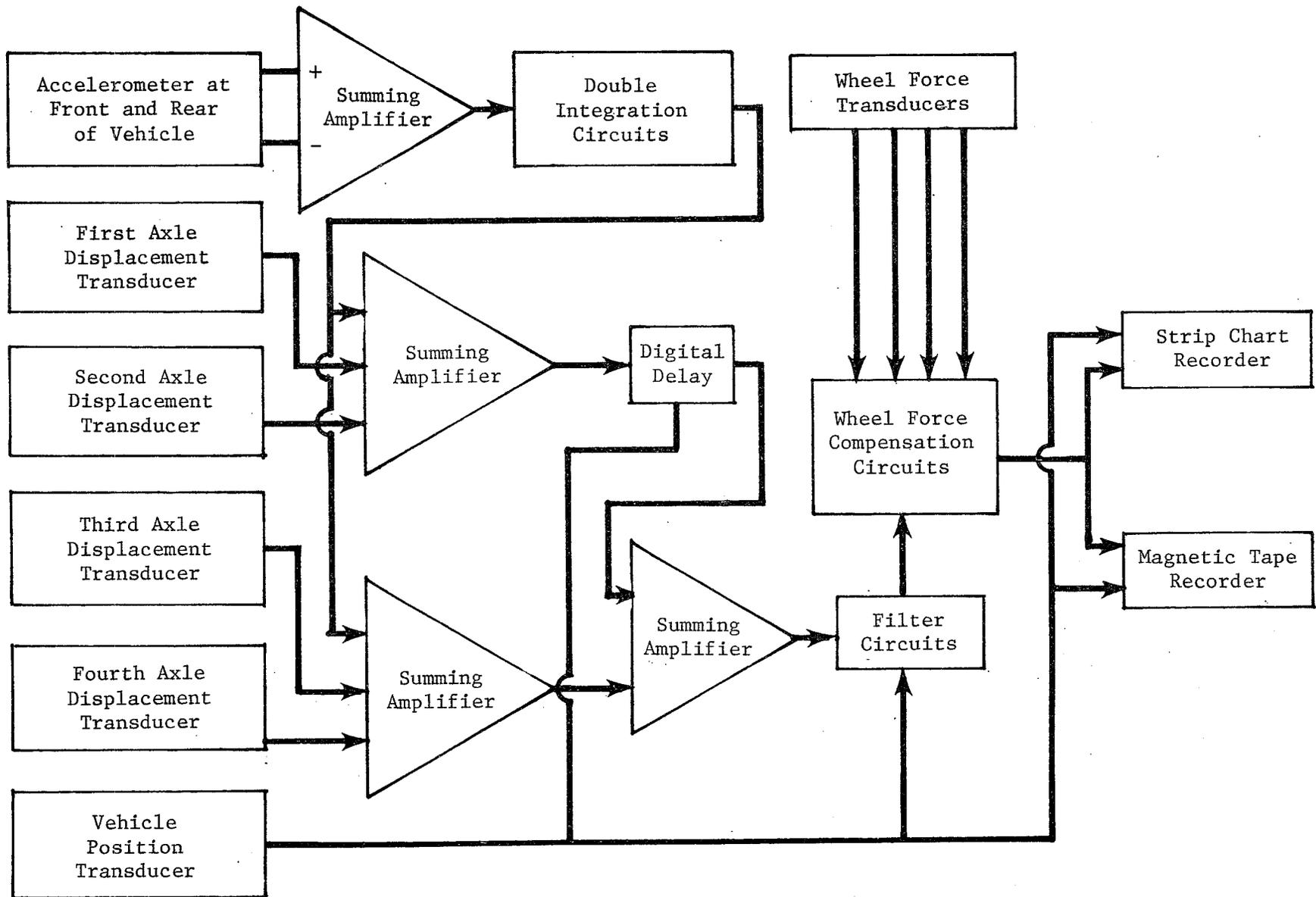


FIGURE 11. SCHEMATIC OF THE STATIC COMPLIANCE MEASURING SYSTEM

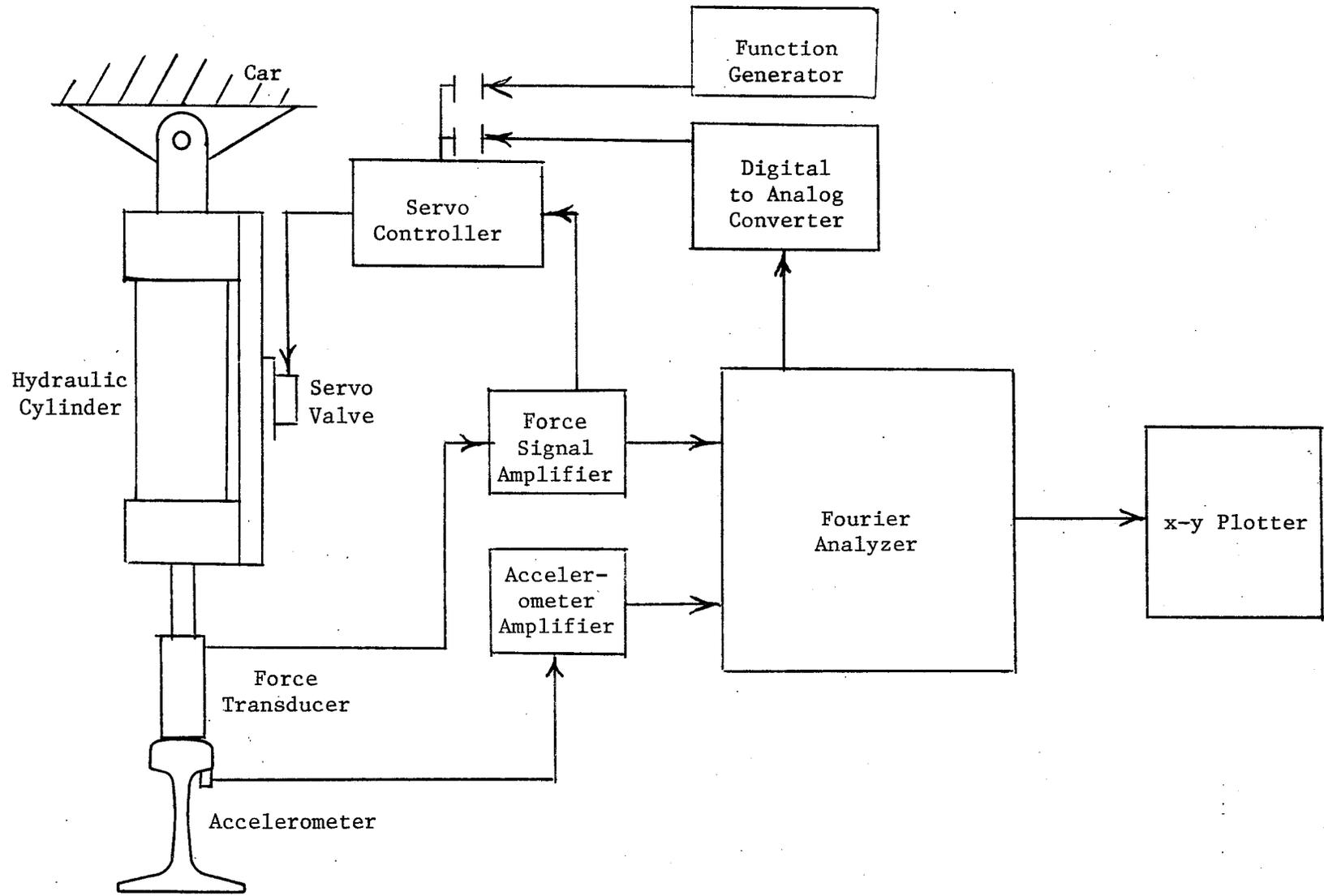


FIGURE 12. SCHEMATIC OF ACTUATOR CONTROL AND DATA ANALYSIS SYSTEM

Provisions would be made so that excitation signals could be produced by the Fourier Analyzer's computer and/or by an external function generator. The external function generator would consist of both general-purpose signal generators and special-purpose circuits designed to produce command signals that simulate moving trains and/or other special-purpose signals. These circuits would be assembled on operational manifolds with plug-in components to provide flexibility in producing command signals.

ESTIMATED TIME AND COSTS

Table 4 shows the costs, broken down by Tasks, to develop the recommended measurement vehicle equipped with the high-speed static stiffness measuring system and the stationary dynamic compliance measuring system. It is estimated that the time required to develop this system would be approximately 24 months. The time, and to some extent the cost, depend upon the availability of a surplus locomotive and its condition.

Large quantities of the locomotive chosen for the recommended system were produced and these are currently being made surplus. If procurement is started before most of these locomotives are scrapped it is believed that one of these units could be located and procured within a period of 6 months.

Cost estimates were also made for systems more complex and simpler than the recommended system. These costs are compared in Table 5.

TABLE 4. ESTIMATED COSTS FOR RECOMMENDED SYSTEM

Purchase of surplus locomotive	\$ 10,000
Engineering to convert from three-axle to two-axle trucks, add wheel loading system, and transducers	15,000
Vehicle structural refurbishing of trucks, couplers, brakes, etc., to meet interchange requirements, repair, and paint	50,000
Control room interior construction	
(1) materials	20,000
(2) labor	80,000
Equipment room - equipment, construction, and installation of equipment	
(1) hydraulic power supply	30,000
(2) motor generator set	10,000
(3) air conditioner	3,000
(4) miscellaneous hardware	2,000
(5) labor	50,000
Design of dynamic compliance actuators and mounting system	20,000
Construction and installation of dynamic compliance actuator system	
(1) parts	10,000
(2) labor	40,000
Select components and design data acquisition and processing system	20,000
Procure and install data acquisition and processing system	
(1) FFT analyzer	50,000
(2) signal conditioners, etc.	30,000
(3) labor	40,000
Procure miscellaneous auxiliary equipment	
(1) parts	10,000
(2) labor	10,000
Payments to railroad for moving equipment and use of equipment	20,000
Travel, management, meeting, and other miscellaneous costs	30,000
Debugging and demonstration	50,000
	<u>\$600,000</u>
20 percent inflation and safety factor	<u>120,000</u>
Total Estimated Cost	<u>\$720,000</u>

TABLE 5. COST COMPARISONS

I. Recommended system with capabilities of:	
A. Measuring vertical static stiffness while moving	
B. Measuring vertical and lateral dynamic compliance with two fixed loading points and two movable motion-measuring points from stationary vehicle	\$ 720,000
II. Minimum system capable of:	
A. Measuring vertical and lateral dynamic compliance with two fixed loading points and two movable motion-measuring points from stationary vehicle	\$ 690,000
III. Minimum system capable of:	
A. Measuring vertical static stiffness while moving	\$ 480,000
IV. Best system capable of:	
A. Measuring vertical and lateral static stiffness while moving	
B. Measuring vertical and lateral direct dynamic compliance while moving	
C. Measuring vertical and lateral dynamic cross compliance from the primary loading axle to a secondary loading axle while moving	\$1,500,000

REFERENCES

1. Nessler, G. L., Prause, R. H., and Kaiser, W. D., "An Experimental Evaluation of Techniques for Measuring the Dynamic Compliance of Railroad Track", Interim Report, prepared for U.S. Department of Transportation by Battelle's Columbus Laboratories, September, 1977.
2. Association of American Railroads, "Field Manual of the AAR Interchange Rules", 1975.
3. Our GM Scrapbook, Kalmbach Publishing Company, Milwaukee (1971), pp 141-142.

APPENDIX A

REVIEW OF CONCLUSIONS FROM THE MEASUREMENT PROGRAM

The measurement program* consisted of an experimental evaluation of techniques for measuring the dynamic compliance of railroad track. The different techniques used were sinusoidal, random, and pulse excitation applied in the vertical and lateral direction superimposed on a static preload using an electrohydraulic servo system. The conclusions reached at the end of the measurement program are as follows.

Track Vertical Stiffness. The track structure displayed very nonlinear behavior with vertical preload. The vertical stiffness increased with increasing vertical preload. This verifies the necessity for using vertical loads representative of typical wheel loads in order to measure track stiffnesses that are valid for actual railroad service.

Track Lateral Stiffness. Track lateral load-deflection measurements with constant vertical preload also show a significant nonlinear behavior. The track lateral stiffness is almost directly proportional to the vertical preload, so realistic wheel loads are also required in the lateral direction.

Track Dynamic Characteristics. Dynamic stiffness, resonant frequency, effective mass, and damping were measured and calculated, based on a simplified model, to characterize the behavior of track under dynamic loading. Typical results for vertical track dynamic characteristics with a 15,000-pound vertical preload were:

Resonant frequency - 30 to 45 Hz

Effective mass - 2500 to 5500 lbm per rail

Damping - 15 to 45 percent critical

* Nessler, G. L., Prause, R. H., and Kaiser, W. D., "An Experimental Evaluation of Techniques for Measuring the Dynamic Compliance of Railroad Track", Interim Report, prepared for U.S. Department of Transportation by Battelle's Columbus Laboratories, September, 1977.

The measurements of track dynamic compliance indicate that data on the stiffness, resonant frequency, and damping are probably adequate to characterize track dynamic response over the frequency range of 0 to 80 Hz. This frequency range is of primary interest for vehicle/track design and analysis. A frequency resolution of 1 to 2 Hz will be adequate for defining the track dynamic characteristics.

Measurement Techniques. A comparison of the results from using sinusoidal, random, or pulse excitation superimposed on a constant preload showed that these three different techniques for making dynamic measurements usually gave similar, but not identical data for track dynamic characteristics. The results from the pulse excitation showed the greatest variation from the other measurements because the force amplitude needed to get sufficient energy in the pulse was much larger than that needed for the random or sine excitation. These larger force amplitudes for the pulse measurements increased the influence from the nonlinear track behavior.

However, a more surprising and significant result was that the vertical track stiffness determined by the low-frequency (5 to 10 Hz) response of the track to any of the three dynamic force excitations was considerably higher than the tangent stiffness determined from the slope of the static load-deflection measurement at the corresponding preload. This stiffness, as measured with these dynamic techniques, was as much as a factor of two greater than the static stiffness as determined by load-deflection measurements. Further investigation of this result indicated a considerable compaction or settling effect when the track is loaded by a constant preload with repeated vertical dynamic excitation. This hysteresis effect was also confirmed by the difference in the results obtained from an unloading and loading pulse. In this test, the only significant variation was that the sequencing of the tension and compression portions of a nearly symmetrical dynamic pulse were reversed. This would not affect the response for a linear system, but the track response was noticeably different.

The conclusion that the track has a significant settling effect is quite important for selecting a measurement technique. When the objective is to measure the vertical track characteristics which are relevant to those seen by a passing wheel, it will be necessary to duplicate the service loading environment for the track in much greater detail than was previously expected. The vertical track loads from a passing train are characterized by a series of loading pulses for each truck. The load variations from

individual axles are noticeable only on some very rigid track, but these variations are relatively small compared to the total truck load.

The important part of the track loading in service is that the track is nearly unloaded between trucks of adjacent cars (when the coupler passes), and it is always unloaded between the passage of the front and rear trucks of each car. The measurement results indicate that this periodic unloading is very important. This was confirmed by changing the constant preload with multiple pulses or random excitation superimposed to an excitation having a slowly varying cyclic preload (1 pulse every 5 seconds), with a single pulse superimposed at the maximum loading point. This type of cyclic preload/pulse excitation unloaded the track between each dynamic measurement and much closer agreement was obtained between the static and dynamic stiffness measurements. It appears that this type of loading produces better simulation of the wheel loads from a moving train. The track hysteresis in the vertical direction makes this more realistic simulation necessary to obtain valid data for analytical models or simulation studies. The results from lateral measurements showed much closer agreement between the static and dynamic stiffness measurements, so it is concluded that settling effects are not as significant in the lateral direction.

The results from this measurement program have revealed that track behavior under dynamic loads is quite complex and that settling effects cannot be neglected. It should also be mentioned that other research investigations where track dynamic measurements have been made, in the U.S. and in Europe, utilize a constant preload with repetitive dynamic loading (usually sinusoidal) superimposed. Results from these measurements may differ considerably from realistic service loading, depending on track conditions. It is important to understand that the data which showed large hysteresis effects were measured on wet track. Some other measurements showed relatively minor settling effects, and it is believed that the ballast and subgrade may have been frozen during this time period. Therefore, it is conjectured that wet track may exhibit maximum settling effects and frozen or quite dry track may have relatively little settling. Additional measurements are needed to fully evaluate the effect of these different climatic conditions.

In summary, the major result from the measurement program is that the ability to apply a cyclic load (representative of an actual wheel load) at a rate simulating a passing truck at some fixed train speed should be designed into the test vehicle.

APPENDIX B

TECHNIQUES FOR CONTINUOUSLY MEASURING STATIC COMPLIANCE

To measure static track compliance, the deflection due to a known load must be measured. In theory this is a simple measurement, but in practice it becomes difficult because of two considerations. When measuring rail deflections from a moving vehicle, the first problem which arises is a lack of an absolute reference from which to measure the desired deflections. The second problem is that all of the vehicles' wheel loads on a common track interact to cause track deflections at the measurement point. This interaction makes it difficult to measure the deflection due to a single known load. These two problems may be overcome in several ways.

One system to measure static track compliance is to use the midchord offset of the center axle of a three-axle truck, as shown in Figure B-1. To explain the basis of this technique, first consider the classical linear model of a rail as an infinite beam continuously supported by an elastic foundation, as shown in Figure B-2. When the beam is subjected to a point load, the deflection in the vertical direction is given by

$$y = \frac{P\beta}{2U} e^{-\beta x} (\cos \beta x + \sin \beta x). \quad (B-1)$$

The static track compliance is therefore given by

$$C = \frac{\beta}{2U} \quad \text{or} \quad \frac{1}{8EI\beta^3}. \quad (B-2)$$

The deflection at any point on the rail due to the three point loads of a rail truck may be calculated using superposition since linearity was assumed. Therefore, we may calculate compliance versus midchord offset curves on the basis of this model.

Figure B-3 shows the static track compliance versus midchord offset for a truck with equal 18,000-pound wheel loads on 80-pound and 140-pound rail. These values for midchord offset are calculated with a truck having an axle spacing of 5.5 feet. Using these curves, it is possible to determine track compliance if it is assumed that the wheel loads remain constant, or are measured, and the midchord offset if measured as a vehicle travels down a continuous uniform track with a known rail weight. If the midchord offset

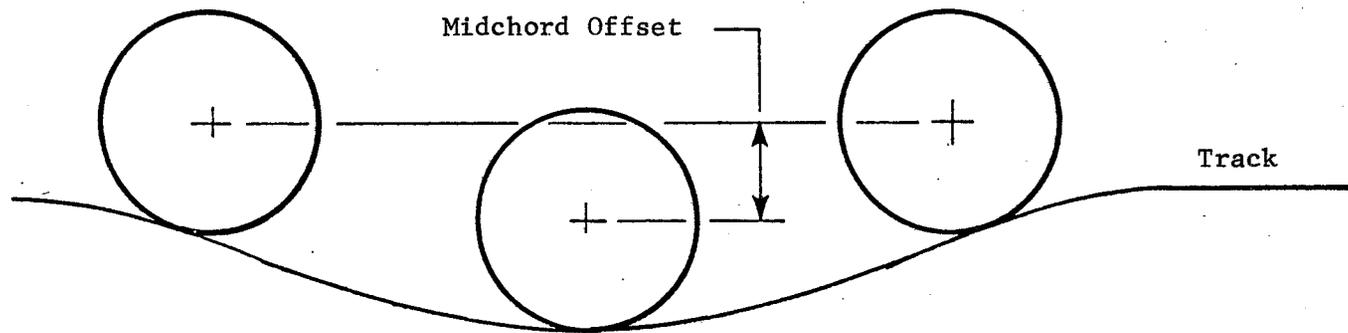
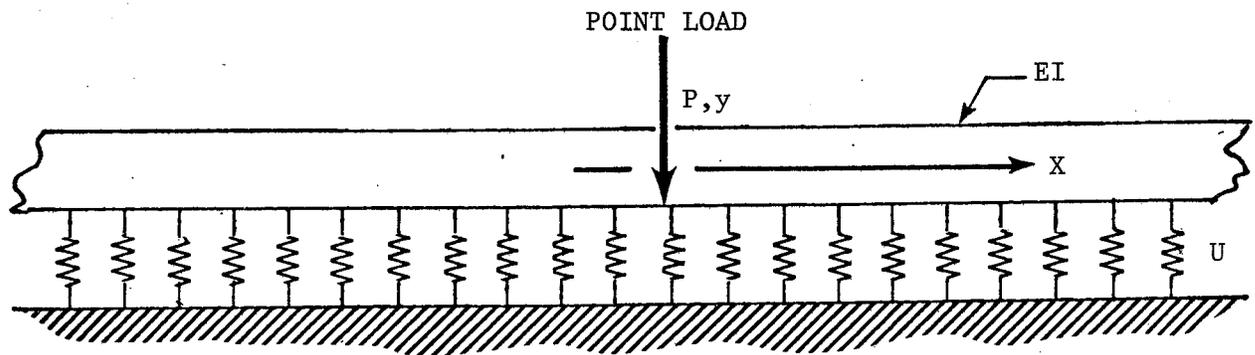


FIGURE B-1. MEASUREMENT OF MIDCHORD OFFSET



$$y = P\beta/2U e^{-\beta x} (\cos \beta x + \sin \beta x)$$

$$y_{\max} = P\beta/2U$$

E = Young's Modulus (lb/in.²)

I = Second Moment of Area (in.⁴)

U = Track Modulus (lb/in./in.)

$$\beta = \sqrt[4]{U/4EI} \text{ (in.}^{-4}\text{)}$$

FIGURE B-2. BEAM ON CONTINUOUS ELASTIC SUPPORT

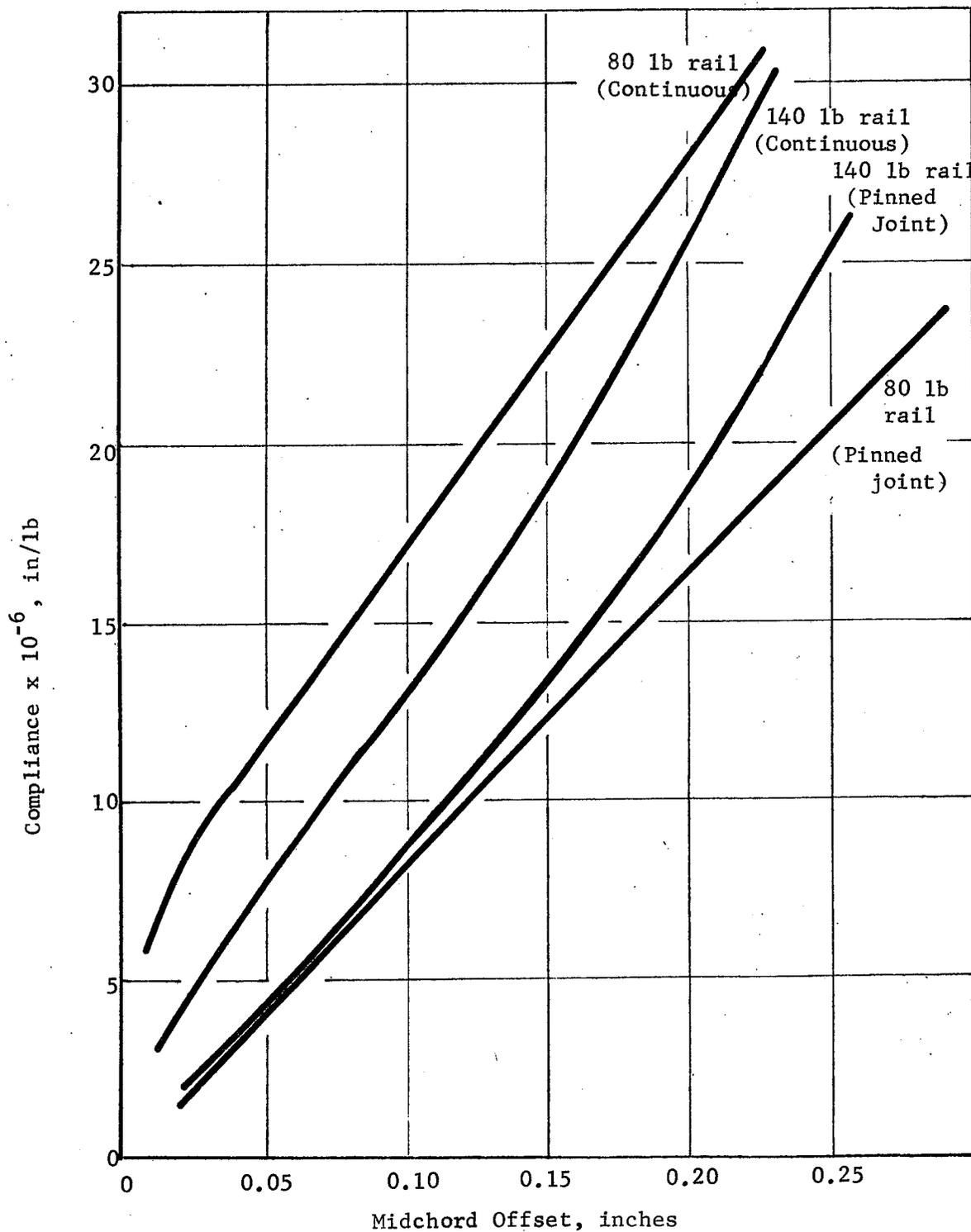


FIGURE B-3. MIDCHORD OFFSET VERSUS COMPLIANCE FOR TRUCK WITH EQUAL 18000 LB WHEEL LOADS

can be measured to an accuracy of ± 0.005 inch, this would give a maximum compliance error of ± 11 percent of true value for the 80-pound rail and ± 13 percent of true value for the 140-pound rail on average track with a compliance of 5×10^{-6} lb/in.

The performance of this method was next evaluated on jointed rail. Because of the discontinuity of the rail at a joint, the track stiffness is lower, giving a higher compliance and greater deflection at joints. In order to evaluate the potential error in the compliance measurement at joints when using this method, the deflection at a joint was estimated by assuming that the joint was a pinned connection between two semiinfinite beams on an elastic foundation. Also shown in Figure B-3 is static track compliance at a joint versus midchord offset for a truck with equal 18,000-pound wheel loads on 80- and 140-pound rail. It can be seen that a calibration curve which is valid for CWR track would not be valid for jointed track over a joint.

If the calibration curves for CWR track are used to calculate the compliance at a joint, the measured compliance is in error by a factor of about 100 percent for a wide range of modulus values. The majority of the midchord offset measurements at the joint are of such a magnitude that they are completely off scale and give compliance values out of a physically realistic range when this method is used to calculate compliance at a joint. If the load on the center axle is increased to 29,000 pounds per wheel and the load on the outer wheels is decreased to 12,500 pound per wheel, the midchord offset versus compliance curves shown in Figure B-4 result. The larger center wheel load produces corresponding larger rail deflections. The maximum compliance error, assuming the midchord offset can be measured to within ± 0.005 inch, is 7 percent of true value for the 80-pound rail and approximately 8 percent of true value for the 140-pound rail on average track. This would indicate that this method of measuring rail compliance is well suited to CWR track. Figure B-4 also shows static track compliance at a joint versus midchord offset when the truck center wheels have a 29,000-pound wheel load and the outer wheels have a 12,500-pound wheel load. Again the difference between the calibration curves is such that joint compliance calculated using the CWR track calibration curve would be almost 100 percent in error. Therefore, this method works well only with CWR track, which follows the calibration

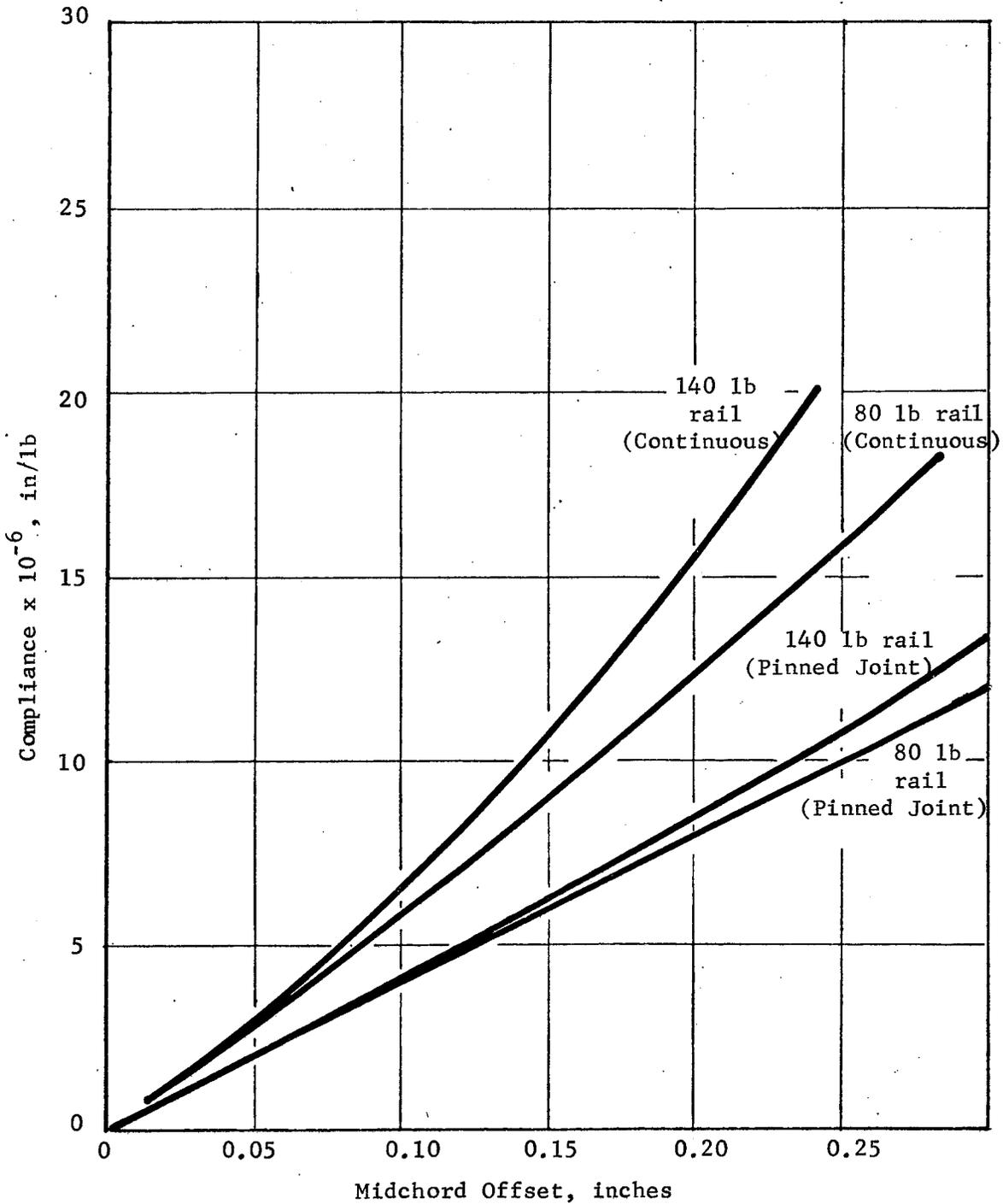


FIGURE B-4. MIDCHORD OFFSET VERSUS COMPLIANCE FOR TRUCK WITH 29,000 POUNDS PER WHEEL ON CENTER AXLE AND 12,500 POUNDS PER WHEEL ON OUTER WHEELS

model closely. The usefulness of this method for determining the compliance of track with discontinuities is therefore limited because a separate "model" would be required with several additional unknown variables. However, the use of a complex model for discontinuities would require a more complicated measuring system, which is considered to be impractical. If the discontinuities are not large, then this method could be used to give some estimate of an average static compliance over sections of track, or measurements would have to be taken only on continuous sections. Great care would have to be exercised in using such averages. The inability to deal with discontinuities or local changes in support condition also limits the usefulness of this method as a detector for defective or missing ties, since this is a localized phenomenon and significant errors would be expected.

A simple and direct method of measuring static track compliance is to measure the midchord offset of two separate, unequally loaded three-axle trucks. The midchord offset for each truck is measured as shown in Figure B-1. The approach is to load all outer wheels of the two three-axle trucks the same, and load the center wheels differently. The midchord offset for each truck is measured with the truck over the same point on the track. The difference in the midchord offset of the two trucks should be due to the difference in the loads on the center axles of each truck. Further explanation of this method is found by writing an expression for the midchord offset in terms of influence coefficients. If we assume symmetry about the center axle, the midchord offset may be expressed as

$$\text{Midchord offset} = 2 C_{12} P_1 - C_{12} P_2 - C_{11} P_1 + C_{22} P_2 - C_{13} P_1, \quad (C-3)$$

where

- C_{11} = influence of an outer axle on the same outer axle
- C_{12} = influence of center axle on outer axle
- C_{22} = influence of center axle on the same center axle
- C_{13} = influence of one outer axle on the other outer axle
- P_1 = load on each outer axle
- P_2 = load on inner axle of the first truck.

If we subtract the resulting midchord offset for the second truck from the midchord offset for the first truck at the same point of track, the following results:

$$\begin{array}{r} \text{Midchord offset} \\ \text{Truck 1} \end{array} - \begin{array}{r} \text{Midchord offset} \\ \text{Truck 2} \end{array} = (C_{22} - C_{12}) (P_2 - P_3), \quad (\text{B-4})$$

where P_3 equals the load on the center axle of second truck, or

$$\frac{\begin{array}{r} \text{Midchord offset} \\ \text{Truck 1} \end{array} - \begin{array}{r} \text{Midchord offset} \\ \text{Truck 2} \end{array}}{P_2 - P_3} = C_{22} - C_{12}. \quad (\text{B-4})$$

Since C_{22} is the true compliance of the track, C_{12} is the inherent error due to the measurement technique. If parameters are such that C_{12} is small when compared with C_{22} , or if the effect of C_{12} is known, then constant compensation can be used so that this method will give accurate results. Figure B-5 shows the ratio of C_{12}/C_{22} versus $1/\beta$ for both continuous rail and a pinned joint for a three-axle truck with a 66-inch axle spacing. The parameter $1/\beta$ is a measure of the track stiffness which is a function of the rail weight and track modulus. Typical values for $1/\beta$ are between 30 and 70 inches. The range for C_{12}/C_{22} over this range of $1/\beta$ is .37 to .85 for continuous rail and from .16 to .57 for a pinned joint. If Equation (B-5) is used directly with C_{12} , assumed equal to zero, this will give compliance errors of 15 to 36 percent for continuous rail, and 43 to 84 percent for a pinned joint. Through the use of a correction curve, the error on continuous rail may be reduced, but if a correction curve is used, then the system suffers the same drawbacks as the single three-axle truck evaluated earlier. Because of the large difference in compliance between continuous and jointed track, it is impossible to generate a correction curve that is accurate for both continuous and jointed.

One way of decreasing C_{12}/C_{22} is to increase the truck axle spacing. The ratio C_{12}/C_{22} versus truck length for $1/\beta = 50$ is shown in Figure B-6. To obtain less than 10 percent error, a truck axle spacing of at least 7.5 feet is required. As a worst case, the ratio C_{12}/C_{22} versus truck length for $1/\beta = 75$ is shown in Figure B-7. To obtain less than 10 percent error with this value of $1/\beta$, a truck axle spacing of at least 11 feet is required. In short, to accurately measure rail deflections needed for calculating track compliance both on continuous track and at discontinuities,

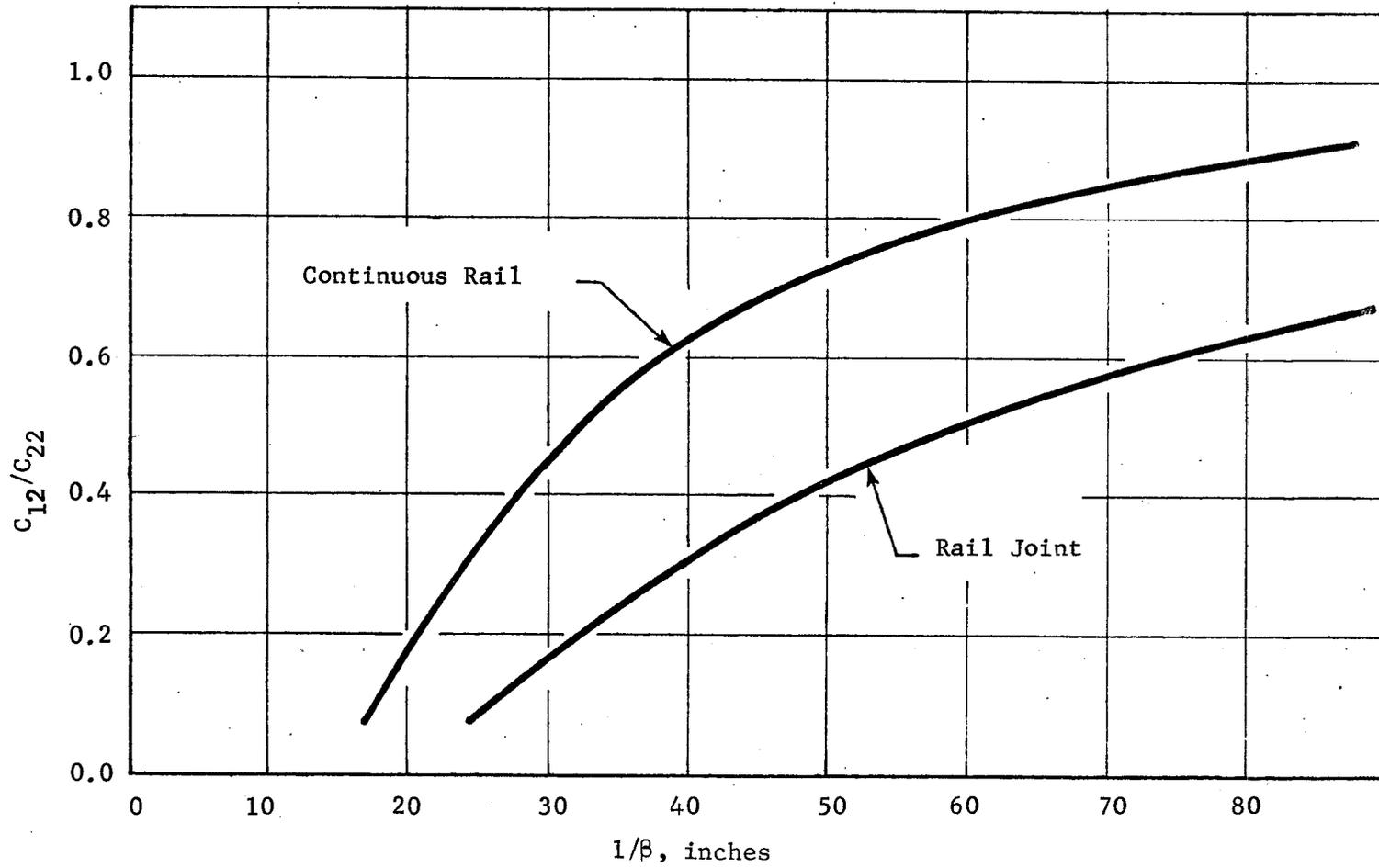


FIGURE B-5. C_{12}/C_{22} VERSUS $1/\beta$

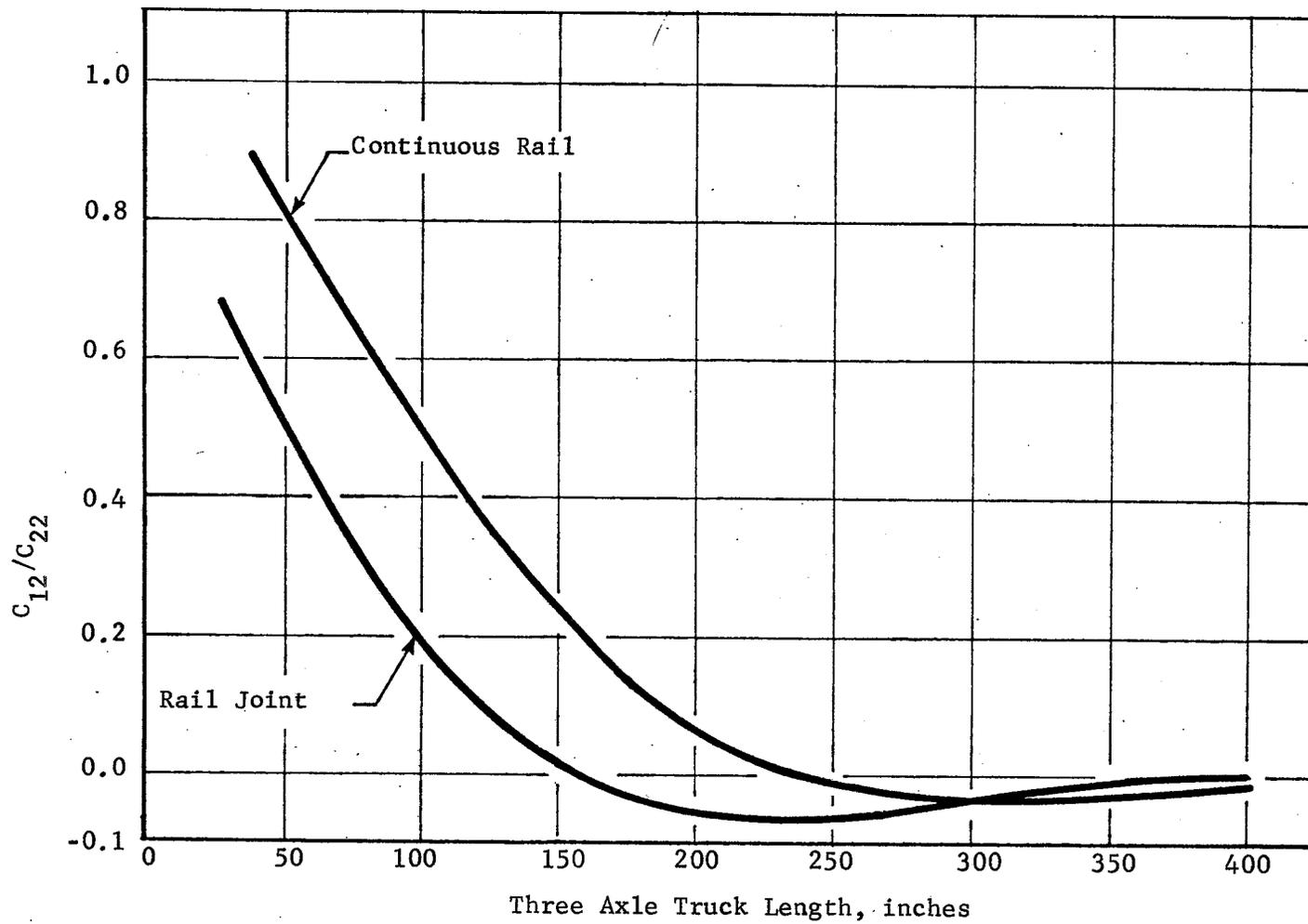
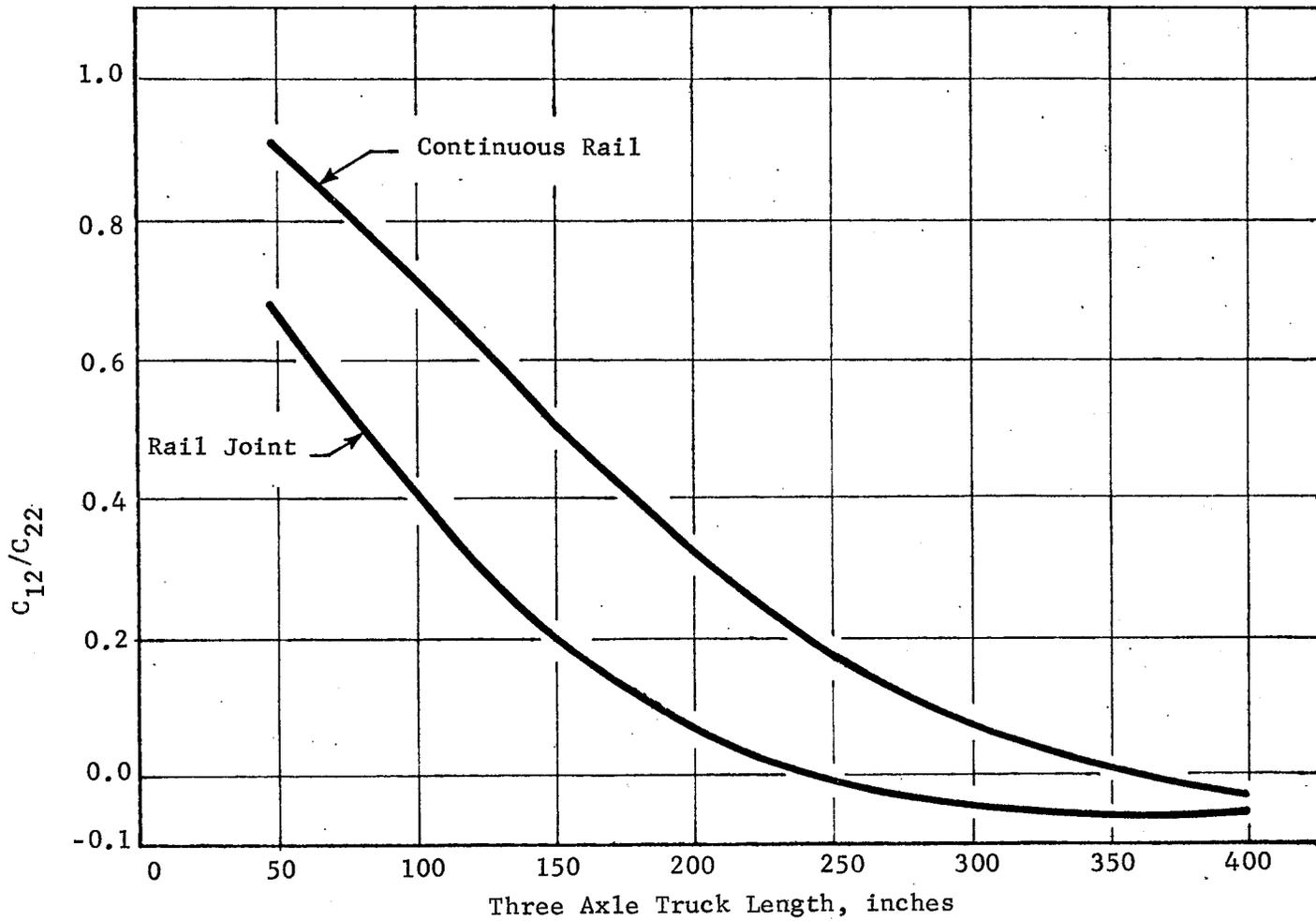


FIGURE B-6. C_{12}/C_{22} VERSUS TRACK LENGTH FOR $1/\beta = 50$ INCHES

FIGURE B-7. C_{12}/C_{22} VERSUS TRUCK LENGTH FOR $1/\beta = 75$ INCHES

a reference system must be chosen which is outside the range of influence of the measurement wheel(s).

Another method which has been proposed for measuring static track compliance makes use of two two-axle trucks with a wide axle spacing. An analysis of this system was made based on a truck with an axle spacing of 9 feet. The wheels are loaded as shown in Figure B-8. The measurements Y_1 , Y_2 , Y_3 , and Y_4 are wheel displacements measured relative to the car body. The measurements for the second truck are taken over the same point of track as the measurements for the first truck. Using these measurements, a calibration curve for CWR track such as shown in Figure B-9 may be drawn. Using this curve the compliance of nominal CWR track may be measured with less than ± 6 percent error if the displacement measurements are accurate to within .005 inch, and the rail weight is known. Using this same calibration curve, the compliance at joints may also be measured. The ratio of the calculated compliance using the calibration curve to the actual compliance versus track modulus for a pinned joint is shown in Figure B-10. The compliance error for a pinned joint is typically less than ± 25 percent. This indicates that this method works well on CWR track and is reasonably accurate at rail joints. The drawback to this system is that the measurements are made relative to the car body. Any pitching motion of the car body will introduce errors. If it is assumed that a vehicle with the dimensions shown in Figure B-11 is used for the measurement system, then to maintain ± 10 percent accuracy the car body pitch must be less than 0.15 inch end to end or compensation for the car pitching motion must be used. One way of compensating for car-body pitch is to use an inertial reference system to obtain car-body-motion information. This system has the drawback of having a minimum speed below which the system will not function. Another way of compensating for car-body pitch is to use a second vehicle in the measurement consist for the measurement of the second truck displacement.

Another problem arises with this system as to what point on the track the compliance measurement is referenced. Because of the way in which the wheels are loaded and the displacement measurements are combined, as the measuring vehicle passes a single pinned joint the output signal contains more than one peak, as shown in Figure B-12.

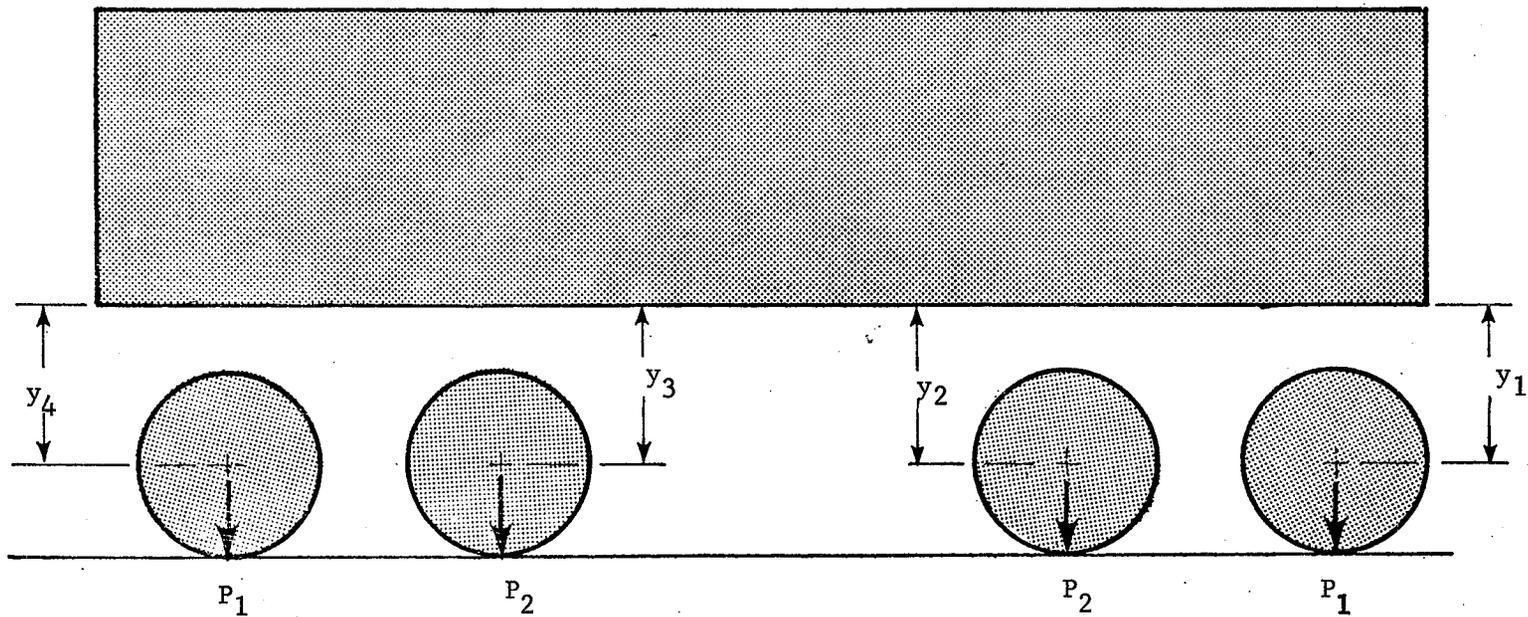


FIGURE B-8. VEHICLE LOADING CONFIGURATION FOR RELATIVE MEASUREMENT SYSTEM

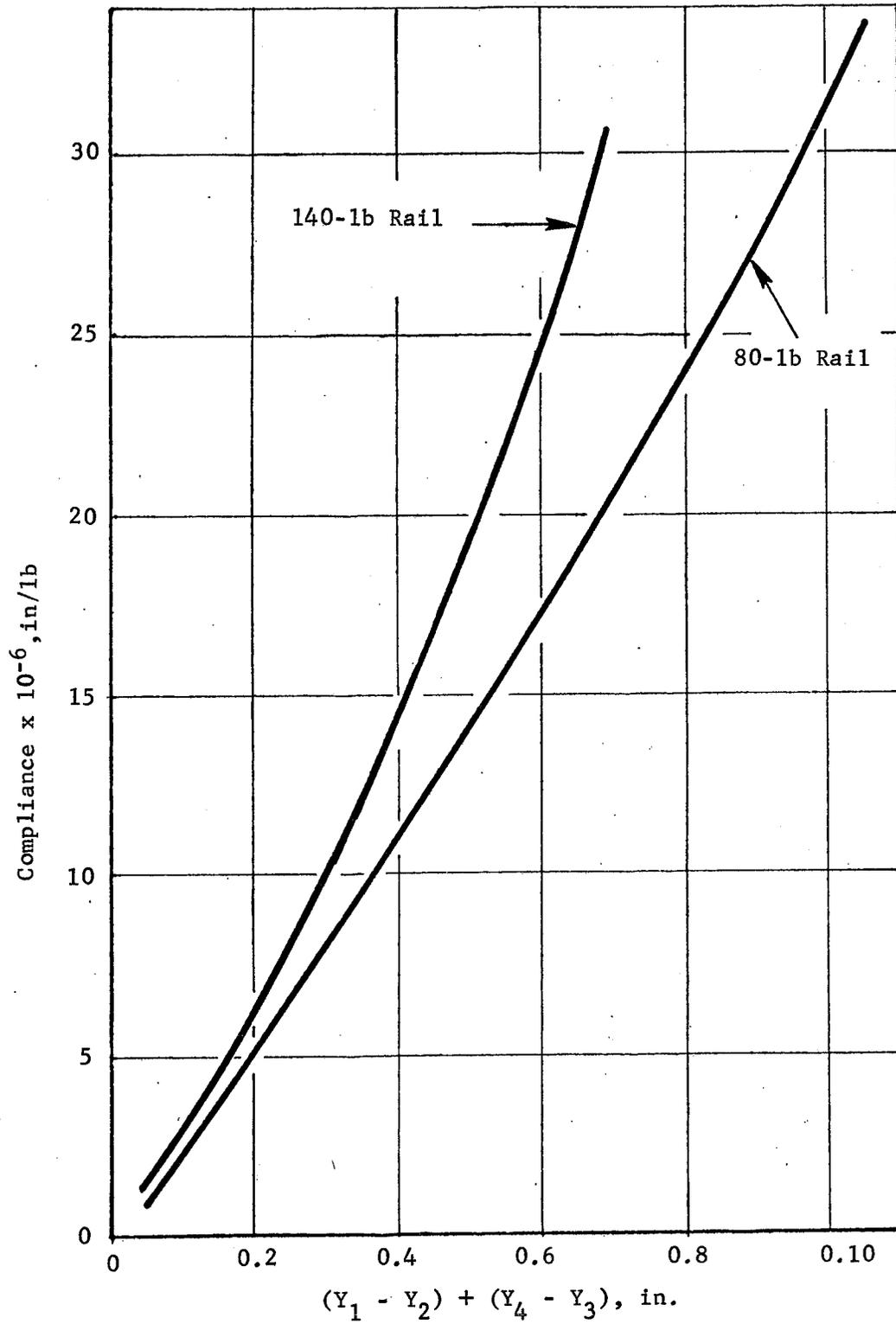


FIGURE B-9. $(Y_1 - Y_2) + (Y_4 - Y_3)$ VERSUS COMPLIANCE WITH 9 FOOT AXLE SPACING

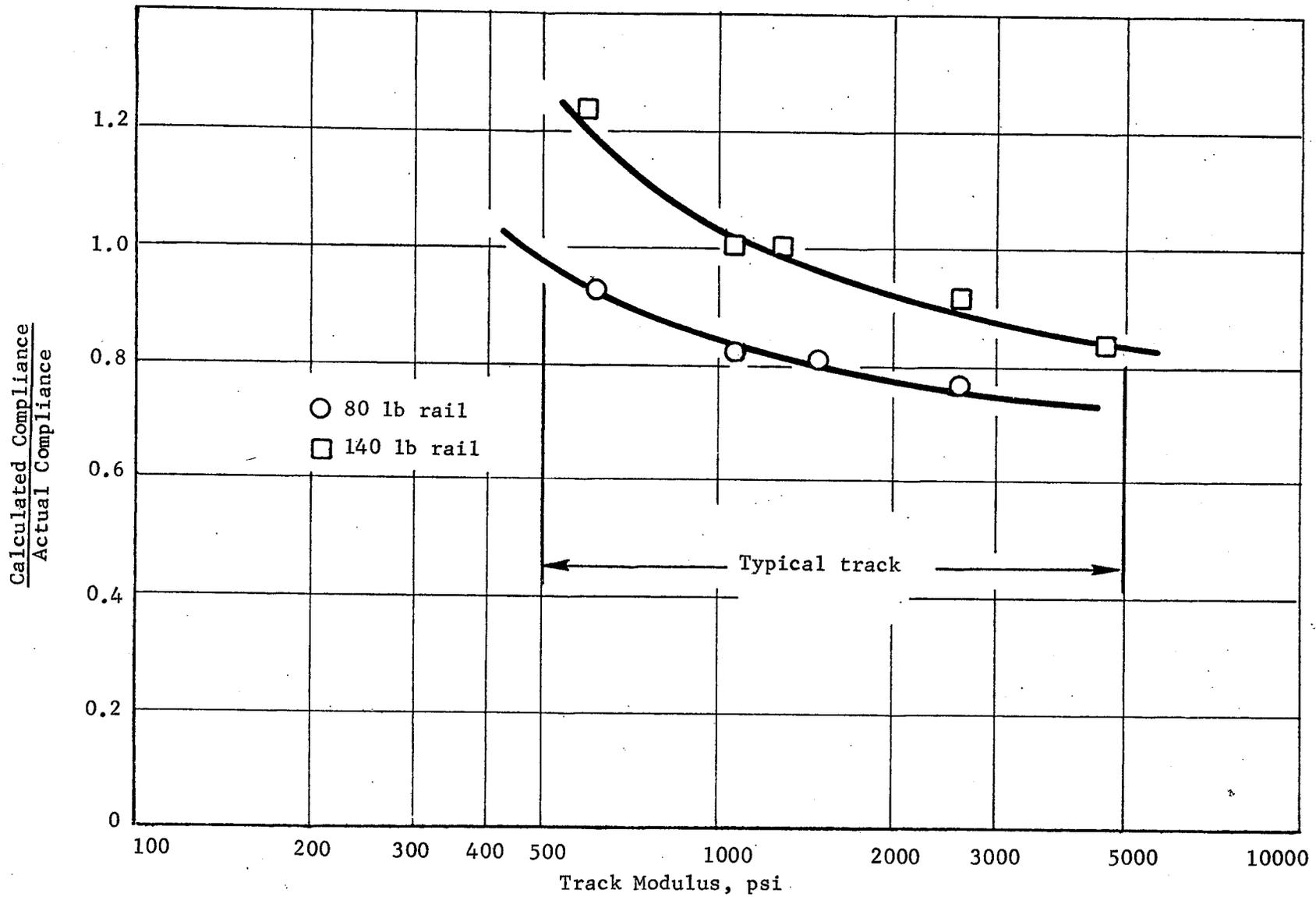


FIGURE B-10. COMPLIANCE RATIO VERSUS TRACK MODULUS AT A PINNED JOINT

B-16

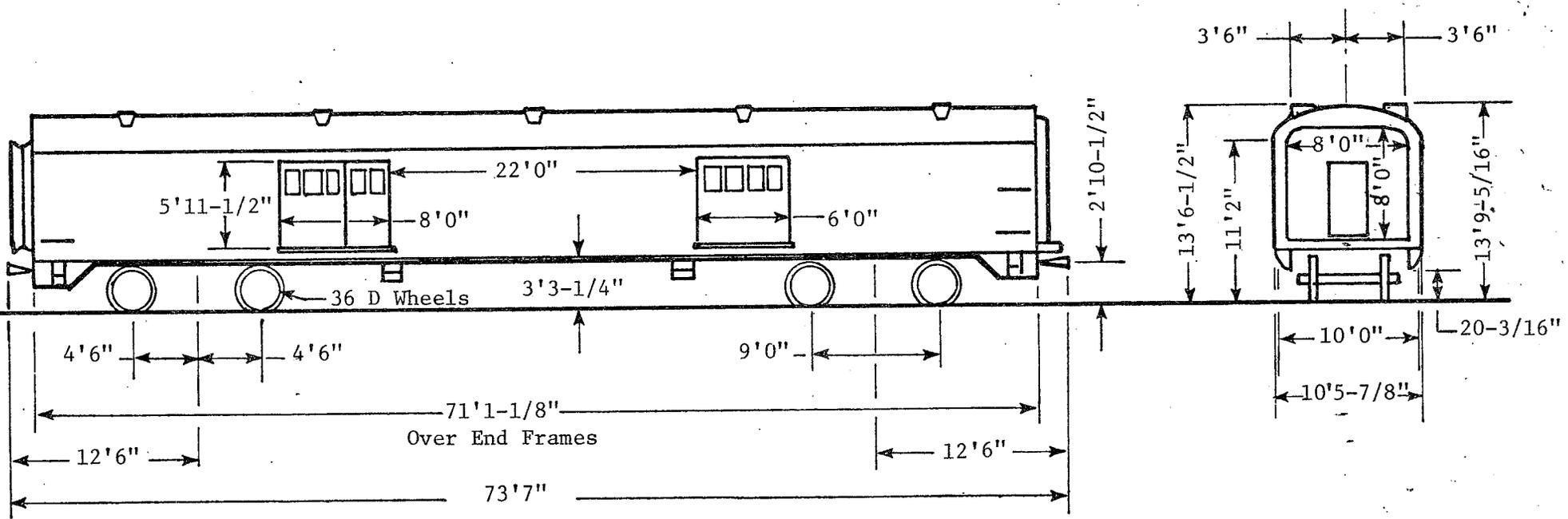


FIGURE B-11. VEHICLE DIMENSIONS

B-17

* Delayed
Signal
 $(Y_1 - Y_2) + (Y_4 - Y_3)$, in.

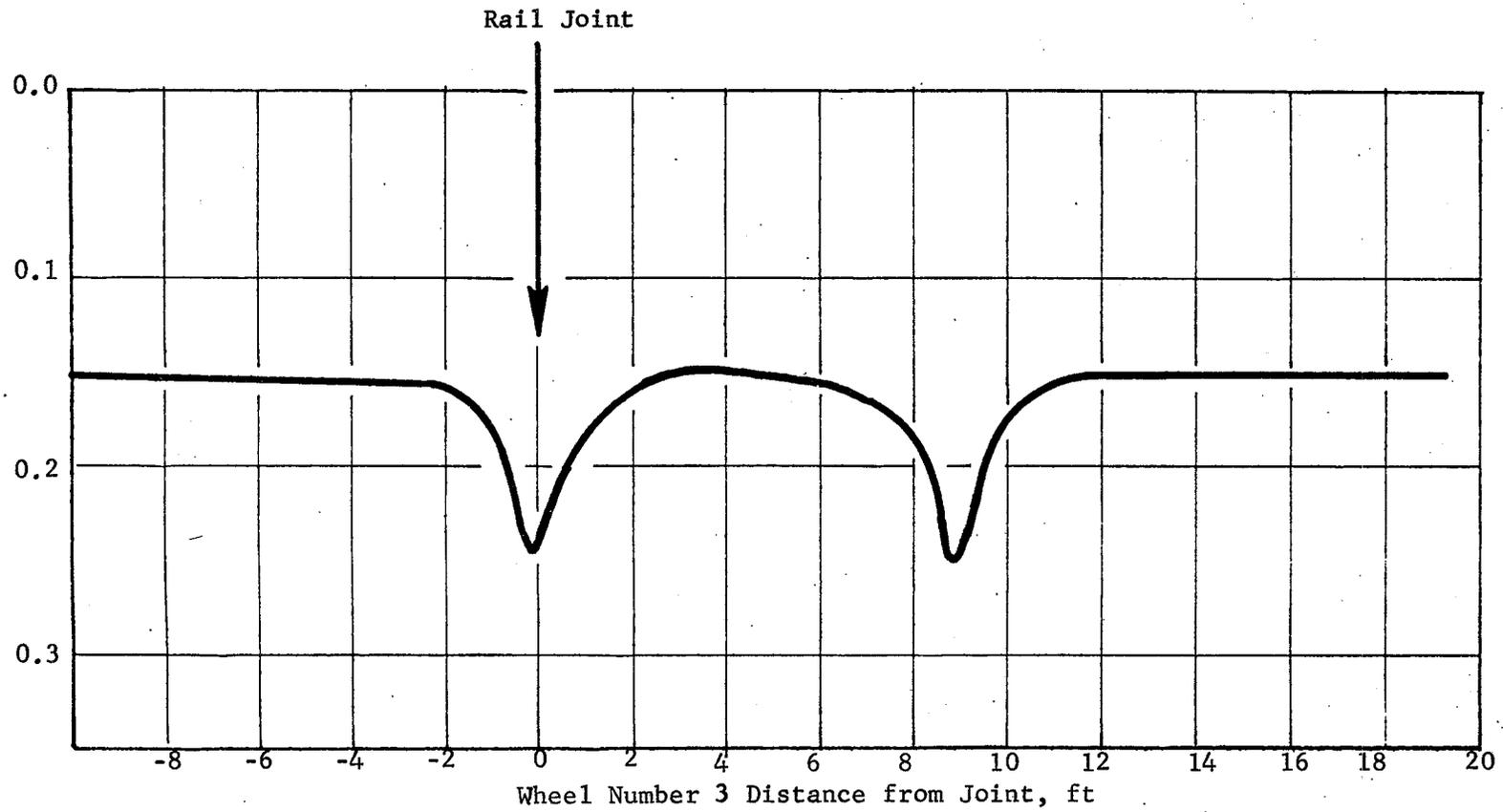


Figure B-12. OUTPUT SIGNAL VERSUS DISTANCE FROM JOINT

For the above reasons, this measurement system would not work well on either CWR track or jointed track.

The second peak problem shown in Figure B-12 can be eliminated by making the loading on the rear (or front) axle of each truck equal. Making these axle loads equal results in the recommended system discussed in the main body of the report.

APPENDIX C

ALTERNATIVE DESIGNS FOR MEASURING DYNAMIC COMPLIANCE

The recommended system for measuring track dynamic compliance requires that the measurement vehicle be stopped, the system set up, and the measurements taken.

In addition to this nonmoving system, three moving excitation systems were analyzed. These were the independent wheel suspension with pivoted arm support, Figure C-1; independent wheel suspension with parallel-arm support, Figure C-2; and modified rigid axle wheel suspension, Figure C-3. A preliminary analysis on the three moving systems showed that the independent wheel suspension with pivoted arm support did not provide any additional advantages over the other two systems and appeared to require more complicated fixtures to make the system function. On the basis of this preliminary analysis, the number of systems to be analyzed was reduced to two, and a detail design and analysis were completed on both systems.

Independent Wheel Suspension With Parallel-Arm Support

The various designs evaluated using the parallel-arm wheel suspension are shown in Figures C-4, C-5, and C-6 and these designs were to be mounted in the frame shown in Figure C-7. The motions desired from each suspension system are a pure vertical movement and a pure lateral movement. In the parallel-arm design, the vertical motion is obtained through the use of parallel arms which constrain the wheel in a vertical plane. The lateral motion is obtained by using hydrostatic bearings which allow both rotation and translation of the wheel on the axle. The lateral cylinder is connected to the wheel through various bearing configurations. These bearing configurations must allow for thrust in both directions. Figure C-4 is a design where the wheel is straddled by the vertical cylinders. Two vertical cylinders are needed to obtain the 35,000-pound vertical force and to reduce the required flow rate. One cylinder applies a constant downward force and the other vertical cylinder either counteracts the force or adds to it. The lateral cylinder is connected to the wheel through an X-roller bearing that can transmit thrust loads in both directions.

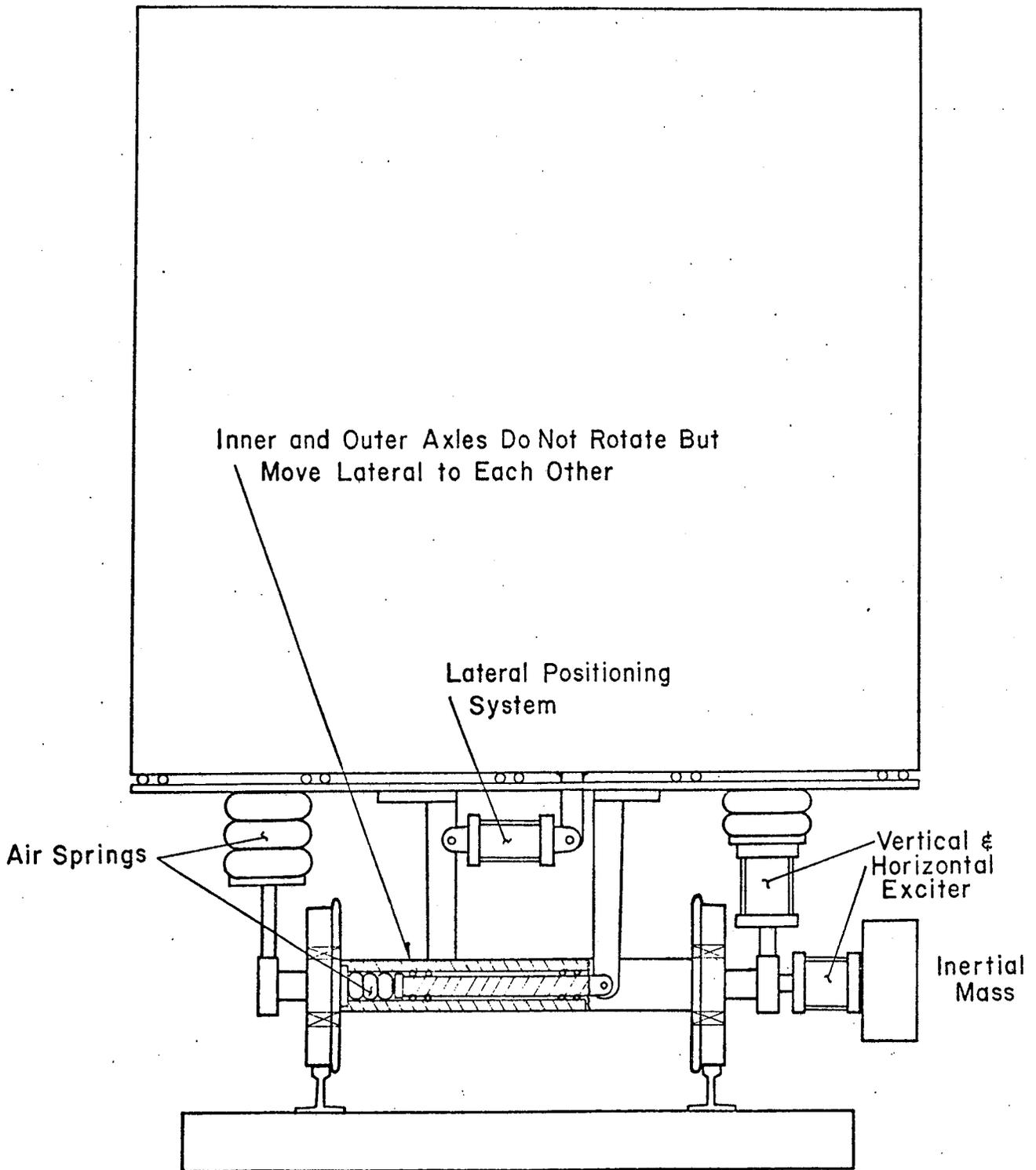


FIGURE C-1. INDEPENDENT WHEEL SUSPENSION WITH PIVOTED ARM SUPPORT

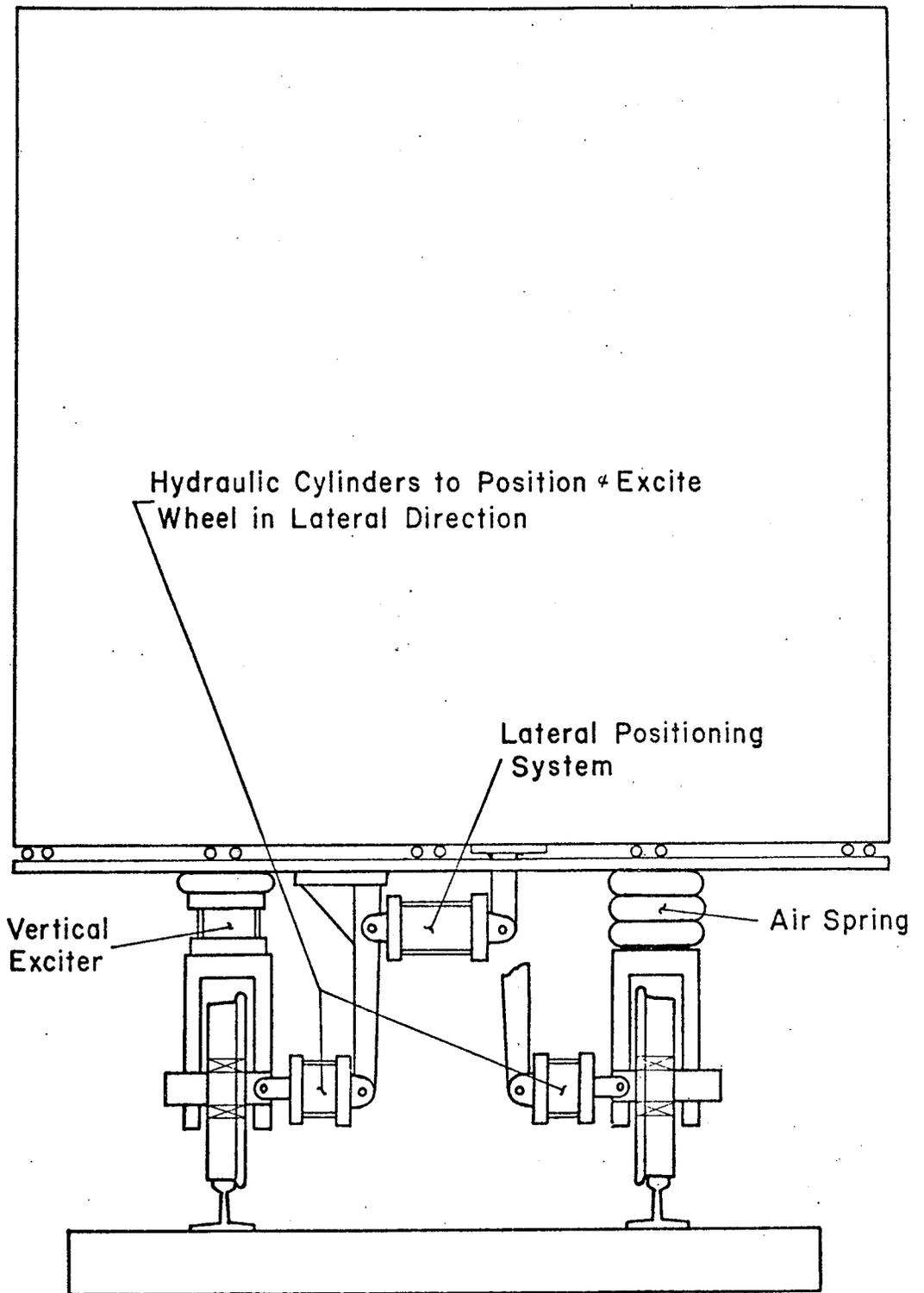


FIGURE C-2. INDEPENDENT WHEEL SUSPENSION SYSTEM WITH PARALLEL ARM SUPPORT

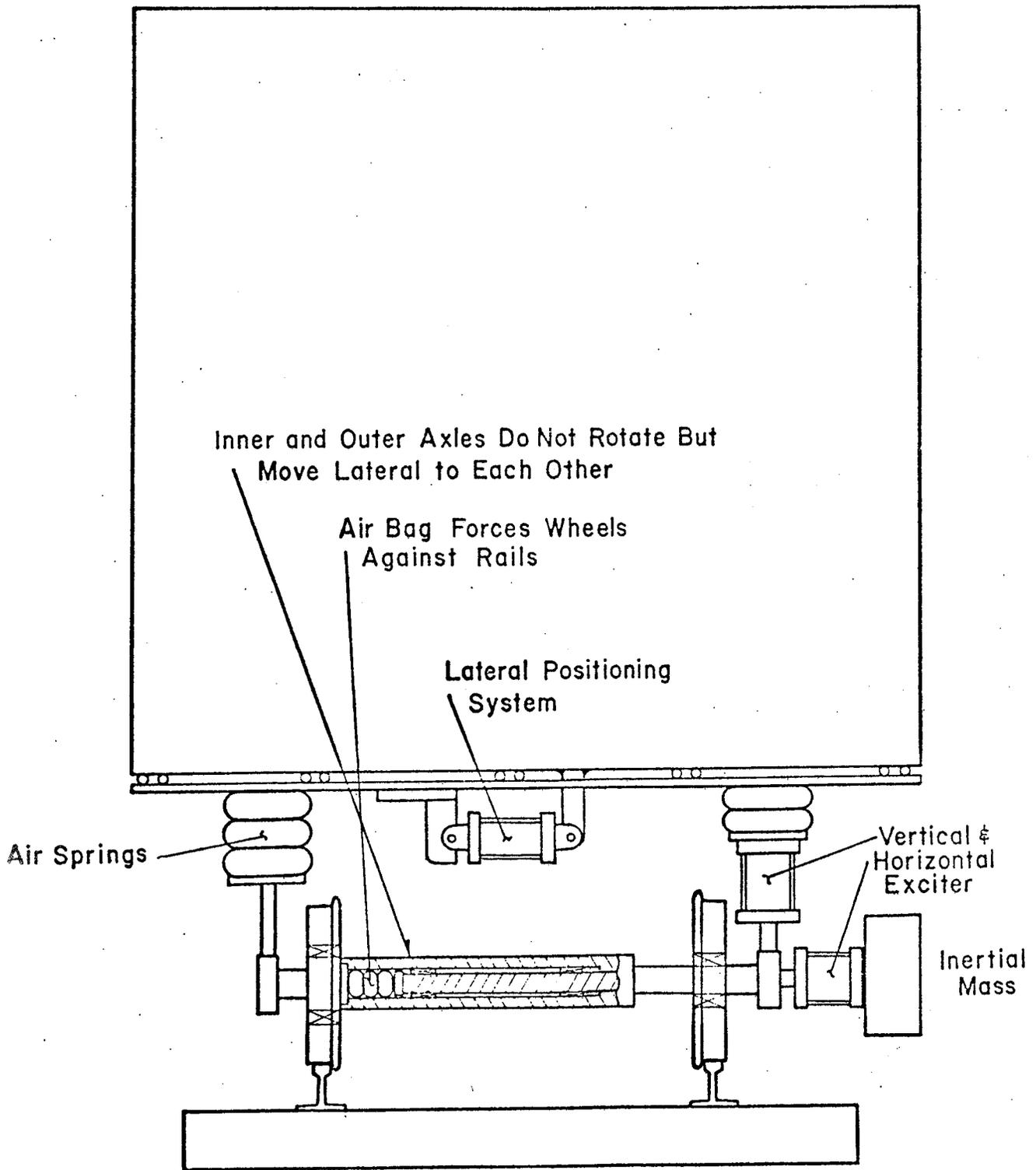
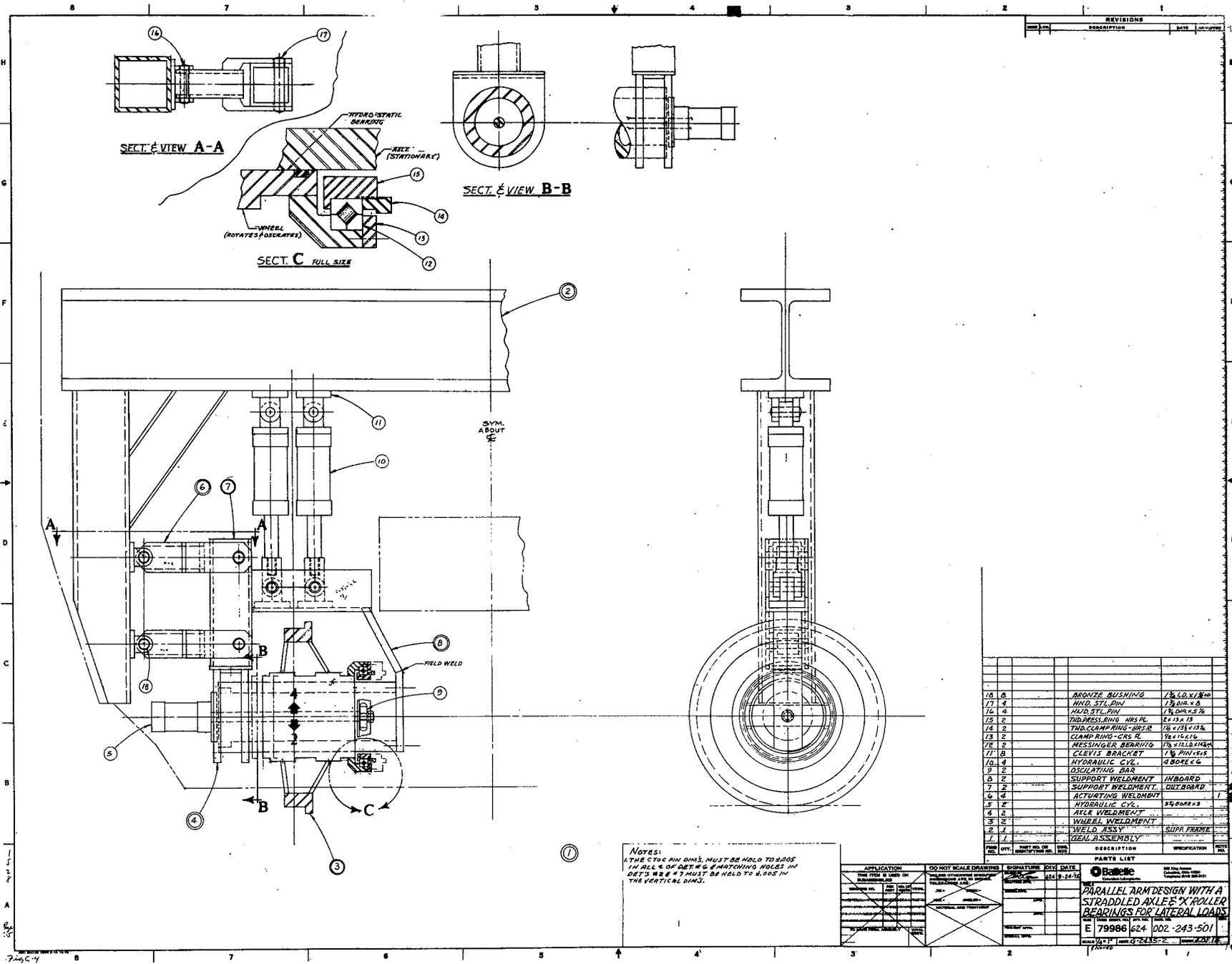


FIGURE C-3. MODIFIED RIGID AXLE WHEEL SUSPENSION

C-5



REVISIONS		
NO.	DESCRIPTION	DATE

QTY.	PART NO. OR IDENTIFYING NO.	DESCRIPTION	SYMBOL	UNIT
16	8	BRONZE BUSHING	1/2 LB. X 1/4"	
17	4	HYD. STL. PIN	1/2" DIA. X 8"	
14	4	HYD. STL. PIN	1/2" DIA. X 5 1/2"	
15	2	TRD. PRESS. RING WRS PL	2" X 1 1/2" X 1/8"	
14	2	TRD. CLAMPING W/RS	1/2" X 1 1/2" X 1/8"	
13	2	CLAMP RING-CRS. R.	1/2" X 1 1/2" X 1/8"	
7	2	MESSINGER BEARING	1/2" X 1 1/2" X 1/8"	
11	B	CLEVIS BRACKET	1/2" PIN X 5"	
10	4	HYDRAULIC CYL.	4 BORE X 4"	
9	2	OSCILLATING BAR	INBOARD	
8	2	SUPPORT WELDMENT	OUTBOARD	
7	2	SUPPORT WELDMENT	OUTBOARD	
6	4	ACTUATING WELDMENT		
5	2	HYDRAULIC CYL.	3/4" BORE X 3"	
4	2	AXLE WELDMENT		
3	2	WHEEL WELDMENT		
2	1	WELD TESS	SUPP. FRAME	
1	1	GEN. ASSEMBLY		

NOTES:
 1. THE CTOR PIN DIMS. MUST BE HELD TO .0005
 IN ALL % OF DET. & MATCHING HOLES IN
 DET'S BORE # 7 MUST BE HELD TO .0005 IN
 THE VERTICAL DIMS.

APPLICATION	DO NOT SCALE DRAWING	SIGNATURE	CHK	DATE

REV.	DATE	DESCRIPTION

Balluff
 PARALLEL ARM DESIGN WITH A STRADDLED AXLE & "X" ROLLER BEARINGS FOR LATERAL LOADS
 E 79986 624 1002-243-501
 1/19/70

FIGURE C-4. PARALLEL ARM DESIGN WITH A STRADDLED AXLE AND "X" ROLLER BEARINGS FOR LATERAL LOAD

C-6

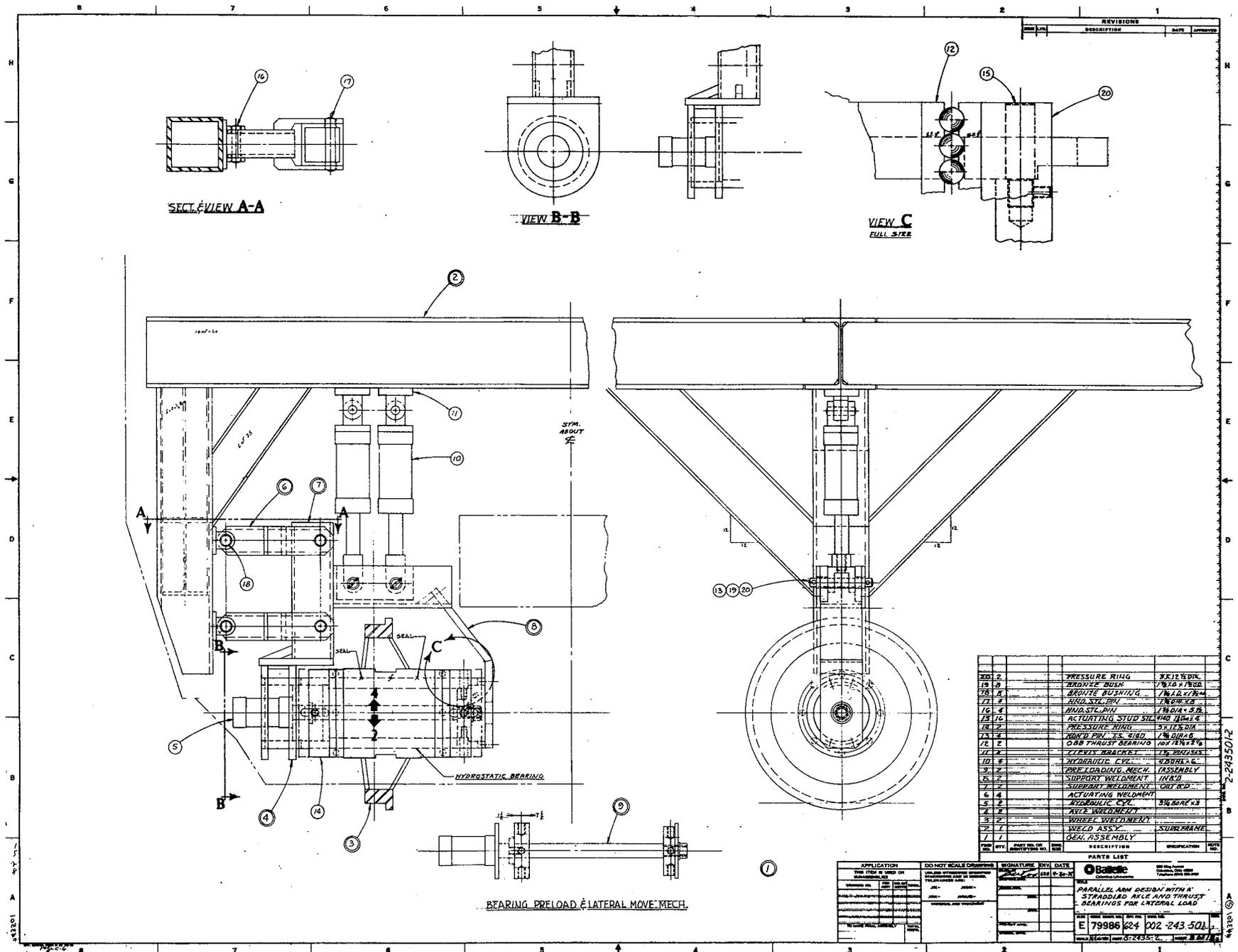


FIGURE C-5. PARALLEL ARM DESIGN WITH A STRADDLED AXLE AND THRUST BEARINGS FOR LATERAL LOAD

C-8

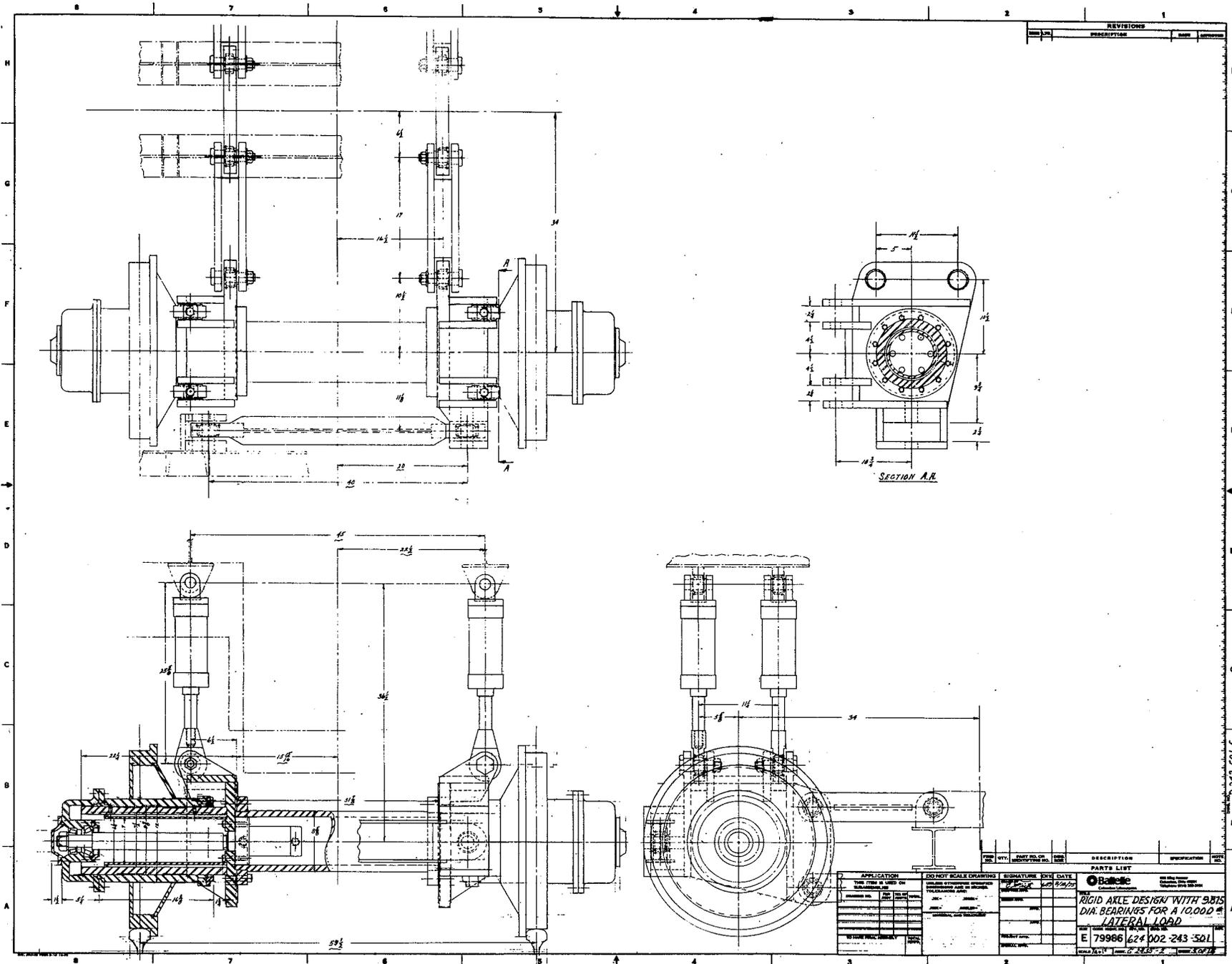


FIGURE C-7. RIGID AXLE DESIGN WITH 9.875 INCH DIAMETER BEARINGS FOR A 10,000 POUND LATERAL LOAD

The second version of this design is shown in Figure C-5. This design again has the cylinders straddling the wheel, the difference being in using two thrust bearings instead of the X-roller bearing to transmit the lateral load. This design still would require more work.

The last version using the parallel-arm design is shown in Figure C-6. This design has the wheel cantilevered and the lateral cylinder transmits the thrust through two tapered roller bearings. This design also requires more development.

An advantage of this parallel-arm system is the simplicity of design and no requirement of high center sills. A disadvantage for the parallel-arm design is the possibility of fixture resonances in the region of excitation, due to the numerous joints, since joint stiffness is difficult to predict.

Modified Rigid Axle Wheel Suspension

The modified rigid axle design is shown in Figures C-7, C-8, and C-9. Figures C-7 and C-8 show the front axle with the vertical and lateral cylinders. There are two versions of the front axle. The first design, Figure C-7, is sized for a lateral force capability of 10,000 pounds and the second version, Figure C-8, is designed for 35,000 pounds lateral load. Both versions were considered to determine the increase in cost for the additional lateral force capability.

Again there are two vertical cylinders per wheel to yield 35,000 pounds vertical force per wheel. In the lateral direction, cylinders mounted inside the axle apply lateral force in both directions. A hydrostatic wheel bearing is used to allow both rotational and translational motion of the wheel. This first axle will have excitation capability in excess of 100 Hz in both directions.

To obtain the independent control of each wheel, the axle is pivoted at two points. The first point is on the axle shown in Figure C-7. The other pivot is on the support frame shown in Figure C-10. With this continuation, one wheel can be held fixed and the other wheel forced vertically to yield independent control of each rail.

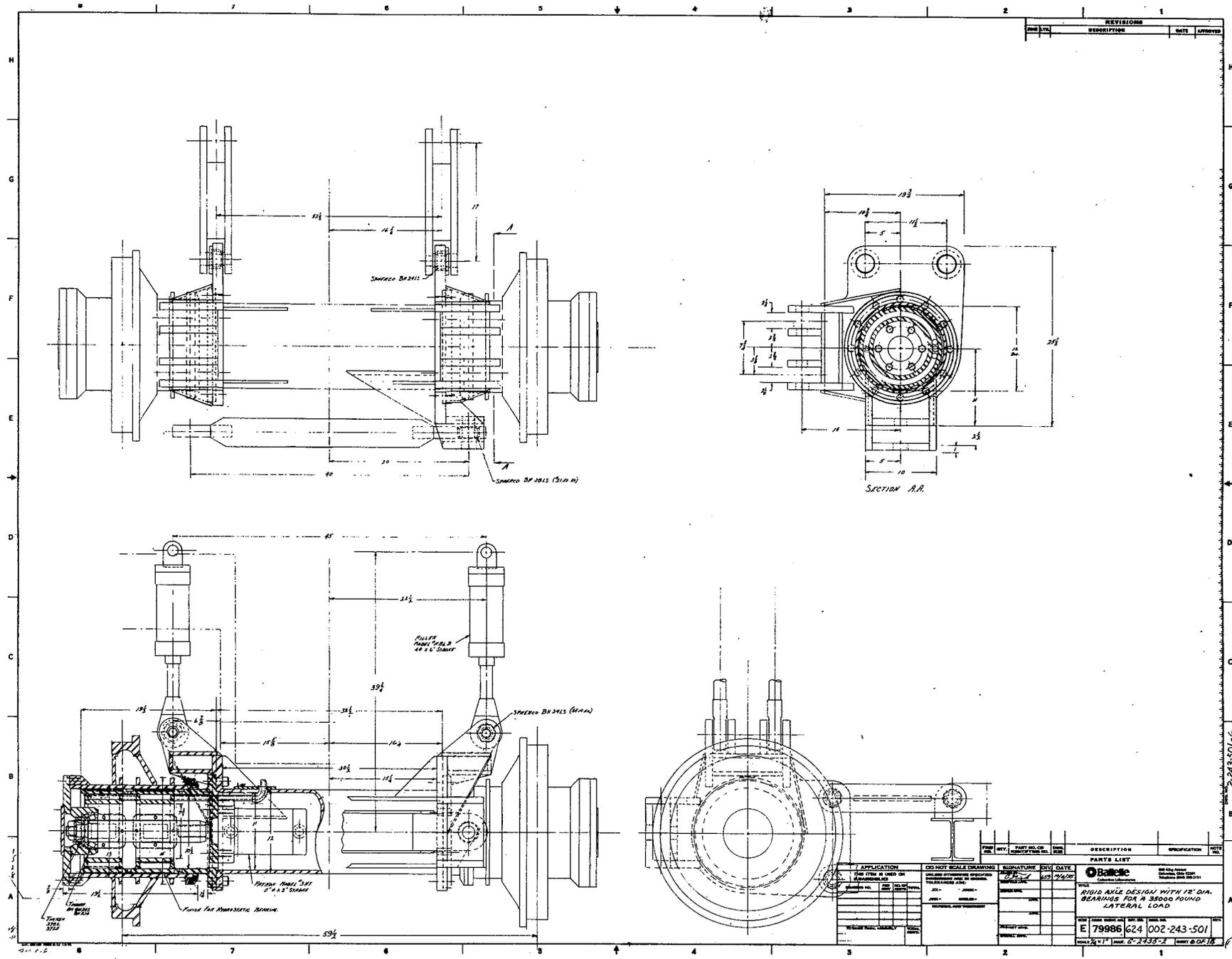
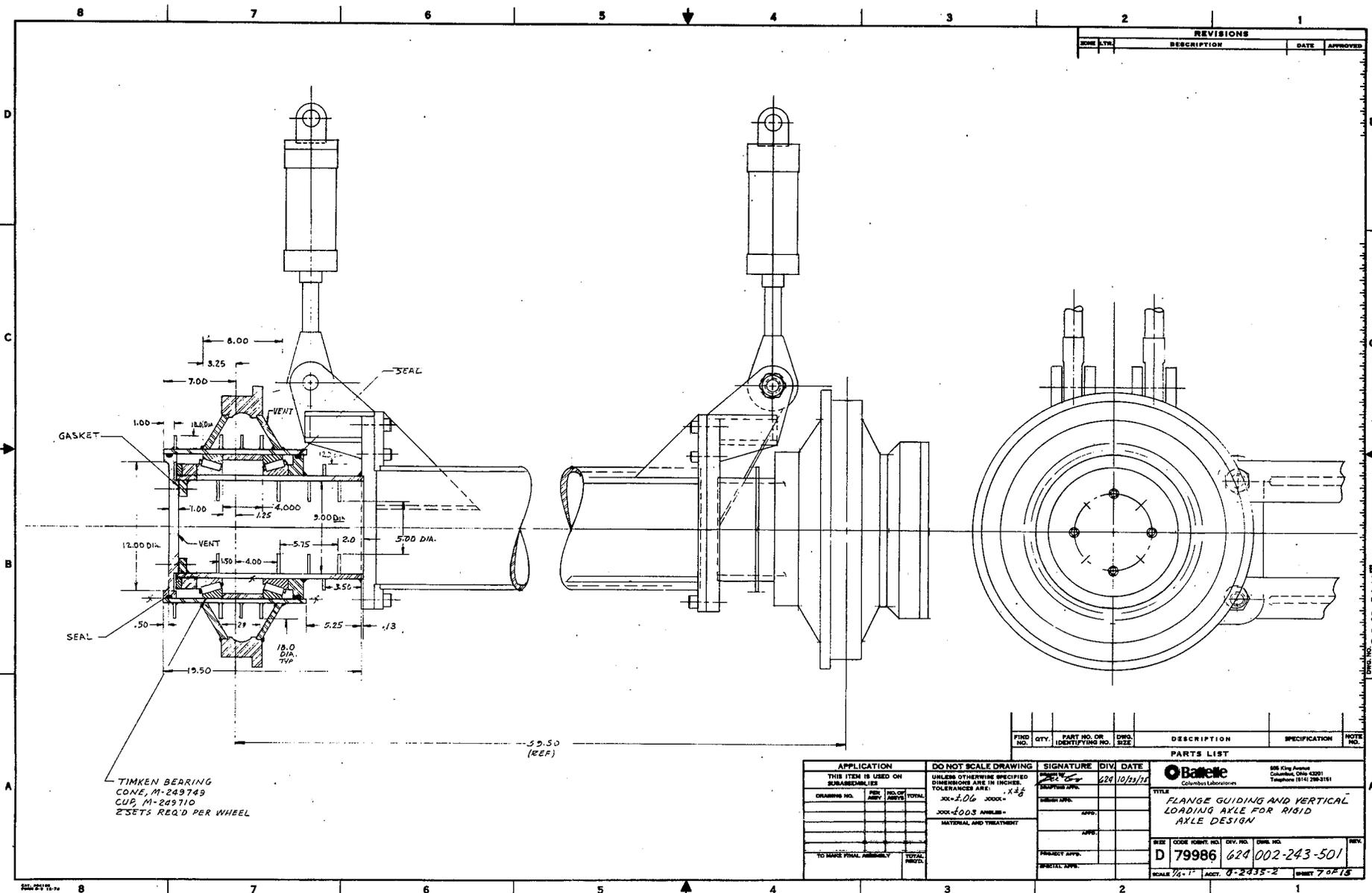


FIGURE C-8. RIGID AXLE DESIGN WITH 12 INCH DIAMETER BEARINGS FOR A 35,000 POUND LATERAL LOAD

2-243501-6

C-11



REVISIONS			
NO.	DATE	DESCRIPTION	APPROVED

QTY.	PART NO. OR IDENTIFYING NO.	DRWG. SIZE	DESCRIPTION	SPECIFICATION	NOTE NO.

APPLICATION THIS ITEM IS USED ON SUBASSEMBLIES		DO NOT SCALE DRAWING UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. TOLERANCES ARE: .001-2.00 .X25 .001-5.00 .X10 .001-50.00 .X10		SIGNATURE [Signature]		DIV. DATE 624 10/23/55	
DRAWING NO. 79986		MATERIAL AND TREATMENT		PROJECT APPR.		SPECIAL APPR.	
TO MAKE FINAL ASSEMBLY		TOTAL REQ'D		SCALE 1/2" = 1"		ACCT. 8-2435-2	

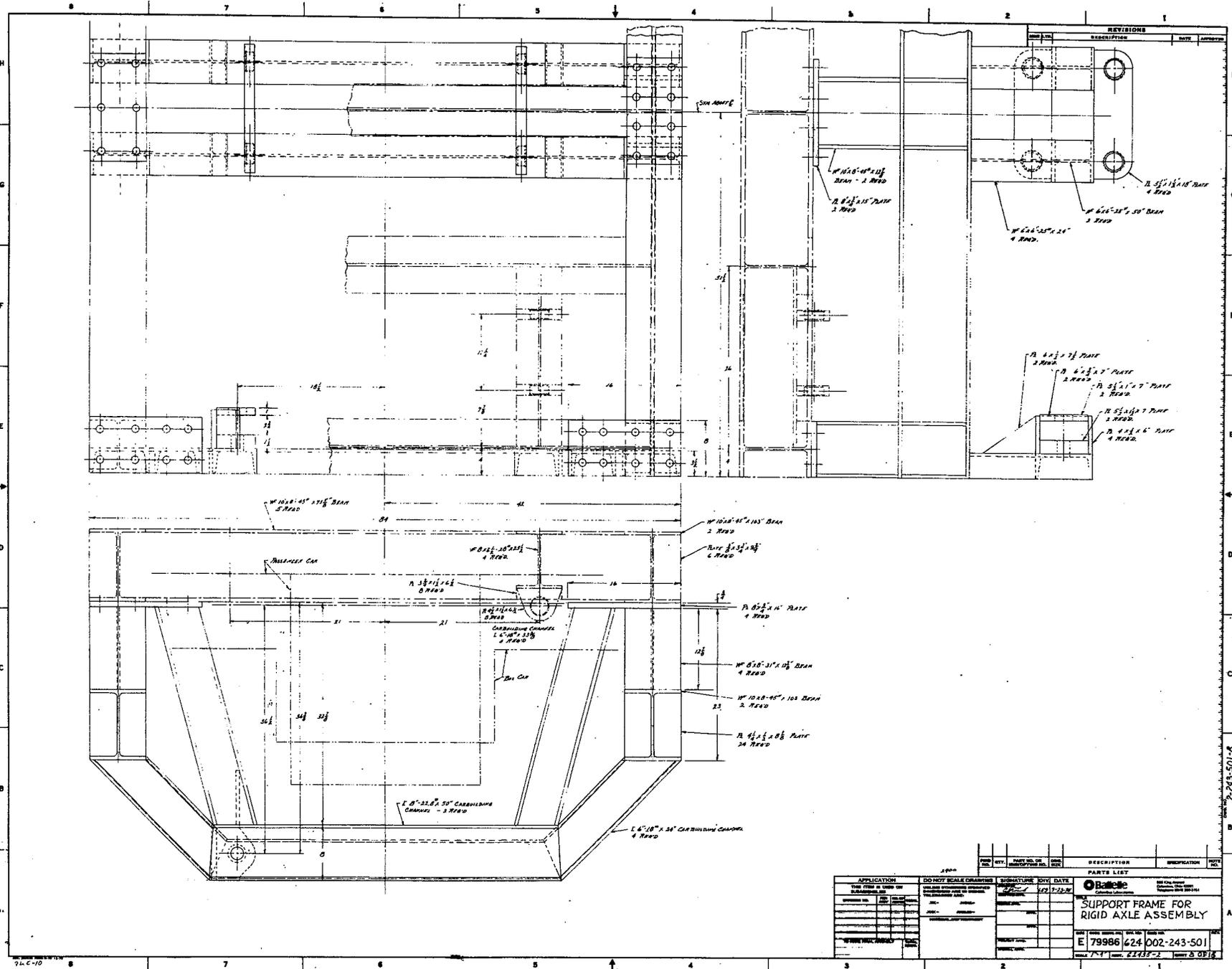
Barclay
 Columbus Laboratories
 688 King Avenue
 Columbus, Ohio 43201
 Telephone (614) 298-3161

TITLE
 FLANGE GUIDING AND VERTICAL LOADING AXLE FOR RIGID AXLE DESIGN

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 SHEET 7 OF 18

FIGURE C-9. FLANGE GUIDING AND VERTICAL LOADING AXLE FOR A RIGID AXLE DESIGN

C-12



The second axle, shown in Figure C-9, is the guiding axle for the lateral positioning system. This axle incorporates tapered roller bearings in the wheel and uses flange contact for guiding the axle. The axle is free to move laterally with the car. There is an angular transducer that senses the angle of the vertical cylinders and signals the lateral positioning system to shift laterally to correct the angle of the vertical cylinders. This lateral positioning frame is driven by hydraulic motors (see Figure C-11). In this manner the excitation system truck will follow the rail and negotiate curves. The second axle will have a vertical force capability of 35,000 pounds up to an 8 cps rate.

Some of the vehicles which were considered in this design study did not have sufficient bending stiffness to limit the car body center deflections under dynamic forcing. For this reason, a truss system was designed to support the lateral positioning carriage. This system is shown in Figure C-11. The framework of the recommended system is stiff enough so that the truss system is not needed here.

