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Administration

DYNAMIC BUFF AND DRAFT TESTING TECHNIQUES

Office of Research and
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Washington D.C. 20590

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Transportation Test Center
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Final Report

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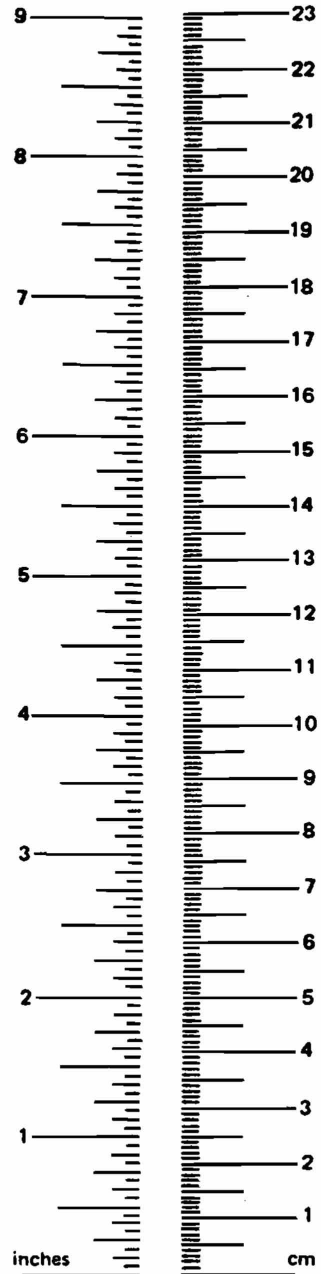
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

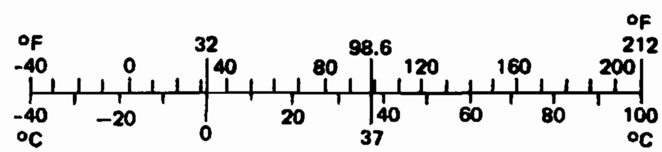
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.50	centimeters	cm
ft	feet	30.00	centimeters	cm
yd	yards	0.90	meters	m
mi	miles	1.60	kilometers	km
		AREA		
in^2	square inches	6.50	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.80	square meters	m^2
mi^2	square miles	2.60	square kilometers	km^2
	acres	0.40	hectares	ha
		MASS (weight)		
oz	ounces	28.00	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.90	tonnes	t
		VOLUME		
tsp	teaspoons	5.00	milliliters	ml
Tbsp	tablespoons	15.00	milliliters	ml
fl oz	fluid ounces	30.00	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.80	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

* 1 in. = 2.54 cm (exactly)



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.40	inches	in
m	meters	3.30	feet	ft
m	meters	1.10	yards	yd
km	kilometers	0.60	miles	mi
		AREA		
cm^2	square centim.	0.16	square inches	in^2
m^2	square meters	1.20	square yards	yd^2
km^2	square kilom.	0.40	square miles	mi^2
ha	hectares (10,000 m^2)	2.50	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.10	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	36.00	cubic feet	ft^3
m^3	cubic meters	1.30	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius* temperature	9/5 (then add 32	Fahrenheit temperature	°F



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16. Abstract

The Federal Railroad Administration and the Association of American Railroads have jointly sponsored a program for developing cost-effective and rational techniques for evaluating the curving performance of any freight car under buff and draft conditions. The procedures are developed as possible inclusions to AAR *Manual of Standards and Recommended Practices*, Specification M-1001, Chapter XI.

Under this program, an analytical method was developed for predicting coupler angles and forces for any given coupling arrangement and operating conditions. The method was validated in a series of full scale track tests. A test vehicle was also designed and constructed for the purpose of applying controlled levels of lateral coupler loads at the coupler pin of any adjacent candidate car.

The test car was used to evaluate the safe curving limits of two candidate cars, namely the Frontrunner and the 89-foot flatcar, under buff and draft loads of up to 250,000 pounds and for a 10-degree curve. The tests, supported by analysis, showed that coupling of the Frontrunner to the 89-foot flatcar reduces the flatcar's ability to negotiate curves safely for draft loads higher than 220,000 pounds and buff loads higher than 180,000 pounds.

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EXECUTIVE SUMMARY

The in-train stability of freight cars is important to the operating railroads. Recent trends of operating heavier and longer mixed trains, combined with the widespread use of dynamic braking in preference to air braking, can lead to derailments due to excessive slack action and the resulting buff and draft loads.

Therefore, an ad hoc committee was formed by the Association of American Railroads (AAR) and the Federal Railroad Administration (FRA) to consider future methods for certification of new freight cars. The committee identified the development of dynamic buff and draft testing techniques as an important requirement. Accordingly, the AAR and FRA agreed to jointly fund the initial phase of a program for the development of cost-effective test procedures. The proposed techniques are to be submitted to the AAR's Car Engineering Committee for their consideration as potential inclusions to Chapter XI of the AAR *Manual of Standards and Recommended Practices*, Section C-II, Specifications for Design, Fabrication, and Consturction of Freight Cars, Chapter XI.

The proposed method, consisting of testing aided by analysis, is based on the fact that when a car is subjected to buff and draft loads, only the lateral and vertical components of these loads affect the stability of its performance. Therefore, a reasonable assessment of the performance of a car may be made if the longitudinal forces are substituted with their effective lateral and vertical components. An advantage of this concept was that the required loads now could be limited, based on the limits of coupler angles, to a more easily achieved range of perhaps, 10 to 60 thousand pounds. However, an additional requirement that this method presented was the need to determine the coupler angles and forces associated with each dynamic event in order to predict the corresponding lateral and vertical components. Based on this alternative concept, the AAR and FRA have agreed to jointly fund Phase I of this program consisting of the following tasks:

- Determination of coupler angles
- Transient versus steady state coupler characteristics
- Design/construction of buff and draft load simulation car
- Pilot dynamic buff and draft testing of a sample car

The relationship between buff and draft forces and their corresponding coupler loads and moments was determined analytically and formulated in an IBM compatible computer program called Coupler Angling Behavior Simulator (CABS), which predicts the coupler angles and forces generated for any given conditions. CABS was validated in a series of track tests in which measured coupler angles were compared to those predicted by CABS. The results of the validation showed that CABS correctly predicted the average coupler angles produced under constant curving conditions.

One of the primary tasks in this project was the design and construction of a vehicle capable of applying controlled levels of lateral force at the coupler pin of any adjoining car. A prototype buff and draft car was designed, constructed, and tested. Preliminary track tests have identified a problem in applying steady state loads to the test vehicle while in motion. This was determined to be the result of an under powered hydraulic system. Another objection to the prototype design was that the coupling between the test vehicle and the buff and draft car was not made through the conventional coupler arrangement. Instead, a specially fabricated section was attached to the test car to transmit the applied loads. This linkage raised questions regarding the similarity of the attachment to the actual coupler/draft gear combination. Another problem with the linkage was the impracticality of uncoupling from one car and coupling to another. In summary, however, the prototype car proved the feasibility of the proposed method.

Sample cars selected for the initial phase of testing with the buff and draft car were the Frontrunner and the 89-foot flatcar. The Frontrunner was selected first. This car was well characterized under a previous FRA project at TTC and therefore accurate modeling of its performance was possible. Both cars were tested in the empty configuration only.

Results from testing the Frontrunner indicated that the maximum lateral coupler load that can be applied safely is 11,000 pounds in either direction. An increase of the coupler load above this limit caused occasional wheel lift between the wheel and rail opposite to the direction of load application. The average vertical wheel loads dropped below the recommended minimum (Chapter XI) of 10 percent of the static load. A simulation of the test conditions with the computer program New and Untried Car Analytic Regime Simulation (NUCARS) showed excellent agreement with the measured forces. The results also proved that the buff and draft car succeeded in applying the required lateral coupler loads in the manner for which it was designed. The simulation also indicated an increasingly positive angle of attack between the outer wheel and rail under buff conditions.

Results of the 89-foot flatcar testing as well as the predictions from NUCARS indicated that the maximum lateral coupler load that can be applied safely is 26,500 pounds in either direction. An increase of the coupler load above this limit caused the lead axle's L/V ratio to exceed 1.0 on the side of the direction of load application. The analysis agreed well with the test data. The predicted lateral shift and yaw for the lead and trail axles of the lead truck were similar to those of the lead axle of the Frontrunner with one notable difference. Unlike the Frontrunner, the simulation predicted a negative angle of attack between both axles and the outer rail under buff conditions. This is primarily due to the differences in reacting the buff loads through a single axle (the Frontrunner) versus reacting them through a conventional three-piece truck.

Based on the limits established from testing with the buff and draft car a number of simulations were conducted on the coupling of Frontrunner to the 89- foot flatcar and the coupling of each to like cars. The analysis considered buff and draft loads of up to 250,000 pounds. The results indicated that for the 10-degree curve examples the Frontrunner will perform safely under both conditions of coupling within the given limits of buff and draft forces. When coupled to an 89-foot flatcar in lead, the Frontrunner experienced considerably lower lateral coupler loads suggesting an adverse effect on the performance of the flatcar. The results indicated a safe performance of the 89-foot flatcar when coupled to a like car. However, coupling of the flatcar to the Frontrunner reduced the ability of the flatcar to safely negotiate curves for draft loads in excess of 220,000 pounds and buff is in excess of 180,000 pounds.

Results obtained from the pilot testing proved the feasibility of the buff and draft car in applying controlled levels of lateral coupler loads to any candidate freight car. It is recommended that the procedure described here be included in Chapter XI, Section 11.7, in order to verify the capability of a new car design to operate in trains with sustained buff and draft loads. It recommended that further effort be invested in developing the specific Chapter XI provisions based on the proposed techniques.

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1.0 INTRODUCTION

The term "Buff and Draft" refers to the longitudinal forces which can develop throughout a train due to the continuous adjustments in tractive or braking effort necessary for its handling, or due to gravitational forces in the presence of a grade. In general, these forces may be initiated at one point along the train and gradually build up and be transmitted through any number of cars within the train. An example is when a dynamic (locomotive only) brake application is made, causing a squeezing action on the train as the freely rolling cars are pushed against the increasingly resisting locomotives. In this example, the extent to which the buff forces will travel will depend on the duration and intensity of the brake application.

The in-train stability of freight cars is of importance to the operating railroad. This has become even more important because of the widespread use of dynamic braking in preference to air braking. Longer mixed consist trains usually include cars having end of car cushioning devices with accompanying slack. This aggravates run-ins and run-outs creating high buff and draft forces. With conventional freight cars having three-piece trucks and a conventional draft system arrangement, a level of tolerance to either steady state or transient buff and draft forces has become expected. Currently this level of expectation is a 200,000-pound steady state buff and draft force on a car operating on a 10-degree horizontal curve (or sharper). The expectation for vertical curves, although not explicitly stated, is that all existing vertical curves will be negotiated. The most extreme condition is the case of a car coupled to another car which is loaded and has a sagging coupler with a coupler height of 31.5 inches. When a car's buff and draft force sustaining capability is exceeded, one or more of the following conditions happen:

- Wheel climb may occur and derailment ensues.
- Lateral jackknifing may occur with the coupler angling at a limit.
- Vertical jackknifing can occur with center plate lift-off.
- Track gage widening may occur in an extreme case.

Buff and draft forces can have an adverse effect on the lateral stability of either the train or the track on which it is rolling. The work described here focuses only on the lateral stability of the train. Of particular interest is the curving performance under such conditions. The negotiation of curves and spirals requires couplers to angle with respect to a car's longitudinal axis. The coupler angling, if combined with buff and draft forces, can reduce the stability of a moving car by causing it to react to some of the buff and draft force at the wheel/rail interface in a direction lateral to the car body. Since buff and draft forces are transmitted through the couplers at a typical height of some 32 inches above the rail surface, their net effect is that of a lateral force and a roll moment as seen by the wheels and rails (see Figure 1).

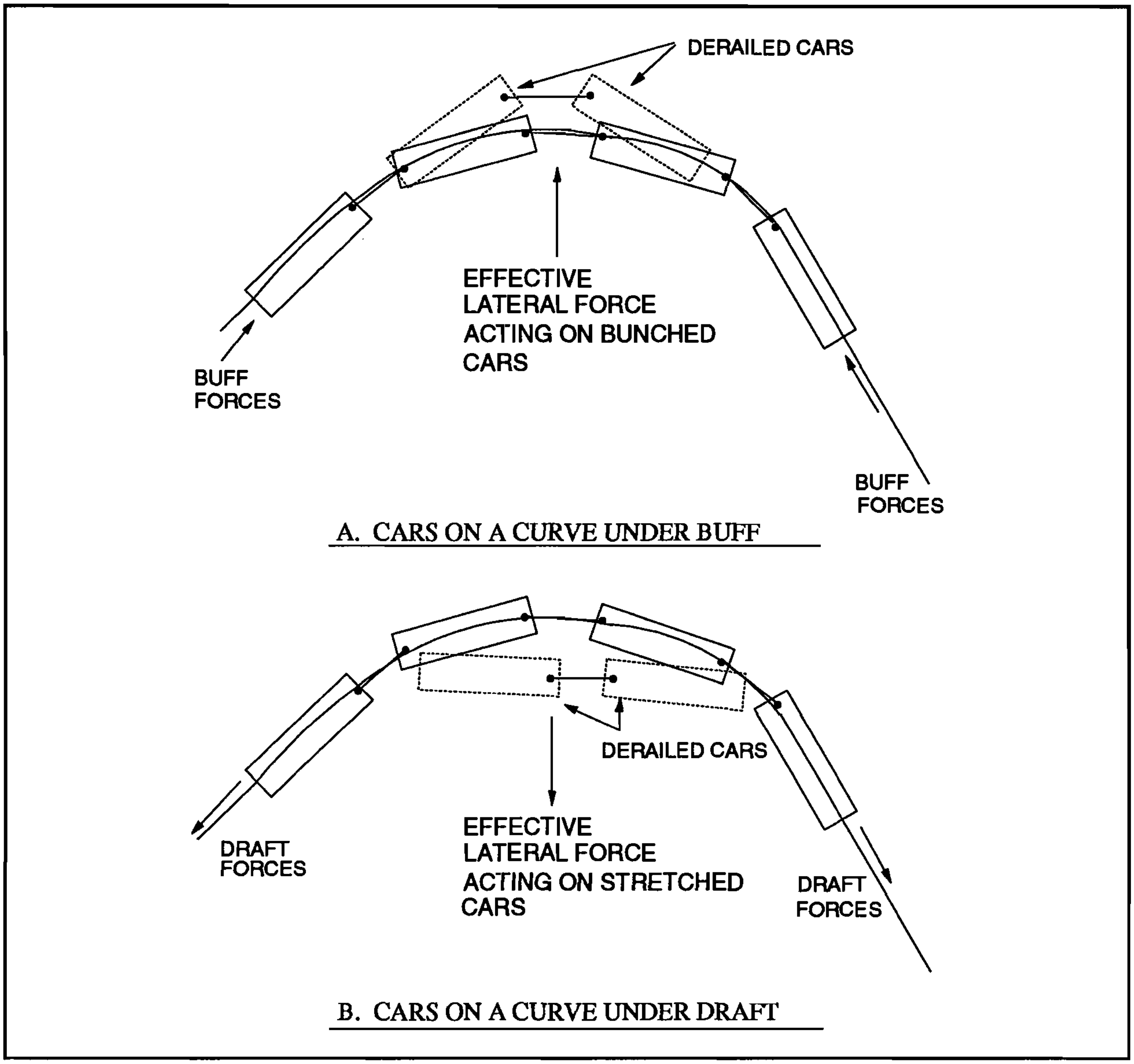

Figure 1. Lateral Force due to Buff or Draft Loads

In 1990, the Federal Railroad Administration (FRA) reported a total of 2,146 derailments at a direct cost to the railroad industry in excess of $150 million (estimated total cost of at least $330 million). Roughly 200 of these incidents were reported to be due to a slack action or an excessive drawbar force at a direct cost of $11 million (estimated total cost of at least $25 million).[1]

From the previous discussion, it is evident that buff and draft forces play a role in causing some derailments. Accordingly, when an ad hoc committee was formed by the Association of American Railroads (AAR) and FRA to consider future methods for certification of new freight cars, the committee identified the development of dynamic buff and draft testing techniques as an important requirement. To achieve this goal, the AAR and FRA jointly sponsored the initial phase of a program for the development of cost-effective test procedures. The proposed techniques will be submitted to the AAR Car Engineering Committee (CEC) for its consideration as inclusions to Chapter XI, Service-Worthiness Tests and Analyses for New Freight Cars, of the AAR *Manual of Standards and Recommended Practices*. Currently, the only specification within Chapter XI of the Manual for buff and draft testing is the static curve stability requirement. In this test, an empty vehicle is subjected to static buff and draft loads of 200,000 pounds for a sustained duration of 20 seconds each, while positioned in a curve of at least 10 degrees. Under such loads, requirements must be met regarding car body-to-truck and wheel-to-rail separation.[2,3] This requires sufficient motive power to apply the force and an equal retarding capability to serve as the reaction means. This approach has been followed for cars with new draft arrangements as well as cars with new light weight car bodies and new trucks. A dynamic test requirement is mentioned in Chapter XI, although the section is currently left blank in anticipation of the development of an acceptable test procedure.

The task of imparting controlled levels of buff and draft loads to a vehicle at user defined intervals, and while in motion, presents a number of practical problems. The traditional approach is to apply the power of a number of locomotives to one end of the test vehicle while retarding at the other end with a number of cars with their air brakes applied, or with additional locomotives under dynamic brake. Testing of this type has been done in the past at the Transportation Test Center (TTC), Pueblo, Colorado. Difficulties in controlling the level of buff and draft applied and the requirement of many locomotives and/or cars rendered this option less practical. Another approach is to utilize a servo controlled electrohydraulic system for imparting the buff and draft loads directly to the vehicle under testing. However, a desired buff and draft load range of 200,000 pounds to 400,000 pounds makes the cost and size of the hydraulic system prohibitive.

An alternative concept to the longitudinal load requirement was suggested based on the fact that when a car is subjected to buff and draft loads, only the lateral and vertical components of these loads affect the stability of its performance. Therefore, a proper assessment of the performance of a car may be made if the longitudinal forces are substituted with their effective lateral and vertical components. An advantage of this concept is that the required loads could be limited, based on the limits of coupler angles, to a range of perhaps 10,000 pounds to 60,000 pounds. However, an additional requirement that this method presents is the need to determine the coupler angles and forces associated with each dynamic event in order to predict the corresponding lateral and vertical components. Based on this alternative concept, the AAR and FRA jointly sponsored Phase I of this program consisting of the following tasks:

- Determination of coupler angles .... AAR Funded
- Transient versus steady state coupler characteristics .... AAR Funded
- Design/construction of buff and draft load simulation car .... FRA Funded
- Pilot dynamic buff and draft testing of a sample car .... FRA Funded
- Buff and draft car simulations versus locomotive testing .... AAR Funded

This report describes the progress made on each of these tasks during the completion of Phase I. The performance period of this phase extends from January 1, 1990 through June 30, 1992.

2.0 OBJECTIVE

The objective of this program is to develop cost-effective and rational techniques for evaluating the curving performance of any freight car under buff and draft conditions. The developed procedures are suggested as possible inclusions to Chapter XI.

3.0 DETERMINATION OF COUPLER ANGLES

The objective of this task was to provide the necessary link between buff and draft forces and their orthogonal force components. In general, vertical curves are considerably shallower than horizontal curves; therefore, it was decided to limit the analysis and testing to the lateral component of the buff and draft force. Since the proposed testing technique replaces the buff and draft load with its lateral component at the coupler, an analytical method was needed to determine the component for a given buff and draft load level, track curvature, and coupling arrangement. A method was developed in the form of an IBM compatible computer program called Coupler Angling Behavior Simulator (CABS), which predicts the coupler angles and forces generated for any given conditions. CABS was validated in a series of track tests in which measured coupler angles were compared to those predicted by CABS.[4] The results of the validation showed that CABS correctly predicted the average coupler angles produced under constant curving conditions.

Figure 2 shows the mechanical model adopted in CABS. The model may consist of any number of cars coupled as shown. A nonlinear response can be assumed at any of the rotational or linear connection elements. A combination of elements may be assumed at a single connection. The model permits the longitudinal travel of the coupler pivot (draft gear travel). The influence of the lateral shift in the coupler pivot due to the presence of any special alignment features can be also modeled.

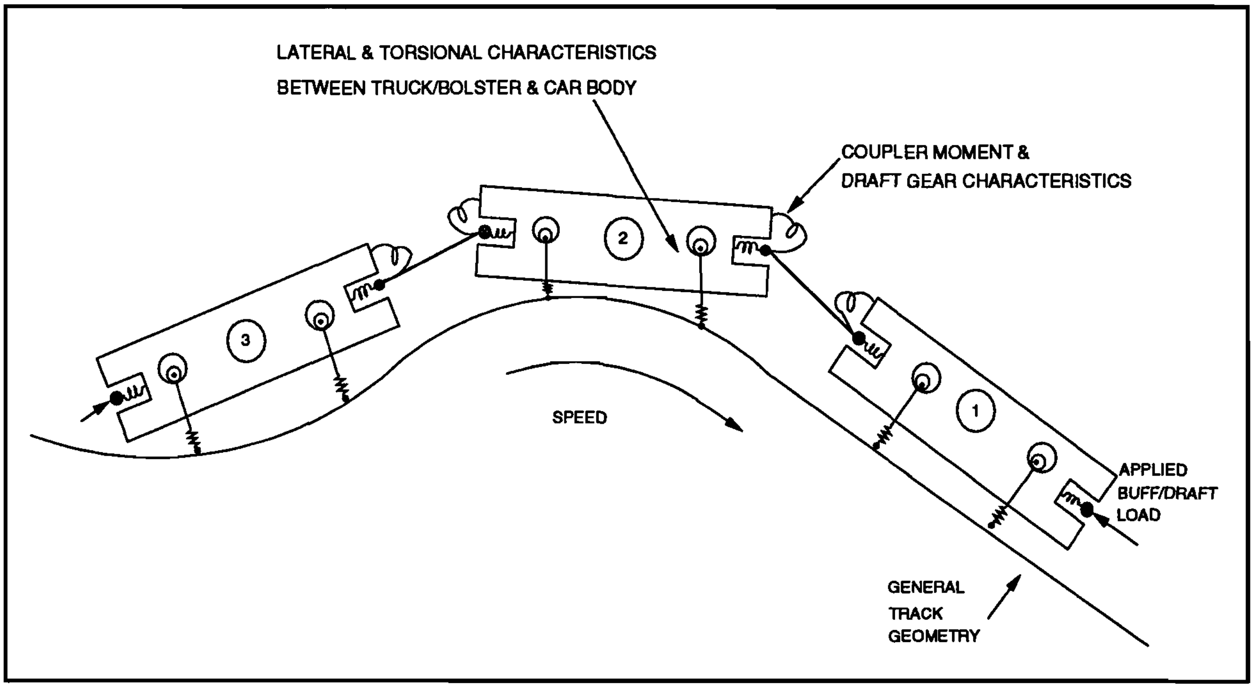

Figure 2. Mechanical Model of CABS

By considering the equilibrium of forces acting on a coupler connection, the net lateral force generated at a coupler pin depends not only on the buff and draft load present, but also on the net coupler moment and length, and the angle of the coupler at that pin. The contribution of the net moment to the lateral load may be comparable to that of a relatively large buff and draft force.

In the proposed test methodology, CABS is used to predict the lateral loads generated in a given car combination, track curvature, and load scenarios. For the candidate car, the coupler yaw characteristics for buff and draft loads of up to 250,000 pounds must be determined. This can be measured by test, an example of which may be found in references 5 and 6. For a new car design, it is recommended that an analysis be made for a 250,000-pound sustained buff and draft load, on a 10-degree curve or tighter, and a coupling arrangement with at least one additional car which provides the longest possible truck base and overhang. For the purpose of analysis or testing, it is proposed that the buff and draft load be replaced with the resulting components acting laterally with respect to the car body.

Since its inception, CABS has been refined to include some of the following features:

- ability to analyze the coupling response of longer consists,
- addition of on-screen interactive graphics to view any of the simulation parameters while computed,
- addition of a library of elements including a special element for modeling dry friction.

Accurate modeling of dry friction is an important aspect since laboratory and field testing indicated that the characteristics of most couplers are dominated by this type of friction. Further details on the mechanical model adopted in CABS and results of validation field testing are found in AAR report R-772.[4]

3.1 TRANSIENT VERSUS STEADY STATE COUPLER CHARACTERISTICS

Initial analysis using CABS revealed that coupler rotational characteristics may influence the lateral coupler forces. The equilibrium of forces and moments acting on a coupler connection show that coupler moments influence the level of lateral loads produced at the coupler pin. The moments depend largely on the degree of coupler rotation or angling, and the characteristics of the coupler shank assembly. Although the testing method proposed in this program is based on evaluating the steady state curving performance under constant coupler loads, a question arose as to the influence of variable or transient in-train forces on coupler characteristics. In order to evaluate coupler performance under a spectrum of buff and draft loads, a series of tests were conducted at the TTC in January 1991. Using the Mini-Shaker Unit, characterization of F and E style couplers were performed. The resistance to coupler angling to quasi-static buff and draft loads was measured under a range of realistic load and boundary conditions. The testing was done on a laterally and longitudinally restrained aluminum gondola. Each coupler was tested for loads ranging from +100,000 pounds to -100,000 pounds. The lateral and longitudinal coupler loads were applied using a pair of servo-controlled hydraulic actuators.

Results of the test indicated that frictional losses were generally higher in the F coupler than in the E coupler. The differences were more pronounced under buff loads. This was explained based on the differences in the mechanical configuration of each coupler type. Visual inspection of both couplers before and after testing showed substantial wear in the spherical butt area of the F coupler. In contrast, minor wear was observed near the edges of the relatively flat E coupler butt. Wear was also observed in the pin area of the F coupler especially towards the rear end of the shank indicating friction under draft loads (Figure 3). For the E coupler, wear was mostly confined to the key and slot in the coupler shank and yoke. Minimum friction did not occur at zero loads, but rather at a buff load in the range of 25,000 pounds to 50,000 pounds. Further details are found in AAR working paper WP-151.[6]

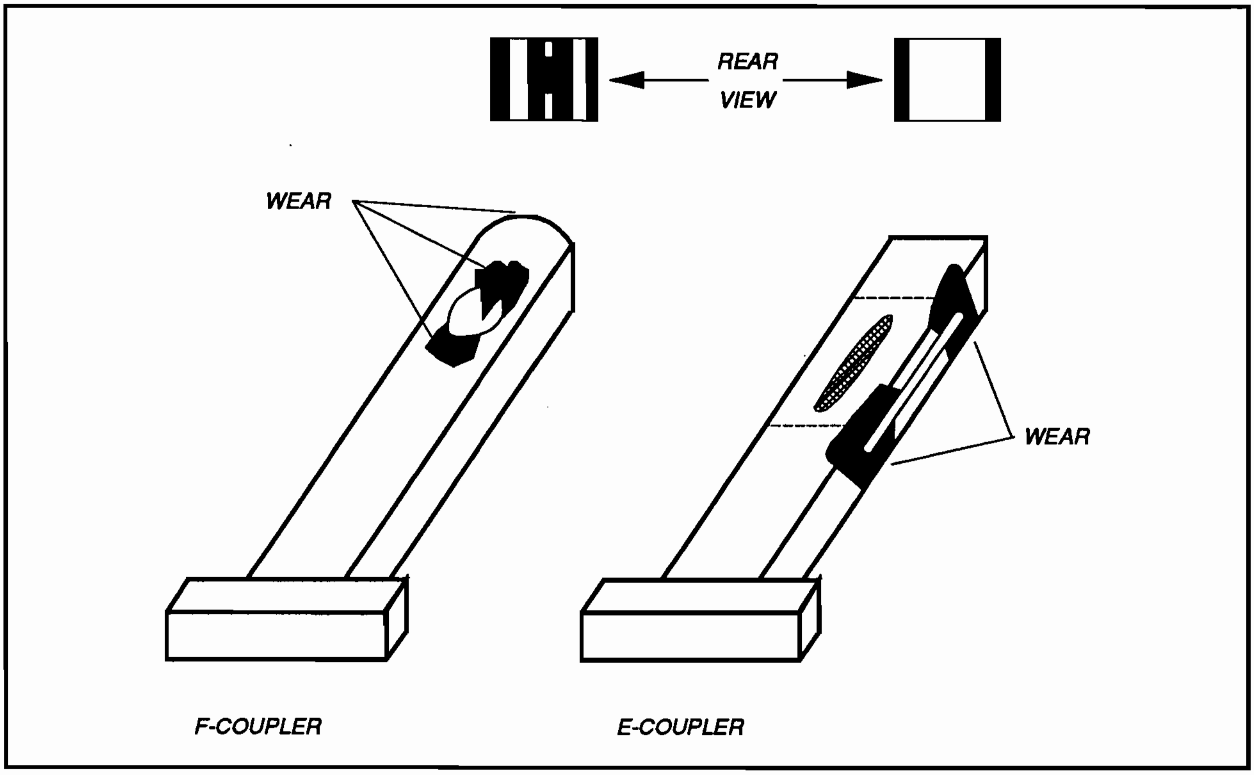

Figure 3. Wear F and E Couplers

4.0 BUFF AND DRAFT LOAD SIMULATION CAR

One of the primary tasks in this project was the design and construction of a vehicle capable of applying controlled levels of lateral force at the coupler pin of any adjoining car. Prior to the start of the performance period of this project, a prototype buff and draft car was designed, constructed, and tested. The following is a brief description of the prototype car and the conclusions made from its testing.

4.1 PROTOTYPE BUFF AND DRAFT TEST CAR

A prototype buff and draft load simulation car was designed and constructed for the purpose of verifying the feasibility of imparting lateral and vertical loads to a moving test vehicle in a controlled environment. The car was equipped with hydraulic actuators capable of applying loads of up to 55,000 pounds to the end of a test vehicle. Each actuator was servo-controlled and operated in a load feedback loop. Preliminary track tests have identified a problem in applying steady state loads to the test vehicle while in motion.

This was determined to be the result of a servo-control load feedback loop mechanism which was too slow to react to the continual load variations during motions. The hydraulic pump used in this prototype was limited to a flow rate of only 30 gpm which limited the dynamic response of each of the servo valves used.

Another objection to the prototype design was that the coupling between the test vehicle and the buff and draft car was not made through the conventional coupler arrangement. Instead, a specially fabricated section was attached to the test car to transmit the applied loads. This linkage raised questions regarding the similarity of the attachment to the actual coupler/draft gear combination. Another problem with the linkage was the impracticality of uncoupling from one car and coupling to another.

In summary, however, the prototype car proved the feasibility of the proposed method. The identified problems were solved in the design of the new buff and draft car.

4.2 DESIGN/CONSTRUCTION OF THE NEW BUFF AND DRAFT CAR

The new buff and draft test car was constructed from a 55-foot-long flatcar which formerly belonged to the U.S. Department of Defense. The car was supported by a pair of three-axle trucks. A request to the U.S. Army was made for permanently donating the car to this project. Once the request was approved by the Army, proprietary work on the car body began. The car was initially covered with a wooden deck which was in a poor condition and had to be completely removed. The car body and trucks were sandblasted in preparation for painting. The removal of rust and old paint was also done in order to facilitate the cutting and welding required by the new design. A new metal flooring was installed on the deck and painted along with the car body and trucks. An aluminum container was then mounted on the deck, creating a room for housing all the hydraulic and electric components. Additional miscellaneous modifications were made to install an access door and windows for lighting and cooling the hydraulic system.

The design and construction of the buff and draft car was divided into three subsystems: mechanical, hydraulic, and electrical. The following is a description of each system.

4.2.1 Mechanical System

The mechanical system consisted primarily of a load transfer linkage with three different mechanical adapters and a swivel joint. The purpose of this system was to transfer the hydraulic power of a pair of actuators into a pure lateral force acting on the coupler pin of any candidate test car. The diagram in Figure 4 shows the basic principle of how the linkage works.

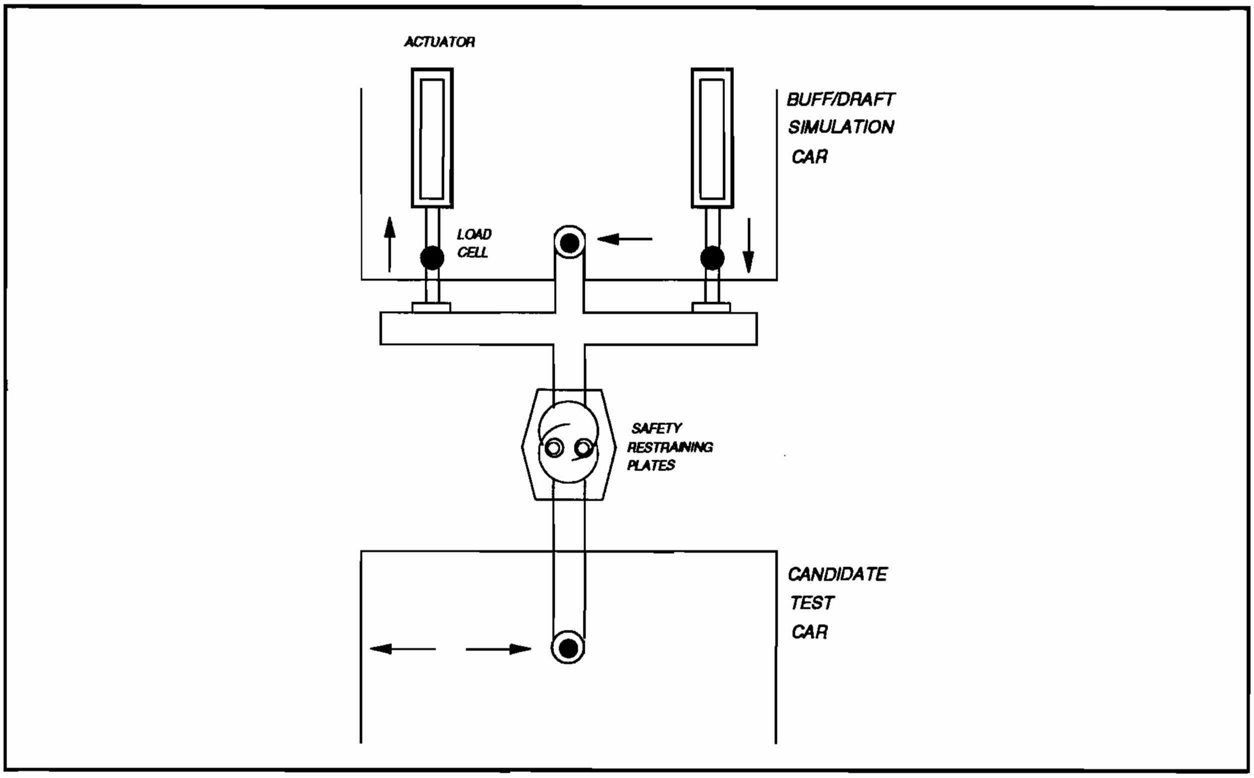

Figure 4. Schematic of Mechanical Linkage (Top View)

The actuators, acting in an out-of-phase mode, generate a pure moment within the linkage that must be reacted by equal and opposite lateral forces, one of which is acting at the coupler pin of the adjacent car. A swivel joint at the base allows the linkage to rotate freely in the horizontal plane while providing some roll motion. This allows safer negotiation of spirals. Three adapters were also fabricated for mating the linkage to the adjacent car. The primary adapter consists of a conventional coupler knuckle for a quick and perhaps more realistic connection. The two other adapters, consisting of 60-inch F and E style coupler shanks, were designed for mating with cars that already have their coupler shanks removed. Figure 5 shows this linkage.

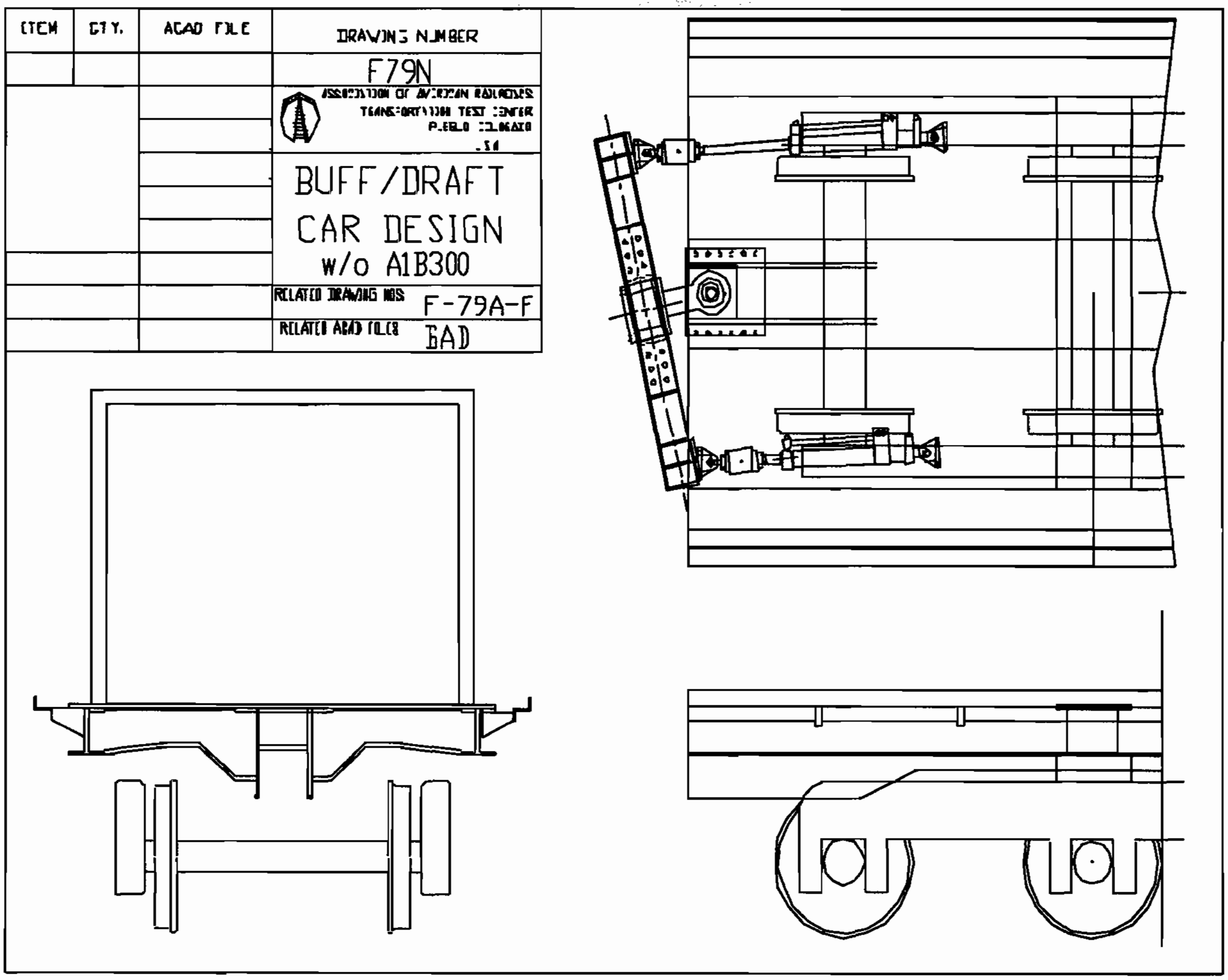

Figure 5. Details of Mechanical Linkage

4.2.2 Hydraulic System

One of the initial recommendations for the design of the new buff and draft car was related to the hydraulic system. An alternative, and perhaps simpler, mechanism was proposed to solve the problem of load fluctuations observed with the prototype car. It consisted of a large accumulator filled to the pressure necessary to produce the desired load. No feedback loop would be required since the system pressure will be unaffected by the minor stroke variations at the actuator due to vehicle interaction with the track. Initially, this idea seemed to provide an effective solution to the load control problem. Further examination, however, revealed that the required accumulator size may be prohibitive and perhaps unsafe if emergency shutdown is ever needed. An additional restriction to this approach was the inability of the system to provide any transient load application.

Although the proposed method requires only steady state load applications, it was felt that a system capable of applying transient loads will be preferred in case it is needed for future verifications, or in case the scope of the buff and draft car itself is expanded.

A third design emerged which, as in the prototype car, utilized a pair of servo-controlled actuators but with two significant modifications. Shown in Figure 5, the new design required the system to be equipped with a considerably larger hydraulic pump with a capacity of 116 gpm and a maximum nominal operating pressure of 3000 psi. The system also included a pair of servo-controlled actuators fatigue rated for a maximum load of 60,000 pounds and with a maximum stroke of 18 inches. Each actuator is controlled by a 2-stage 40 gpm servo valve. The addition of a much larger pump and a better and faster acting servo valve was designed to provide an enhanced system response to track anomalies. The design criteria for this system were to provide adequate response to track perturbations of 0.25 inches peak-to-peak at a maximum limiting frequency of 5 Hz. A second modification, borrowed from the passive control system suggested earlier, was the installation of a single 10-gallon accumulator between the pump and the service manifolds to provide power storage in case power was demanded by the actuators. These design modifications were made to correct the problem of load fluctuation observed with the prototype car without sacrificing the ability to impart transient loads if necessary.

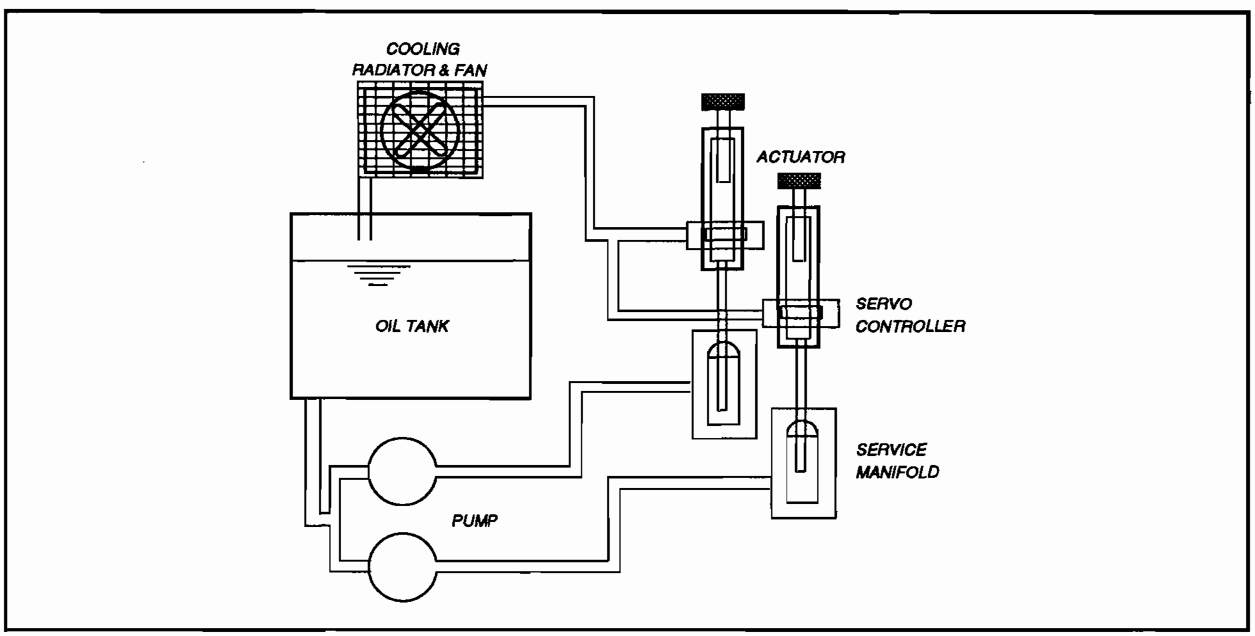

Figure 5. Schematic of Hydraulic System

Based on this design, specifications were written for a complete hydraulic power unit consisting of a hydraulic pump, an electric motor, a reservoir, and various pressure valves and remote control switches. The specifications were then submitted for general commercial bidding by outside suppliers. Representatives from the three lowest bidders were invited to appear separately at the TTC and to present their proposals in person. When each of the three representatives was on site, the details of the proposed system were fully discussed and reviewed in light of the specifications set by TTC. Based on these reviews the contract was awarded to the lowest bidder.

Once all the hydraulic components were delivered, they were mounted on top of one end of the base flatcar. A container room was erected for housing all the hydraulics and the outside was painted. Two windows were installed, one for access and another for a radiator cooling outlet. An access door also was installed.

4.2.3 Electrical System

A government owned 300 Hewlett Packard (HP) diesel generator was donated to the buff and draft car. A diesel tank of a 300 gallon capacity was also mounted next to the diesel unit. Additional plumbing and wiring were also completed providing fuel to the generator and electric power to the hydraulic room. The exteriors of the diesel generator and tank were painted.

4.3 PROCEDURE FOR OPERATING THE BUFF AND DRAFT CAR

Due to the relative complexity of the buff and draft car design, a procedure was developed to ensure proper and safe start-up and operation of its various systems. This procedure is included in the appendix.

5.0 PILOT TESTING OF TWO SAMPLE CARS

The sample cars selected for the initial phase of testing with the buff and draft car were the Frontrunner and the 89-foot flatcar. The Frontrunner was selected first. This car was well characterized under a previous FRA project at TTC and therefore accurate modeling of its performance was possible. Recent derailments, in which buff and draft forces were suspected, involved the coupling of the Frontrunner to the 89-foot flatcar. Therefore, it was natural to select the 89-foot flatcar as the second sample car. Both cars were tested in only the empty configuration because they are less stable laterally under buff or draft loads when empty than when loaded.

The Frontrunner is a two-axle car and the one tested was equipped with yaw dampers and a leaf-spring type of suspension. Testing was conducted at TTC on the Wheel/Rail

Mechanics (WRM) loop on the 10-degree curve for speeds of 12, 24, and 32 mph in the clockwise direction (Figure 7). These speeds correspond to -3, 0, and +3 inches off the balance speed for this curve.

5.1 TEST INSTRUMENTATION

The instrumentation used in the tests can be divided into the following three groups.

5.1.1 Instrumentation for Hydraulic System Performance and Control

This group consisted of a pair of load cells for measuring the forces generated in each actuator and for the feedback force control loop, a pair of displacement transducers for measuring actuator displacements, and gages for continuous monitoring of oil temperature and pressure

5.1.2 Instrumentation for Measuring Vehicle Performance

This group consisted of a single instrumented wheel set placed always nearest to the load linkage for measuring the wheel/rail forces, and string potentiometer devices for measuring truck (or axle) lateral, vertical, and yaw motions with respect to the car body.

5.1.3 Video Equipment

This group consisted of four video cameras, two TV monitors, a video recorder, and an image splicer that allows mixing of the four views into a single screen, if desired. The cameras provided continuous viewing of the wheel/rail interface under the instrumented axle, in addition to a top view of the linkage coupling to the test car and the view from the locomotive cab.

5.2 TEST PROCEDURES

The test procedure was as follows:

1. Position test consist outside the 10-degree curve as shown in Figure 7.
2. Set hydraulic actuators in force control mode at zero force.
3. Accelerate consist to maintain test speed through the curve.
4. Turn on the data acquisition system prior to curve entry.
5. Once positioned well into the curve, increase the actuator load to the fixed prescribed value for the test run. Maintain the load until end of curve.
6. Bring actuator load back to zero prior to curve exit.
7. Turn off the data acquisition system after completion of curve exit.
8. Stop the consist gradually and position for another run.
9. Evaluate real-time L/V ratios from instrumented wheel set to determine whether its safe to increase actuator load.
10. If safe, repeat above procedure for an actuator load increment of 2,500 pounds.

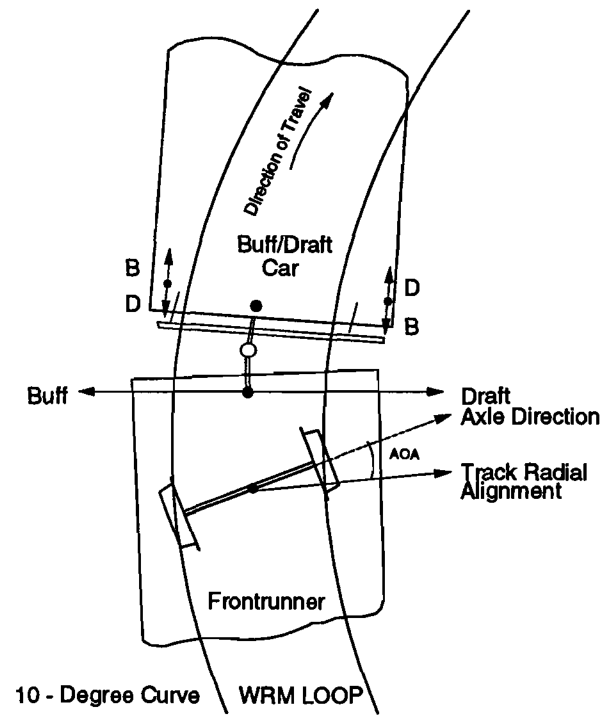

Figure 7. Position for Buff and Draft Test Consist

5.3 RESULTS OF THE FRONTRUNNER TESTS

The instrumented wheel set raw strain signals from all the test runs were post- processed for accurate determination of the resulting wheel/rail forces. The wheel/rail forces were then averaged for each run for the curve portion through which the applied actuator load was constant. This constituted a distance of roughly 600 feet for each run. The results from the cases corresponding to the balance speed (24 mph) will be detailed first.

Figure 8 shows the average measured vertical wheel loads plotted versus the net applied lateral coupler load. The lateral coupler load was calculated based on the measured actuator loads and the ratio of the actuator spacing to the effective distance from the Frontrunner's coupler pin to the load application beam's pin. A negative coupler load denotes a buff condition in which the end of the Frontrunner car was pushed towards the outer rail. Positive coupler loads correspond to draft conditions in which the car end was pushed towards the inner rail. The individual symbols are the measured vertical forces with the solid lines representing the best line fit. Figure 7 indicates that the applied lateral coupler load resulted in a linear variation in the vertical wheel loads. The net effect is that of loading one side while evenly unloading the opposite side. Lateral coupler loads in excess of 11,500 pounds caused the unloaded side to drop below the minimum 10 percent static weight criteria set by Chapter XI. This limit is indicated by the dashed line. Testing beyond this limit in either direction produced occasional wheel lifts on the unloaded side as observed through a pair of video cameras set up to monitor the wheel/rail interface and as evident from the measured wheel/rail forces.

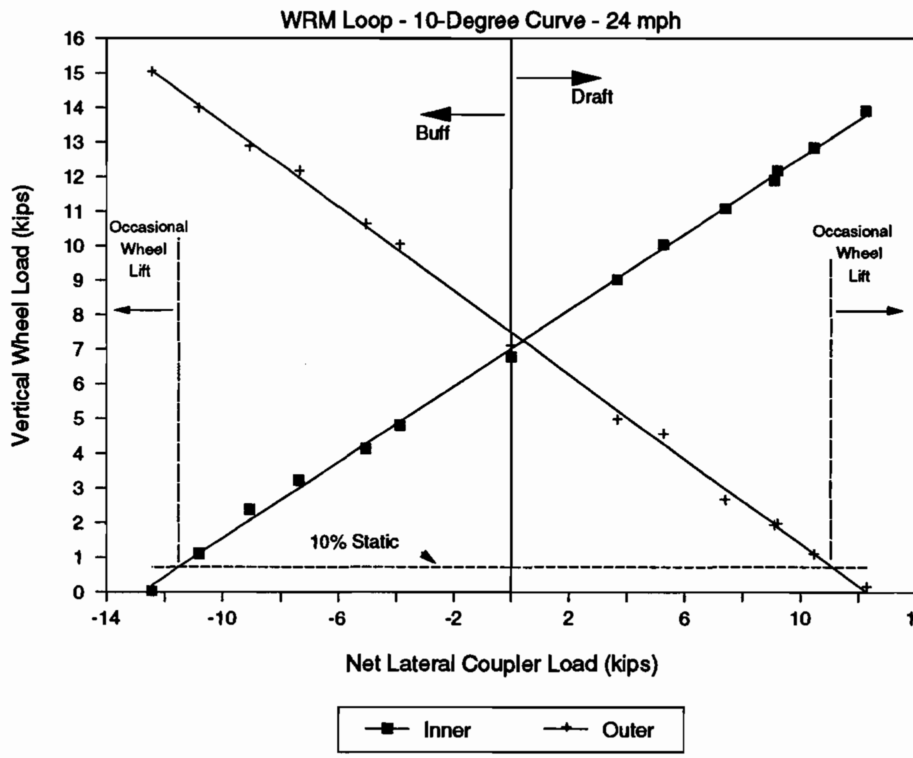

Figure 8. Average of Measured Vertical Wheel Loads

Figure 9 shows the average measured lateral wheel loads plotted versus the net applied lateral coupler load. The sign convention adopted in this plot, and throughout the remainder of this report, is that of the instrumented wheel set; i.e., positive if pushing against the wheel flange. The variation through the range of loads applied may be approximated by a bilinear fit for each wheel as indicated by the solid lines. The reason for the change in the rate of linear variation, which occurred for both wheels at a draft coupler load of approximately 7,000 pounds, was not clear from the initial examination of the data. Moreover, both lateral wheel loads for the zero coupler load run are offset from the fitted lines. Initially it was judged as a possible inaccuracy in the data collected in this run.

Figure 10 shows the average measured L/V ratios plotted versus the lateral coupler load. An average L/V ratio of 0.8 was measured for the loaded side in each case. At this level the potential for the wheel flange to climb the rail depends largely on the angle of attack.

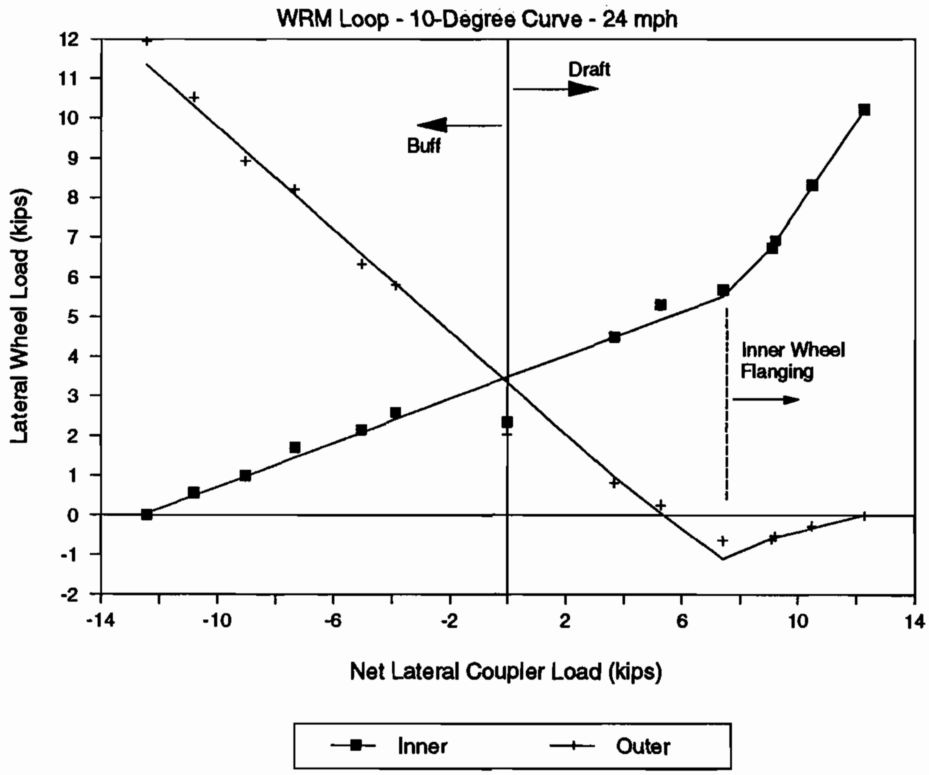

Figure 9. Average of Measured Lateral Wheel Loads

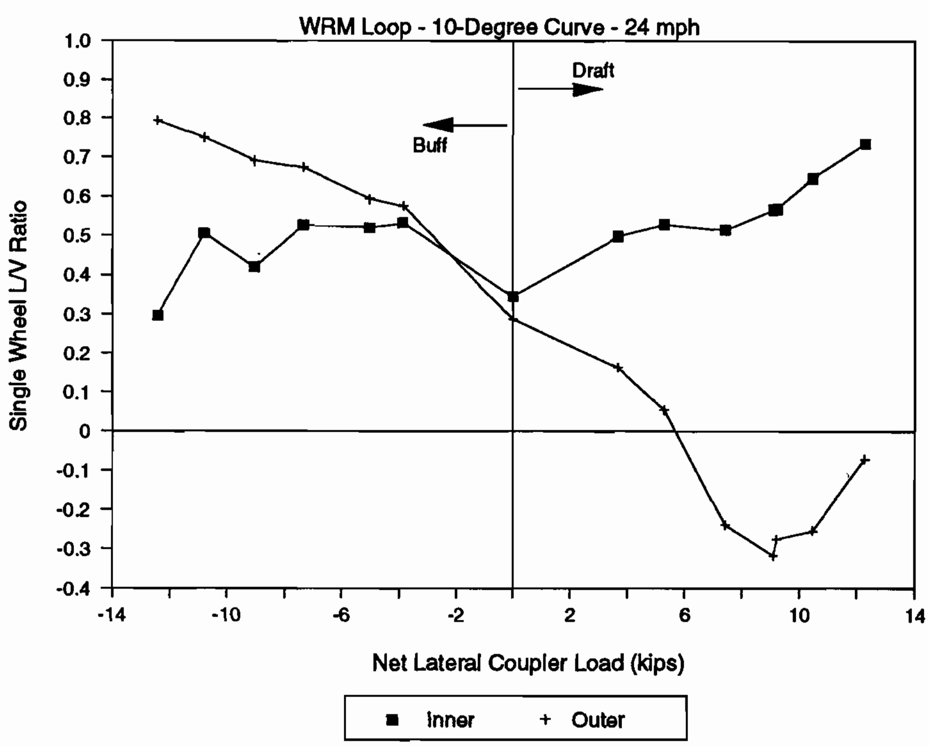

Figure 10. Average of Measured L/V Ratios

Figure 11 shows the average measured vertical wheel loads for all the three speeds considered. The influence of the variation in speed on the level of coupler load at which wheel unloading below the 10 percent static level occurs seems negligible. Measured L/V ratios on the loaded side did not vary significantly from the 0.8 ratio measured for the balance speed. This is to be expected given the relatively light weight of the tested vehicle.

Results of these tests indicate that the maximum coupler load that can be applied safely in either direction is 11,000 pounds.

It is of interest to compare the established test limit of 11,000 pounds for the empty Frontrunner to a value computed from simple static equilibrium. Based on a coupler height of 32 inches and a measured total car weight of 27,600 pounds, equilibrium predicts that a lateral coupler load of 12,200 pounds will be required to produce wheel lift on one side o the car. This indicates that the 11,000 pound test limit is 10 percent lower than the static equilibrium value.

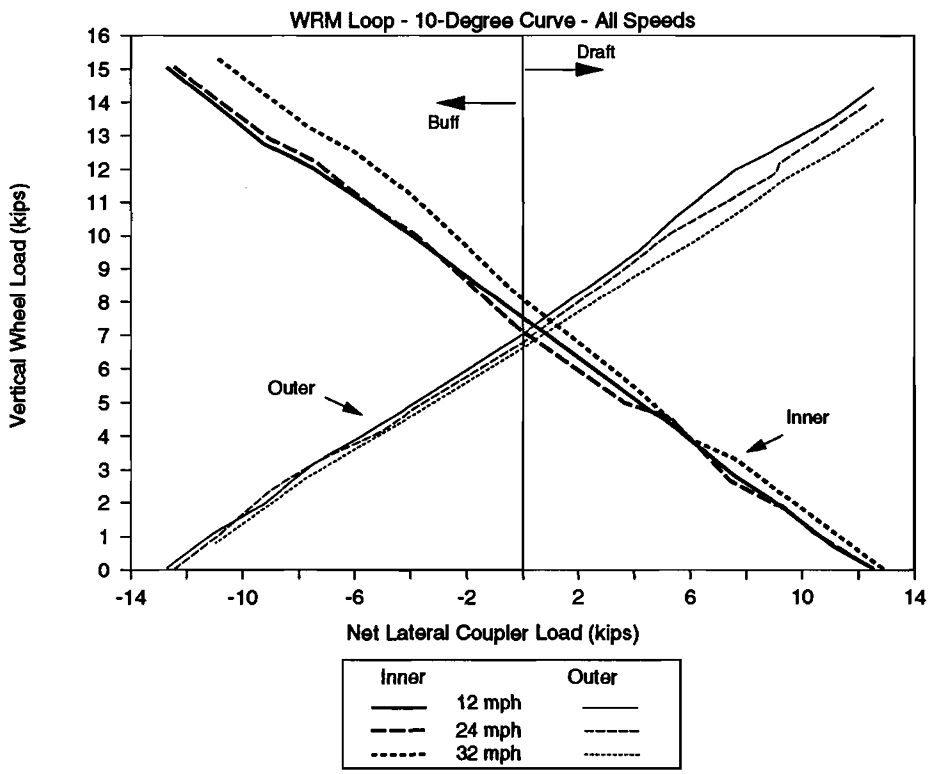

Figure 11. Average of Measured Vertical Wheel Loads -- All Test Speeds

5.4 NUCARS SIMULATION OF FRONTRUNNER TESTS

Since the angle of attack was not continuously measured (currently there are no instruments that provide a reliable measurement), and to clarify some of the questions raised in the previous section, a NUCARS simulation of the test conditions was performed. The analysis consisted of applying an external lateral load to the lead coupler pin of a single Frontrunner model car moving at a constant speed of 24 mph on a steady 10-degree curve. The coupler load was varied slowly from -11,000 to +11,000 pounds over a track distance of 5,000 feet to combine all the steady state curving runs in a single simulation.

Figure 12 shows the predicted vertical wheel loads versus the externally applied coupler load. The apparent noise in the output is due to the choice of the integration time-step and is not considered to be a realistic response. The figure confirms the same linear variations measured from the tests. Figure 13 shows the predicted lateral wheel loads for the same simulation. The analysis shows lateral load variations similar to the measured forces (Figure 9). Note the increase in rate of change of lateral load around the 7,000-pound draft load mark and the drop in load levels at zero coupler load.

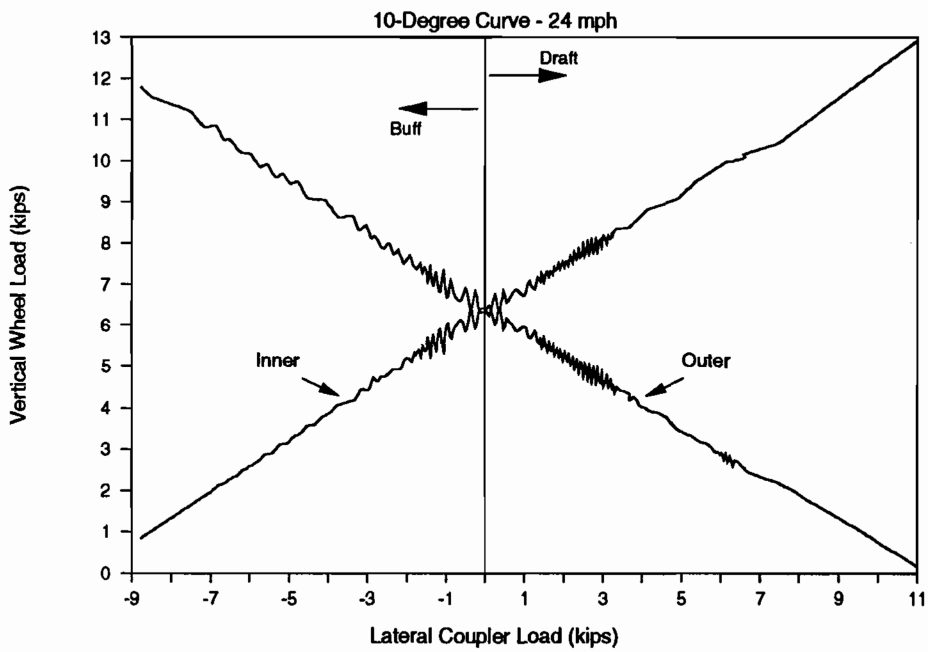

Figure 12. NUCARS Predicted Vertical Wheel Loads

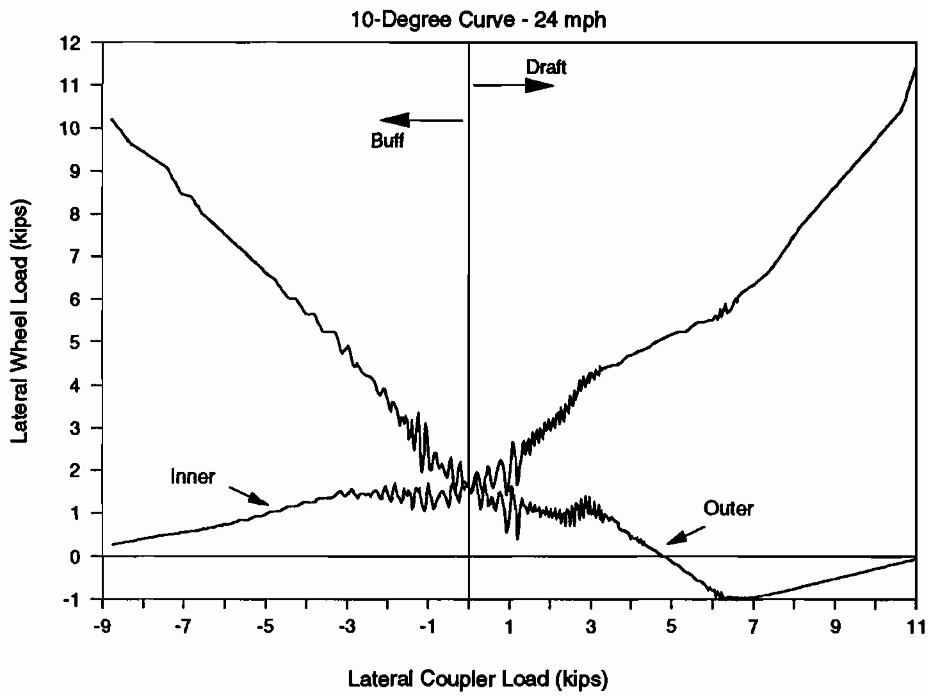

Figure 13. NUCARS Predicted Lateral Wheel Loads

Figure 14 shows the predicted lateral displacement and yaw angle of the lead axle plotted versus the coupler load. The figure indicates that the axle suddenly shifted laterally around the 7,000-pound draft load mark causing the inner wheel to be in flange contact with the inner rail. This explains the sudden increase in the lateral inner wheel load beyond this load level, as seen in the test data and as predicted by the simulation. A positive axle yaw angle (Figure 5) was predicted for the full range of coupler loads considered. The axle yaw was at a minimum around zero coupler load which may explain the lower lateral wheel loads measured, as well as predicted, under those conditions. When the buff (or negative) coupler load was increased, the axle yaw angle was increased to a maximum angle of 30 milliradians. This constitutes a relatively large positive angle of attack between the outer wheel and rail. At this angle, the risk of a flange climb derailment is relatively high for L/V ratios greater than 0.8. When the draft (or positive) coupler load was increased, a sharper increase in the axle yaw was predicted up to a 7,000-pound point marking the flange contact between the inner wheel and rail. Short of flanging, this is the expected axle response in order to react the applied lateral load. However, once the inner wheel is in hard flange contact with the rail, the axle yaw gradually tappers off as the axle begins to react to the applied load through the flange. Since a positive axle yaw constitutes a negative angle of attack between the inner wheel and rail, the potential for a flange climb derailment under these conditions (draft) is less than those under buff, given equal coupler loads.

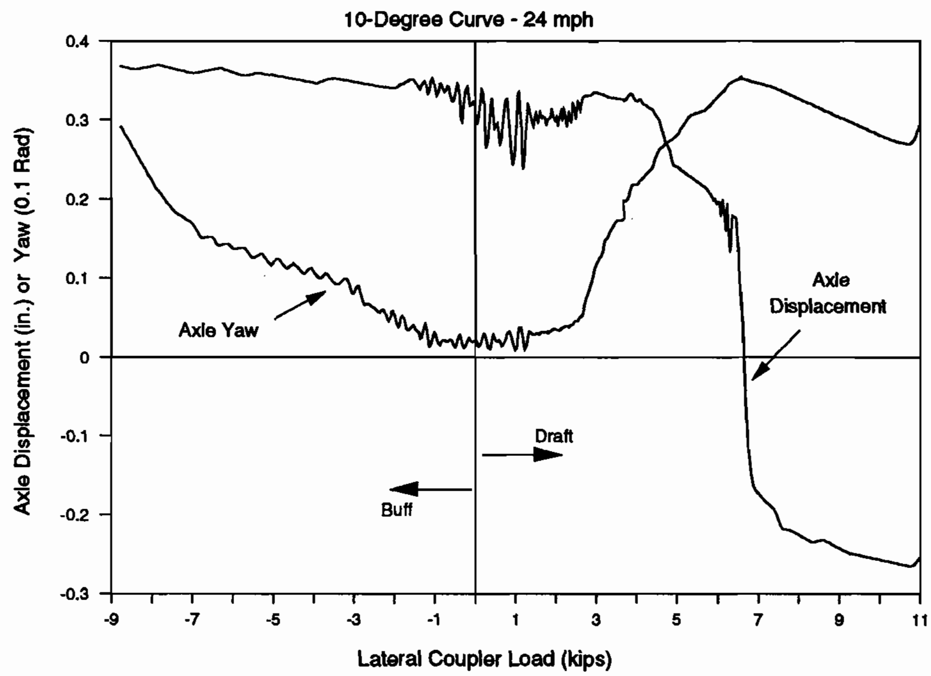

Figure 14. NUCARS Predicted Lead Axle Lateral Shift and Yaw Angle

The results of the simulation indicate excellent overall agreement with the test data and prove that the buff and draft car succeeded in applying the required lateral coupler loads in the manner for which it was designed.

5.5 TEST RESULTS AND NUCARS SIMULATION OF THE 89-FOOT FLATCAR

Results of the 89-foot flatcar testing as well as the predictions from NUCARS are shown in Figures 15 through 17, for the leading axle of the lead truck. Only one instrumented wheel set was used measuring the wheel/rail forces on that axle. The testing was conducted at the balance speed of only 24 mph on the WRM loop's 10-degree curve. Lateral coupler forces up to a level 26,500 pounds were applied in both the buff and draft directions. Testing was terminated at this load level due to the loaded side of the leading axle occasionally registering a 1.0 L/V ratio. In the NUCARS simulation, the range of the lateral coupler load was extended to 30,000 pounds. All three figures indicate an overall good agreement between the test data and the NUCARS predictions. Figure 15 shows that the vertical wheel loads on the leading axle are linearly proportional to the applied coupler load. Although the analysis results in Figure 15 indicate that an L/V ratio did not exceed 1.0 for coupler loads less than 30,000 pounds, the test data and experience suggest a maximum safe coupler load of 26,500 pounds. It should be noted again that the measured L/V ratios shown in Figure 15 represent an averaged L/V over a curve distance of approximately 600 feet.

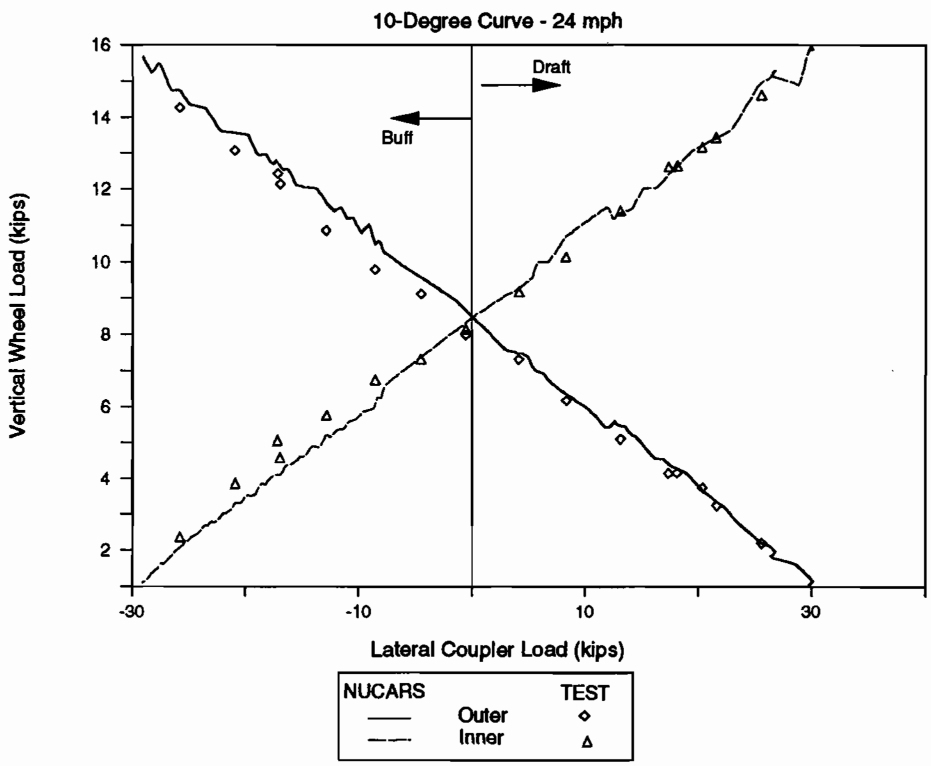

Figure 15. Predicted Versus Measured Vertical Wheel Loads

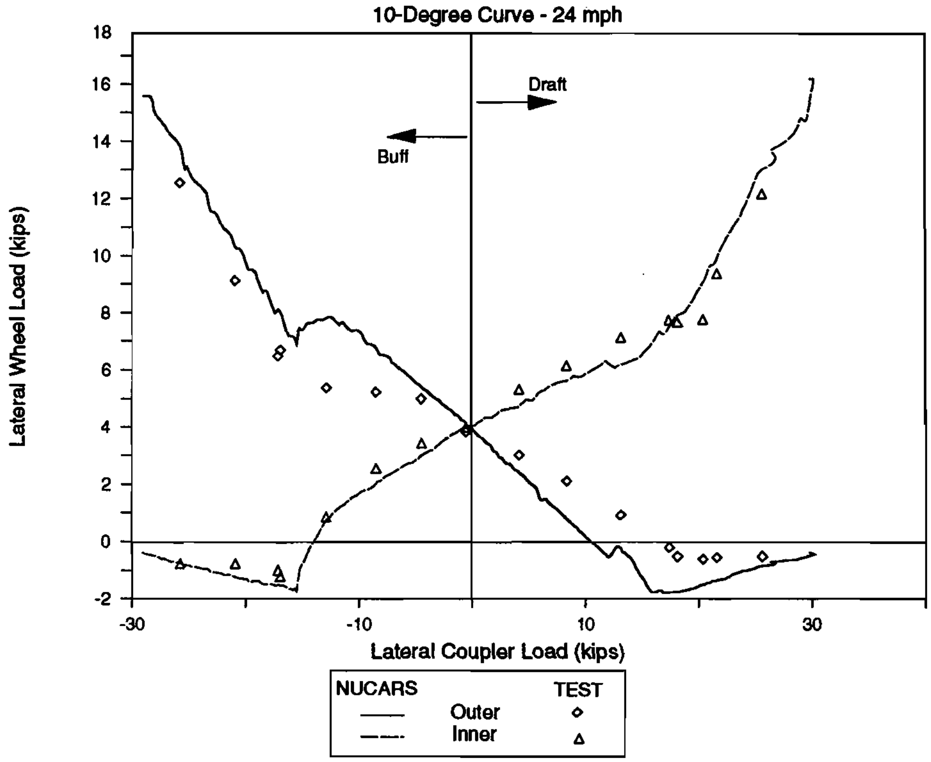

Figure 16. Predicted Versus Measured Lateral Wheel Loads

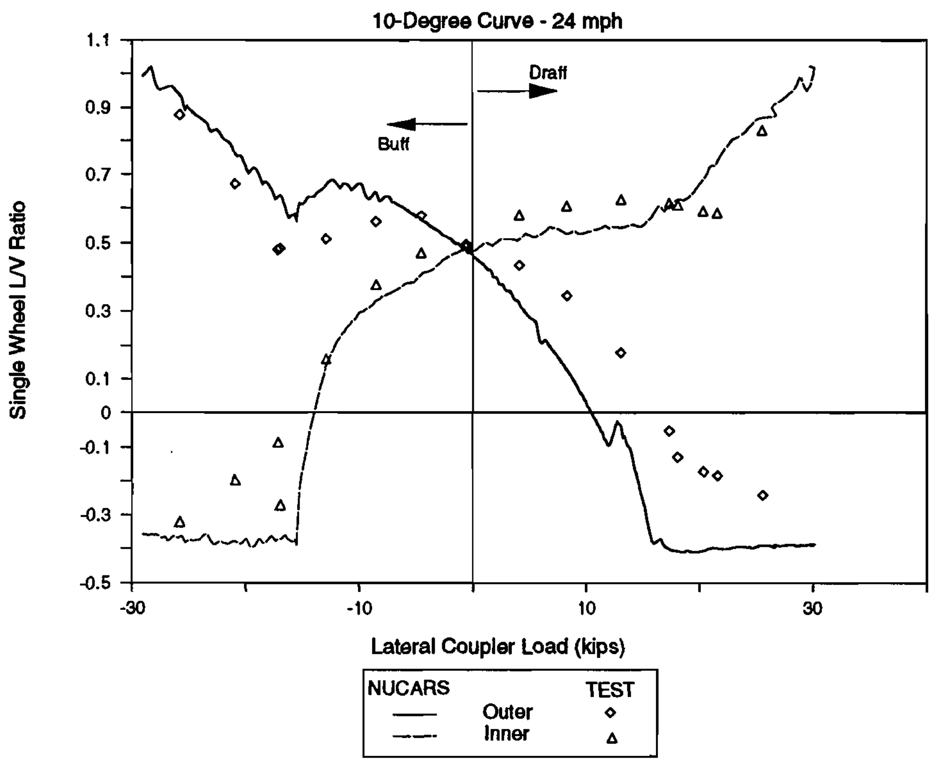

Figure 17. Predicted Versus Measured L/V Ratios

The predicted lateral shift and yaw for the lead and trail axles of the lead truck are shown in Figures 18 and 19. The predicted trends are similar to those of the lead axle of the Frontrunner with one notable difference. Unlike the Frontrunner, the simulation predicts a negative angle of attack between both axles and the outer rail under buff conditions. This is primarily due to the differences in reacting the buff loads through a single axle (the Frontrunner) versus reacting them through a conventional three-piece truck.

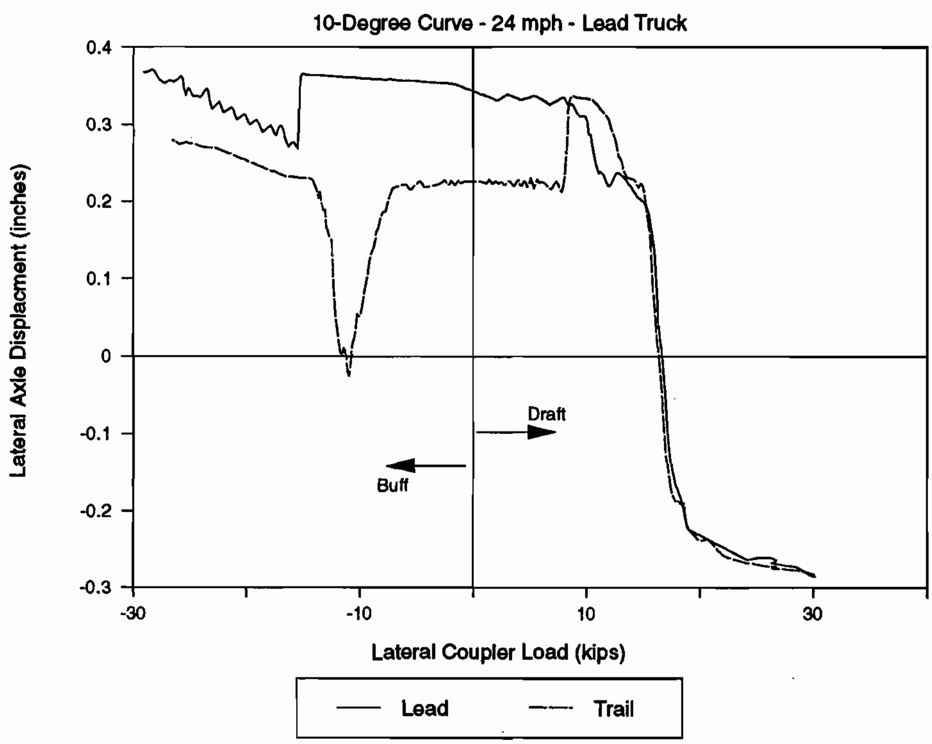

Figure 18. Predicted Lateral Axle Shift -- Lead Truck

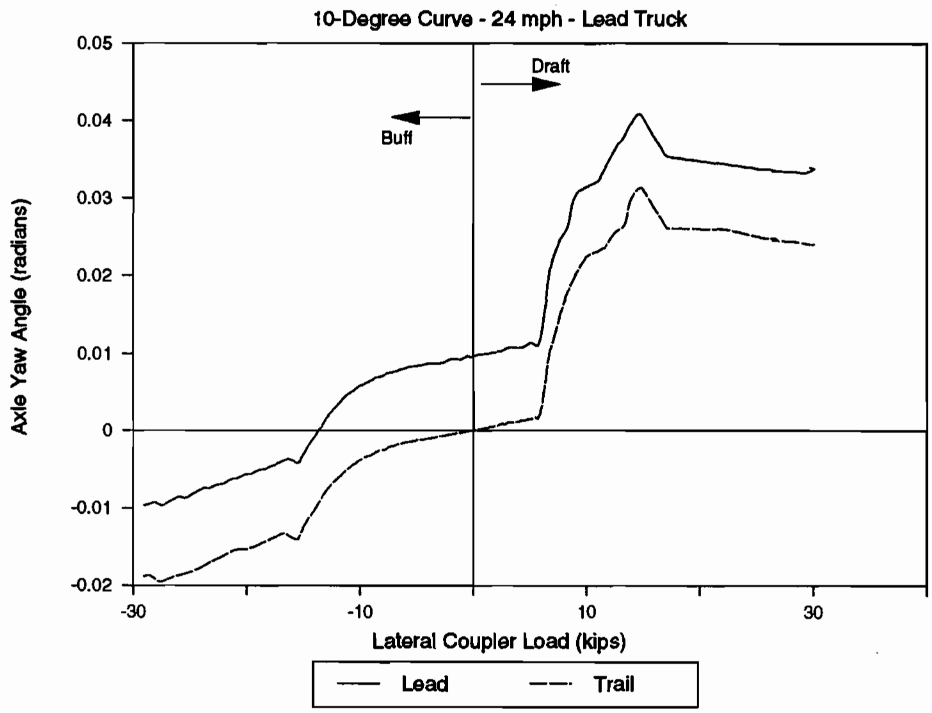

Figure 19. Predicted Axle Yaw Angle -- Lead Truck

5.6 BUFF AND DRAFT FORCES AND THE LATERAL COUPLER LOADS

In order to translate the limits of lateral coupler load established by the previous testing and analysis into limits on actual buff and draft forces, additional analysis is required. This analysis utilizes the program CABS which was developed under the first sub-task of this program. A number of simulations were conducted on the coupling of Frontrunner to the 89-foot flatcar and the coupling of each to like cars. During each simulation, the consist was run over a long section of a constant 10-degree curve and at a speed of 24 mph. During the steady state curving, the net longitudinal external loads were slowly varied from a pure buff load of 250,000 to a draft load of 250,000 pounds. Under the proposed test methodology, cars will be expected to safely negotiate a constant 10-degree curve under buff and draft loads of up to 250,000 pounds.

Figure 20 shows the results of the simulation for the Frontrunner car. The figure shows the variation in the lateral coupler load as a function of the applied buff and draft load for the two cases of coupling to a like car as well as coupling to an 89-foot flatcar. The two horizontal lines indicate the safe limits identified from the previous testing results. The figure indicates that the Frontrunner will perform safely under both conditions of coupling within the given limits of buff and draft forces. Note, however, that when coupled to an 89-foot flatcar, the Frontrunner experiences considerably lower lateral coupler loads. This may suggest an adverse effect on the performance of the 89-foot flatcar.

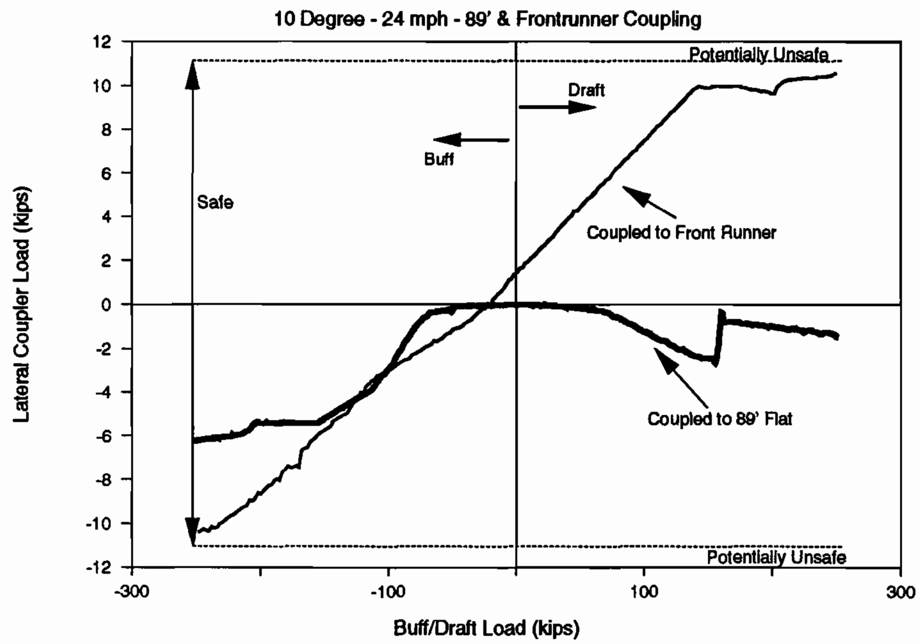

Figure 20. Buff and Draft Forces Versus Lateral Coupler Load -- Frontrunner

Figure 21 shows the results of the simulation for the 89-foot flatcar. Similar to the previous figure, the two horizontal lines indicate the safe limits of performance determined from the test data. The results indicate a safe performance of the 89-foot flatcar when coupled to a like car. However, coupling of the 89-foot flatcar to the Frontrunner reduces the ability of the 89-foot flatcar to safely negotiate curves for draft loads in excess of 220,000 pounds and buff loads in excess of 180,000 pounds.

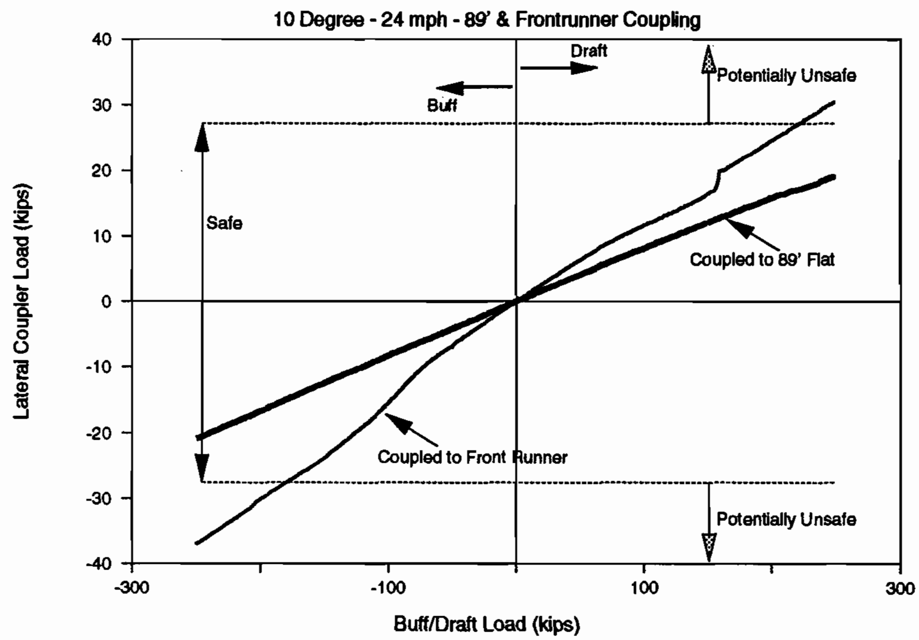

Figure 21. Buff and Draft Forces Versus Lateral Coupler Load -- 89-Foot Flatcar

6.0 CONCLUSIONS AND RECOMMENDATIONS

Results obtained from the pilot testing described show the feasibility of the buff and draft test vehicle in applying controlled levels of lateral coupler loads to any candidate freight car. The use of this tool was successfully demonstrated by the example of coupling of the Frontrunner 2-axle car to the 89-foot flatcar.

It is recommended that the procedure described here be included in Chapter XI, Section 11.7, in order to verify the capability of a new car design to operate in trains with sustained buff and draft loads. In many instances testing would not be necessary if the yaw moment at the coupler butt were known and any other aligning features could be modeled in CABS. The lateral load could be predicted through the curve and a NUCARS simulation could provide the required analysis of the curving performance.

A static load application in a load frame is not sufficient verification of a car's buff and draft load sustaining capability. The influence of factors associated with the dynamics of a rolling wheel set (such as angle of attack) on the lateral stability of a moving car must be considered.

The lateral load predictive capability of CABS has been demonstrated through validation tests. The CABS program can be offered to all carbuilders or designers. The results from the CABS/NUCARS programs could be submitted for evaluation by the AAR CEC in much the same manner as the curve negotiability results. The recommendation for testing could be determined by the CEC.

The logistics and cost of obtaining four locomotives and associated trailing tonnage to conduct a running buff and draft test are prohibitive compared with the single locomotive unit required with the special test car proposed here. The proposed test procedure could determine the margin of safety by extending the upper limit of the test lateral load.

The train operations community has come to expect that a 250,000-pound load can be safely sustained in most trains. The Train Make-Up Manual provides limitations on trailing tonnage for car combinations that do not meet the 250,000-pound criteria.[7] On the other hand, the carbuilders community has a 200,000-pound load cited in the specifications. Cars that do not meet the 250,000-pound criteria are subject to train placement limitations. Raising the design limit value to 250,000 pounds would simplify train make-up requirements.

The potential application of the methodology presented here to a candidate new car design can be summarized as follows:

1. Proponent would forward drawings and other details of the car in the usual manner. If the car's draft arrangement is claimed to be substantially different, the proponent would be asked to provide details including a yaw moment (lateral load) versus coupler angle curve under the specified buff and draft loads.
2. Proponent would be requested to forward the results from the prescribed analysis which entails the use of CABS to predict the lateral loads and the corresponding L/V ratios from the associated NUCARS runs for a 10-degree curve.
3. The CEC would determine if testing is required.
4. Following testing, the car would be deemed safe, marginal, or unsafe. For marginal cars, a trailing tonnage restriction may be recommended. If like coupled cars cannot be operated without exceeding criteria, the car would be deemed unsafe.

It is concluded that the CABS model and its interface with NUCARS can provide an evaluation capability under buff and draft conditions which did not exist before. Aside from its potential use in Chapter XI, this methodology would be useful in derailment investigation and in developing car placement in train restrictions. Carbuilders and designers will also be offered the opportunity to evaluate an important aspect of the design's safety.

The proposals set forth here are intended as a "strawman" for discussion. It is recommended that further effort be invested in developing the specific Chapter XI provisions based on the proposed techniques. This would entail CEC deliberations and further testing/development to produce the most practical, economical, and verifiable provisions. It is also recommended that CEC review the current requirement for a 200,000-pound buff and draft capability and raise it to 250,000 pounds, since it represents current operating practice.

Failure to meet the 250,000-pound buff and draft requirement would not disqualify the car from interchange revenue service operation. Rather, a trailing tonnage restriction should be placed on this car as described in the Train Make-Up Manual.[7] Failure of like coupled cars to operate safely under the recommended limits would be cause for rejection.

REFERENCES

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2. Track Train Dynamics Program. "Field Tests of the Track-Train Dynamics Program," Association of American Railroads Report R-183, 1975.
3. Association of American Railroads, Mechanical Division. *Manual of Standards and Recommended Practices*, Vol. C-II, Section 2.1.6.
4. El-Sibaie, Magdy. "Coupler Angling Under In-Train Loads: Modeling and Validation," Association of American Railroads, R-772, 1991.
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6. El-Sibaie, Madgy. "Coupler Characterization Tests on the RDL Mini-Shaker Unit," Association of American Railroads, Working Paper WP-151, 1991.
7. Association of American Railroads. "The Train Make-up Manual," R-802.

APPENDIX

OPERATING PROCEDURE -- BUFF AND DRAFT CAR

1. PRELIMINARY

a. Start generator on test car (Section 2).
b. Turn AC power ON to service manifold and actuator control consoles.
c. Ensure service manifolds are OFF (neither LOW nor HIGH lights lit).

2. BUFF AND DRAFT CAR GENERATOR

a. Ensure main generator circuit breaker is OFF. This circuit breaker is located below generator start/monitor panel.
b. Check OIL and COOLANT levels in generator engine. Top off if required.
c. Visually check generator for any signs of leaking, physical damage, or any other noticeable signs.
d. Check FUEL level (fuel gage located on filler end of tank). Refuel if required.
e. Start generator by moving the ENGINE AUTO CONTROL SELECTOR from STOP to MAN. Do not allow starter motor to crank generator over 30 seconds. If generator does not start within 30 seconds, try using starter fluid or call maintenance for assistance.
f. After generator engine has started and stabilized, allow it to WARM up 10 minutes before applying LOAD.
g. Generator voltage output should be 208 VAC at 60 Hz. If not, use governor motor raise/lower switch to set frequency to 60 Hz and voltage level knob to set voltage.
h. Turn ON main generator circuit breaker (UP position).
i. Turn ON center breaker located between generator and fuel tank (UP position).

3. BUFF AND DRAFT CAR PUMP

a. Place both HIGH/OFF/LOW switches in the OFF position. This will limit pump pressure to approximately 300 psi when pumps are started.
b. Press pump START button. Pumps should start and pressure gages should read approximately 300 psi.

NOTE: The hydraulic actuators should be powered UP at this point. Let the PUMPS operate at approximately 300 psi and go to the actuator control console.

4. SERVICE MANIFOLD & ACTUATOR CONTROL

a. While monitoring the load cell output on the LOAD DVM, adjust the coarse and fine balance adjustments on the DAYTRONIC signal conditioners for 0.00 VDC for BOTH load cells, Actuators 1 and 2.

NOTE: The following procedure is to RCAL the load cell. This procedure need not be accomplished every time the system is shut down, but should be done at least once every other day when testing is continuous or before any use after a lay off of several days or at the discretion of the test or instrumentation engineer if any doubt arises over the validity of the calibration. <u>Caution</u>!! Make sure service manifolds are <u>OFF</u> before performing this calibration. If hydraulic power is applied to the actuators, they will move equal to the calibration value when the load cell is shunted with the calibration resistor.

b. On the left side of the controller console, remove the 18-pin cable from Actuator 1 load cell connector and insert an 18-pin break out box. Recheck load cell BALANCE and readjust to 0.00 VDC, if necessary. Insert a 59.88 k ohm resistor between pins B and L on breakout box. Adjust coarse and fine SPAN controls on Daytronic Signal Conditioner 1 for_10.0 volts, which is equal to_50.0 kip. Scaling factor is 0.2 volts per kip. This is the calibration for load cell number 1 .

c. For Actuator 2 load cell repeat the above procedure.

d. Remove the 18-pin breakout box, connect the cables directly to the controller and rebalance both load cells if required.

5. POWERING UP ACTUATORS

a. <u>Pre-Sets</u>

1. HYDRAULIC master control (top bay), power switch ON, SM 1&2 OFF.
2. Buff and draft car controller (lower bay) power switch ON, DISP/LOAD switches, (both actuators) to LOAD. FWD/REV switch Actuator 1 to REV. FWD/REV switch Actuator 2 to FWD. Run ON/run OFF switch to run OFF. Run OFF light should be lit. MASTER GAIN pot to 0.00, (full CCW). Actuator 1 and 2 SPAN pots to approximately 9.0. These SPAN pots will be set precisely after system is powered up. Actuator 1 OFFSET pot adjusted until VALVE DVM reads 0.0 VDC. Actuator 2 OFFSET pot adjusted with VALVE DVM reads 0.0 VDC.

b. On hydraulic master control (top bay) press MASTER LOW pressure switch. Amber colored low pressure light should come on for both service manifolds. Check LOAD CELL outputs on LOAD DVM. They should have stayed at 0.00. If not, carefully adjust the OFFSET pots for each actuator until the load cell output is 0.00 VDC. Now press the MASTER HIGH pressure switch on the Hydraulic Master Control (top bay). The amber colored lights for high pressure should come on for both service manifolds and the low pressure lights should go off. Again, verify load cell outputs are 0.00 VDC, carefully adjust if necessary.

c. With both service manifolds on HIGH pressure go to the pump control panel in the pump room and move the HIGH/OFF/LOW switches for both pumps from OFF to LOW. Pressure should increase to approximately 1000 psi. On the actuator control panel (lower bay), again verify load cell outputs are 0.00 VDC and carefully adjust if necessary.

d. When satisfied positive, stable actuator control is established go to pump control panel in the Pump Room and switch the HIGH/OFF/LOW switches for both pumps from LOW to HIGH. Pressure should increase from 1000 to approximately 3000 psi. Again, verify load cell outputs are 0.00 VDC and not changing. **NOTE:** A small change in the setting of the OFFSET pot should cause the actuator to move and apply a load on load cell.

e. The system is now fully powered up and ready for testing.

NOTE: The buff and draft car consist should never be moved without the actuators being powered up. The above start up procedure should be completed through steps 5b prior to moving car so the actuators can "steer" through any curves encountered.

CAUTION: Due to the mechanical makeup of the buff and draft car, the actuators cannot be operated in phase. Never operate unless one actuator is set for FWD and the other REV. The Controller has been designed as a universal unit capable of operating in a variety of modes and configurations, however, for buff and draft car operation, the actuators must be OUT OF PHASE and operated only in FORCE control mode. Due to the nature of the TJ (Vickers) displacement transducers, realignment of the Vickers EM-D-30 Servo Amplifier board is necessary to operate in STROKE control. (See Vickers Technical Information Sheet on the EM-D-30 Servo Amplifier board with PID.)

NOTE: The EM-D-30 Servo Amplifier is being operated with only proportional feedback. DIP switches on boards should be as follows:

S1(C1) B		S1(2) A		S1(C3) A	
S2-1:0	S2-2:0	S2-3:1	S2-4:0	S2-5:0	S2-6:1
S3-1:0	S3-2:1	S3-3:0	S3-4:1	S3-5:1	S3-6:0

Complete setup and alignment information for the EM-D-30 Servo Amplifier board is given on the Vickers Tech Information Sheet. In normal day to day operation, the potentiometers on the Servo Amplifier board should not need adjusting.

6. TEST OPERATION

a. Quasi-Static Operation

1. Set DATEL voltage calibrator output to 0.000 VDC and connect output to BNC input jack on actuator control panel. Press RUN ON switch.
2. Carefully roll in the MASTER GAIN pot. Load cell outputs should remain at 0.00 VDC. If not, using the OFFSET adjust on the DATEL and the two individual actuator OFFSET adjusts, balance each actuator for 0 force with a 0 voltage input. Return MASTER GAIN to 0.00 (full CCW).
3. Set DATEL output to +3.000 VDC. Carefully roll in the MASTER GAIN pot until Actuator 1 load cell reads -0.50 on LOAD DVM. Check Actuator 2 LOAD DVM. If both channels are balanced, Actuator 2 load should read +0.50 on its LOAD DVM. If this is not the case, adjust the individual actuator SPAN pots to attain an equal but out of phase load on each actuator.
4. Set MASTER GAIN to 0 again (full CCW), change polarity on DATEL, and roll MASTER GAIN back in (CW) until Actuator 1 load cell reads +0.50 VDC. Actuator 2 load cell should read -0.50 VDC.
5. Repeat above steps (6a1, 2, 3 and 4) until both actuators are balanced at 0 and push and or pull at equal and opposite amounts for a given setting of the MASTER GAIN control. Inform TEST ENGINEER you are ready for quasi-static testing.

NOTE: The polarity of the DATEL for testing will be determined by whether or not you want buff and draft action and by whether you are in a left or right hand curve. Verify polarity with TEST ENGINEER.

b. Dynamic Operation

1. Connect output from FUNCTION GENERATOR to BNC input jack on actuator control panel. Make sure MASTER GAIN control is full CCW and press RUN ON switch.
2. Set desired frequency on FUNCTION GENERATOR thumb switches, set output level at 3.000 volts, set OFFSET to 0.00. Monitor load cell outputs on oscilloscope or strip chart.
3. Carefully roll in master gain until Actuator 1 load cell output is approximately $\pm$0.50 volts. Adjust FUNCTION GENERATOR OFFSET adjust and the two individual actuator OFFSET and SPAN adjusts until each actuator is pushing and pulling equal and opposite amounts, symmetrical around zero, for a given setting of the master gain control. Set MASTER GAIN to 0.00 (full CCW) and inform TEST ENGINEER you are ready for dynamic testing.

NOTE: All of the above setup and balancing procedures should be accomplished while stopped on tangent track. If the system is properly set up, the only control that should be needed to accomplish the desired test parameter is the MASTER GAIN control.

7. SYSTEM SHUT DOWN

a. Master gain control to full CCW.

b. Run ON/run OFF to run OFF.

c. Service manifolds to OFF.

d. Pump HIGH/OFF/LOW switches to OFF.

e. Stop pumps.

f. Generator main circuit breaker OFF (down).

g. Generator ENGINE AUTO CONTROL SELECTOR switch to STOP.

h. Test car generator OFF.

SYSTEM IS NOW SECURE.

ABBREVIATIONS:

AC	alternating current
BNC	bayonet connector
CCW	counterclockwise
CW	clockwise
DISP	displacement
DVM	digital volt meter
FWD	forward
HSM	hydraulics service manifold
k ohm	kilo ohms
MAN	manual
pot	potentiometer
PID	proportional integral differential
RECAL	recalibrate
REV	reverse
VAC	voltage alternating current
VDC	voltage direct current