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HIGH CANT DEFICIENCY TEST OF THE F40PH LOCOMOTIVE AND THE PROTOTYPE BANKING AMCOACH

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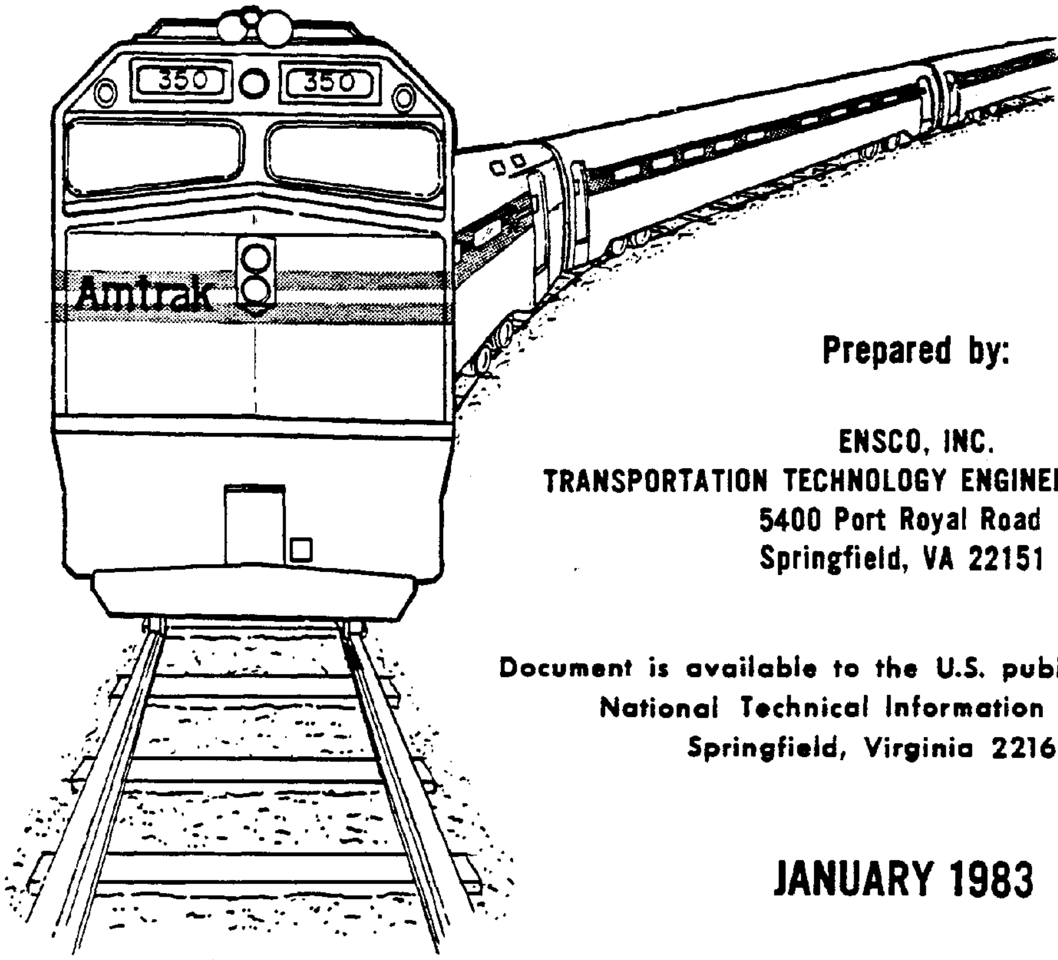

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16. Abstract

Increasing the speed of passenger trains in existing curves has been proposed as an alternative to changing curve radii for the purpose of reducing trip times on the Northeast Corridor. Tests were performed on the F40PH diesel-electric locomotive and an Amcoach modified for banking, with and without the banking system in operation. Safety at high cant deficiency was evaluated by comparing direct wheel/rail force measurements to safety criteria from world-wide sources.

Cant deficiency limits for these and other lightweight four-axle passenger vehicles (tested previously) were set by concern for vehicle overturning (wheel lift) rather than wheel climb or catastrophic track damage. An allowance for safety in high cross wind operation caused the coach to be the limiting factor. A typical curve limited to 60 mph by the current 3 inch cant deficiency limit could be negotiated by a train consisting of F40PH locomotives and banking Amcoachs at up to 76 mph (8 inch cant deficiency) while satisfying ride comfort and safety criteria. A few rough curves were identified as exceptions to the general rule.

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1.0 EXECUTIVE SUMMARY

This report describes high speed curving tests performed by ENSCO, Inc. under the joint sponsorship of the Federal Railroad Administration (FRA) and Amtrak. Tests were performed to evaluate the safety of the F40PH locomotive and the prototype banking Amcoach at elevated cant deficiency and the effectiveness of the banking system in maintaining ride quality at higher curve speeds. The ground work for testing these vehicles was established during a previous effort involving the same organizations (reference: High Cant Deficiency Testing of the LRC Train, the AEM-7 Locomotive and the Amcoach, NTIS Report No. 82213018).

Four mechanisms which can lead to derailment were identified from available literature in North America, Europe and Japan during the previous project. Quantitative criteria were defined to indicate the limits of safe operation with respect to the mechanisms of vehicle overturning, wheel climb, rail rollover and lateral track panel shift.

A set of criteria developed by Japanese National Railway (JNR) researchers was used to evaluate safety against vehicle overturning. One criterion is a limit on short time duration transient load transfer which reduces the vertical load on the low rail wheels during curving. The criterion requires that at least 20% of the nominal vertical wheel load remains on the low rail wheels. The effects of vehicle dynamics and track irregularities as well as steady state load transfer and the effect of lateral wind force are considered. A second criterion evaluates safety on the basis of only steady state load transfer and the effect of lateral wind force. It requires that 40% of the nominal vertical wheel load remains on the low rail wheels when only steady state events are considered. The assumption that the component of transient load transfer caused by track irregularity and vehicle dynamics is less than 20% of the static wheel load is implicit in the steady state criterion.

The advantage of the steady state overturning safety criterion is that steady state forces may be expressed as functions of cant deficiency leading to a general cant deficiency limit applicable to all curves. Also, simple computational models are of value in predicting the general cant deficiency limit of a vehicle. When the transient criterion is applied to a particular vehicle, it assigns a unique cant deficiency limit at each curve as a result of geometric pecularities. Although the transient criterion is more rigorous, it is not as useful as the steady state criterion in defining general vehicle limitations.

The allowance for track related transient load transfer implicit in the use of the steady state overturning criterion by JNR is not of obvious applicability to larger U.S. vehicles and differing track conditions. The method of investigation was to define general cant deficiency limits for the F40PH locomotive and the banking Amcoach with respect to the steady state overturning criterion and to identify exceptions to the general rule by applying the transient criterion at a large representative sample of curves. The number and character of the exceptions determine the usefulness of the general cant deficiency limits.

A major conclusion of this test and the one preceeding it was that there were few exceptions to the general cant deficiency limits based on the steady state overturning criteria for the F40PH, the prototype banking Amcoach, the standard Amcoach, the AEM-7, the LRC coach or the LRC locomotive. All of the curves where the transient criterion set a significantly lower cant deficiency limit for the F40PH and the Amcoach than the steady state criterion were associated with switches, undergrade bridges or grade crossings.

Another important conclusion common to all of the above vehicles is that of the four derailment mechanisms, the danger of vehicle overturning (wheel lift) limited the amount of cant deficiency. It appears that cant deficiency safety limits set by concern for vehicle overturning rather than wheel climb or catastrophic track damage is a general characteristic of modern lightweight passenger vehicles with four axles.

The overturning safety of the above coaches rather than locomotives sets the general cant deficiency safety limits of trains. The reason is that light vehicles with large side areas are most influenced by the assumption of worst case lateral winds in the overturning criteria. A wind speed of 56 mph was used in the calculation of safety criteria because it is the greatest expected on the Northeast Corridor within 15 feet of the ground for a 10 year mean recurrence interval.

The criterion used to evaluate safety against wheel climb was taken from the Amtrak acceptance specification of the AEM-7 locomotive. Transient measurements of the ratio of the simultaneous lateral and vertical forces on the most heavily loaded wheel were compared to the criterion.

The rail rollover safety criterion is the result of research by the AAR. It uses transient measurements of the L/V ratio for the high rail side of the most heavily loaded truck to evaluate the safety of a vehicle.

The criteria used to evaluate safety against lateral track panel shift are based on research by the French SNCF with modifications for American tie spacing and the internal thermal forces of welded rail as proposed by Battelle Columbus Labs. The basic criterion makes use of transient lateral axle force, and a corollary criterion evaluates safety on the basis of transient truck lateral force. The more inclusive truck force measurements were used, and a conservative assumption of superimposed wind forces was made.

Table 1-1 enumerates the various safety criteria for a range of passenger vehicles. Differences in the criteria when applied to

TABLE 1-1

SUMMARY OF SAFETY CRITERIA LIMITS FOR SPECIFIC TEST VEHICLES

		Maximum Permissible Test Measurement					
Derailment Mechanism	Measurement	F40PH Locomotive	Banking Amcoach	Standard Amcoach	AEM-7 Locomotive	LRC Locomotive	LRC Coach
Vehicle Overturning	Steady State Weight Vector Intercept	15.7 in	12.8 in	12.8 in	16.2 in.	16.3 in	12.5 in
	Transient Weight Vector Intercept	21.7 in	18.8 in	18.8 in	22.2 in	22.3 in	18.5 in
Wheel Climb	Transient Wheel (L/V) $T \geq 50$ ms	0.9	0.9	0.9	0.9	0.9	0.9
Rail Rollover	Transient Truck Side (L/V) $T \geq 50$ ms	0.57	0.65	0.65	0.59	0.57	0.65
Track Panel Shift	Transient Lateral Axle Force	41,900 lb	18,700 lb	18,700 lb	34,000 lb	41,300 lb	18,200 lb
	Transient Lateral Truck Force	59,800 lb	27,300 lb	27,300 lb	48,400 lb	58,900 lb	26,900 lb

various vehicles result from differences in their weight and cross wind exposure area. The vehicle overturning criteria are expressed in terms of weight vector intercept. Weight transfer usually is conceptualized in U.S. literature by the movement of a resultant inertial force vector from the centerline of the track.

Knowledge of vertical and lateral wheel/rail forces is required for the application of the quantitative safety criteria, and instrumented wheelsets existing in the FRA inventory were calibrated and used at the lead trucks of the F40PH locomotive and banking Amcoach. The data acquisition system developed under the previous program was used both for real-time data processing necessary for monitoring the safety of the test crew and for more detailed post processing of the recorded data.

Both steady state and transient measurements are necessary for the determination of the maximum safe cant deficiency of a vehicle. Steady state performance refers to the average force and acceleration levels existing in the constant radius part of a curve, and transient performance is concerned with extreme forces and accelerations of around 50 millisecond time duration due to curve irregularities and vehicle dynamcis.

Two types of tests were conducted. In the first part of the program runs over the same pair of left and right curves were repeated at increasing speed. Investigation of steady state performance was the objective, and a cant deficiency of approximately 12 inches was achieved (corresponding to 97 mph at a test curve normally limited to 70 mph). The rest of the program consisted of road tests intended to reach the highest possible speed at as many curves as possible between New Haven and Providence on the Northeast Corridor. Cant deficiency targets were increased incrementally and key safety measurements monitored continually. Measurements were taken at over one hundred curves in the test zone, and cant deficiencies of nine inches or more were achieved at many. The object was to investigate transient performance over a wide range of typical curve features and perturbations.

Both types of tests were performed with and without operating the prototype Amcoach banking system. The worst case condition for each derailment mechanism was required for the evaluation of safety.

The results of the repetitive tests and the road tests were processed using statistical routines. The objective of the data reduction was to organize the data so that the results could be compared to safety and comfort criteria.

The data analysis effort was concentrated on the following principal measurements:

- Speed
- Carbody Lateral Acceleration
- High Rail Lead Wheel Lateral Force
- Low Rail Lead Wheel Vertical Force
- High Rail Lead Wheel Vertical Force
- High Rail Side Truck Lateral Force
- High Rail Lead Wheel L/V Ratio
- High Rail Side Truck L/V Ratio
- Weight Vector Intercept

When time consuming examination of analog strip charts is performed, the force (or other measurement) level which was exceeded for 50 milliseconds is a good indication of the intensity of transient phenomena. The analog strip chart data from the repetitive runs on the same curve were compared to the results obtained from the computer derived statistical data reduction. The comparison showed that the 95th percentile level from the statistical analysis (95% of the data points are less than this value) agreed closely with the transients (50 ms exceedance) identified by chart readings. Therefore, the 95th percentile statistical level was used for comparison to safety criteria where transient measurements were appropriate in order to automate the data reduction.

RESULTS

Table 1-2 summarizes the test results of the F40PH locomotive and prototype banking Amcoach. Whether the higher measurements occur in the banking or non-banking mode of the modified Amcoach is identified. Results from the previous test of other passenger vehicles are included for comparison.

CONCLUSIONS

1. A train consisting of F40PH locomotives and Amcoaches satisfies the operating safety criteria at up to 8 inches cant deficiency at all but 3 curves tested on the NEC between New Haven and Providence. The maximum cant deficiencies set by the transient overturning safety criterion are 6.3 inches for the locomotive and 7.2 for the coach at one curve, 7.6 inches for the locomotive at another and 7.8 inches for the coach at the third.

2. The general cant deficiency limit for the train is useful because exceptions caused by transient weight transfer are very few, and they all occurred at curves with switches or undergrade bridges. No exceptions limiting safe cant deficiency below 6 inches were identified.

3. The general cant deficiency limit, obtained by rounding down to the nearest inch the limit imposed by the steady state overturning criterion, is 8 inches for both the banking Amcoach and the standard Amcoach. It is useful because the transient overturning criterion identified only two exceptions in over 100 test curves, and both were associated with switches in undergrade bridges.

4. The general cant deficiency limit of F40PH locomotive, obtained by rounding down to the nearest inch the limit imposed by steady state overturning, is 9 inches. It is less useful than the general limit of the Amcoach because

TABLE 1-2

SUMMARY OF TEST RESULTS

	F40PH Locomotive	Banking Amcoach (Worst case)	Standard Amcoach	AEM-7 Locomotive	LRC Locomotive	LRC Coach
Recomended General Cant Deficiency Limit	9 in	8 in (non-banking)	8 in.	10 in.	12 in.	9 in.
Cant Deficiency Limit Set by Steady State Overturning Criterion	9.5 in.	8.3 in.* (non-banking)	8.3 in.	10.5 in	12.2 in.	9.3 in
Lowest Cant Deficiency Limit Set by Transient Overturning Criterion at a curve without a special feature**	9.1 in.	8.6 in.* (non-banking)	8.5 in.	8.5 in.	10.6 in.	8.7 in
Lowest Cant Deficiency Limit Set by Transient Overturning Criterion at any Test Curve	6.3 in.	7.2 in.* (non-banking)	8.5 in	4.7 in	10.6 in	6.8 in
Estimated Maximums at General Cant Deficiency Limit						
Transient Wheel (L/V) Ratio***	.45	.64 (banking)	.60	.75	.60	.60
Transient Truck Side (L/V)**** Ratio	.36	.45 (banking)	.40	.50	.40	.40
Transient Lateral Truck Force	41,000 lb	18,000 lb (both)	18,000 lb	32,000 lb	33,000 lb	15,000 lb
Steady State Lateral Acceleration	.19g	0.10 (banking 0.18 (non-banking)	0.15g	0.18g	0.25g	0.09g

*Including allowance for typical static load asymmetry, see Section 6.6.
**Switches, undergrade bridges or grade crossings in curves are special features.
***Safety criterion in .9.
****Safety criterion is .57 to .65, see Table 1-1.

eight exceptions were identified by the transient overturning criteria. Switches, undergrade bridges or grade crossings were associated with all curves where exceptions occurred.

5. The Amcoach rather than the locomotive set the general cant deficiency limit of the train because the assumption of worst case lateral wind is more restrictive for the coach.

6. The JNR overturning criterion, based on steady state considerations only, was generally applicable to U.S. passenger trains. However, the F40PH locomotive exhibited greater transient force increases than the Amcoach at curve irregularities.

7. All curves which caused exceptionally high transient weight transfer in the test vehicles also induced harsh transient lateral acceleration, but the presence of high transient lateral acceleration did not guarantee exceptional transient weight transfer.

8. The banking system of the modified Amcoach was successful in maintaining a low level of steady state carbody lateral acceleration at high cant deficiency. The AAR ride comfort criterion was satisfied up to 8 inches cant deficiency.

RECOMMENDATIONS

1. The general cant deficiency limit of a train using F40PH locomotives and banking Amcoaches should be considered 8 inches. Steady state ride quality requirements are achieved and conservative safety criteria are satisfied at all but a very few identifiable curves.

2. Lateral acceleration surveys (especially when measurements are made in the locomotive) can be used to identify curves where exceptions to the general cant deficiency limit exist. Exceptions are much more likely at curves with switches, undergrade bridges, or grade crossings. The frequency of surveys should be based on local track degradation history.

3. Fail-safe devices are required to prevent one truck of a banking coach from operating while the other is disabled.

4. The general cant deficiency limit of a train using F40PH locomotives and standard Amcoaches should be considered 6 inches, purely for passenger ride comfort requirements. No exceptions to the safety criteria were identified below 6 inches cant deficiency.

2.0 INTRODUCTION

An efficient way to reduce trip times in Amtrak's Northeast Corridor (NEC) is to increase train speed in curves. It has been demonstrated that AEM-7 electric locomotives and standard Amcoaches may operate at substantially higher than present curving speeds while satisfying established safety criteria. Tilting the Amcoach body in curves was proposed as a method of maintaining passenger comfort during faster curving, and a retrofit system was designed and installed by the Budd Company on an existing Amcoach (No. 21183). A test was then conducted to evaluate the high speed curving safety of the prototype banking Amcoach and F40PH diesel-electric locomotive used on the non-electrified portion of the NEC between New Haven and Boston. Measurements were also made to indicate the effectiveness of the banking system in maintaining ride comfort at high curving speed.

2.1 DESCRIPTION OF TEST VEHICLES

The Amcoach weighs about 120,000 lb with load and has light fabricated trucks designed for high speed stability. The primary suspension of an Amcoach I consists of very firm rubber rings between the axle journal bearings and the clamping devices at the corners of the truck frame. The test coach was upgraded to an Amcoach II by the use of a truck frame with larger and softer elastomeric elements in the primary suspension and the addition of tread brakes. The secondary suspension spring units are coil springs in series with air springs placed between the carbody floor and a transverse bolster (Figure 2-1). The bolster is restrained longitudinally to the body with brake reaction struts and the truck frame is attached to it with a center pin. The banking system tilts the body toward the inside of the curve by overcoming the secondary airsprings with the air actuated torsion bar device shown in Figure 2-2. The ends of the torsion bar are supported by bearings secured to the carbody. The tilt arms which flex the torsion bar are linked to the bolster. The air

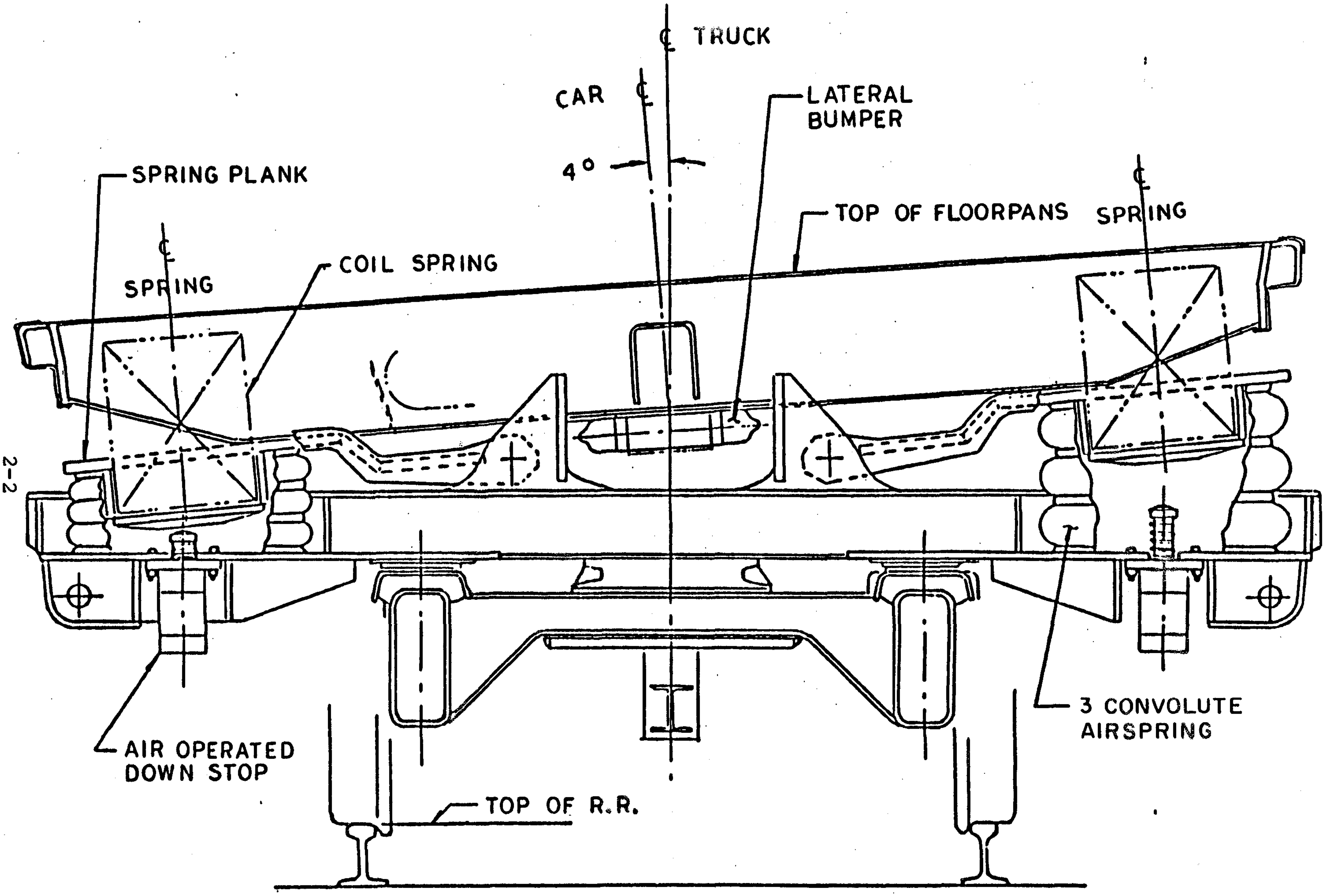


FIGURE 2-1

TRANSVERSE SECTION THRU TRUCK BOLSTER

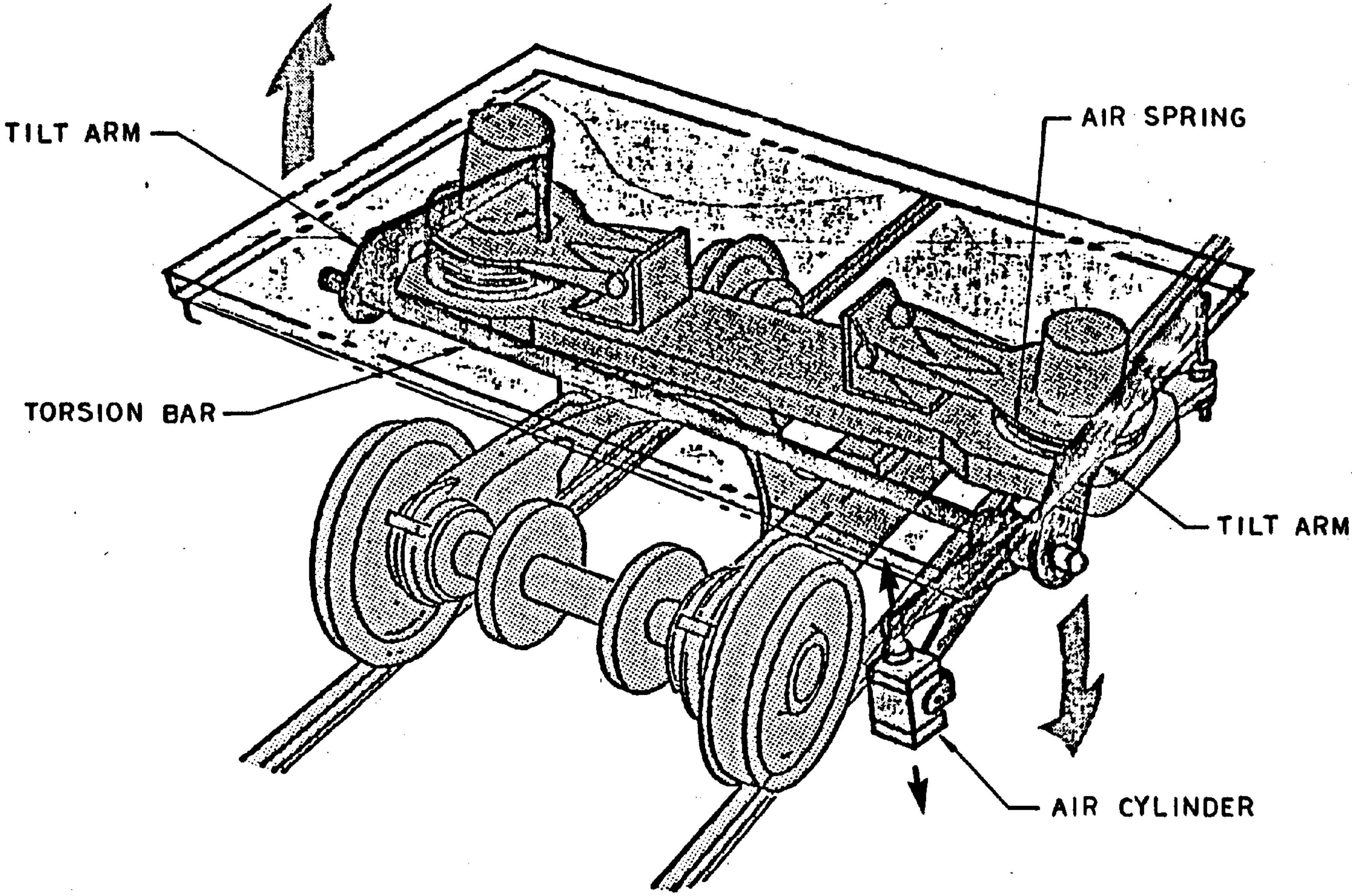


FIGURE 2-2
TILT ARM & TORSION BAR ARRANGEMENT

cylinder rotates one tilt arm out of plane with the other to cause the body to tilt with respect to the bolster. When the damped lateral acceleration of the truck frame reaches 0.04g, an electronic controller initiates the full 4° body tilt. The air spring stops are retracted also, at controller command, to accommodate the 4° carbody movement. Table D-1 contains Amcoach II specifications.

The F40PH is a four axle locomotive weighing about 256,000 lb with supplies. The primary suspension consists of coil springs between the truck frame and sliding axle bearing housings. Most of the total vertical deflection is at the primary suspension. The secondary suspension is shown in Figure 2-3. The bolster, which has a center-pin connection to the body, is supported at each end by a wedge between a pair of inclined elastic blocks acting as very stiff secondary springs. Early F40PH locomotives with GP suspension use even stiffer single vertical blocks. The secondary springs rest on the spring plank which is suspended under the truck frame rails by a pair of swing hangers. The swing hangers allow secondary lateral displacement. Tables D-2 and D-3 contain specifications of F40PH locomotive with inclined rubber suspension and GP suspension respectively. A test vehicle with inclined rubber suspension was chosen because slightly greater body roll and load transfer was expected with its softer secondary springs.

2.2 DEFINITION OF CANT DEFICIENCY

The test program focused on measurements required for the evaluation of safety and comfort at higher curving speeds. The measurements were recorded as functions of cant deficiency in order to compare the results at many curves. Cant deficiency expresses the intensity of curving by considering the curve radius and banking as well as vehicle speed. Figure 2-4 illustrates cant deficiency in the simplified case where the vehicle center of gravity is fixed on the track center line. In Figure 2-4A the

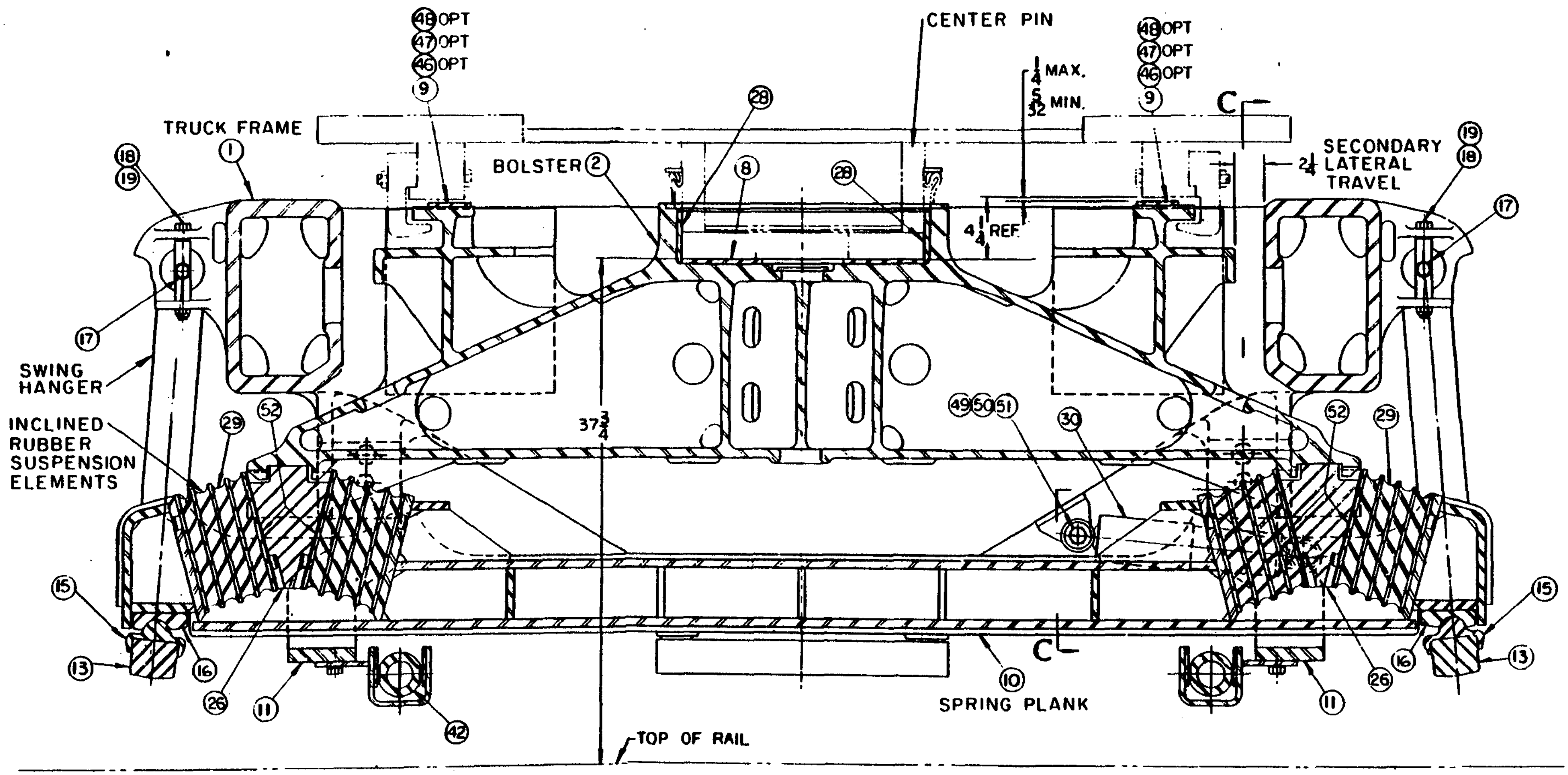
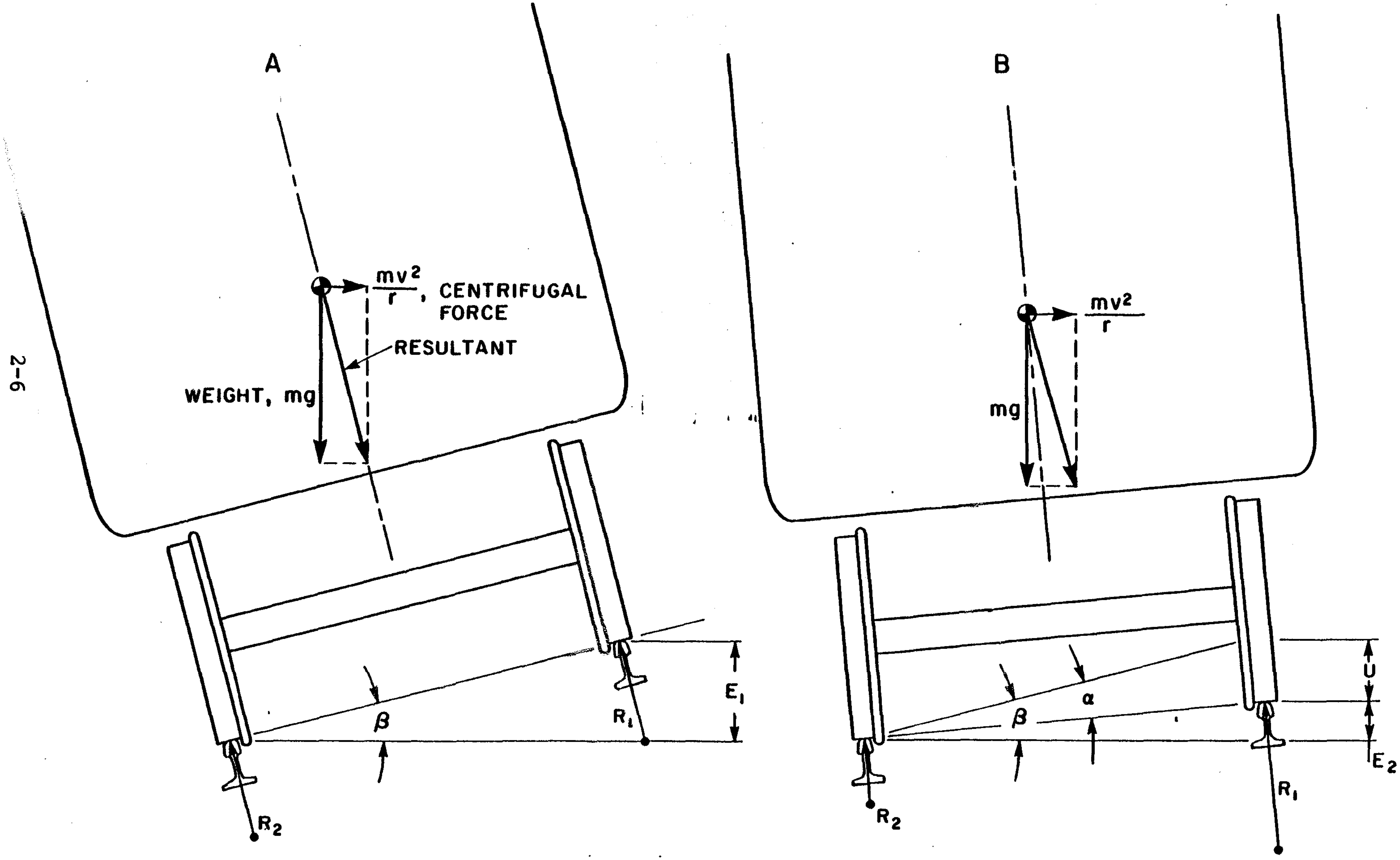


FIGURE 2-3

TRANSVERSE SECTION THROUGH F40PH LOCOMOTIVE TRUCK BOLSTER

CANT DEFICIENCY



BALANCE CONDITION AT SPEED V AND CROSSLEVEL E_1
(VERTICAL RAIL FORCES R_1 AND R_2 ARE EQUAL)

UNBALANCE AT SAME SPEED AND CROSSLEVEL E_2
CANT DEFICIENCY, U, EQUALS $E_1 - E_2$

track has sufficient crosslevel (or cant), E_1, so that the resultant of the weight and centrifugal force vectors lies along the centerline of the track. Therefore, the rail forces, R_1 and R_2 are equal, and curving is said to be balanced with zero cant deficiency. In Figure 2-4A the weight vector, mg, and the centrifugal force vector, mv^2/r, are the same as in Figure 2-1A. The quantity, m, is the vehicle mass; g, the gravitational acceleration; v, the vehicle speed; and r, the curve radius. However, in Figure 2-4B the crosslevel, E_2, is less. The resultant no longer lies on the center line, and the outer rail force is larger than the inner. The cant deficiency is U, the difference between the crosslevel for balance and the actual crosslevel. This paper expresses cant deficiency in inches according to the American convention. The European convention is to express cant deficiency by the angle, α.

2.3 OVERVIEW

An evaluation of operating safety addresses the risks of vehicle overturning, wheel climb, rail rollover and track panel shift due to increased wheel/rail forces at higher curving speed. A mini-computer based data acquisition system and force sensing instrumented wheelsets developed for previous research of the Federal Railroad Administration were used to obtain wheel/rail force, suspension displacement, and carbody acceleration measurements. The data acquisition system (DAS) recorded test data on digital tapes and computed real-time wheel/rail forces, vehicle rollover indicators and truck and wheel lateral to vertical force ratios, and displayed them on strip charts used to monitor the safety of the test crew. Statistical analysis and graphical displays of data contained in the four volume Appendix A were obtained using the same DAS for post processing.

The test train consisted of F40PH locomotive #350 and Amcoach #21183, modified for banking, followed by three standard Amcoaches. The lead truck of the locomotive and banking Amcoach were equipped with instrumented wheels.

Repetitive runs at increasing speed were made at a right curve and a left in Part A of the test. Measurement of steady state performance up to 12 inches cant deficiency was accomplished.

Road tests between New Haven and Groton with increasing cant deficiency targets were performed in Part B of the test. The emphasis was on transient measurements depending on irregularities of curves, and many curves were tested at 9 inches of cant deficiency.

In Part C the route of the road tests was extended to Providence. Measurements at over 100 curves were made at elevated cant deficiency.

The route between New Haven and Providence has the highest concentration of curves on the Northeast Corridor, and F40PH locomotives and Amcoaches are the standard passenger equipment serving this route. The results of this test program have been used to support Amtrak's petition to operate trains at higher curving speeds.

3.0 DESCRIPTION OF TESTS

The test was designed to investigate the curving safety limits by a series of orderly steps. A preliminary step was the computation of steady state weight transfer, lateral acceleration, and lateral truck force as a function of cant deficiency based on manufacturer's specifications using the model in Appendix D. This step established the first estimate of speed limitations for test planning. The first step, in the field, was a static lean test of the actual test vehicles on approximately 6 inches of crosslevel to estimate their "as installed" suspension constants. The results of the static lean tests were used to confirm the previous computation. These pre-test computations provided a set of reasonable expectations used for quality control of the initial full scale measurements.

3.1 ROAD TESTS

The curving safety criteria detailed in Appendix B and summarized in Section 4.0 pertain to both steady state and transient measurements of wheel forces. Average forces throughout the constant radius curve body were considered to be steady state, and extreme measurements (high or low) having time durations of about 50 milliseconds occurring at the curve body or the spirals were referred to as transient.

Part A of the road tests consisted of repeated runs at increasing speed at a right curve and a left curve to establish steady state cant deficiency limits and to indicate the gradient of peak forces with increases in cant deficiency. Curve 67 track 2 eastbound ($2^{\circ}20'$, 5.88" having wooden ties) at Bradford, RI on the Northeast Corridor was the right curve, and curve 67 track 1 westbound ($2^{\circ}36'$, 5.25" having concrete ties) was the left curve. Curving speeds were increased incrementally for cant deficiencies from 2 inches to nearly 12 inches. Transient and steady state measurements were examined after each run to assure safety at the next cant deficiency step. Right and left curves were tested to investigate possible asymmetry of performance, and to base conclusions concerning safety on worst case situations. Likewise, the tests were repeated with an without banking operation of the coach to seek the worst case situation for each safety parameter. The steady state measurements could be applied to all curves by expressing them as functions of cant deficiency because steady state performance is independent of perturbations and spirals particular to individual curves.

Part B was the first series of over-the-road tests designed to sample transient measurements over a wide range of track conditions. The purpose of these tests was to determine whether the more conservative limit of safe cant deficiency was set by steady state performance or by transient performance. Round trips between New Haven and Groton were made progressively increasing the cant deficiency targets from 5 to 9 inches. Tests were repeated with and without the operation of the coach banking system. A computer program, which considered the acceleration and braking rates and top speed of the train, was used to calculate a speed profile to achieve the target cant deficiency at the greatest number of curves in the test zone. Figure 3-1 is a sample of the displays provided to the engineer to assist him in maximizing the number of curves tested at the target cant deficiency. The curve number and target speed are specified in a box tagged to each curve location on the graph.

Part C was an extension of the test zone to Providence. About 112 curves were available for high speed testing in the full test zone. The same procedures were used, and the highest speed runs were chosen for analysis without distinction between Parts B and C.

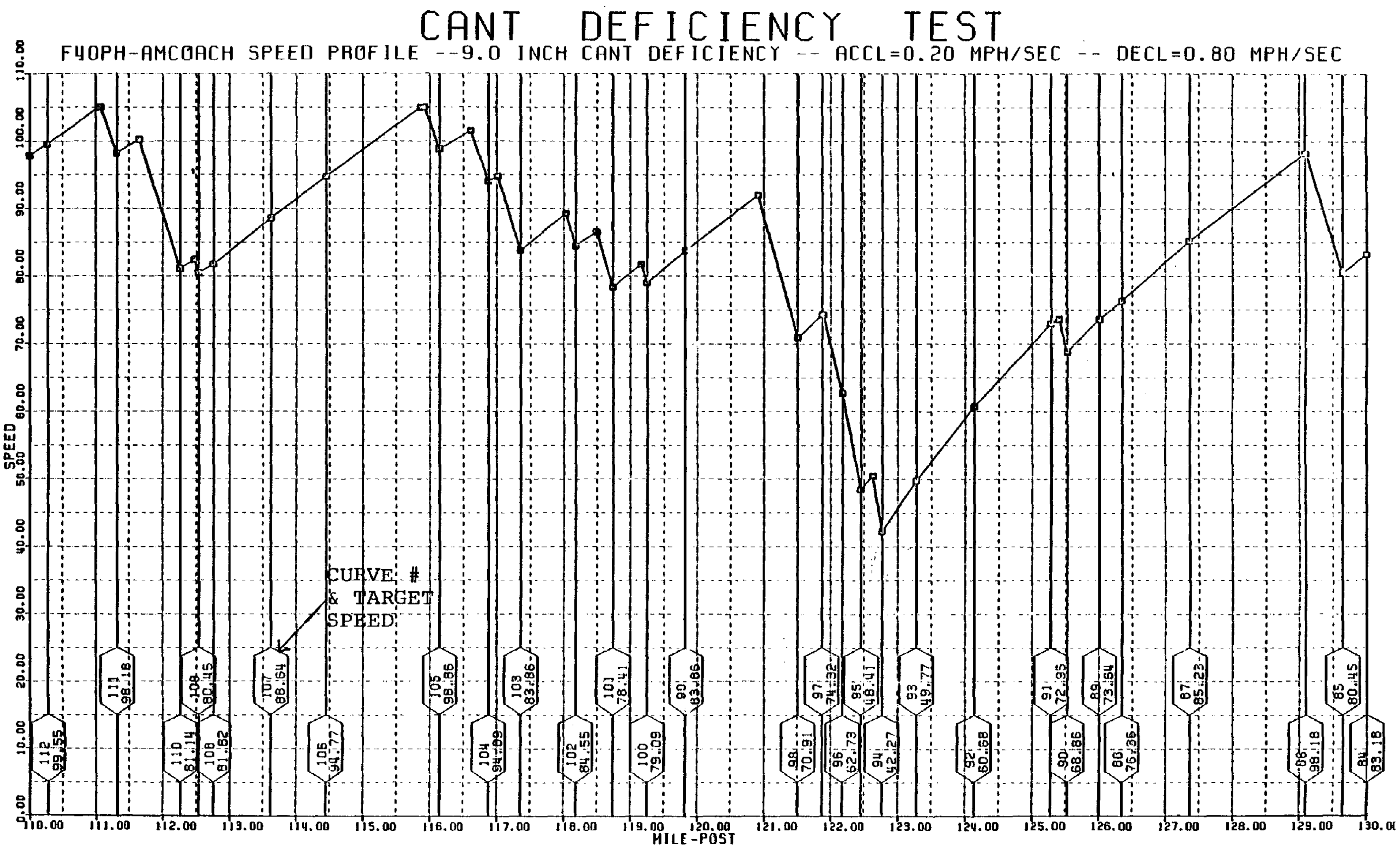


Figure 3-1. Sample Segment of Target Speed Profile for Road Test Runs

3.2 MEASUREMENT CHANNELS

The principal measurements for determining safety at high cant deficiency were the continouus measurements of vertical and lateral force at each wheel of the lead trucks of the F40PH locomotive and the prototype banking Amcoach. Measurements using instrumented rails show that higher lateral truck forces in general occur at the lead truck and confirm that vertical load transfer is equivalent at both trucks (ref. 25). Appendix C describes the instrumented wheelsets. The banking Amcoach was coupled to the locomotive, and three additional coaches were used to make up a representative test train (and to provide extra braking since the instrumented wheels were not braked). Other sensors were installed to measure accelerations, primary and secondary suspension travel, banking movements, speed and other parameters. Table 3-1 lists the array of sensors.

A minicomputer based data acquisition system was used to compute and display several key safety parameters in real time to monitor the safety of the test train. This system also recorded the raw data digitally on magnetic tape and performed the post processing computations and statistical analysis. Generating the speed profiles for the tests between New Haven and Providence was another of its chores.

TABLE 3-1

TEST PARAMETERS

Parameter	Sensor	No. of Channels
LOCOMOTIVE:		
Vertical Wheel Force	Wheel Strain Gage Bridge	8
Lateral Wheel Force	Wheel Strain Gage Bridge	4
Truck Side L/V	Computed	
Wheel L/V	Computed	
Weight Vector Intercept	Computed	
Primary Vertical Displacement (1 Axle)	Displacement Potentiometer	2
Secondary Vertical Displacement	Displacement Potentiometer	2
Secondary Lateral Displacemwent (& Truck Yaw)	Displacement Potentiometer	2
Cab Lateral Accelerations	Servo Accelerometer	1
	Loco Data Channels	19
COACH:		
Vertical Wheel Force	Wheel Strain Gage Bridge	8
Lateral Wheel Force	Wheel Strain Gage Bridge	8
Truck Side L/V	Computed	
Wheel L/V	Computed	
Weight Vector Intercept	Computed	
Speed	Decelostat	1
Location	ALD	1
Primary Vertical Displacement (1 Axle)	Displacement Potentiometer	2
Secondary Vertical Displacement	Displacement Potentiometer	2
Secondary Lateral Displacement (& Truck Yaw)	Displacement Potentiometer	2
Banking Motions	Displacement Potentiometer	1
Carbody Lateral Accel.	Servo Accelerometer: test coach & ref. coach	2
Carbody Vertical Acceleration	Servo Accelerometer: test coach & ref. coach	2
Truck Lateral Accel.	Tilt System Test Point; 6.8 v/g	
Tilt Command Signal	Tilt System Test Point; +2V,0, or -2V	1
Primary Lateral Displacement	LVDT	1
Torsion Bar Stress	Budd Strain Gage Bridge	1
Cylinder Pressure	Budd Sensors	2
	Coach Data Channels	35

4.0 SUMMARY OF SAFETY AND COMFORT CRITERIA

The safety criteria recommended by world-wide sources for determining operating cant deficiency limits are discussed in Appendix B. The detailed discussion is exerpted from the LRC test report (Ref 23). The criteria address four principal hazards associated with high speed curving: vehicle overturning, wheel climb, rail rollover, and track panel shift. The operating safety criteria are a means of using wheel force measurements made under test conditions to determine safe curving speeds. The safety criteria includes allowances for maximum wind forces.

VEHICLE OVERTURNING

Criteria a) and b) below are based on the JNR load ratio standards and are useful for comparison to steady state models and to steady state and transient measurements. The effect of potential high cross winds is factored into the criteria. Transient measurements of about 50 ms duration should be used for comparison to the transient overturning criteria. The lower of the two cant deficiency limits derived from steady state and transient criteria should be taken when both measurements are available. The criteria may be stated by the following two conditions expressed in terms of weight vector intercept.

a) Steady state criterion:

$$\text{Steady State Vector Intercept} \leq 18 - (.0153V^2Sh_{cp}/W) \text{ inches}$$

and

b) Transient criterion:

$$\text{Peak Vector Intercept} \leq 24 - (.0153V^2Sh_{cp}/W) \text{ inches}$$

where:

V = the lateral wind speed in miles per hour

S = the lateral surface area of the vehicle in square feet

h_{cp} = the height of the center of wind pressure in feet

W = one-half of the unloaded weight of the vehicle in pounds

Figure 5-1 is a graphical representation of the criteria for the F40PH locomotive.

WHEEL CLIMB

The criterion of safety against wheel climb used by Amtrak and EMD is recommended because it clearly specifies the maximum permissible measurement as a function of time duration and was developed using the AAR flange angle of 68°. It may be expressed as:

$$\text{Peak Wheel } (L/V) < 0.056T^{-0.927} \qquad \text{for } T < 50 \text{ ms}$$

and

$$\text{Peak Wheel } (L/V) \leq 0.90 \qquad \text{for } T \leq 50 \text{ ms}$$

The peak measurement for a particular time duration is the maximum level that was exceeded for that time duration. Because of a lack of full scale measurements, this criterion is based on Nadal's formula (reference 1) with conservative judgements of friction coefficient and angle of attack.

RAIL ROLLOVER

The rail rollover criterion was based on rail section geometry and AAR measurements of the torsional support of the surrounding rail, and it assumes zero pull out strength of the fasteners. It is expressed as follows:

$$\text{Peak truck } (L/V) \leq 0.5 + 2{,}300/P_w$$

for peaks of 50 ms or greater duration where P_w is the nominal wheel load. For peaks of less than 50 ms duration greater levels can be endured safely as given by the rule:

$$\text{Peak truck } (L/V) \leq .113\ (0.5 + 2300/P_w)T^{-0.728}$$

for $T < 50$ ms

TRACK PANEL SHIFT

A criteria for determining the maximum lateral axle force on wood tie track with compacted ballast which takes into account the internal forces in CWR due to temperature changes and the lateral carbody forces caused by unfavorable high crosswinds is:

$$F_{max}(\text{wheel}) = \left[1 - \frac{A\Delta\theta}{22320}\ (1 + .458D)\right]\left[.7P+6600\right] - (1.28\text{x}10^{-3}SV^2)$$

where

A = rail section area, in^2
$\Delta\theta$ = max temperature change after rail installation, $^{\circ}F$
D = track curvature, degrees
P = vertical axle load, lbs
S = lateral surface area of vehicle, ft^2
V = lateral wind speed, mph

and it is assumed that a single axle bears half the entire wind load. For typical NEC conditions of 140-pound rail ($A = 13.8$ in^2), $\Delta\theta$ max of 70°F and D max of 4°.

$$F_{max} \text{ (wheel)} = .61P + 5800 - 1.28 \times 10^{-3}\ SV^2$$

A maximum truck force criteria for the CWR example can be expressed:

$$F_{max} \text{ (truck)} = .7N\left[.61P + 5800 - (1.28/.7N)10^{-3}SV^2\right]$$

where N is the number of axles per truck and one truck supports half the total wind load. The allowable truck force is less than the number of axles times the maximum allowable axle force because the zones of ties loaded laterally by each axle overlap.

Transient measurements of high rail wheel and high rail truck side lateral forces having durations of about 50 ms are suitably conservative measures of maximum axle and truck lateral loads, respectively. This conservativeness is warranted because the only track shift measurements in the literature were taken on French 92 lb/yd rail and even the best criteria in use is an extrapolation from the French National Railway (SNCF) experiments.

RIDE QUALITY

Current AAR standards limit steady state lateral acceleration to 0.1 g and "jerk" to 0.03 g/sec. The JNR criteria of ±.08 g maximum additional transient component upon entering and exiting curves should be considered, especially for tilt body cars. Low frequency measurements filtered at about 1 Hz are appropriate for comparison to these comfort standards.

5.0 RESULTS OF TESTING THE F40PH LOCOMOTIVE AT HIGH CANT DEFICIENCY

Two types of testing were performed with the F40PH Locomotive. In the first type, repetitive runs at increasing speeds (over the same curve) were made to determine steady state performance. Steady state measurements were obtained by averaging over the constant radius curve body. Tracks 1 and 2 at curve 67 (Bradford, RI) were used to obtain data at both left and right curves.

The second type of testing was a series of runs over NEC track between New Haven and Boston to determine transient performance for a wide variety of track conditions. A transient measurement is the extreme which can result from the effects of vehicle dynamics and track perturbations superimposed on the steady state measurement. The steady state test results at one curve can be applied to all curves if they are expressed as functions of cant deficiency. However, transient measurements depend on track geometry, and they are unique to each curve. A large sample of test curves is required to gauge accurately the range of transient forces and accelerations to be expected.

As the term suggests, a transient event has a very short time duration. An important question is how long must the transient event endure for it to be of consequence. A very large pulse of lateral wheel force existing for only one thousandth of a second would not derail a car because the energy associated with such a short transient force is small. A way to classify the magnitude of a transient measurment, which is arbitrary but thought by researchers to be conservative, is illustrated in Appendix B on page . This method defines the value of the transient measurement as that which was exceeded for 50 milliseconds. It was used to monitor safety during the tests because it is a rule which is easy to apply to strip chart recordings. The transient measurements presented in this section and in Appendix A were reduced from digital tape recordings with the aid of a computer by taking the 95th percentile of the filtered digitized points sampled at each curve. A comparison using the repetitive runs indicated data reduced by the computer with the 95th percentile statistic was equivalent to hand reduction using the 50 ms exceedance rule.

5.1 VEHICLE OVERTURNING

A set of criteria developed by the Japanese National Railway (JNR) researchers was used to evaluate safety against vehicle overturning. One criterion is a limit on short time duration transient load transfer which reduces the vertical load on the low rail wheels during curving. The criterion requires that at least 20% of the nominal vertical wheel load remains on the low rail wheels. The effects of vehicle dynamics and track irregularities as well as steady state load transfer and the effect of lateral wind force are considered. A second criterion evaluates safety on the basis of only steady state load transfer and the effect of wheel wind force. It requires that 40% of the nominal vertical wheel load remains on the low rail wheels when only steady state events are considered. The assumption that the component of transient load transfer caused by track irregularity and vehicle dynamics is less than 20% of the static wheel load is implicit in the steady state criterion. Experimental evidence was necessary to determine the applicability of this assumption to the larger vehicles and different track conditions of the U.S. Northeast Corridor.

The advantage of the steady state overturning safety criterion is that steady state forces may be expressed as functions of cant deficiency leading to a general cant deficiency limit applicable to all curves. Also, simple computational models are of value in predicting the general cant deficiency limit of a vehicle. When the transient criterion is applied to a particular vehicle, it assigns a unique cant deficiency limit at each curve as a result of geometric pecularities. Although the transient criterion is more rigorous, it is not as useful as the steady state criterion in defining general vehicle limitations.

Historically, transient measurements have been made with instrumented track sites, and steady state information has been the domain of computational models. More recently developed instrumented wheelsets, such as those used during this test, provide continuous force measurements. Thus steady state and transient information could be gathered at a large number of curves. Steady state measurements from right and left curves were compared to the steady state overturning criterion and transient measurements at over 100 curves were compared to the transient criterion. The determination of the maximum cant deficiency, maintaining safety against vehicle overturning, was made for each curve using the more restrictive criteria. The usefulness of the steady state criterion to NEC track and equipment was investigated by noting the number and character of curves at which the transient criterion set the lower cant deficiency limit.

The JNR vehicle overturning criteria, which permit a maximum reduction of static wheel load of 60% for steady state computations and 80% for transient measurements may be stated in terms of vector intercept as follows:

Steady State Vector Intercept $\leq 18 - (.0153V^2Sh_{cp}/W)$ inches

and

Transient Vector Intercept $\leq 24 - (.0153V^2Sh_{cp}/W)$ inches

where:

V is the anticipated lateral wind speed in mph

S is the lateral surface area of the vehicle in ft^2

h_{cp} is the height of the center of wind pressure in ft

W is one half of the weight of the vehicle in lbs

For the F40PH Locomotive:

$S \simeq 730 \ ft^2$

$h_{cp} \simeq 8\text{-}1/4 \ ft$

$W = 128{,}000 \ lb.$

The second term in the above equations is an allowance for the maximum detrimental effect of wind speed. The equations are plotted in Figure 5-1. They express the maximum transient and the steady state test measurements of weight vector intercept permitted by the overturning criteria when a given crosswind speed is anticipated in service. Note that the tests are performed without crosswind. The criteria for comparison to the test data includes a computed factor to allow for additional wind forces in service.

5.1.1 WIND SPEED EFFECT

The overturning criteria are extremely sensitive to wind speed especially over 50 mph. However, small heavy vehicles such as the F40PH locomotive are the least affected by crosswinds. The operational cant deficiency should be chosen to allow for sudden unexpected lateral winds. Train speed, however, need not be limited by overturning considerations which include heedless operation in gales and hurricanes. Train speeds must be reduced under those circumstances to meet other conditions such as reduced visibility or the danger of debris on the track. The operational cant deficiency chosen will provide safe operation at the maximum wind speed corresponding to the 10 year mean recurrence interval. This cant deficiency is sufficiently conservative to provide for safety during unexpected winds. The level of wind speed for locations along the NEC is greatest in Boston

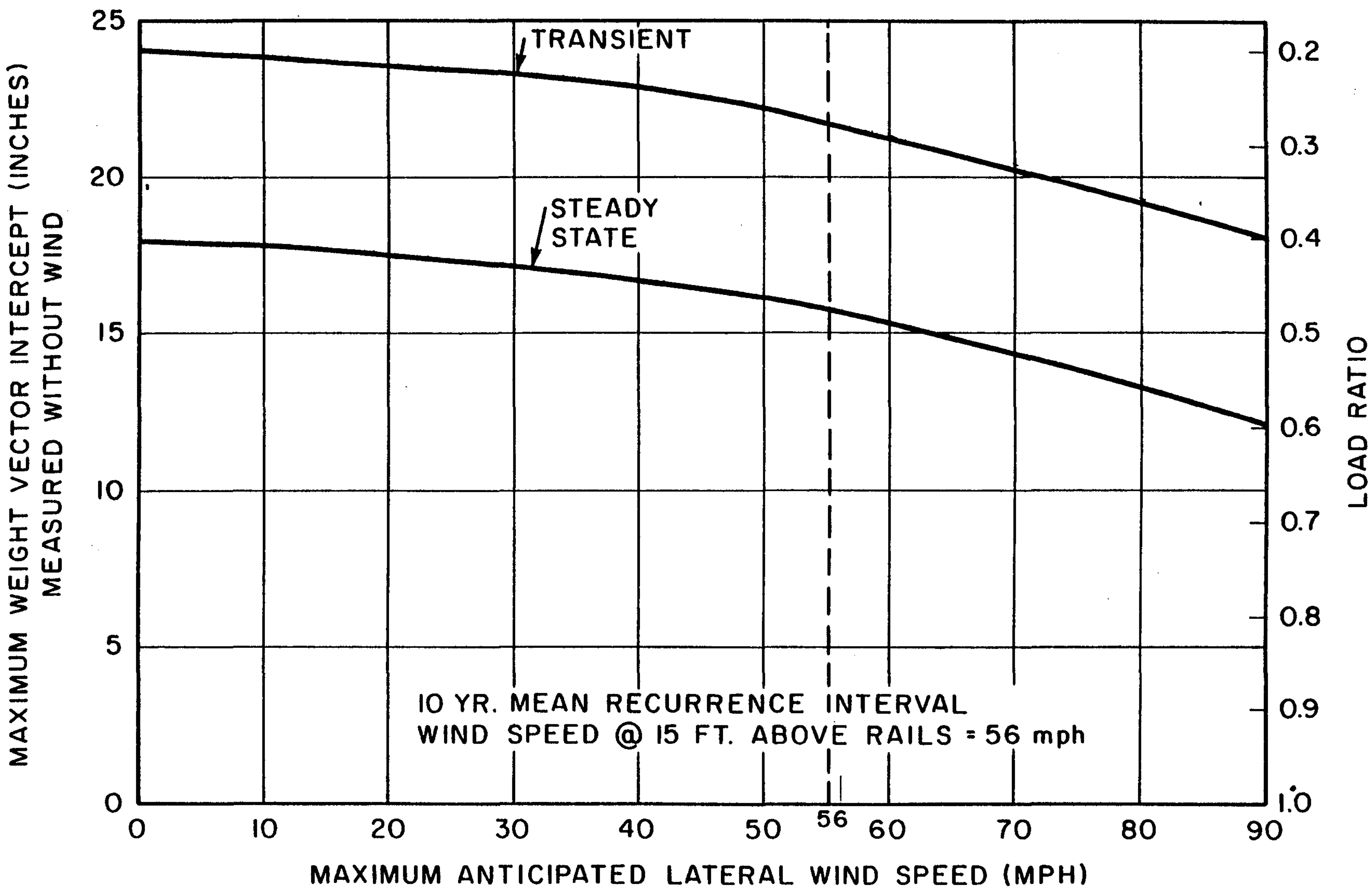


FIGURE 5-1

MAXIMUM SAFE WEIGHT VECTOR INTERCEPT MEASUREMENT (ZERO WIND) FOR THE F40PH LOCOMOTIVE AS A FUNCTION OF ANTICIPATED LATERAL WIND SPEED BASED ON JNR RECOMMENDED LOAD RATIOS where it is 70 mph measured 30 ft. above the ground (Ref 24). Reference (24) also provides a factor to adjust wind speed measurements for other distances above ground level. At 15 feet above the ground the 10 year mean recurrence interval wind speed is 0.8 x 70 mph = 56 mph. The cant deficiency safe for crosswinds up to 56 mph must be chosen to limit the weight vector intercept of the F40PH locomotive to 15.7 inches steady state and 21.7 inches transient when measured in still air as indicated in Figure 5-1.

5.1.2 STEADY STATE CURVING MEASUREMENTS

Figure 5-2 gives steady state test results which relate weight vector intercept to cant deficiency. The weight distribution on the instrumented truck was offset from the geometric centerline about 1-1/2 inches to the left. Consequently, the vector intercept was expected to be greater for right hand curving, and this was confirmed. The test results for the right hand curve represent the worst case and should be used for comparison to safety criteria. The worst case test data indicate that operation at up to about 9-1/2 inches of cant deficiency can be achieved without exceeding the steady state overturning safety criterion of 15.7 inches vector intercept for curving with up to 56 mph crosswinds.

5.1.3 TRANSIENT CURVING MEASUREMENTS

The most conservative way to use the vehicle overturning criteria is to compute the cant deficiency limits imposed by the steady state criterion applied to steady state measurements and by the transient criterion applied to transient measurements and choose the most restrictive. Since the steady state weight transfer of a given vehicle depends on cant deficiency and static load distribution and not on track geometry or suspension dynamics, the results of tests on a single pair of right and left curves may be used to determine the steady state performance at any curve. By considering the curve direction for which the vehicle has an unfavorable static weight distribution, a single cant deficiency

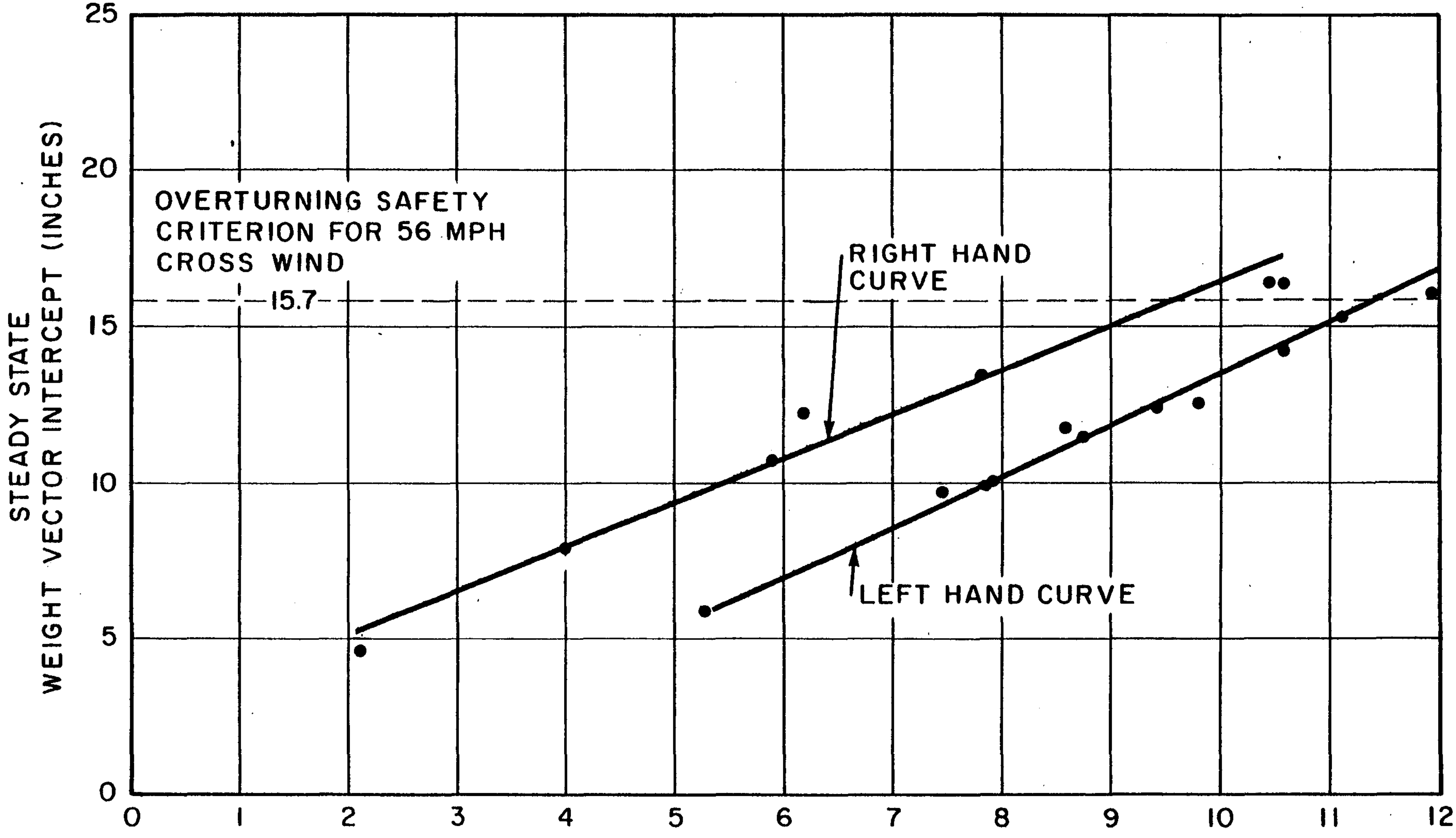


FIGURE 5-2

STEADY STATE MEASUREMENT OF F40PH LOCOMOTIVE OVERTURNING SAFETY limit may be determined which is safe for all curves when only steady state performance is at issue. This limit for the F40PH was 9-1/2 inches of cant definciuency as shown in Figure 5.2. The application of the vehicle overturning criteria is thus reduced to determining whether the transient criterion of 21.7 inches vector intercept (56 mph lateral wind assumed) will be exceeded at less than 9½ inches of cant deficiency.

A direct examination of the restrictiveness of the transient overturning criteria would require transversing each of the curves in the test zone at 9½ inches cant deficiency. Although most curves were tested at increased speed, the target of 9 inches cant deficiency was achieved at only a few because of speed restrictions, braking and accelerating distance requirements and top vehicle speed. A method of estimating the transient weight vector intercept at 9½ inches of cant deficiency from measurements at lower curving speed is required in order to use the measurements from all the curves in the test zone. As large a sample of test curves as possible is needed to determine which geometry features lead to transient weight transfer great enough to limit cant deficiency below the limit set by steady state weight transfer. Also since the test zone is on the primary route of the F40PH in NEC service, it would be desirable to know which specific curves now limited to low cant deficiency by braking or acceleration would become problems at higher speeds made possible by future improvements to the rest of the system.

The trend lines for the test data of transient and steady state vector intercept versus cant deficiency are plotted in Figure 5-3 for the same curve. The rates of change of the transient measurements and the steady state measurements with cant deficiency are nearly equal (i.e., the slope of the trend lines are similar). The measurements at the left curve (not plotted) also indicated similar slopes. Similarity of slopes between transient and steady state measurements may appear counterintuitive because it implies a constant difference between them, whereas the

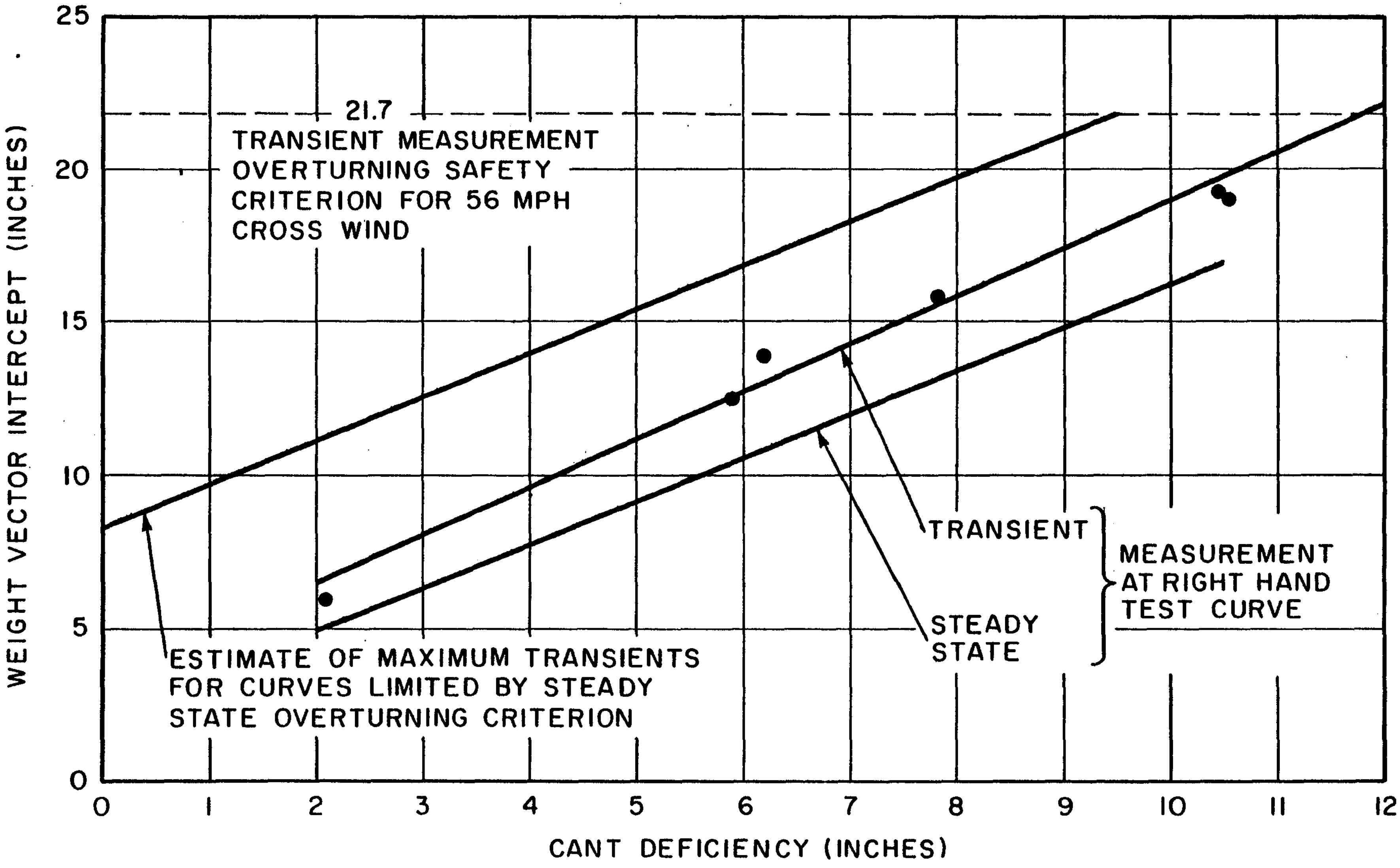


FIGURE 5-3

COMPARISON OF TRANSIENT WEIGHT VECTOR INTERCEPT MEASURED ON THE RIGHT HAND TEST CURVE TO THE ESTIMATED MAXIMUM TRANSIENTS FOR CURVES LIMITED BY STEADY STATE OVERTURNING CRITERION difference would be expected to increase with speed. However, the increase in cant deficiency from 5.9 inches to 10.4 inches, a 76% increase, resulted from an increase in speed of only 17% at the test curve of 2.6° curvature. Since the percent change in speed is much less than the percent change in cant deficiency, the known slope of steady state vector intercept versus cant deficiency can be used to estimate the unknown slope of the transient measurements. Using this estimate of the slope and a transient measurement at any cant deficiency, the transient vector intercept at the steady state cant deficiency limit (9½ inches) may be predicted. If the prediction exceeds the transient overturning criterion, then that criterion is the more restrictive.

At a right curve of maximum "roughness" for the use of the steady state overturning criterion, the transient vector intercept measurement of the F40PH locomotive would be 21.7 inches at 9½ inches cant deficiency. In Figure 5-3 a line having the slope of the steady state measurements (to estimate the slope of the transient measurements at the hypothetical curve) is drawn passing through the point (9.5 inches cant deficinecy, 21.7 inches vector). This line estimates the maximum transient vector intercept measurements of the F40PH as a function of cant deficiency for curves at which the steady state overturning criteria can be used.

Figures 5-4 through 5-7 present transient measurements of vector intercept of the F40PH at a large sample of right and left curves tested at up to 11 inches of cant deficiency. The line estimating the maximum transient measurements at a curve for which the steady state overturning criterion is the more restrictive is included in each figure. The transient overturning criteria is the more restrictive at any curve where the measurement falls above this line, and the cant deficiency limit set for this curve by the transient criteria is the intersection of a parallel line passing through the data point (representing the increase with

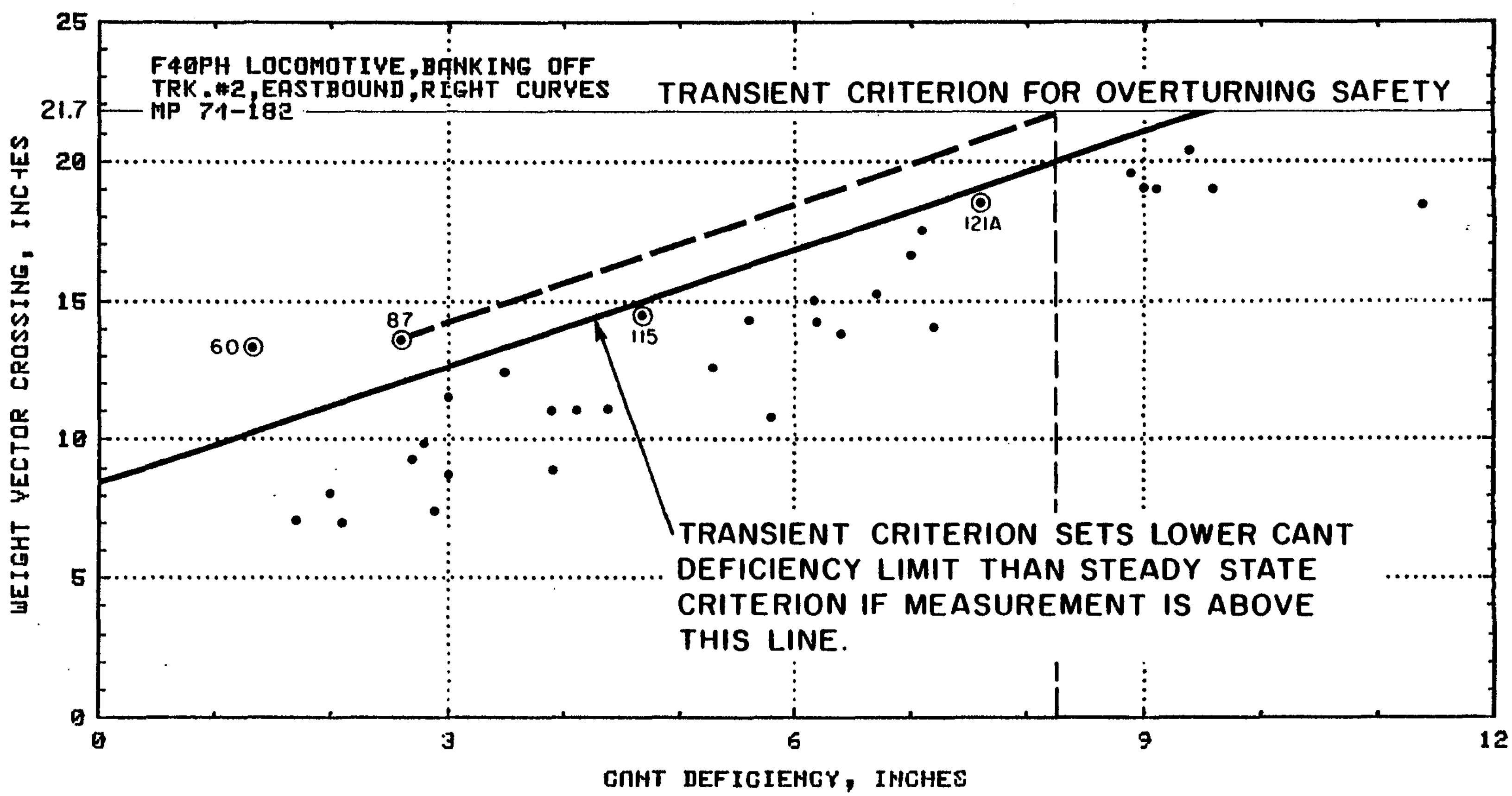


FIGURE 5-4

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT AT CURVES IN NEC TEST ZONE

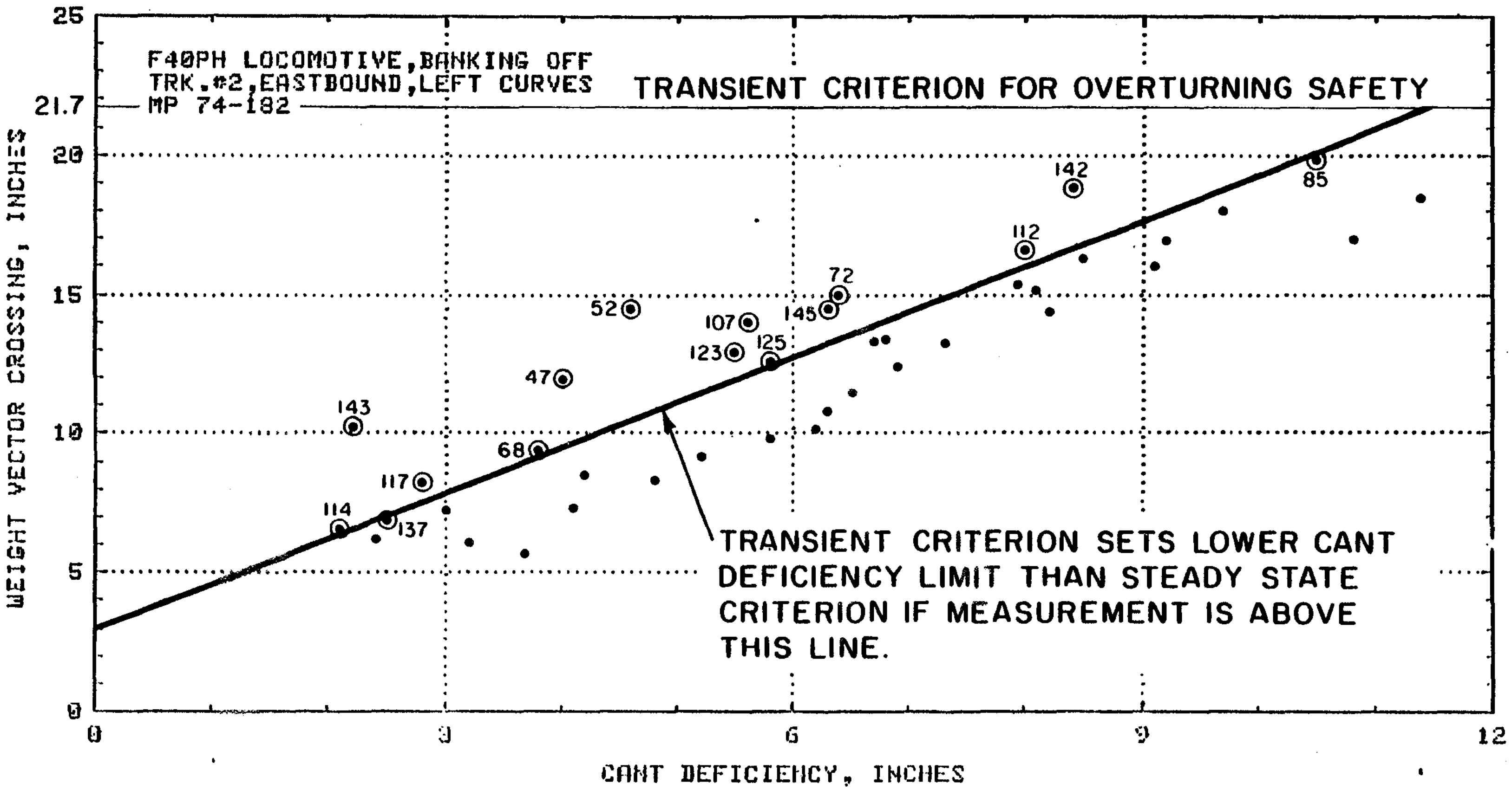


FIGURE 5-5

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT AT CURVES IN NEC TEST ZONE

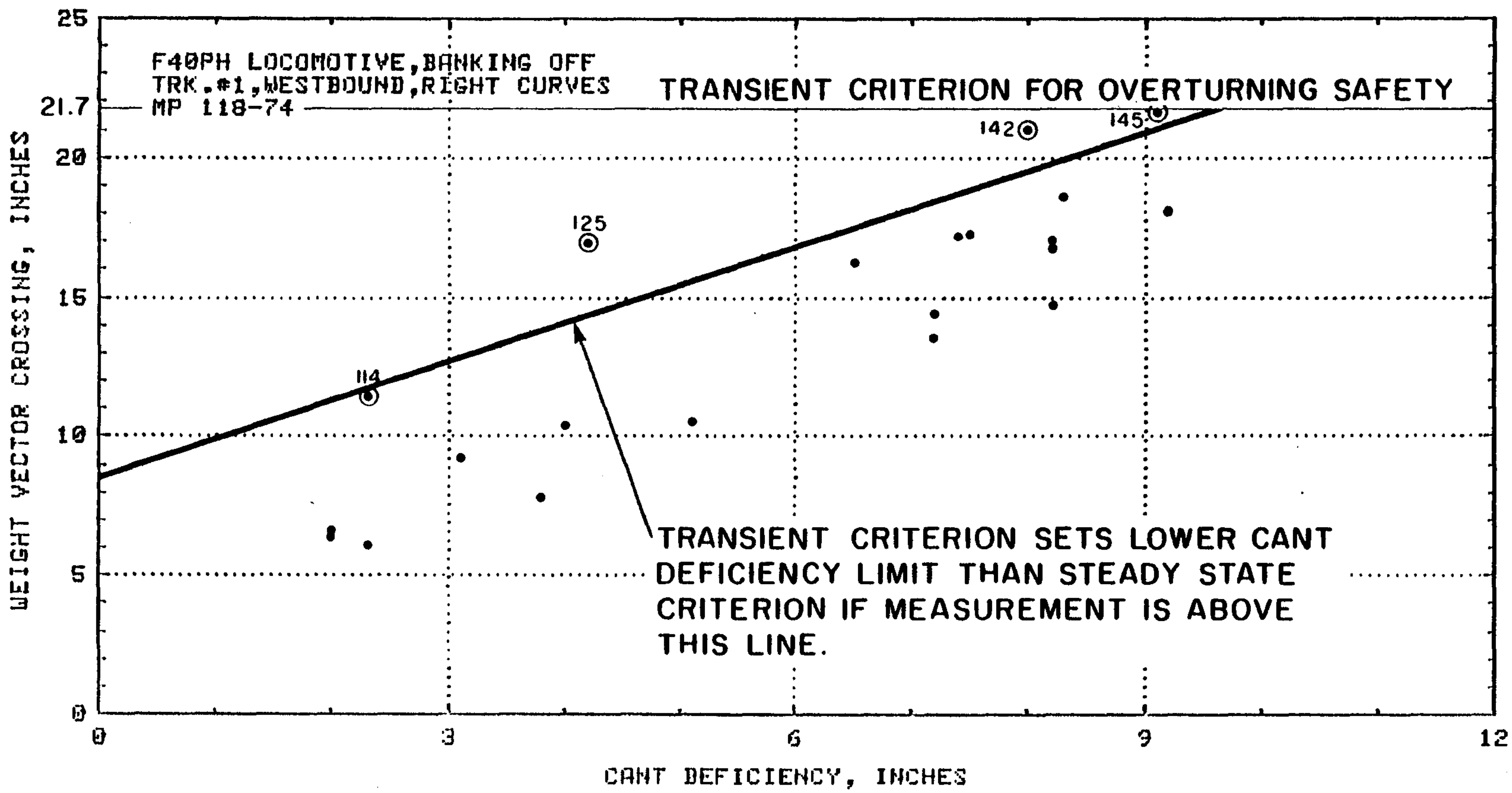


FIGURE 5-6

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT AT CURVES IN NEC TEST ZONE

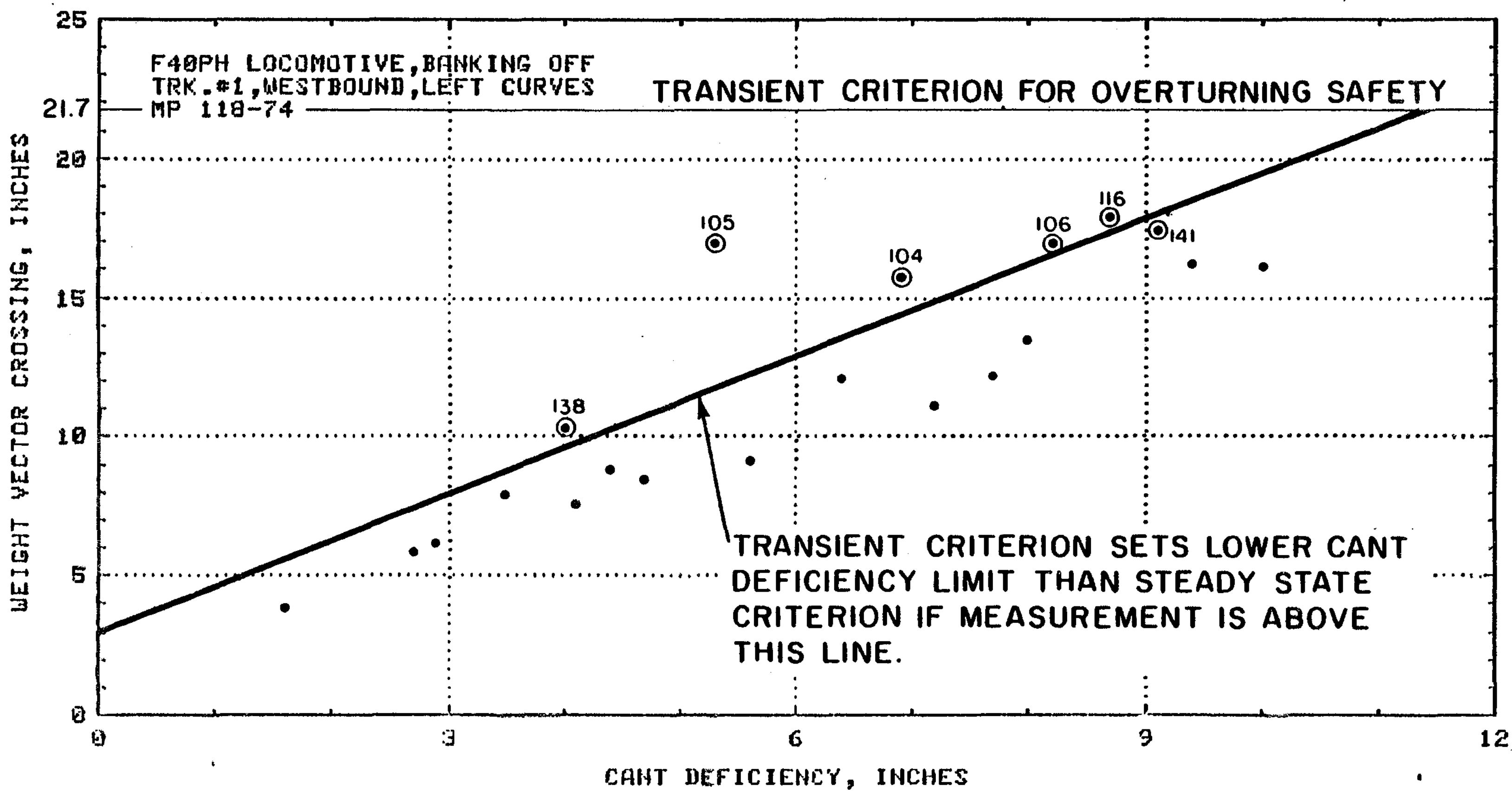


FIGURE 5-7

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT AT CURVES IN NEC TEST ZONE

higher cant deficiency) and the horizontal line at 21.7 inches of vector intercept. The cant deficiency limit of 8.3 inches at curve 87 track 2 is shown as an example in Figure 5-4.

A small static weight offset to the left over the instrumented truck of the test locomotive was detrimental to weight transfer during right curving and beneficial during left curving. The steady state overturning criterion limited cant deficiency to 9½ inches at right curves and 11½ inches at left curves. A general cant deficiency limit of 9½ inches was used as a consequence of steady state weight transfer because asymmetric static loading must be regarded as typical, but the presumption of its benefit in one direction should be avoided. The transient vector intercept measurements should also be analyzed in a way that removes the assumption of beneficial load asymmetry. The line separating curves limited by transient vector intercept from those limited by steady state overturning was drawn passing through the point (11.5 inches cant deficiency, 21.7 inches vector) for left curves (Figures 5-5 and 5-7). The relative restrictiveness of the steady state and transient overturning criteria for each particular left curve can be determined validly by this dividing line because it includes the same effect of static load bias that would occur in both steady state and transient measurements.

Table 5-1 lists the curves at which the transient measurement of vector intercept was high relative to cant deficiency (i.e., all circled points of Figures 5-4 to 5-7). The general cant deficiency limit of 9½ inches set by the steady state criterion is the more restrictive for the F40PH in regard to vehicle overturning safety at all curves not listed in the table. The tabulated curves were examined for the possibility that the cant deficiency limit indicated by the transient overturning criterion or the maximum class 6 track speed was less than 9½ inches. The measurement of peak weight vector intercept, the test speed and cant deficiency, the estimated maximum safe cant deficiency and

TABLE 5-1

CURVES AT WHICH THE TRANSIENT WEIGHT VECTOR INTERCEPT OF THE F40PH LOCOMOTIVE WAS HIGH IN RELATION TO CANT DEFICIENCY

Curve Number	Track Number	Peak Vector Intercept (in)	Speed (mph)	Cant Deficiency (in)	Estimated Maximum Cant Deficiency (in)	Limiting* Factor	Unusual* Features
Category I - Curves (w/o unusual features) limited by maximum track speed.							
143	2	10.2	85	2.2	6.8	110 mph	-
123	2	12.9	106	5.5	6.2	110 mph	-
Category II - Curves with unusual features.							
145	1	21.6	92	8.1	9.1	TOC	UGB
145	2	14.5	83	6.3	8.8	TOC	UBG
142	1	21.0	74	8.0	8.5	TOC	SC
142	2	18.8	81	8.4	8.2	TOC	SC
141	1	17.4	75	9.1	9.5	SSOC	UGB
138	1	10.3	99	4.0	5.6	110 mph	UGB
137	2	6.8	80	2.5	9.5	SSOC	UGB
125	1	16.9	93	4.2	7.6	TOC	SC
125	2	12.6	102.1	5.8	7.5	110 mph	SC
117	2	8.2	89	2.8	5.3	110 mph	SC
116	1	17.9	70	8.7	9.2	TOC	SC
115	2	14.5	82	4.7	9.5	SSOC	UGB
114	1	11.4	99	2.3	3.7	110 mph	UGB
114	2	6.5	92	2.1	4.6	110 mph	UGB
107	2	14.0	90	5.6	8.4	TOC	UGB
105	1	16.9	88	5.3	6.3	TOC	UGB
104	1	15.7	86	6.9	8.6	TOC	UGB
87	2	13.6	83	2.6	8.3	TOC	UGB
85	2	19.8	85	10.5	9.5	SSOC	SC
72	2	15.0	95	6.4	8.6	TOC	GC
68	2	9.4	86	3.8	8.5	110 mph	UGB
60	2	13.3	104	1.3	1.5	110 mph	SC
52	2	14.5	106	5.2	5.3	110 mph	GC
47	2	11.9	94	4.0	7.6	110 mph	SC
Category III - Curves (w/o unusual features) limited by vehicle overturning safety.							
121A	2	18.5	100	7.6	9.5	SSOC	-
106	1	16.9	97	8.2	9.3	TOC	-
112	2	16.6	97	8.0	9.1	TOC	-

*Legend:
- SC - Switch in Curves
- UGB - Undergrade Bridge
- GC - Grade Crossing
- TOC - Transient Vehicle Overturning Criterion
- SSOC - Steady State Vehicle Overturning Criterion

limiting factor, and the presence of unusual features are identified for each curve.

There are three categories of curves listed in Table 5-1. The curves in the <u>first</u> category was tested at a very high speed relative to a low cant deficiency because of their low curvature. Considerable side to side dynamic weight transfer occurs at moderate track geometry deviations because of the speed. However the 110 mph class 6 track speed limit prevents operation at cant deficiencies high enough to produce critical transient weight vector intercepts. Many of the curves in all categories may be limited in speed by braking and accelerating rates from stations and slow curves, but only the overturning criteria and the maximum class 6 track speed are considered for a worst case evaluation of the effect of track condition on overturning safety.

Curves of the <u>second</u> category have switches, grade crossings, or undergrade bridges as specific unusual features in their immediate vicinity. The discontinuity in track stiffness may promote perturbations and hinder maintenance at such sites. Many of these curves have curvature so slight that the cant deficiency remains low at 110 mph. The transient overturning safety criterion limits the safe operation of the F40PH on others. Curves 145, 142, 125, 116, 106, 105 and 104 westbound and curves 145, 142, 112, 107, 87, 86, and 72 eastbound cause dynamic weight transfer of the F40PH great enough to limit its safe speed below that imposed by the steady state overturning criterion or the maximum class 6 track speed. The overturning safety criterion based on transient measurements restricts curves 105 and 125 on track to 6.3 and 7.6 inches of cant deficiency, respectively. Other curves restricted by the transient criterion to 8.5 inches or less cant deficiency are curves 87, 107 and 142 on track 1 and curve 142 on track 2. Of these six curves, restricted much more severely by transient weight transfer than by steady state weight transfer, three have undergrade bridges and three have switches in the curve.

Curves of the <u>third</u> category are limited in cant deficiency by side to side weight transfer rather than by the maximum track speed (as in category I). Although the track chart does not identify unusual features, the transient measurements of vector intercept were high at these curves. Curves 112 track 2 and 106 track 1 would be limited to 9.1 and 9.3 inches of cant deficiency respectively by the transient overturning criteria while the general rule of 9.5 inches set by the steady state criteria remains the more conservative at curve 121A track 2. It is significant that transient weight transfer did not restrict cant deficiency on any curves without specific unusual features to less than 9.1 inches as indicated by the curving data in category III. The overturning safety criteria for the F40PH for comparison to both steady state and transient measurements are in harmony for curves without specific unusual features.

However, a number of curves in the test zone between New Haven and Providence have geometry perturbations which limit the safe cant deficiency of the F40PH by dynamic load transfer before the steady state limit is reached. Severe disturbances often occur in curves with undergrade bridges, switches and road crossings.

5.2 RAIL ROLLOVER

Rail rollover is related to transient truck side L/V ratios. The transient rather than steady-state measurements should be considered because this mode of derailment may be rapid and pulses of relatively low energy are capable of turning the rail whereas a great amount of energy is required to overturn a vehicle with its great interia.

A suitably conservative rail rollover criterion should assume zero pull out resistance of the fasteners. Only the geometry of the rail section and the torsional stiffness of the surrounding rail should be considered for a general safety criterion.

Appendix B describes a rail rollover criterion based on the above two factors. It can be expressed as maximum truck side L/V = 0.5 + $2300/P_w$, where P_w is the single wheel nominal vertical load. This criterion may underestimate the effect of rail section geometry and overestimate the effect of torsional stiffness of the surrounding rail but it appears to be based on the best available information. It should be interpreted as a restriction based on measurements having time duration of at least 50 milliseconds. Higher measurements are permitted for lesser time durations.

For the F40PH Locomotive, $P_w \approx$ 32,000 lb, and a limiting value of 0.57 should be compared to the transient measurements of truck side L/V ratio. Figure 5-8 compares the measurements taken at the right and left test curves to the criterion. At these relatively smooth curves, the measurements were far below the critical level. Truck L/V ratio measured at the high rail side of the F40PH locomotive did not increase rapidly with cant deficiency because of the moderating influence of the vertical load transfer, and the performance was virtually identical for right and left curving.

A few higher measurements occured in the NEC test zone between New Haven and Providence. The highest measurement was 0.35 at 9.7 inches of cant deficiency on curve 109 track 2. The next highest measurement of 0.34 was recorded at curve 88 track 2 at 8.1 inches cant deficiency. Since even the highest measurements from a test zone of 112 curves were well below the limiting value, the rail rollover criterion does not limit the operational cant deficiency of the F40PH locomotive.

5.3 LATERAL TRACK SHIFT

The time duration of the force required to move the track structure laterally is much greater than that required for rail rotation, and the use of transient forces to evaluate safety is quite conservative. Using a single wheel force rather than axle force introduces another estimate on the conservative side because the net axle force which moves the track is usually less than a

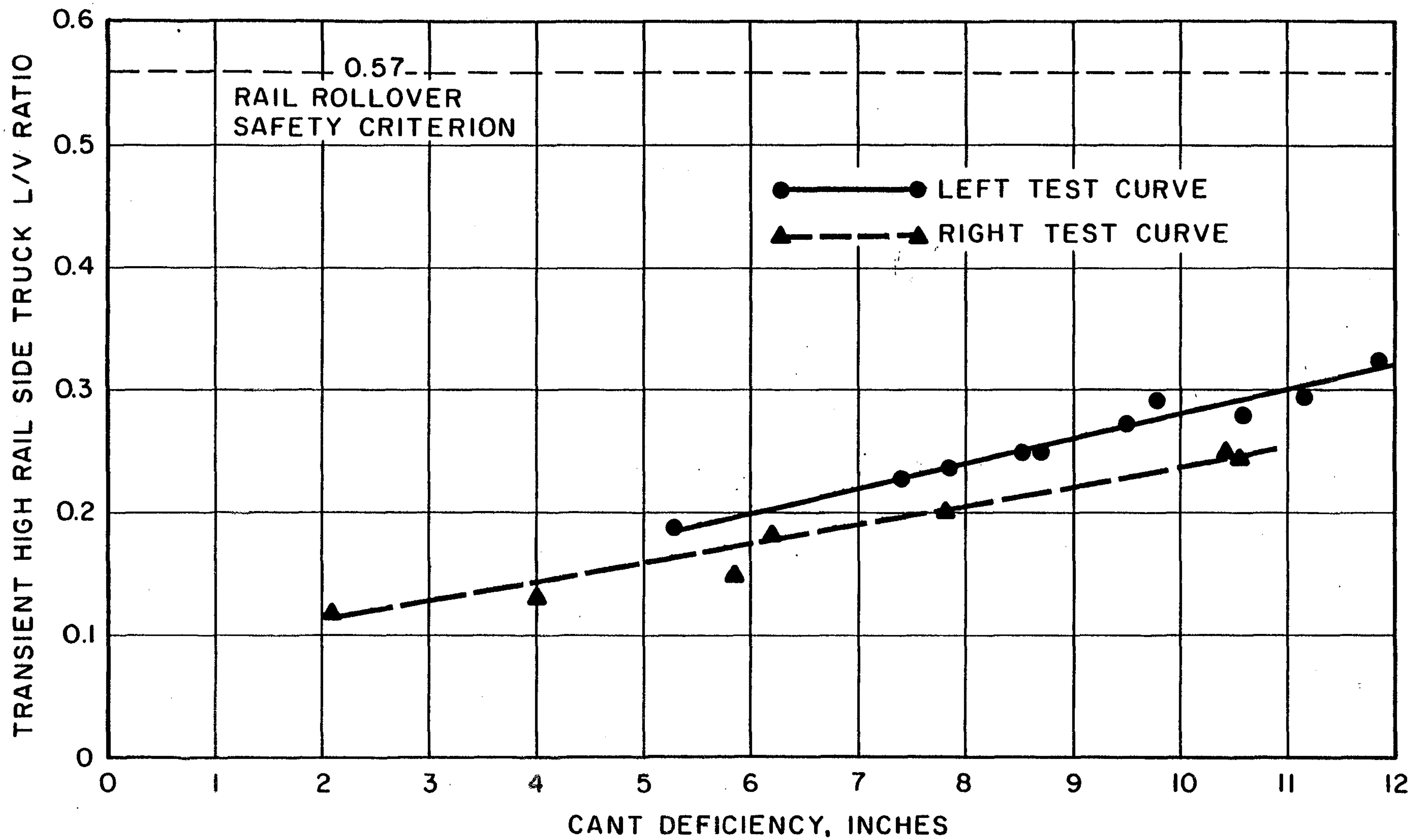


FIGURE 5-8

TRANSIENT TRUCK L/V RATIO OF THE F40PH LOCOMOTIVE AS A FUNCTION OF CANT DEFICIENCY MEASURED AT RIGHT & LEFT TEST CURVES single wheel peak force as the wheel forces on the same axle tend to oppose one another.

The safety criterion discussed in Appendix B assumes compacted ballast for full operational cant deficiency, and it allows for the force of crosswinds encountered in service. For the conservative assumption of wood ties, the criterion can be expressed as follows:

$$F_{max} = \left(1 - \frac{A\Delta\theta}{22320}(1 + .458D)\right)(.7P + 6600) - (1.28 \times 10^{-3} SV^2)$$

A = rail cross section area, in^2

$\Delta\theta$ = max temperature change after rail installation, ^{o}F

D = track curvature, degrees

P = vertical axle load, lbs.

S = lateral surface area of vehicle, ft^2

V = lateral wind speed, mph

and it is assumed that a single axle bears half the entire vehicle wind load. For typical NEC conditions of 140 lb rail (A = 13.8 in^2), $\Delta\theta$ max of 70^{o}F and D max of 4^{o}.

$$F_{max} = .61P + 5800 - (1.28 \times 10^{-3})\ SV^2$$

For the F40PH locomotive axle load of 64,000 lb, body side area of 730 ft^2, and an allowance for 56 mph crosswinds the maximum permissable lateral axle force is 41,900 lb. The maximum truck lateral force for a two axle truck should be limited to only 1.4 times the single axle maximum lateral force since fewer than twice the number of ties support the lateral force. The lateral track shift criterion permits a maximum truck lateral force of track shift criterion permits a maximum truck lateral force of 59,800 lb allowing for one half of the 56 mph crosswind vehicle side load at each truck.

Figure 5-9 compares the high rail lead wheel and truck side transient lateral force measurements on the right and left test curves to the rail rollover criterion. Both measurements remain at about half the critical levels at 10 inches of cant deficiency on the relatively smooth test curves.

The transient measurements of truck side lateral force taken on the NEC test zone, including many rough curves, were also well below the safety criterion. The highest transient measurement of truck side lateral force was 37,300 on curve 109 track 2 at 9.7 inches cant deficiency. Only two measurments of transient truck lateral force greater than 30 kips (about ½ the critical level) were made at less than 8 inches cant deficiency. The greatest truck force relative to cant deficiency was 34,100 kips at 6.9 inches at curve 104 track 1. Even at this curve no more than 45 kips would be expected at 9.5 inches cant deficiency. Since all measurements of truck lateral force were well below the critical level of 59.8 kips, the lateral track shift safety criterion does not limit the safe cant deficiency of the F40PH locomotive.

5.4 WHEEL CLIMB

The most appropriate criterion of safety concerning wheel climb for comparison to the test results of the F40PH locomotive is that used by Amtrak in its acceptance specification for the AEM-7. This criterion takes into account the flange angle specified by the AAR wheel and axle manual, and it appears to have been based on conservative judgements. It states that the wheel (L/V) ratio must be less than $0.056/T^{-0.927}$ where T is the duration of the transient in seconds and that the maximum wheel (L/V) ratio for transients of duration greater than 50 milliseconds is 0.90.

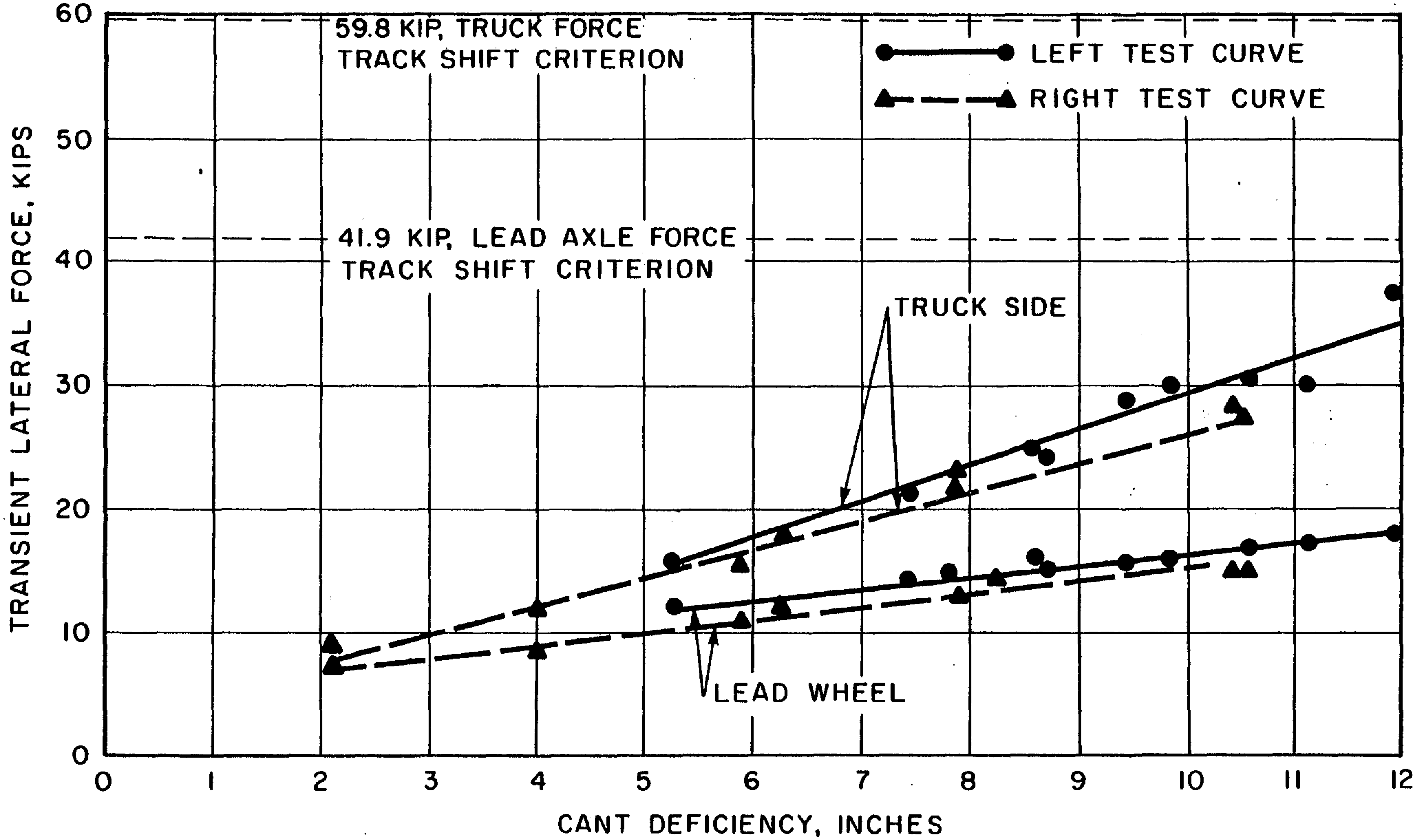


FIGURE 5-9

COMPARISON OF TRANSIENT LATERAL FORCE MEASUREMENTS OF LEAD WHEEL AND TRUCK SIDE OF THE F40PH LOCOMOTIVE TO TRACK SHIFT CRITERIA

Figure 5-10 shows that the wheel L/V ratio of the F40PH is very low with respect to the wheel climb safety criterion on unperturbed curves. It also shows that wheel L/V ratio is less sensitive than truck L/V to increases in cant deficiency indicating that the rear wheel bears a greater portion of the high rail lateral force as cant deficiency increases. The increasing trailing axle force implies flange contact at the trailing wheel and a reduction of the angle of attack of the leading wheel. Safety against wheel climb is increased by a reduction in the angle of attack.

The curves in the test zone between New Haven and Providence which produced the greatest truck side L/V ratios were curves 178 and 88A eastbound traversed at 11.4 and 8.1 inches of cant deficiency respectively. The transient wheel L/V ratio at both curves was 0.43, less than half the critical level. The weight transfer of the high c.g. locomotive was beneficial in limiting the wheel L/V ratio, and the highest values were measured at left curves in which vector intercept was reduced by static weight offset.

The wheel climb safety criterion does not limit the curving speed of the F40PH locomotive because the measurements of wheel L/V remain well below the critical level even at the harshest curves at cant deficiencies permitted by the vehicle overturning safety criteria.

5.5 RIDE COMFORT

Appendix B includes a variety of criteria for lateral acceleration used by organizations throughout the world as indices for ride comfort evaluation. The most commonly recognized standards in the U.S. are the AAR recommendations of 0.1g maximum steady state lateral acceleration and 0.03g/sec maximum time rate of change of acceleration (jerk). The least restrictive standards are those of SNCF which permit 0.15g and 0.10g/sec respectively.

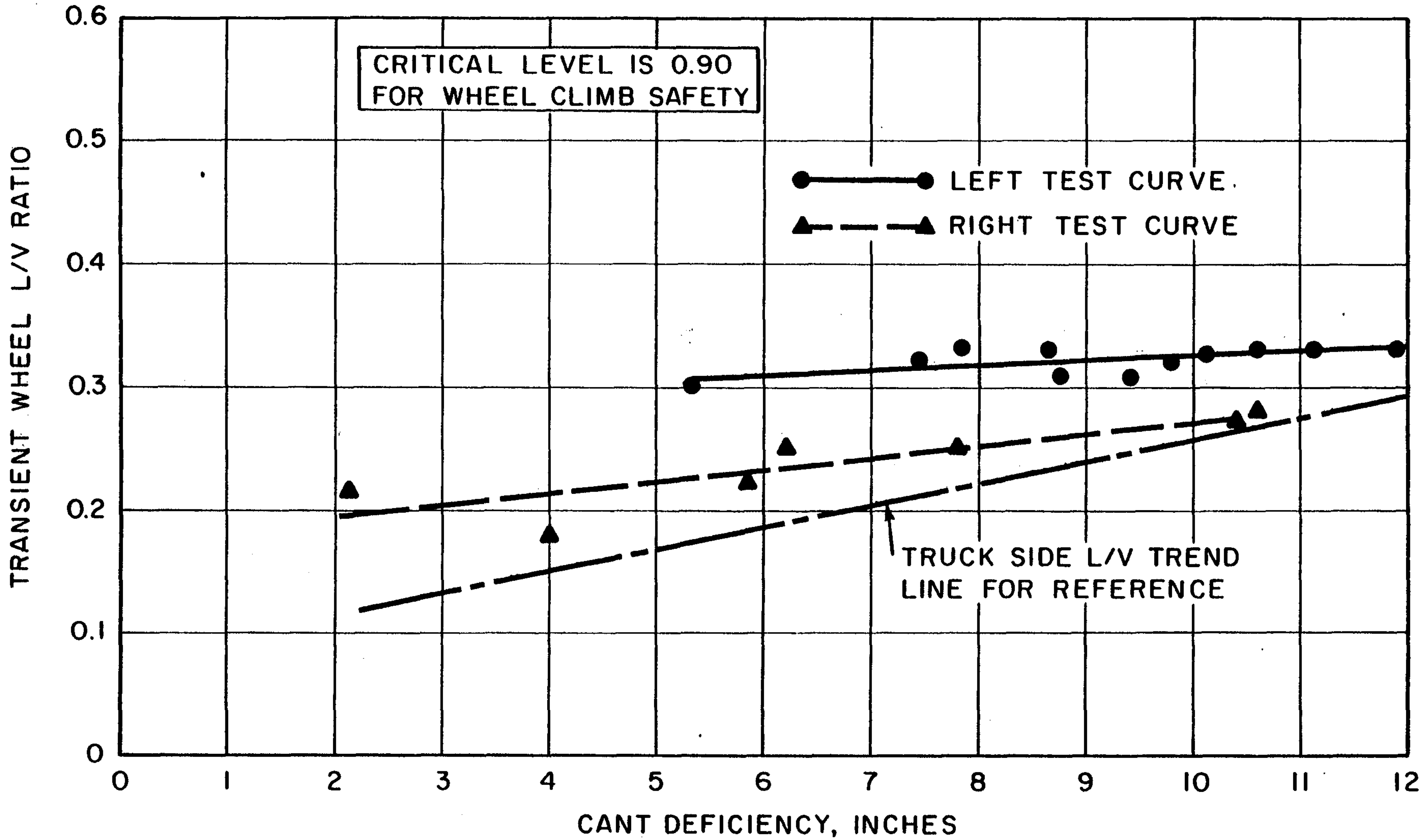


FIGURE 5-10

TRANSIENT WHEEL L/V RATIO OF THE F40PH LOCOMOTIVE AS A FUNCTION OF CANT DEFICIENCY, MEASURED AT RIGHT AND LEFT TEST CURVES

When the AAR study was undertaken in the early 1950's, coach suspensions were designed such that large body roll angles occurred in curves. The component of gravitational acceleration in the plane of the coach floor added substantially to the centrifugal acceleration as the coach body rolled toward the outside of the curve. At three inches of cant deficiency, the body roll of contemporary coaches was usually sufficient to cause a total of 0.1g lateral acceleration in the plane of the floor.

Figure 5-11 shows that the F40PH locomotive suspension controls steady state body roll well. The AAR coach steady state ride comfort criterion is reached at 4 inches cant deficiency and the SNCF criterion at 6-3/4 at the right hand test curve. At 6 inches cant deficiency the steady state lateral acceleration in the F40PH locomotive cab of less than 0.15g would not be considered harsh even by passenger coach standards. The appreciable difference in lateral acceleration between left and right curving appears to be the result of a systematic error in calibration. It is possible that the site of the daily calibration had cross-level at the locomotive cab that was not apparent at the instrumentation coach. The systematic error of about 0.025g should be compensated for if a more detailed analysis of locomotive ride quality is undertaken.

5.6 EFFECT OF COACH BANKING ACTION

The F40PH locomotive was coupled to the prototype banking Amcoach during these tests. It was hypothesized that the banking action of the coach could impart a restoring moment to the locomotive body through the coupler that would reduce its lateral weight transfer and increase its L/V ratios. Consequently, only the tests in which the coach banking system was <u>not</u> operated were used in the evaluation of the maximum safety cant deficiency of the F40PH. However, the results of repetitive tests at right and left curves with and without coach banking (Appendix A, Volume I) show that the coach banking action did not influence the locomotive performance.

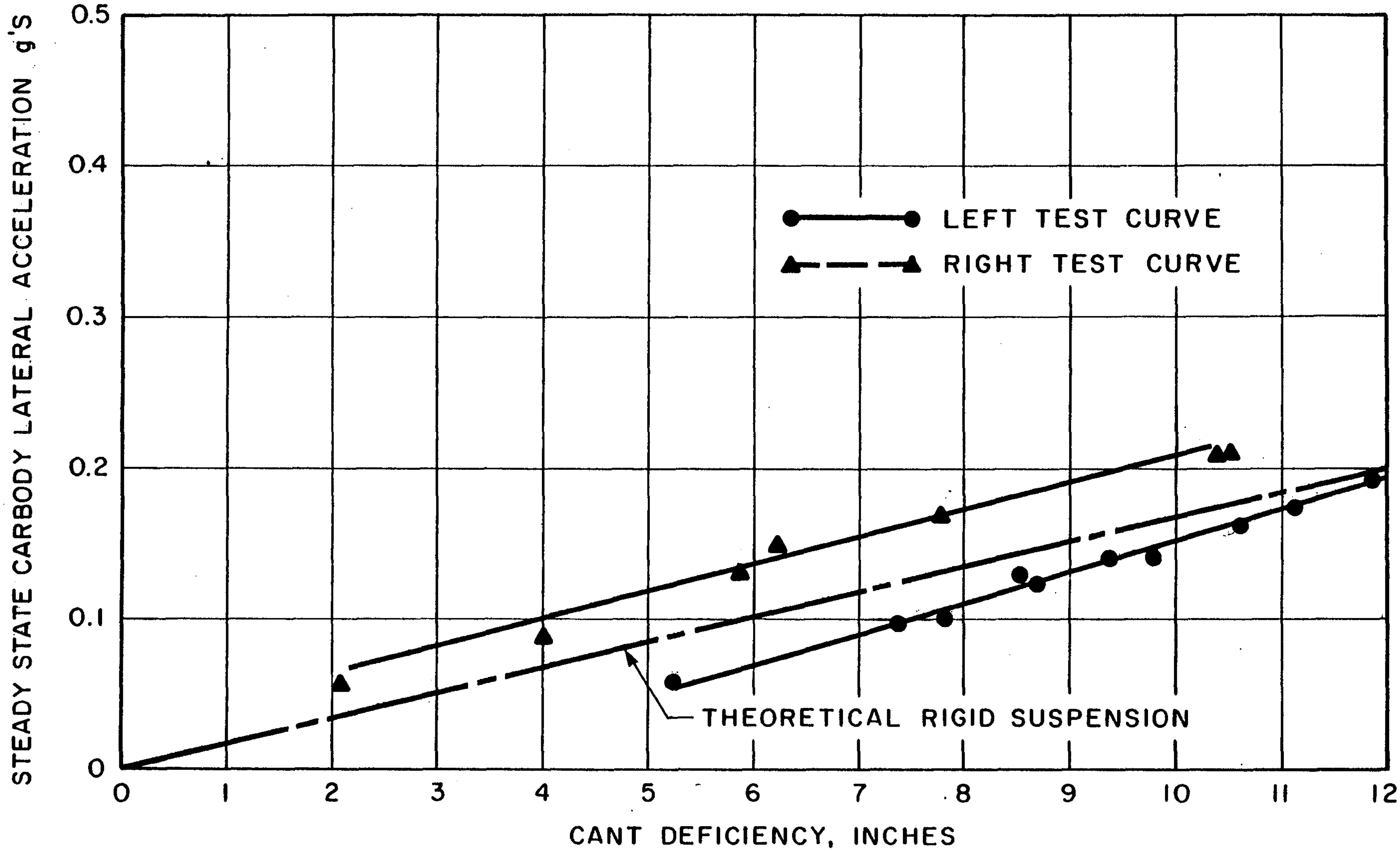


FIGURE 5-11

STEADY STATE LATERAL ACCELERATION OF THE F40PH LOCOMOTIVE
COMPARED TO A VEHICLE WITH ZERO ROLL ANGLE

5.7 <u>MAXIMUM OPERATIONAL CANT DEFICIENCY</u>

The operational cant deficiency of the F40PH locomotive is first limited by the vehicle overturning safety criteria. The critical level dictated by the steady state overturning safety criterion is reached at 9½ inches of cant deficiency allowing for a simultaneous lateral wind of 56 mph. The measurements indicating safety against rail rollover, lateral track shift and wheel climb are well below critical levels at 9½ inches of cant deficiency. However, the safety criterion against vehicle overturning based on transient measurements sets a lower limit of cant deficiency at some perturbed curves in the New Haven-Providence test zone. Most curves that cause high transient load transfer in the F40PH include an unusual feature such as a switch or undergrade bridge. Curve 105 track 1 was the most objectionable, and the safety limit probably would be exceeded at 6.3 inches of cant deficiency. Curves without special track work were not limited below 9 inches of cant deficiency by the transient vehicle overturning safety criterion, and a general cant deficiency limit of 9 inches is recommended for the F40PH locomotive.

It is not reasonable to limit the F40PH locomotive by the worst perturbation in the test zone. Steady state overturning safety provided the first limit at over 9 inches of cant deficiency for "normal" curves. The abnormal curves, which may include others not measured in this test should be identified individually. Many may be limited by factors such as acceleration and braking distances or proximity to stations. The remainder should be either repaired if practical or given special speed restrictions. The measurements indicate safe operation at up to eight inches of cant deficiency at all but a few curves. The identification and repair of problem curves could increase the safe cant deficiency to over 9 inches, but the coaches may well limit the <u>train</u> cant deficiency to a lower level due to the more restrictive wind force factor on vehicles of lighter weight and greater surface area.

6.0 RESULTS OF TESTING THE PROTOTYPE BANKING AMCOACH AT HIGH CANT DEFICIENCY

The Banking Amcoach was coupled to the F40PH Locomotive and was given the same test of steady state and transient performance. The same criteria, summarized in Section 4.0 and detailed in Appendix B, were used to evaluate its maximum safe cant deficiency. Steady state performance, which can be expressed as a function of cant deficiency independent of track geometry perturbations, was measured over a range of cant deficiency at right and left test curves. Transient measurements, which include the effects of vehicle dynamics and track perturbations superimposed on the steady state component, were taken at over one hundred curves on the Northeast Corridor between New Haven and Boston.

6.1 VEHICLE OVERTURNING

The JNR overturning criteria limit side to side weight transfer such that the unloaded wheels retain at least 40% of the nominal static load under steady state conditions and 20% under adverse transients, including the effect of lateral wind forces. Since instrumented wheels were used, both steady state and transient measurements were available. The more restrictive of the dual criteria will be applied at each curve, as in the locomotive evaluation in Section 5.0, for a conservative interpretation.

Weight vector intercept is the common indicator of vehicle overturning in American railroad literature although it is an awkward term to describe load transfer. The JNR overturning criteria may be stated in terms of vector intercept as follows:

Steady State
Vector Intercept $\leq 18 - (.0153V^2SH_{cp}/W)$ inches

and

Transient Vector Intercept $\leq 24 - (.0153V^2Sh_{cp}/W)$ inches

where:

V is the anticipated lateral wind speed in mph

S is the lateral surface area of the vehicle in ft^2

h_{cp} is the height of the center of wind pressure in ft

W is one half of the unloaded weight of the vehicle in pounds

For the Amcoach:

$S \simeq 765\ ft^2$

$h_{cp} \simeq 7.5$ ft

$W = 52{,}200$ lb

The overturning criteria are plotted in Figure 6-1. The steady state criterion is 12.8 inches vector intercept and the transient criterion is 18.8 inches for the anticipated lateral wind speed of 56 mph (corresponding to a 10 year mean recurrance interval on the NEC @ 15 ft above ground). The numerical values of the criteria given as functions of anticipated crosswinds in Figure 6-1 reflect the potential additional forces of crosswinds. Test measurements without crosswinds can be compared to the safety criteria because the criteria contains the allowance for the maximum potential crosswind force.

Comparison of Figures 6-1 and 5-1 shows that the overturning criteria of the coach diminish more rapidly with wind speed allowance than those of the locomotive. The weight transfer caused by lateral wind is much greater relative to static wheel load for the coach because its weight is less than half of that of the lcomotive while the moment of its side area about the rail is nearly equal. Typically the safe cant deficiency of a train

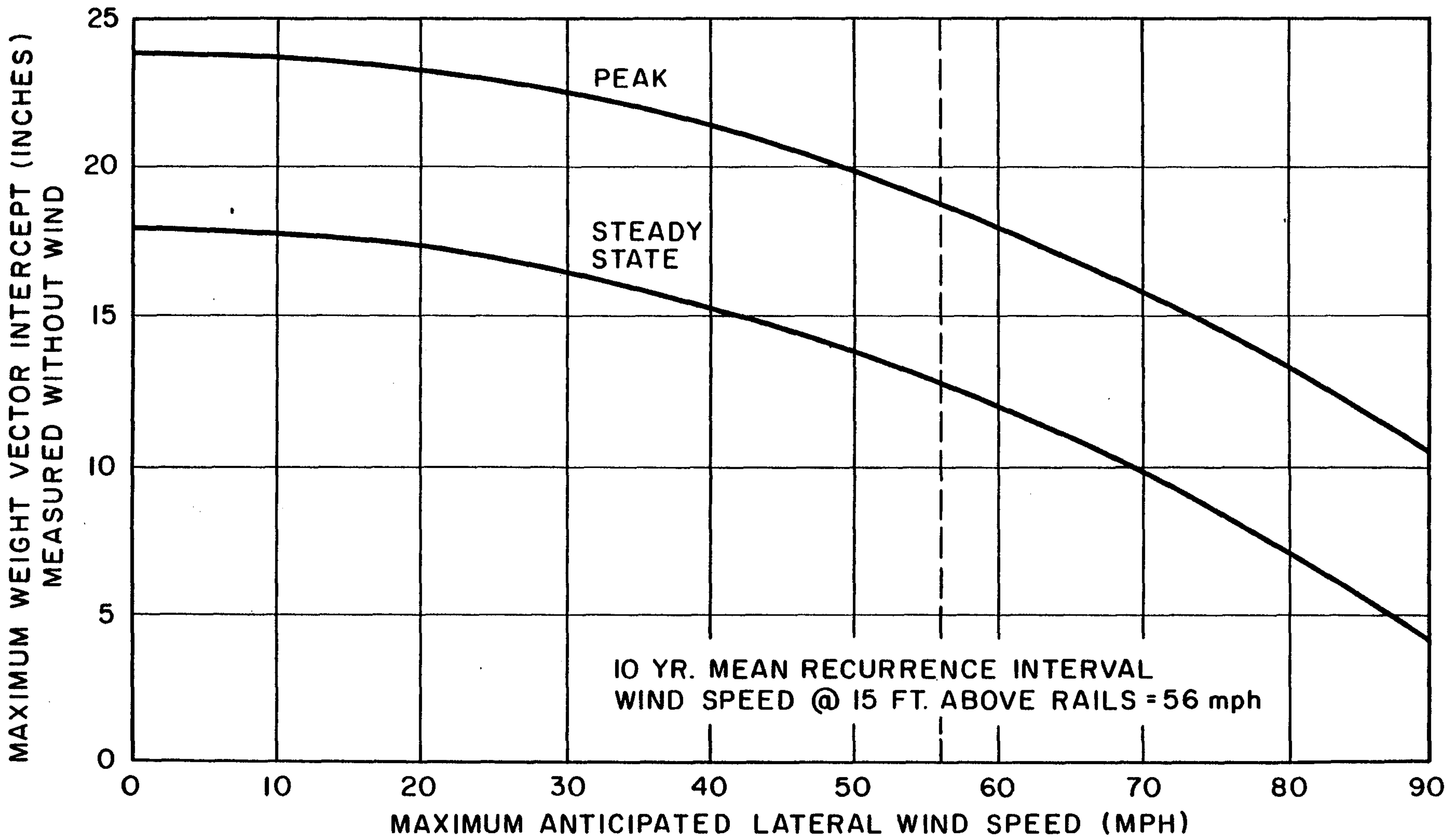


FIGURE 6-1

MAXIMUM SAFE WEIGHT VECTOR INTERCEPT MEASUREMENT (ZERO WIND) FOR THE PROTOTYPE BANKING AMCOACH AS A FUNCTION OF ANTICIPATED LATERAL WIND SPEED BASED ON JNR VEHICLE OVERTURNING SAFETY CRITERIA is limited by overturning safety of the coach rather than the locomotive because the high weight transfer of the coach in crosswinds is a greater disadvantage than the high center of gravity of the locomotive.

6.1.1 STEADY STATE LOAD TRANSFER MEASUREMENTS

The tilting action of the prototype banking Amcoach had a great effect on lateral weight transfer. Its banking is accomplished by an air actuated torsion bar device which overpowers the secondary suspension air springs. A solenoid valve acting on the tilt command signal allows the depressed bag to vent to the extended bag. Sufficient travel for 4° of body roll is gained by retracting the air bag stops during banking. During high speed curving the body rotates toward the low rail about a point at the secondary air springs, well below the body center of gravity. Hence the body c.g. moves laterally toward the low rail reducing considerably the weight transfer to the high rail. Figure 6-2 indicates steady state weight transfer as a function of cant deficiency of the Amcoach with and without banking action. At any cant deficiency the weight vector intercept is reduced appreciably by the banking system operation as a consequence of body c.g. movement toward the low rail. When the coach was operated as an ordinary car in the non-banking mode, the steady state overturning criterion of 12.8 inches vector intercept was reached at approximately 9 1/3 inches cant deficiency at both right and left test curves. In the banking mode, the vector intercept was generally lower by about 4 1/2 inches at the left curve and about 3 inches at the right curve. The slight asymetry in load transfer at the front truck between right and left curves during banking is probably the result of a slight difference in banking angle between front and rear trucks. If the banking angle differs slightly between front and rear, the body acts as a great torsional spring and alters the side to side load distribution at each truck. The asymetry of load distribution of the banking Amcoach was modest, and the load transfer during curving was decreased by banking operation under virtually all test conditions.

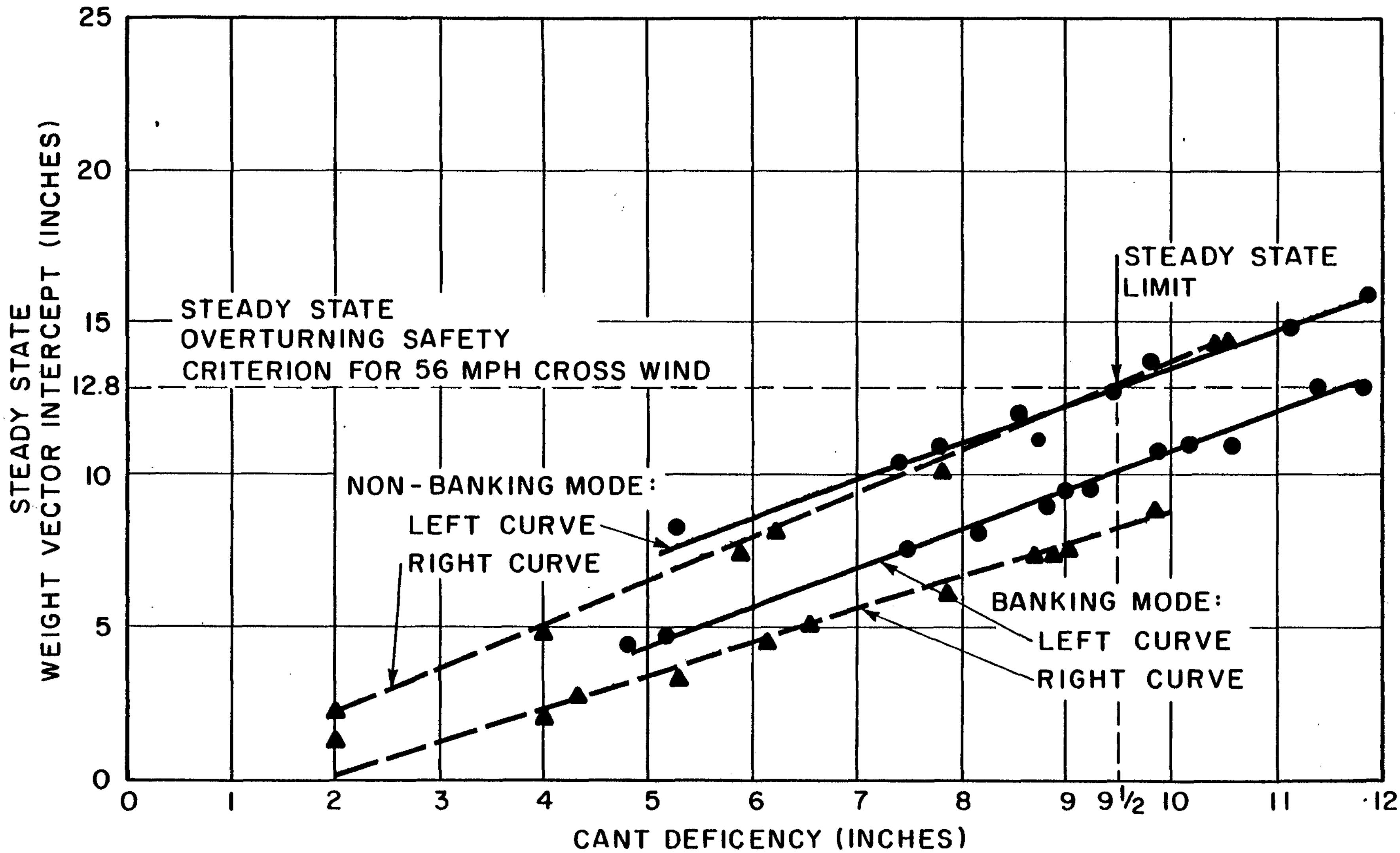


FIGURE 6-2

STEADY STATE MEASUREMENT OF OVERTURNING SAFETY OF PROTOTYPE BANKING AMCOACH WITH AND WITHOUT BANKING ACTION

Although the banking system was beneficial with regard to steady state weight transfer, its operation should not be assumed in the determination of the maximum cant deficiency safe for passenger service. Banking has been proposed only to improve passenger comfort, and its state of development is not advanced enough for dependable benefits in controlling weight transfer. The more advanced Canadian LRC coach rotates its body about a point near the c.g. specifically to avoid changes in load transfer during banking. The modification of the Amcoach for banking was deemed successful with respect to safety because its weight transfer during normal operation, was not greater than would be expected for an unmodified Amcoach.

Assuming the proper fail-safe features, the worst case weight transfer of the prototype banking Amcoach occurs in the nonbanking mode, and safety against overturning should be evaluated using the non-banking test results. Non-banking measurements at both the right and left test curves reached the steady state overturning criterion at approximately 9 1/3 inches cant deficiency. Equal weight transfer during left and right curving is unusual. Greater weight transfer in one direction caused by a small static imbalance of wheel load distribution, such as that measured for the locomotive in Section 5.0, is typical. The cant deficiency, limited by overturning safety, is dictated by the unfavorable curving direction rather than the average weight transfer. The banking coach experienced continual problems with air spring deflation and many test runs were performed with the body listing badly to one side. The near perfect weight transfer symmetry at the cant deficiency limit is merely a chance occurrance. A standard Amcoach, equipped with the same instrumented wheels, was tested as the baseline vehicle during the high cant deficiency test of the LRC train at the same curves (ref. 23). Figure 6-3 compares the weight transfer of the banking Amcoach in the non-banking mode to that of the standard Amcoach. The standard Amcoach exhibits the usual static load offset (the rear

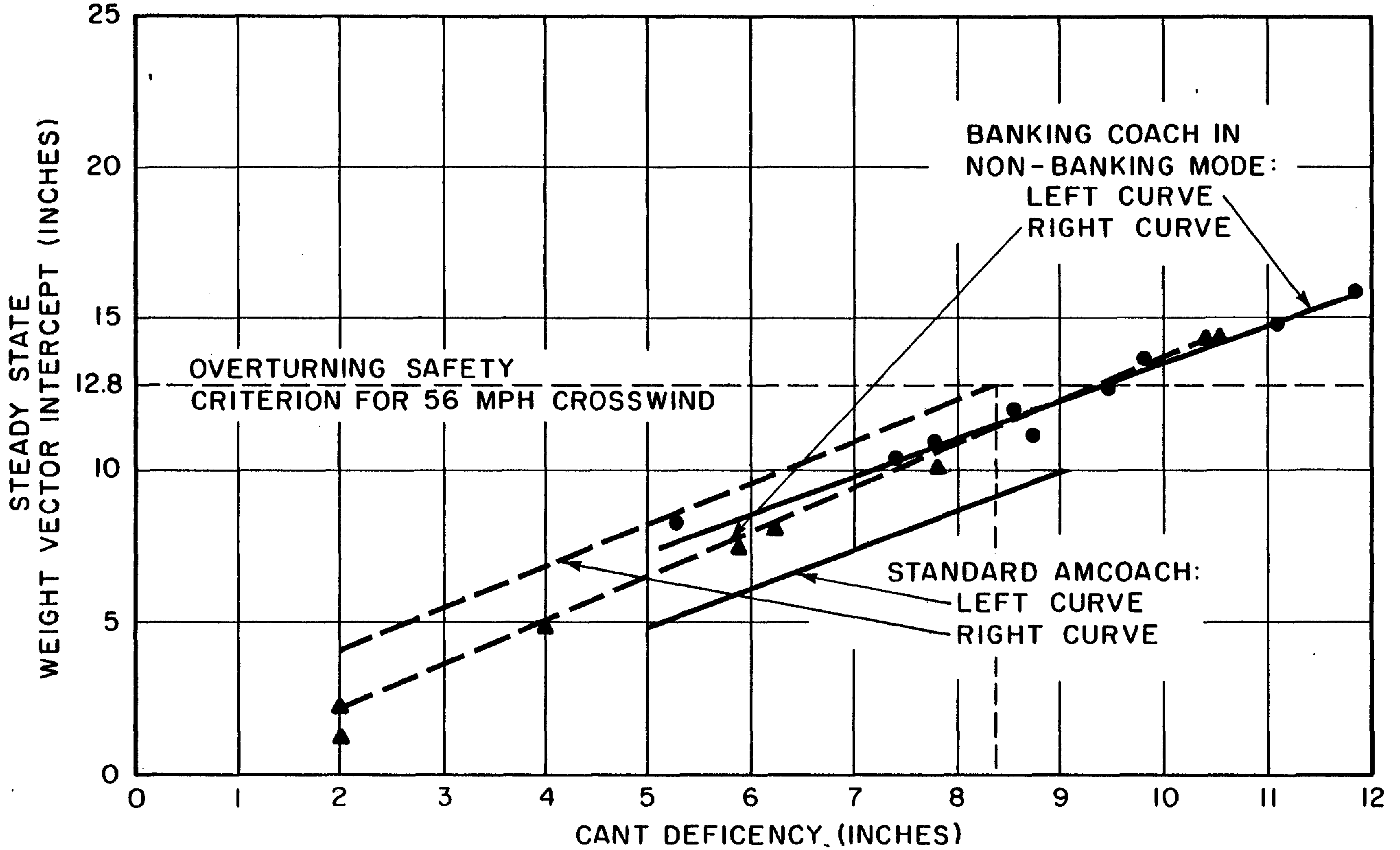


FIGURE 6-3

COMPARISON OF STEADY STATE WEIGHT TRANSFER BETWEEN THE STANDARD AMCOACH (TESTED 1980) AND THE BANKING AMCOACH IN THE NON-BANKING MODE.

truck may have an opposite offset), and the steady state overturning safety criterion sets the cant deficiency limit at 8 1/3 inches in the unfavorable curving direction. The average vector intercept of right and left curves at a given cant deficiency is nearly equal for the standard Amcoach and the banking Amcoach in the non-banking mode. The softer elastomeric elements in the primary suspension and reduced secondary roll stiffness of the three convolute air spring of the banking Amcoach did not change its steady state load transfer characteristic significantly from that of the standard Amcoach. Therefore, a cant deficiency limit of 8 1/3 inches is valid for the banking Amcoach as well as standard Amcoach with regard to the steady state overturning criterion, allowing for 56 mph lateral winds and a typical static imbalance of wheel loads.

6.1.2 TRANSIENT LOAD TRANSFER MEASUREMENTS

As discussed in Section 5.1.3, the cant deficiency limit for each curve with regard to safety against vehicle overturning is taken from the more restrictive of the steady state or transient overturning criteria. The steady state limit of 8 1/3 inches cant deficiency is valid for any curve, but the transient limit is different at each curve because it depends on irregularities. A line shown in Figure 6-4 may be constructed by the method discussed in 5.1.3 to identify curves for which the transient criterion is the more restrictive, on the basis of a transient measurement of any cant deficiency. Figure 6-4 gives the transient measurements of weight vector intercept at the left test curve where runs were performed at many speeds. The steady state criterion is the more restrictive because its limit was reached at 9 1/3 inches cant deficiency while the transient limit was not reached below 11 inches cant deficiency. Had only the measurement at 5.3 inches cant deficiency been available, comparison of the single measurement to the line estimating the maximum transients for curves limited by the steady state criterion would have predicted correctly which criterion was the more restrictive. If the construction line had indicated that the transient criterion

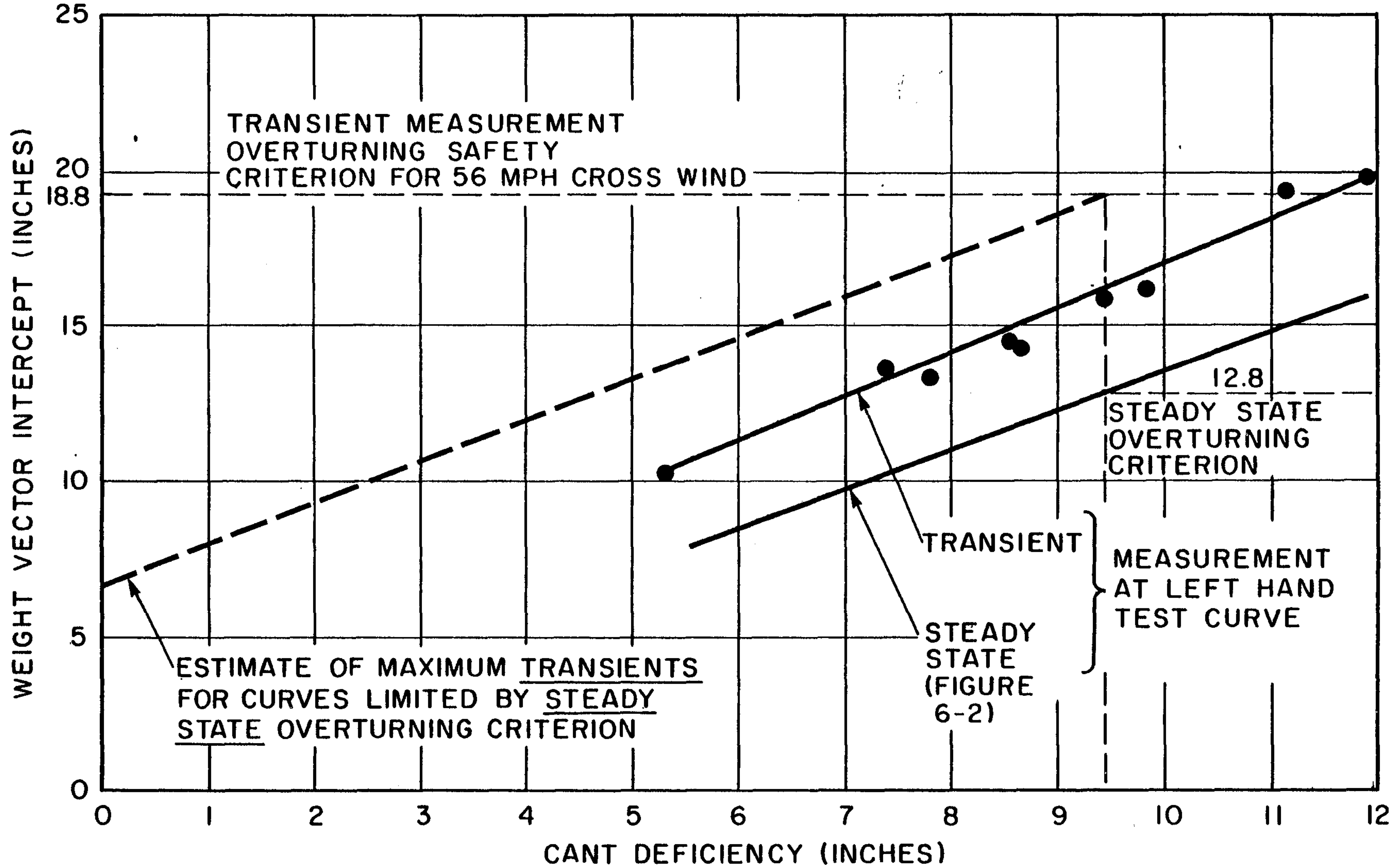


FIGURE 6-4

COMPARISON OF TRANSIENT WEIGHT VECTOR INTERCEPT MEASURED ON THE LEFT HAND TEST CURVE TO THE ESTIMATED MAXIMUM TRANSIENTS FOR CURVES LIMITED BY STEADY STATE OVERTURNING CRITERION of cant deficiency, the transient limit at the particular curve would be computed by subtracting the increment from 8 1/3 inches cant deficiency rather than 9 1/3 to allow for the possibility of the static load offset typical in a fleet of coaches.

Figures 6-5 to 6-8 present single transient measurements of weight vector intercept at many curves in the NEC test zone at the highest test speeds possible. The coach was operated in the non-banking mode, and locomotive measurements in Figures 5-4 to 5-7 were recorded simultaneously. Curves at which the transient vector intercept measurement was high relative to cant deficiency are identified on the figures and listing in Table 6-1.

Table 6-1 divides the curves into three categories. Category I curves have low curvature but no unusual features, and the speed is high even at low cant deficiency. Minor perturbations taken at high speed cause transient vector intercepts that are large relative to cant deficiency, but they remain well below the overturning transient criterion at the maximum class 6 track speed.

Category II curves include specific unusual features such as switches, grade crossings or undergrade bridges. Some are limited by the overturning criteria to speeds less than the class 6 maximum, but the transient criterion is more restrictive than the steady state criterion at only two of these curves.

Category III curves are limited by the vehicle overturning criteria rather than maximum track speed, but they do not include unusual track features. The transient weight vector intercept of the banking Amcoach (non-banking mode) was not high at any category III curve.

The banking Amcoach operated without banking is less sensitive to track perturbations than the F40PH locomotive. The transient overturning criterion was more restrictive than the steady state criterion at 13 curves in the test zone for the locomotive but at

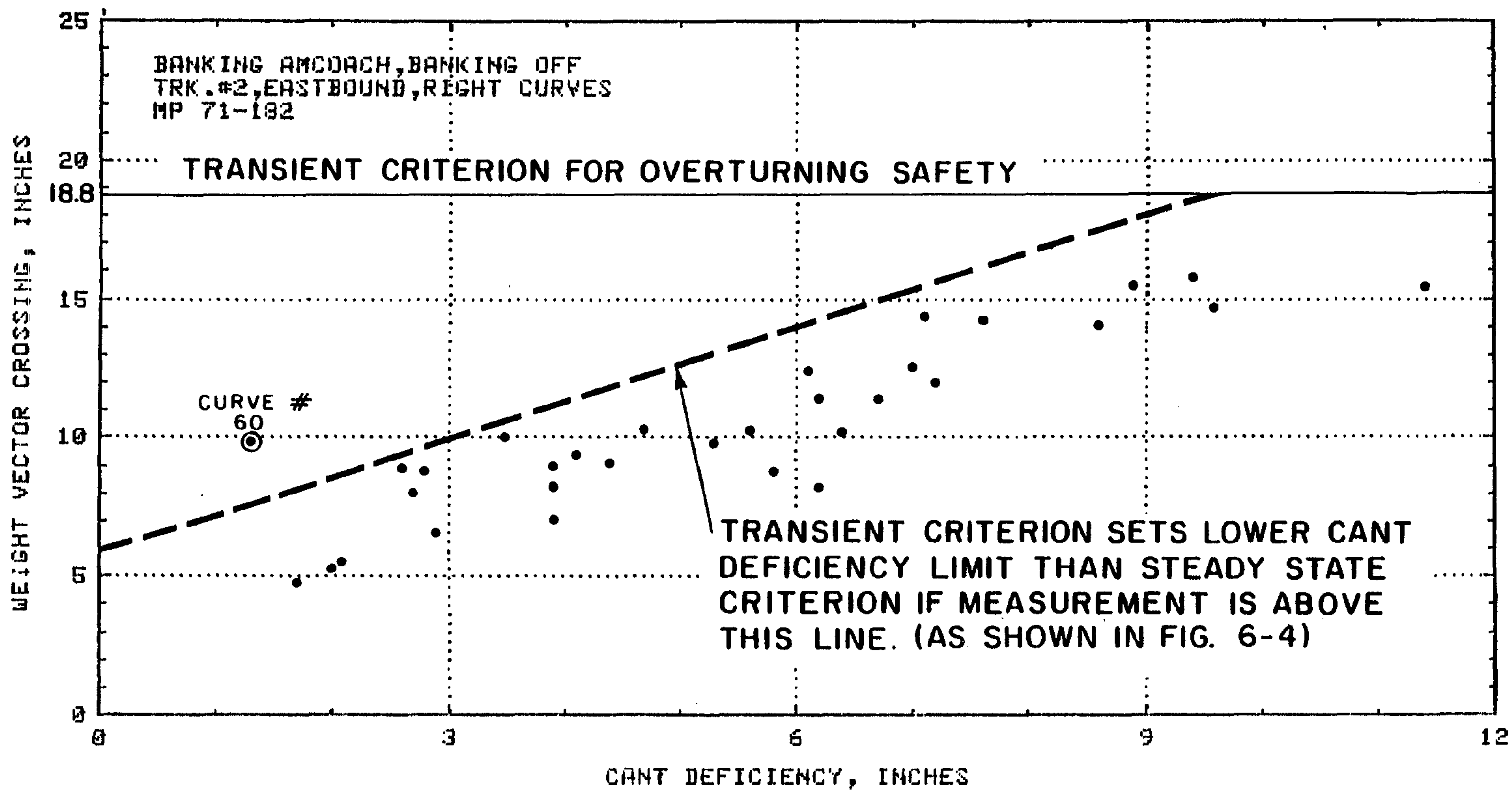


FIGURE 6-5

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT OF THE COACH (NON-BANKING MODE) AT CURVES IN NEC TEST ZONE

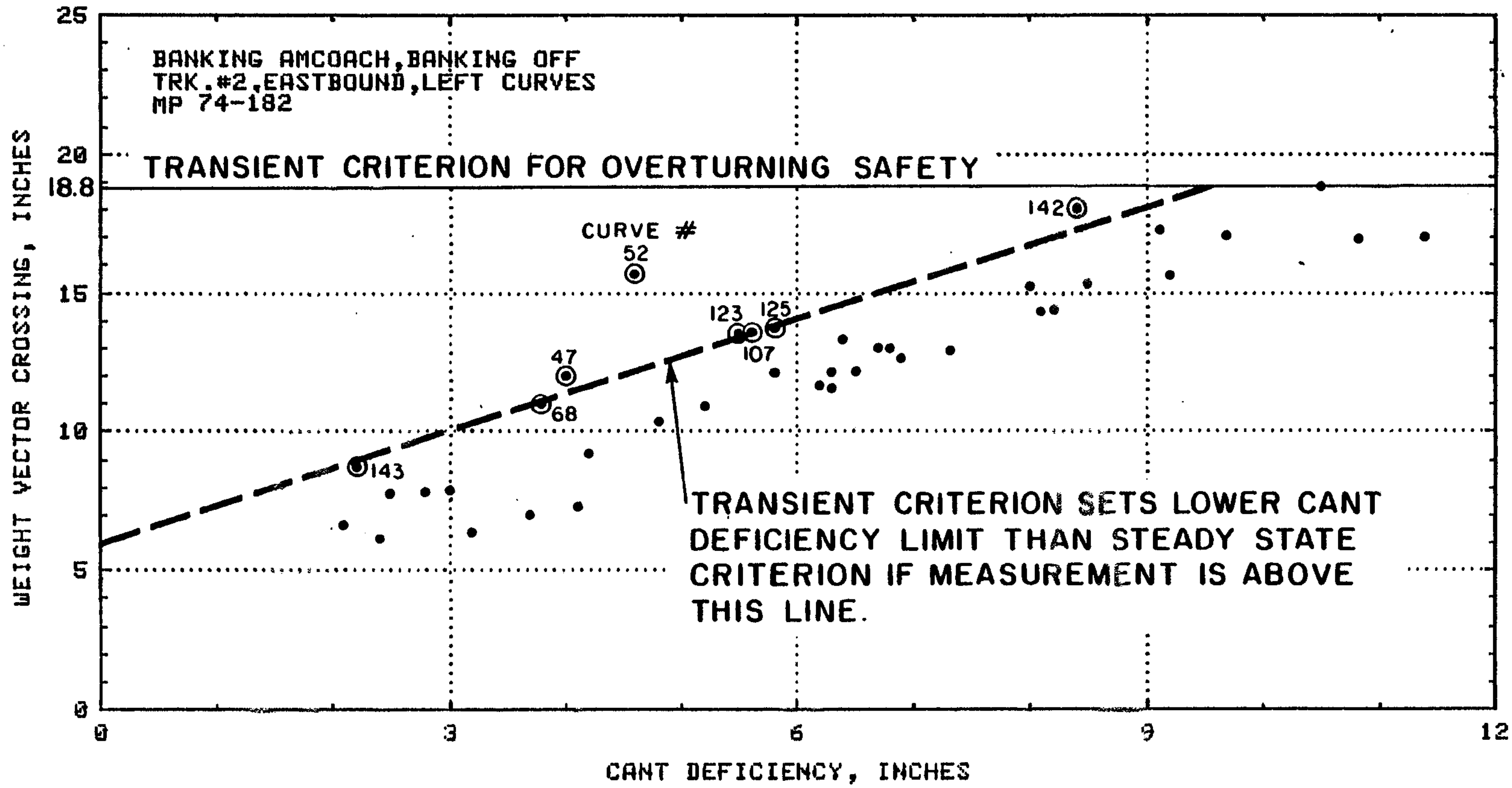


FIGURE 6-6

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT OF THE COACH (NON-BANKING MODE) AT CURVES IN NEC TEST ZONE

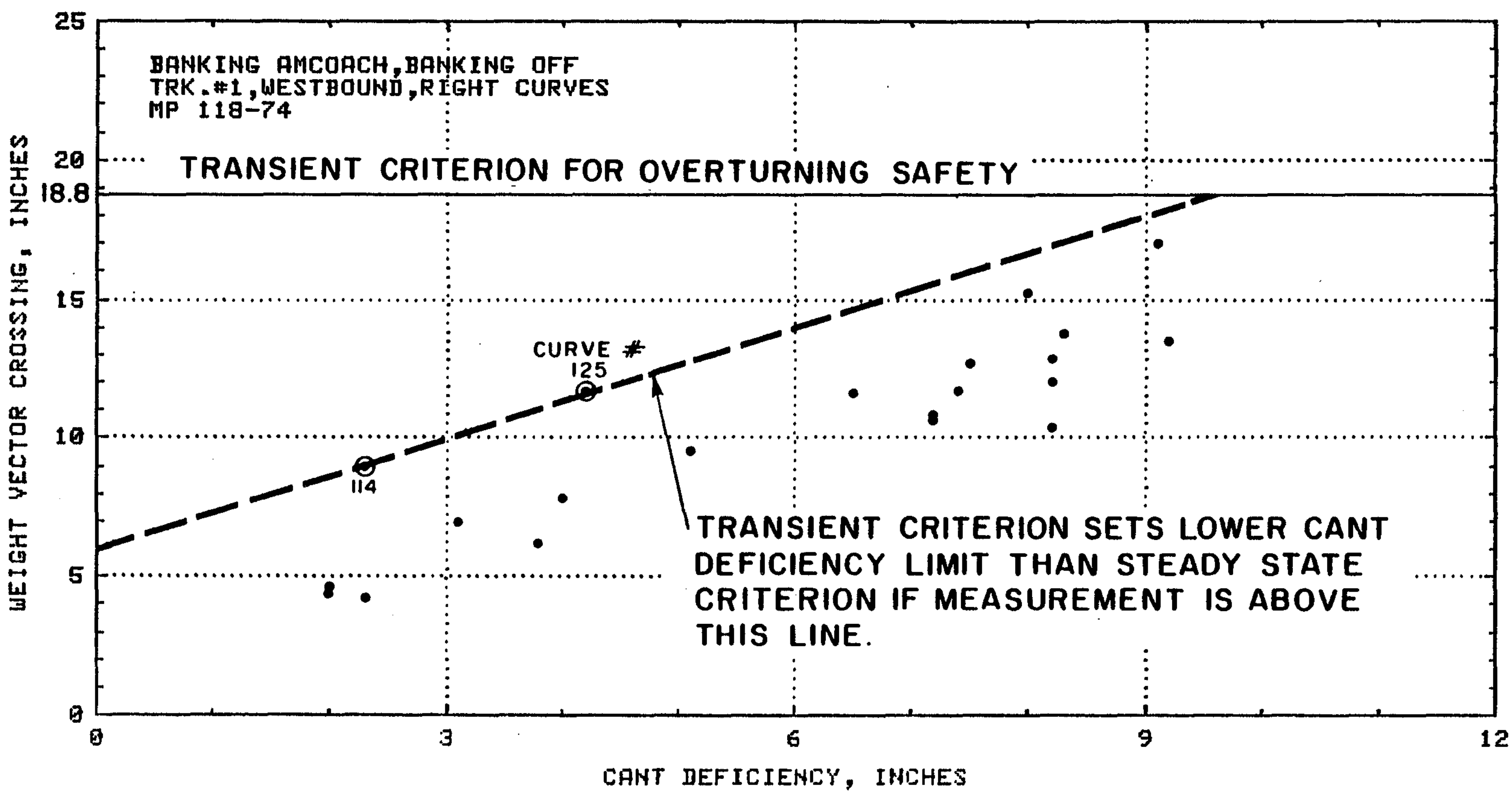


FIGURE 6-7

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT OF THE COACH (NON-BANKING MODE) AT CURVES IN NEC TEST ZONE

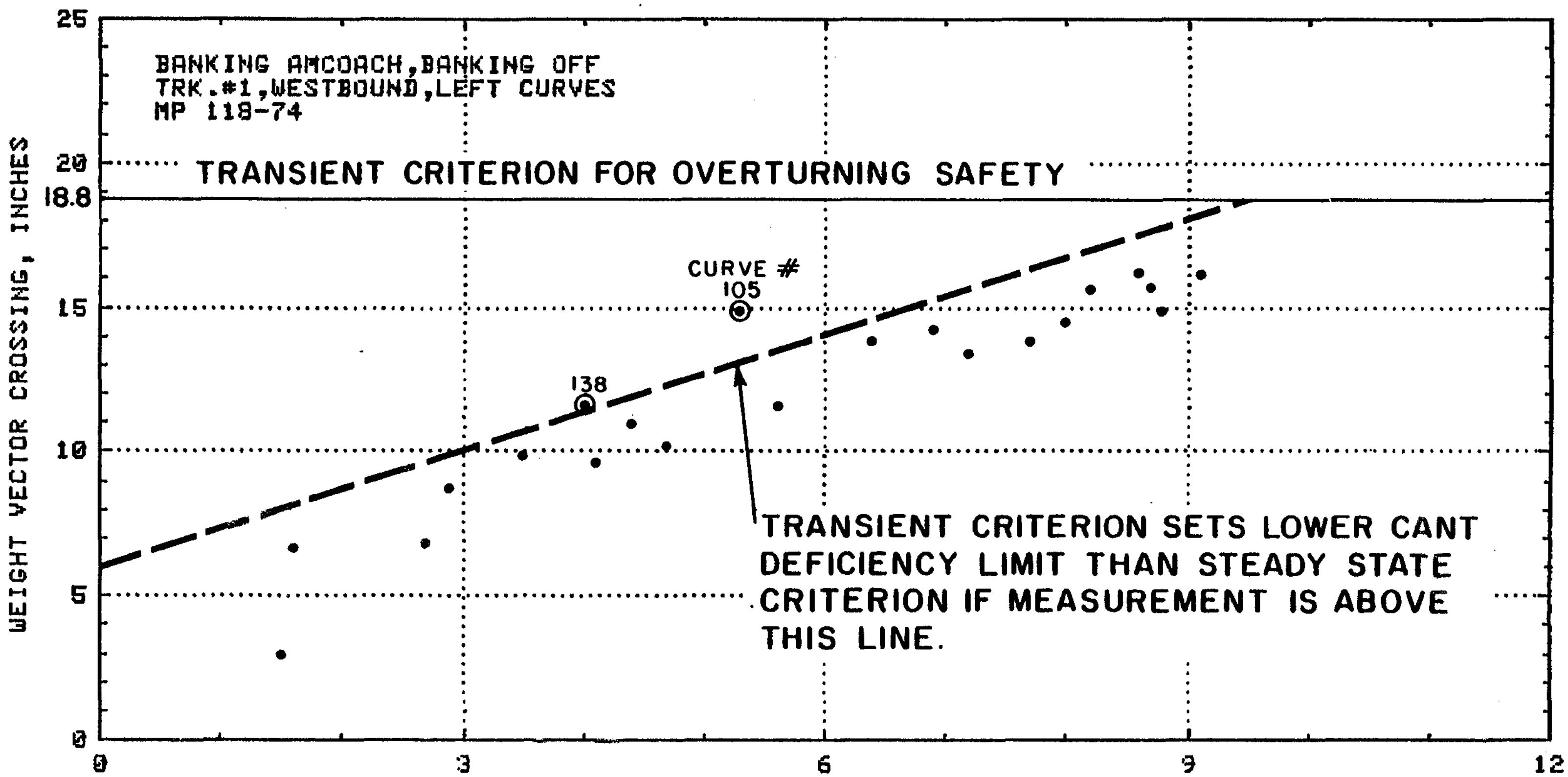


FIGURE 6-8

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT OF THE COACH (NON-BANKING MODE) AT CURVES IN NEC TEST ZONE

TABLE 6-1

CURVES AT WHICH THE TRANSIENT WEIGHT VECTOR INTERCEPT OF THE PROTOTYPE BANKING AMCOACH WAS HIGH IN RELATION TO CANT DEFICIENCY

Curve Number	Track Number	Peak Vector Intercept (in)	Speed (mph)	Cant Deficiency (in)	Estimated Maximum Cant Deficiency (in)	Limiting*** Factor	Unusual*** Features
Category I - Curves (w/o unusual features) limited by maximum track speed.							
*143	2	8.7	85	2.2	6.8	110 mph	-
**143	2	9.5	89	2.6	6.8	110 mph	-
*123	2	13.6	106	5.5	6.2	110 mph	-
Category II - Curves with unusual features.							
*142	2	18.1	81	8.4	7.8	TOC	SC
*138	1	11.6	99	4.0	5.6	110 mph	UGB
**138	1	11.8	101	4.3	5.6	110 mph	UGB
*125	1	11.6	93	4.2	8.3	SSOC	SC
**125	2	13.7	102.1	5.8	7.5	110 mph	SC
**117	1	10.1	84	1.9	4.8	110 mph	SC
*114	1	9.0	99	2.3	3.7	110 mph	UGB
**114	1	10.3	100	2.4	3.7	110 mph	UGB
*107	2	13.6	90	5.6	8.3	SSOC	UGB
*105	1	14.9	88	5.3	7.2	TOC	UGB
*68	2	10.9	86	3.8	8.3	SSOC	UGB
*60	2	9.9	104	1.3	1.5	110 mph	SC
*52	2	15.6	106	5.2	5.3	110 mph	GC
*47	2	12.0	94	4.0	7.6	110 mph	SC
**47	2	11.1	91	3.5	7.6	110 mph	SC
Category III - Curves (w/o unusual features) limited by vehicle overturning safety.							
None							

* Coach Not Banking
** Coach Banking
***Legend:
- SC - Switch in Curves
- UGB - Undergrade Bridge
- GC - Grade Crossing
- TOC - Transient Vehicle Overturning Criterion
- SSOC - Steady State Vehicle Overturning Criterion

only two for the coach. However, the cant deficiency limit set by the steady state overturning criterion was lower for the Amcoach than the F40PH locomotive because of the greater effect of the cross wind allowance.

Operating the banking system had no adverse effect on transient weight transfer. Tests were repeated at the same test zone with the same cant deficiency targets while the coach banking system operated. Figure 6-9 is equivalent to Figure 6-6 with the addition of banking activation. The transient vector intercept measurements were generally lower with banking although there was no perceptable change at some curves. The banking system is subject to lag time between curve recognition and body tilting, and very little improvement in weight transfer would be expected in the entry spiral.

The prototype banking Amcoach is safe against overturning at cant deficiencies up to the same 8 inch general limit set by the steady state criterion for the standard Amcoach, including the adverse effects of 56 mph cross wind and typical static wheel load imbalance. Curves 105 track 1 with an undergrade bridge and 142 track 2 with a switch were the only curves tested in which the transient overturning criteria set the lower limit. Over 7 inches cant deficiency is permitted even at these perturbed curves.

6.2 RAIL ROLLOVER

Rail rollover is related to transient truck side L/V ratio. The transient rather than steady-state measurements should be considered because this mode of derailment may be rapid. Short lateral force pulses may contain the relatively low energy required to roll over the rail, in contrast to the much greater amount of energy required to overturn a vehicle with its large inertial mass.

Appendix B describes the rail rollover criterion. It is based on

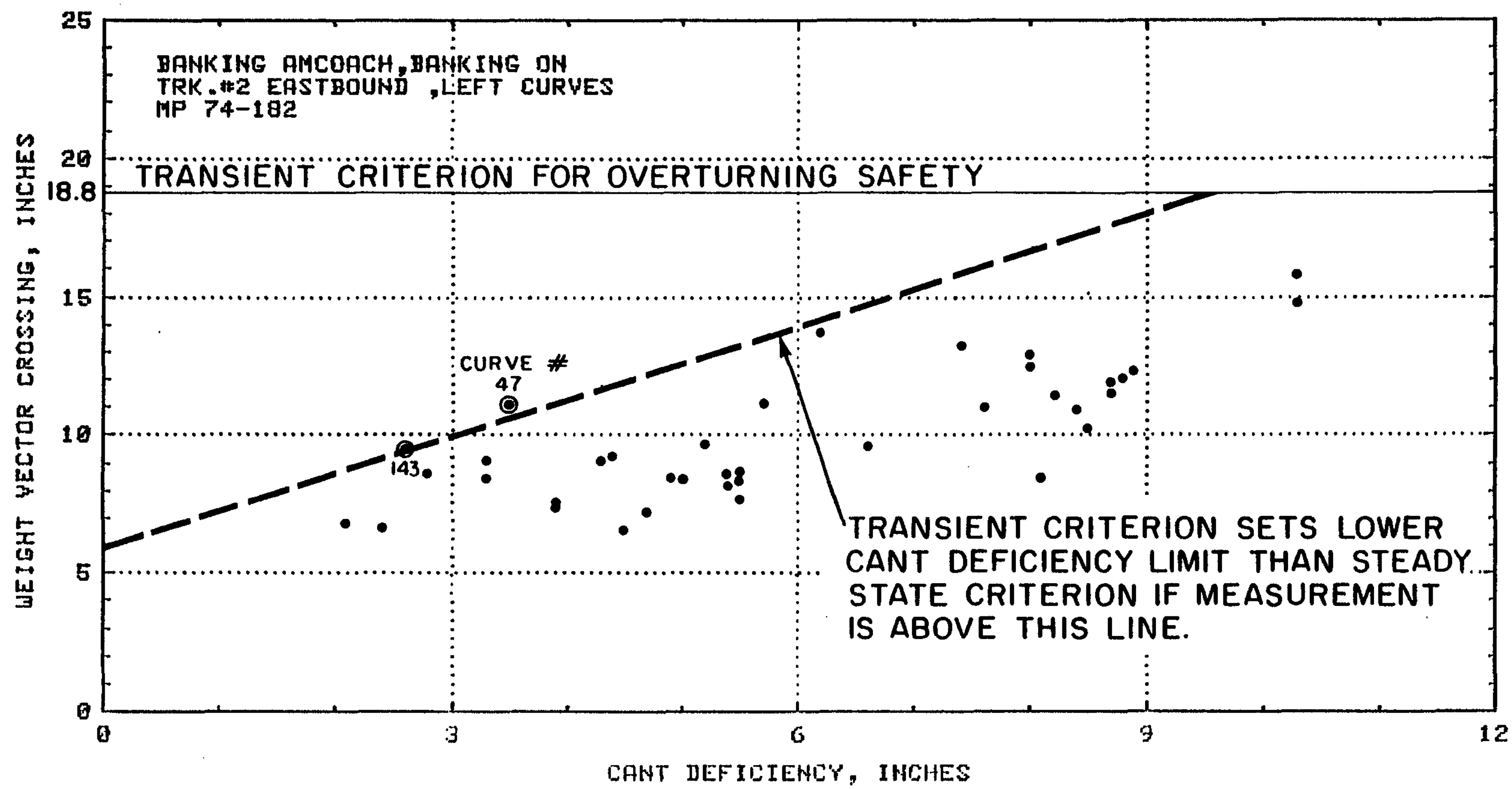


FIGURE 6-9

TRANSIENT MEASUREMENTS OF VECTOR INTERCEPT OF THE COACH IN THE BANKING MODE AT CURVES IN NEC TEST ZONE (SAME CURVES AS FIG. 6-6) the geometry of the rail section and the torsional stiffness of the surrounding rail without assuming pull out resistance of the fasteners. It can be expressed as maximum truck side L/V = 0.5 + $2300/P_w$, where P_w is the single wheel nominal vertical load. It should be interpreted as a restriction based on measurements having time duration of at least 50 milliseconds. Higher measurements are permitted for lesser time durations.

For a loaded Amcoach, $P_w \simeq 15{,}000$ lb, a limiting value of 0.65 should be compared to the transient measurements of truck side L/V ratio. Figure 6-10 compares the measurements taken at the right and left test curves to the criterion. At these relatively smooth curves, the measurements were far below the critical level. Data from tests with the banking operating was used as the worst case because the c.g. movements that decrease vector intercept increase L/V ratio.

A few higher measurements occurred in the NEC test zone between New Haven and Providence. The highest measurements were 0.43 at curves 75A and 79 on track 2, 0.42 at curve 85 track 2, and 0.40 at curve 116 track 1. Only one of these measurements was at less than 8 inches cant deficiency; curve 75A was tested at 6.6 inches cant deficinecy. Even at over 8 inches cant deficiency the truck L/V ratio at curve 75A would remain below 0.50 as projected using the slope of truck L/V ratio with cant deficiency shown in Figure 6-10. Since even the highest measurements from a test zone of 112 curves were well below the limiting value, the rail rollover criterion does not limit the operational cant deficiency of the prototype banking Amcoach.

6.3 LATERAL TRACK SHIFT

The inertia and pulse energy required to move the track structure laterally is much greater than that required for rail rotation, and the use of transient forces to evaluate safety is quite conservative. Using a single wheel force rather than axle force introduces another estimate on the conservative side because the

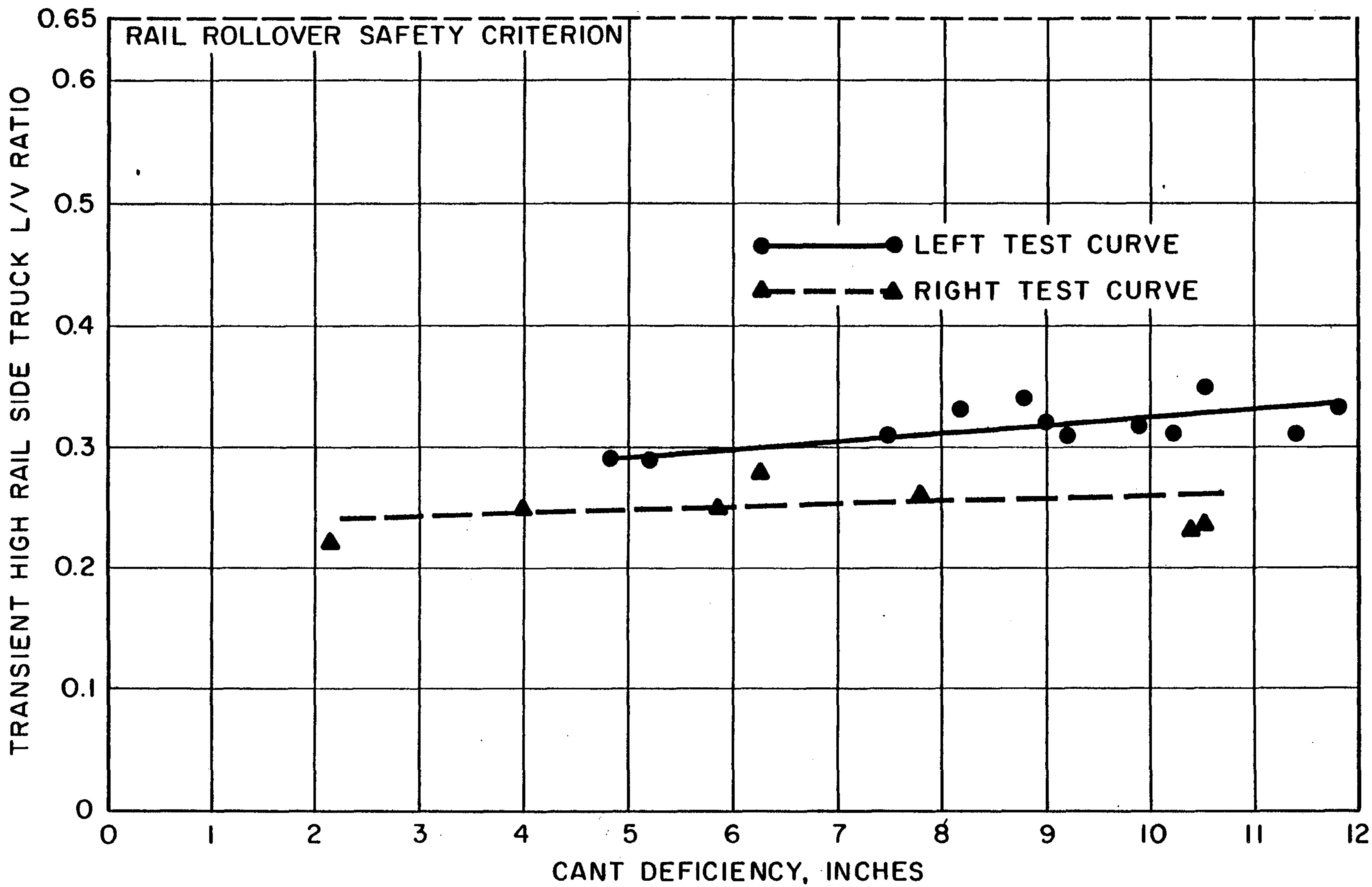


FIGURE 6-10

TRANSIENT TRUCK L/V RATIO OF THE PROTOTYPE BANKING COACH AS A FUNCTION OF CANT DEFICIENCY, MEASURED IN THE BANKING MODE AT RIGHT AND LEFT TEST CURVES net axle force which moves the track is usually less than a single wheel peak force because the wheel forces on the same axle tend to oppose one another.

The safety criterion discussed in Appendix B assumes compacted ballast for full operational cant deficiency, and it allows for the force of crosswinds encountered in service. For the conservative assumption of wood ties, the criterion can be expressed as follows:

$$F_{max} = (1 - \frac{A\Delta\theta}{22320} (1 + .458D)) (.7P + 6600) - (1.28 \times 10^{-3} SV^2)$$

A = rail cross section area, in^2

$\Delta\theta$ = max temperature change after rail installation, ^{o}F

D = track curvature, degrees

P = vertical axle load, lbs.

S = lateral surface area of vehicle, ft^2

V = lateral wind speed, mph

and it is assumed that a single axle bears half the entire vehicle wind load. For typical NEC conditions of 140 lb rail (A = 13.8 in^2), $\Delta\theta$ max of $70^{o}F$ and D max of 4^{o}.

$$F_{max} = .61P + 5800 - (1.28 \times 10^{-3}) SV^2$$

For the unloaded Amcoach axle load of 26,100 lb, body side area of 765 ft^2, and an allowance for 56 mph crosswinds, the maximum permissable lateral axle force is 18,600 lb. The maximum truck lateral force for a two axle truck should be limited to only 1.4 times the single axle maximum lateral force since fewer than twice the number of ties support the lateral force. The lateral track shift criterion permits a maximum truck lateral force of 27,300 lb allowing for one half of the 56 mph crosswind body side load at each truck.

Figure 6-11 presents transient measurements of lead wheel lateral force at the right and left test curves for banking and non-banking tests. The operation of the banking system made no difference, and similar transients were measured at the left and right curves. The wheel lateral force (usually greater than net axle force) increased to only about half the critical level for axle force at about 8 inches cant deficiency and then leveled off. The leveling off indicates that the trailing wheel made flange contact at about 8 inches cant deficiency.

Figure 6-12 presents transient measurements of truck side lateral force at the same curves with and without banking. Slightly higher transients were measured at the left curve, but banking caused no significant difference. These measurements also were only about half the critical level for net truck force.

The transient measurements of truck side lateral force taken on the NEC test zone, including many rough curves, were also well below the safety criterion. In the banking mode, the highest transient measurements of truck side lateral force were 17.0 kips at curve 75A at 6.6 inches cant deficiency and 16.9 kip at curve 110 at 11.2 inches cant deficiency. Both curves were on track 2. Using the greatest slope of the data trends in Figure 6-12, the measurement at curve 75A would be projected to no more than about 18 kips at 8 inches cant deficiency. The measurements of truck side lateral force at even the harshest curves in the test

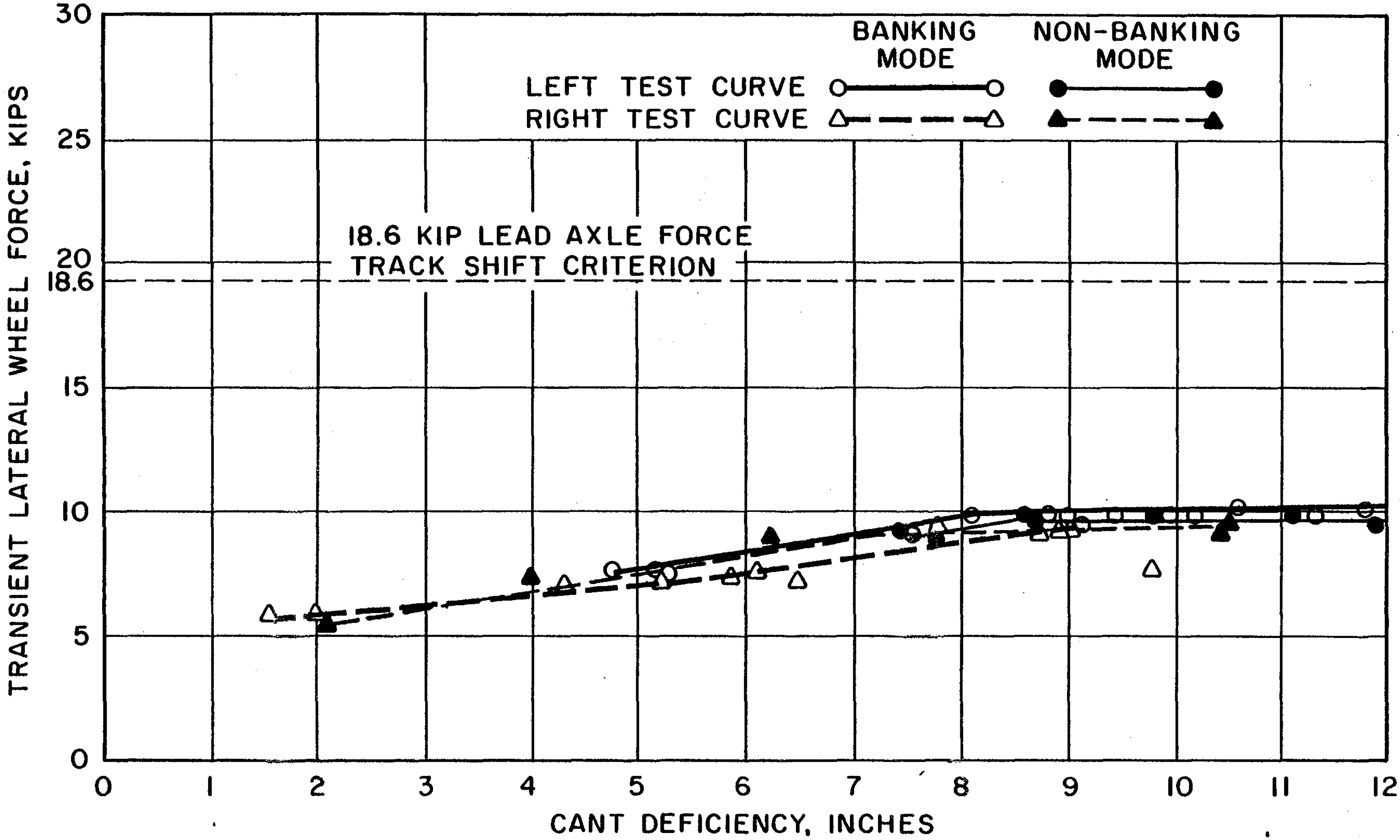


FIGURE 6-11

COMPARISON OF TRANSIENT LEAD WHEEL FORCE MEASUREMENTS OF THE PROTOTYPE BANKING AMCOACH TO THE LATERAL TRACK SHIFT CRITERION

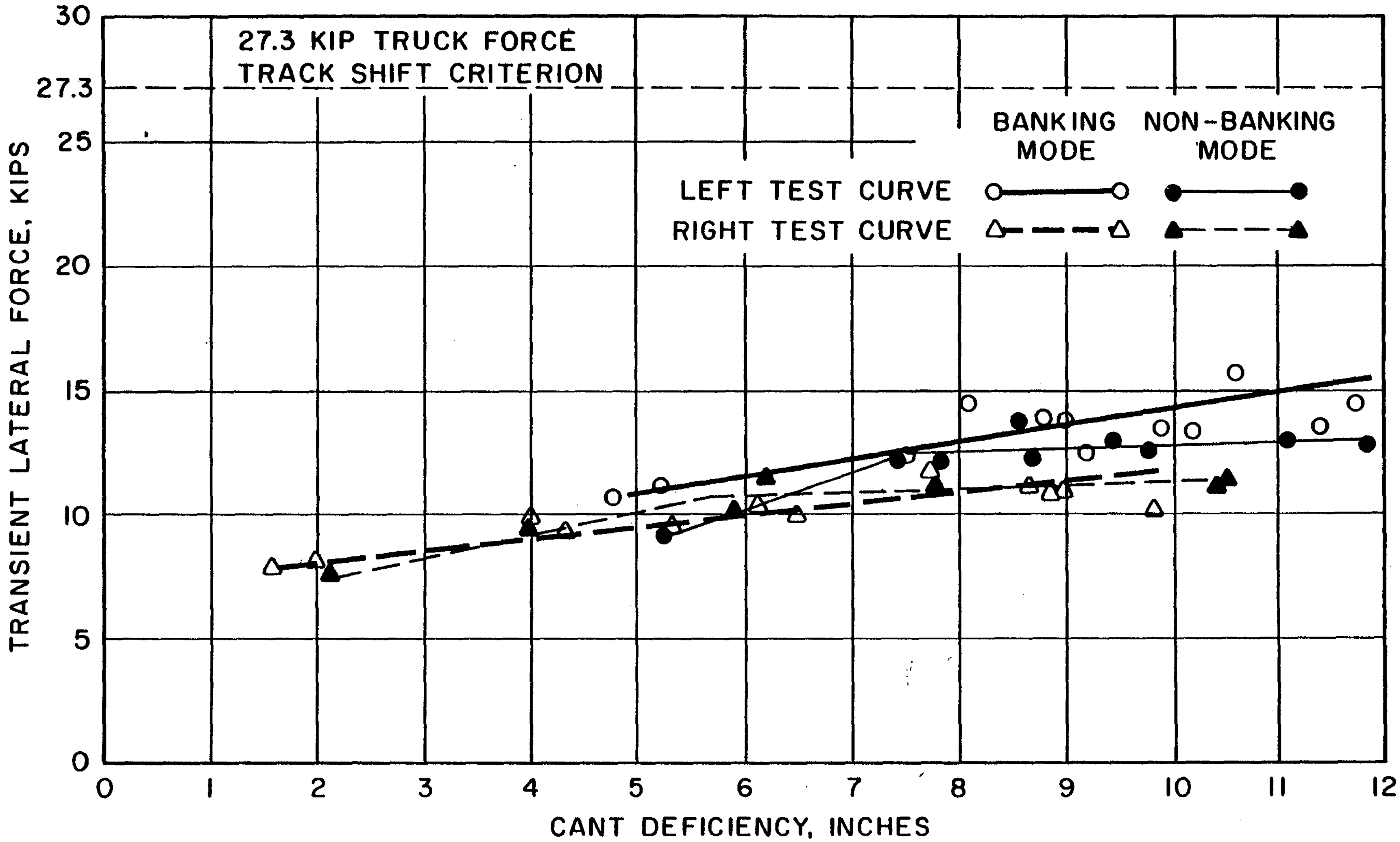


FIGURE 6-12

COMPARISON OF TRANSIENT TRUCK SIDE LATERAL FORCE MEASUREMENTS OF THE PROTOTYPE BANKING AMCOACH TO THE LATERAL TRACK SHIFT CRITERION zone were well below the lateral truck shift safety criterion of 27.3 kip, and the cant deficiency of the banking Amcoach should not be limited by concern for lateral track shift.

6.4 WHEEL CLIMB

The most appropriate criterion of safety concerning wheel climb for comparison to the test results of the prototype banking Amcoach is that used by AMTRAK in its acceptance specification for the AEM-7. This criterion takes into account the flange angle specified by the AAR Wheel and Axle Manual and it appears to have been based on conservative judgements. It states that the wheel (L/V) ratio must be less than $0.056/T^{-0.927}$ where T is the duration of the transient in seconds and that the maximum wheel (L/V) ratio for transients of duration greater than 50 milliseconds is 0.90.

Transient measurements of lead wheel L/V ratio at the relatively smooth test curves are given in Figure 6-13 for the Amcoach in the banking mode. The banking mode is the worst case because the vertical force on the high rail wheels is minimized. The transient L/V ratio is between 0.4 and 0.5, only about half the limiting value, for cant deficiencies between 2 and 12 inches. The increase with cant deficiency is very slight because the lateral and vertical forces increase in unison.

Transient measurements of L/V ratio of the high rail lead wheel were made at over one hundred curves between New Haven and Providence. The highest measurement was 0.62 recorded at both curves 75A and 79 on track 2 at 6.6 and 8.1 inches cant deficiency respectively. Since the greatest transient L/V ratio measurements in a large test zone do not approach the critical level, the wheel climb safety criterion does not limit the safe cant deficiency of the banking Amcoach.

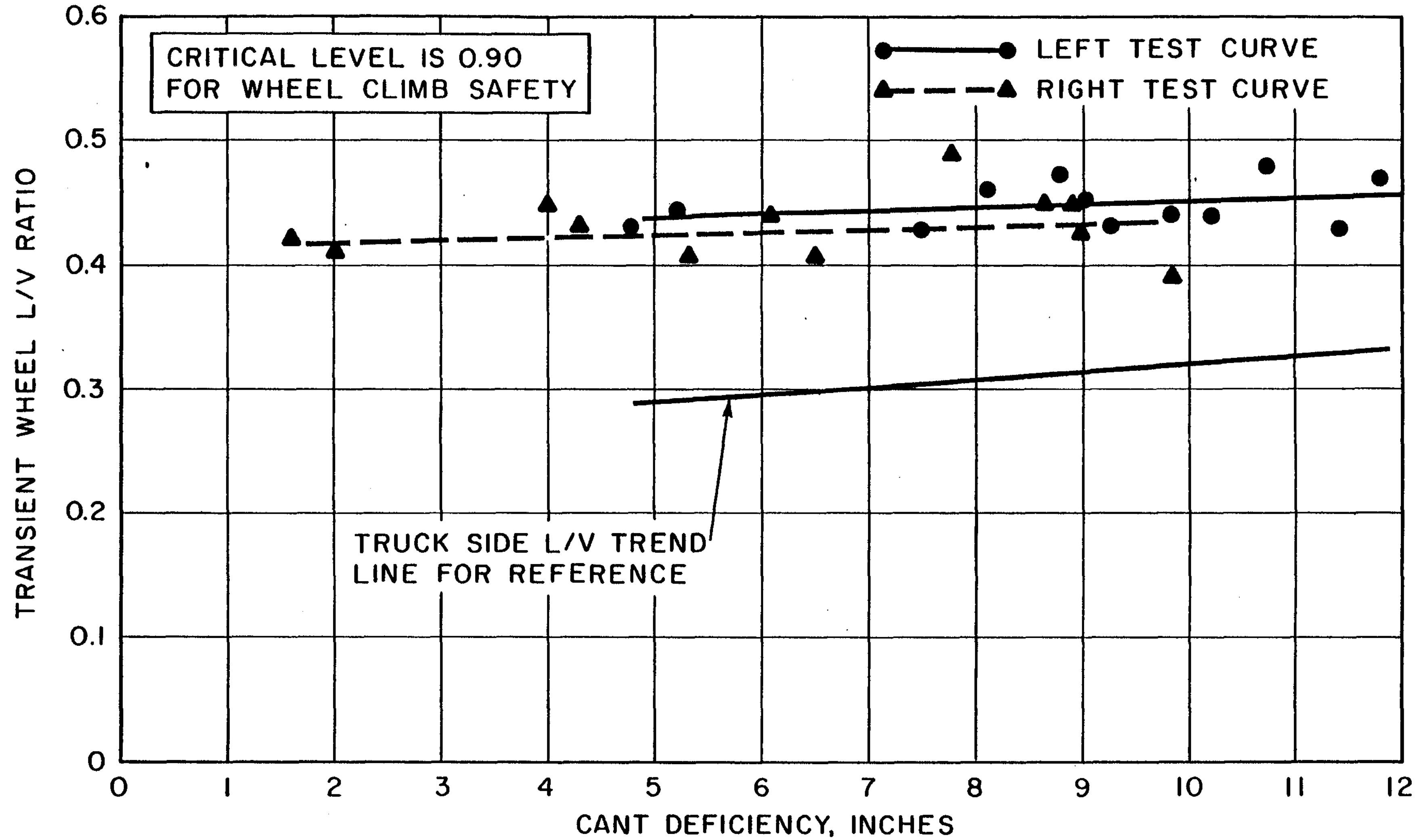


FIGURE 6-13

TRANSIENT WHEEL L/V RATIO OF THE PROTOTYPE BANKING AMCOACH AS A FUNCTION OF CANT DEFICIENCY, MEASURED IN THE BANKING MODE AT RIGHT AND LEFT TEST CURVES

6.5 RIDE COMFORT

Appendix B includes a variety of criteria for lateral acceleration used by organizations throughout the world as indices for ride comfort evaluation. The most commonly recognized standards in the U.S. are the AAR recommendations of 0.1g maximum steady state and 0.03g/sec maximum "jerk". The least restrictive standards are those of SNCF which permit 0.15g and 0.10g/sec respectively.

When the AAR study was undertaken in the early 1950's, coach suspensions were designed such that large body roll angles occurred in curves. The component of gravitational acceleration in the plane of the coach floor added substantially to the centrifugal acceleration as the coach body rolled toward the outside of the curve. At three inches of cant deficiency, the body roll of contemporary coaches was usually sufficient to cause a total of 0.1g lateral acceleration in the plane of the floor.

Figure 6-14 compares the steady state lateral acceleration of the prototype banking Amcoach with and without the use of the banking system. The characteristics of the standard Amcoach, the LRC banking coach, and of a hypothetical vehicle without suspension are included for reference. The floor of a vehicle without suspension remains parallel to the plane for the rail heads, and the steady state lateral acceleration is simply the centrifugal component. The body roll angle of the coach when operated without banking was about equal for left and right curving, and it caused an additional component of lateral acceleration. At 8 inches cant deficiency the steady state lateral acceleration of the banking coach in the non-banking mode was about 0.175 g, exceeding the SNCF ride comfort criterion and greatly exceeding the more conservative AAR criterion. The steady state lateral acceleration of the standard Amcoach was about 0.16 g at the same cant deficiency. It was lower because of the higher roll stiffness of the standard truck but it still exceeded both ride comfort criteria. The low lateral accelerations of the banking coach at

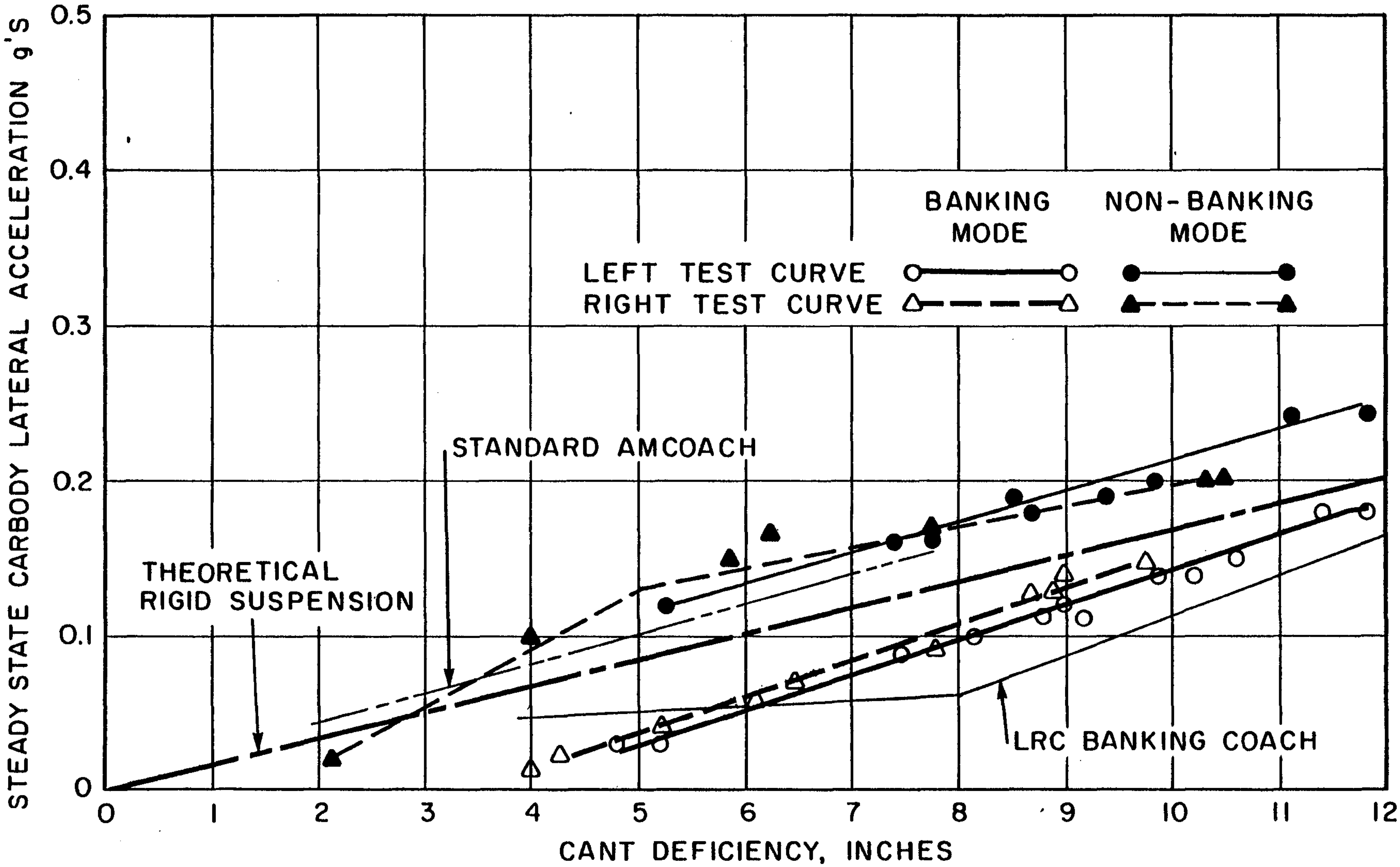


FIGURE 6-14

STEADY STATE LATERAL ACCELERATION OF THE PROTOTYPE BANKING AMCOACH COMPARED TO A VEHICLE WITH ZERO ROLL ANGLE right curves below 5 inches cant deficiency was due to an initial listing of the body to the right. At higher cant deficiencies, the effect of the static list was removed by the bottoming of the air suspension. The banking system malfunction causing the body list was corrected by the manufacturer before the over the road tests at highest speeds.

The banking system rolled the body about 4° toward the low rail after sensing the threshold level of 0.04 g at the bolster. This single banking increment was more than enough to cancel the body roll due to the deflection of the primary suspension and the secondary coil springs up to 12 inches cant deficiency. The body tilt was used successfully to counteract part of the centrifugal acceleration, and the steady state lateral acceleration remained below the AAR ride comfort criteria up to about 8 inches cant deficiency.

The no tilt or full tilt (one increment) banking control of the prototype banking Amcoach was conceived as a lower cost and lower maintenance alternative to the more sophisticated servo controlled proportional banking system of the Canadian LRC coach. The LRC coach had softer primary and secondary suspensions but a greater body tilting angle. As shown in Figure 6-14, it is able to adjust the body angle to maintain about 0.05g steady state lateral acceleration at the coach floor up to approximately 8 inches cant deficiency. Above 8 inches cant deficiency the tilt control is at full range, and further suspension deflections cause body roll relative to the track. Although the simple Amcoach system did not achieve lateral accelerations as low as the LRC coach between 6 and 8 inches cant deficiency, its results satisfied the AAR ride comfort criterion. If the potential benefits in cost and reliability are great enough, the simple approach may be the more attractive. A second important parameter of ride comfort, the rate of change of lateral acceleration called jerk, has not yet been addressed for either banking system. A supplemental investigation of jerk data is anticipated.

6.6 MAXIMUM OPERATIONAL CANT DEFICIENCY

The modification of the Amcoach for banking did not change its maximum safe cant deficiency. The particular test coach experienced the maximum steady state load transfer permitted by the safety criterion at approximately 9-1/3 inches cant deficiency, but its average load transfer characteristic was the same as the standard Amcoach as shown in Figure 6-3. Small imbalances in static weight distribution, as measured for the standard Amcoach, the F40PH locomotive and other vehicles tested previously, are typical of vehicles in service. The favorable weight symmetry of the test coach along with its problems of body lean and air spring deflation were particular to it and were not characteristic of the banking modification. A general cant deficiency safety limit of 8 inches is recommended for fleets of Amcoaches with or without the banking modification. It is based on the steady state overturning criterion allowing for the possibilities of slightly imbalanced static wheel loads and high crosswind.

The general cant deficiency limit is useful because very few exceptions caused by transient load tranfer were identified. Operation at only two curves in the test zone was limited to less than 8 inches cant deficinecy (but more than 7 inches) by the vehicle overturning criterion for transient measurements of load transfer. These curves were 105 track 1 and 142 track 2 which contain an undergrade bridge and a switch respectively. The measurements indicating safety against rail rollover, lateral track shift and wheel climb were considerably below the critical values at every curve in the test zone for cant deficiency well above 8 inches. The cant deficiency limit of the coach is lower than that for the locomotive because the crosswind allowance is more restrictive, but the coach is less sensitive to track perturbations with regard to transient weight transfer (overturning). Since very few curves cause high transient weight transfer, the 8 inch limit set by the steady state criterion should be regarded as a general rule, and local maintenance or slow orders should be implemented at a few trouble spots.

7.0 GENERAL CONCLUSIONS

The purpose of the test program was to evaluate the maximum safe cant deficiency of the F40PH locomotive and the prototype banking Amcoach and to determine if the retrofit banking system maintained ride comfort at high cant deficinecy. It is important to avoid distorting the evaluation of vehicle capability by focusing solely on a few known problem curves. The vehicle capabilities are described by general cant deficiency limits with the recognition that exceptions to the general rule exist. The number and character of the exceptions determine the usefulness of the general cant deficiency limits. The exceptions are enumerated in Tables 5-1 and 6-1. The first eight conclusions speak to the specific purpose of this test program. Other conclusions general to the topic of high cant deficiency passenger service are also included.

	Reference
1. A train consisting of F40PH locomotives and Amcoaches satisfies the operating safety criteria at up to 8 inches cant deficiency at all but 3 curves tested on the NEC betwen New Haven and Providence. The maximum cant deficiencies set by the transient overturning safety criterion are 6.3 inches for the locomotive and 7.2 for the coach at one curve, 7.6 inches for the locomotive at another and 7.8 inches for the coach at the third.	Section 5.7; 6.6; Appendix E
2. The general cant deficiency limit for the train is useful because the exceptions caused by transient weight transfer are very few, and they all occurred at curves with switches or undergrade bridges. No exceptions limiting safe deficiency below 6 inches were identified.	Section 5.1; 6.1

3. The general cant deficiency limit, obtained by rounding down to the nearest inch the limit imposed by the steady state overturning criterion, is 8 inches for both the banking Amcoach and the standard Amcoach. It is useful because the transient overturning criterion identified only two exceptions in over 100 test cruves and both were associated with switches or undergrade bridges. — Section 6.6

4. The general cant deficiency limit of F40PH locomotive, obtained by rounding down to the nearest inch the limit imposed by the steady state overturning criterion is 9 inches. It is less useful than the general limit of the Amcoach because eight exceptions were identified by the transient overturning criteria. Switches, undergrade bridges or grade crossings were associated with all curves where exceptions occurred. — Section 5.7

5. The coach rather than the locomotive set the general cant deficiency limit of the train because the assumption of worst case lateral wind is more restrictive for it. — Section 5.1.1; B.1.1.3

6. The JNR overturning criterion, based on steady state considerations only, was generally applicable to U.S. passenger trains. However, the F40PH locomotive exhibited greater dynamic activity than the Amcoach at curve irregularities. — Tables 5-1 and 6-1

7. All curves which caused exceptionally high transient weight transfer in the test vehicles also induced harsh transient lateral acceleration, but the presence of high transient lateral acceleration does not guarantee exceptional transient weight transfer. — Appendix A; Tables 5-1 and 6-1

8. The banking system of the modified Amcoach was successful is maintaining a low level of steady state lateral acceleration at high cant deficiency. The AAR ride comfort criterion was satisfied up to 8 inches cant deficiency. — Section 6.5

9. Four mechanisms which can lead to derailment have been identified from available literature in North America, Europe and Japan. These mechanisms are vehicle overturning, wheel climb, rail rollover and lateral panel shift. — Section 4.0; Appendix B

10.	For each mechanism, a quantitative criterion has been defined to indicate the limits of safe operation.	Section 4.0; Appendix B
11.	Lateral wind speed produces lateral rail forces that can add to derailment tendency. The worst case assumption of lateral wind speed based on a 10-year mean recurrence interval for the Boston area has been used in all criteria. This level of lateral wind speed is 56 mph measured at 15 feet above ground level.	Section 5.1.1; B.1.1.3
12.	Light vehicles with large side areas are most influenced by the assumption of worst case lateral winds.	Section 5.1.1; B.1.1.3
13.	Overturning weight transfer set the limit of safe cant deficiency of both the F40PH Locomotive and prototype banking Amcoach.	Section 6.1
14.	Two criteria are associated with the vehicle overturning mechanisms. The first is related to transient behavior. The second criteria is associated with steady-state behavior. These criteria are based on JNR research and reports.	Section 5.1; B.1
15.	The worst case safety evaluation with regard to overturning of the prototype banking Amcoach was done in the nonbanking mode because any influence of the banking system on weight transfer was beneficial. However, it had little effect on transient measurements at entry spirals.	Section 6.1
16.	The difference between the transient and the steady state overturning criteria is a load transfer factor of 20% of the static wheel load. It may be viewed as an allowance for track irregularities which induce additional side to side load transfer over the steady state level having a short time duration.	Section 5.1; B.1.1.3
17.	The transient overturning criterion appears to be the limiting criterion for modern passenger vehicles only at curves having significant irregularities, usually associated with switches, bridges and grade crossings.	Section 5.1.3; 6.1.2; reference 23

18.	The vehicle overturning safety criteria set the cant deficiency limits of the LRC locomotive, the AEM-7 locomotive, the LRC coach, and the standard Amcoach, as well as the F40PH Locomotive and the prototype banking Amcoach. It appears that cant deficiency safety limits set by concern for vehicle overturning rather than wheel climb or catastrophic track damage is a general characteristic of modern light weight vehicles with four axles.	Section 5.7; 6.6; reference 23
19.	The simple quasistatic model in Appendix D for computing steady state load transfer is useful in predicting the cant deficiency limit for modern passenger vehicles. Figure 7-1 shows the close agreement betwen predictions and test results.	Appendix D
20.	The important parameters of the model are suspension roll center height, roll stiffness, and lateral suspension travel as well as center of gravity height.	Same
21.	Ride quality under high cant deficiency is influenced by steady state lateral acceleration, time rate of change of acceleration (jerk) and the frequency content of acceleration.	Section B.5
22.	Lateral movement of the body c.g. during, banking, which is a design variable of banking systems, is of major importance to vehicle weight transfer.	Section 6.1

8.0 RECOMMENDATIONS

1. The general cant deficiency limit of a train using F40PH locomotives and banking Amcoaches should be considered 8 inches. Steady state ride quality requirements are achieved and conservative safety criteria are satisfied at all but a very few identifiable curves.

2. Lateral acceleration surveys (especially when measurements are made in the locomotive) can be used to identify curves where exceptions to the general cant deficiency limit exist. Exceptions are much more likely at curves with switches, undergrade bridges, or grade crossings. The frequency of surveys should be based on local track degradation history.

3. Fail-safe devices are required to prevent one truck of a banking coach from operating while the other is disabled.

4. The general cant deficiency limit of a train using F40PH locomotives and standard Amcoaches should be considered 6 inches; purely for passenger ride comfort requirements. No exceptions to the safety criteria were identified below 6 inches cant deficiency.

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APPENDIX A

TEST DATA SUMMARY

(Under separate cover)

APPENDIX B

SAFETY AND COMFORT CRITERIA

Various criteria are used in North America, Europe and Japan for the determination of curving speed. Concern for the possibility of derailment by vehicle overturning, wheel climb, rail rollover, and lateral track panel shift and of passenger discomfort has led to a multiplicity of criteria. The curving criteria have been reviewed as part of the Improved Passenger Equipment Project by Battelle Columbus Laboratory and many of their findings published in Reference (1) are included. The original sources are referenced except for conclusions by the BTL authors. Vehicle overturning is considered in the greatest detail because it appears to be the first limiting factor for the vehicles in this study.

B.1 VEHICLE OVERTURNING CRITERIA

When the overturning moments about the high rail caused by the lateral inertial forces acting at the vehicle center of gravity and by the wind force acting at the center of pressure equal the restoring moment due to the weight, the vehicle is balanced about the high rail. Figure B-1 demonstrates the rollover computations. All computations that follow are based on half vehicle models which have half the mass and surface area of a vehicle and only one truck. For simplicity of illustration, all vehicle mass is considered to be concentrated at the body center of gravity (c.g.) in Figure B-1. The geometry and stiffness of the primary and secondary suspension components result in a roll center (R.C.) about which the body c.g. has rotated through the angle δ. The roll center defined in this way can be considered to translate with the body a distance ϕ due to wheel flanging and lateral deflection of the primary and secondary suspensions. At greater than balance speed the effect of suspension deflections (and passive tilt motions) is to move the c.g. closer to the high rail thereby reducing the restoring moment.

The concept of weight vector intercept is used frequently to express the risk of vehicle rollover. Momentarily neglecting the wind force, Figure B-1 may be used to directly visualize the weight vector intercept. The lateral inertial forces and vertical gravitational force form a resultant force vector which may be projected to the plane of the railheads. The intersection of the line of action of the resultant force with the railhead plane is the point about which the vehicle, with lateral curving loads, would balance at the instant the loads were measured. The distance from the track centerline to this point is called the weight vector intercept (or vector crossing). A symetrically loaded vehicle at rest on a level track would have zero as a weight vector intercept, and at a weight vector intercept of 30 inches (assuming 60 inches between wheel contact with left and right rails) the vertical load on the low rail wheels would be

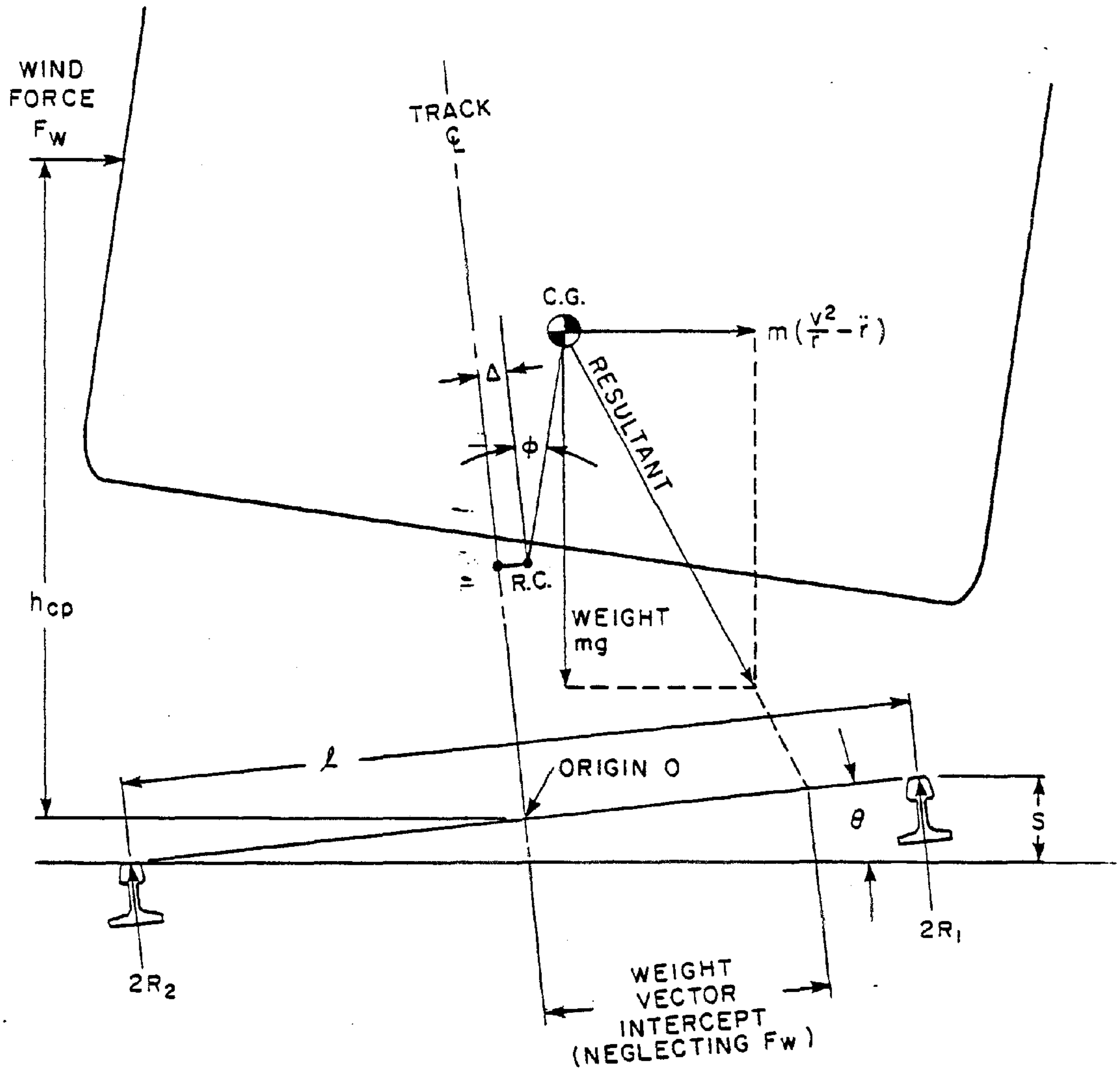


Figure B-1. General Vehicle Overturning Model reduced to zero. The definition of weight vector intercept as the balance point of the vehicle is also valid for a vehicle subjected to wind loads although the resultant force vector is harder to visualize because it does not pass through the c.g.

Most overturning criteria are concerned primarily with the forces acting through the c.g. The wind force is used as a modifying factor because its effect on the balance point can be computed separately and applied additively and it is not a controlled variable during testing.

The weight vector intercept measurements taken during this test program were a direct computation of the vertical balance point of the lead truck of each vehicle which was fully equipped with force measuring instrumented wheels. The computation was:

$$\text{Weight Vector Intercept} = 30 \text{ inches} \left[\frac{(R_{lf} + R_{lr}) - (R_{rf} + R_{rr})}{R_{rf} + R_{rr} + R_{lf} + R_{lr}} \right]$$

where R_{rf}, R_{rr}, R_{lf}, and R_{lr} are the vertical loads of the right front, right rear, left front and left rear wheels of the lead truck.

This measurement includes the effects of static imbalance of the vehicles, all suspension and tilt motions and typical coupler forces. It does not include the gross lateral motion of the vehicle which occurs if both outside wheels of the truck are flanged against the high rail, but this motion of the c.g. contributes very little to side to side load transfer. Only effects of gravitational and inertial forces contribute significantly to the test data because the wind speed was negligible.

The lateral inertial force has two components. Steady state curving criteria assume that r is constant so that only mV^2/r remains. Measurements averaged over the body of a curve approximate a steady state. Transient curving criteria also include lateral inertial forces resulting from the $-m\ddot{r}$ term. The effects of transition spirals and alignment deviations are described by $\ddot{r}$, the second derivative of the path radius with respect to time, which is also a function of speed. The negative sign is required because a decrease in radius results in an increase of force. In order to access the risk of overturning using the transient weight vector intercept criteria, the time duration of the measurement must be considered. Even when the weight vector intercept as shown in Figure B-1 is at the high rail and the low rail vertical force is zero the vehicle does not actually overturn. An even higher lateral force (implying a transient weight vector intercept greater than 30 inches) and time for this force to act are required for actual overturning because of the c.g. must rise as it is pivoted outward and a finite amount of time is required for the net overturning moment to rotate the c.g. outside the rail.

Vehicle overturning criteria specify two essential factors: (1) the total allowable side to side weight transfer ratio and (2) the cross wind velocity whose effect on weight transfer must be allowed for in advance. The ratio of the moment of the vehicle lateral surface area to its weight determines the portion of the gross allowable weight transfer consumed by the wind allowance. The weight transfer ratio due to lateral inertial forces and lateral movement of the c.g. is compared to the net allowable weight transfer ratio. A greater net weight transfer ratio is usually allowed for locomotives than coaches under the same criteria due to their lower ratio of surface area to weight.

In order to compare various criteria stated in terms of weight vector intercept, moment safety factor, or load ratio with differing cross wind speed allowances, it is necessary to reduce them to a common basis. Weight vector intercept will be chosen as a common basis of weight transfer ratio because of its intuitive concept as the instantaneous vehicle balance point and because the data contained in Appendix A includes measurements of weight vector intercept.

B.1.1 OVERTURNING CRITERIA IN USE

1. ONE THIRD RULE - (AAR)

The "one third rule" states that the weight vector intercept computed from the vertical gravitational force and the lateral centrifugal force must remain within the center one third of the track (1). It is a common rule of thumb but it is vague and poorly documented. The description of the lateral forces as centrifugal implies that the steady state rather than transient weight vector intercept should be considered. The wind force is not considered. The middle one third of the track is usually considered to be ± 10 inches about the track centerline although it has been interpreted in one instance (2) as 20 inches from the gage side of each rail (± 8-1/4-inch from centerline). It is believed that the one third rule is actually an earlier rule of thumb for the design of chimneys to withstand wind loads that was applied to railroad vehicles by analogy (3).

2. OVERTURNING MOMENT SAFETY FACTOR (Association of German Locomotive Manufacturers) (Ref. 4)

This factor of safety against overturning can be expressed: $SF = M_r/M_o \geq 1.2$ where M_o is the sum of the overturning moments including the maximum effect of wind pressure at 12 pounds per square foot (68 mph crosswind) and M_r is the restoring moment based on the laterally shifted c.g. location. Presumably only the quasistatic lateral inertial force is included since there is no mention of time duration or transient loads.

The factor of safety criteria is translated into weight vector intercept as follows, where the free body diagram of the vehicle is Figure B-2:

W = weight acting through the vehicle c.g.

X = lateral movement of the c.g.

F_L = resultant lateral force including wind and inertial forces

H = vertical height of the line of action of F_L. It is usually higher than the c.g. because of the wind force component

R_1 = outer rail vertical wheel force

R_2 = inner rail vertical wheel force

Considering Figure 5-2A:

$$M_o = HF_L$$

$$M_r = (30 - X)W$$

Applying the criteria limit:

$$1.2 = \frac{M_r}{M_o} \Rightarrow HF_L = \frac{(30 - X)W}{1.2}$$

Taking moments about the outer rail:

$$(30 - X)W - HF_L - 60(2R_2) = 0$$

$$120R_2 = \left(1 - \frac{1}{1.2}\right)(30 - X)W$$

Taking moments about the inner rail

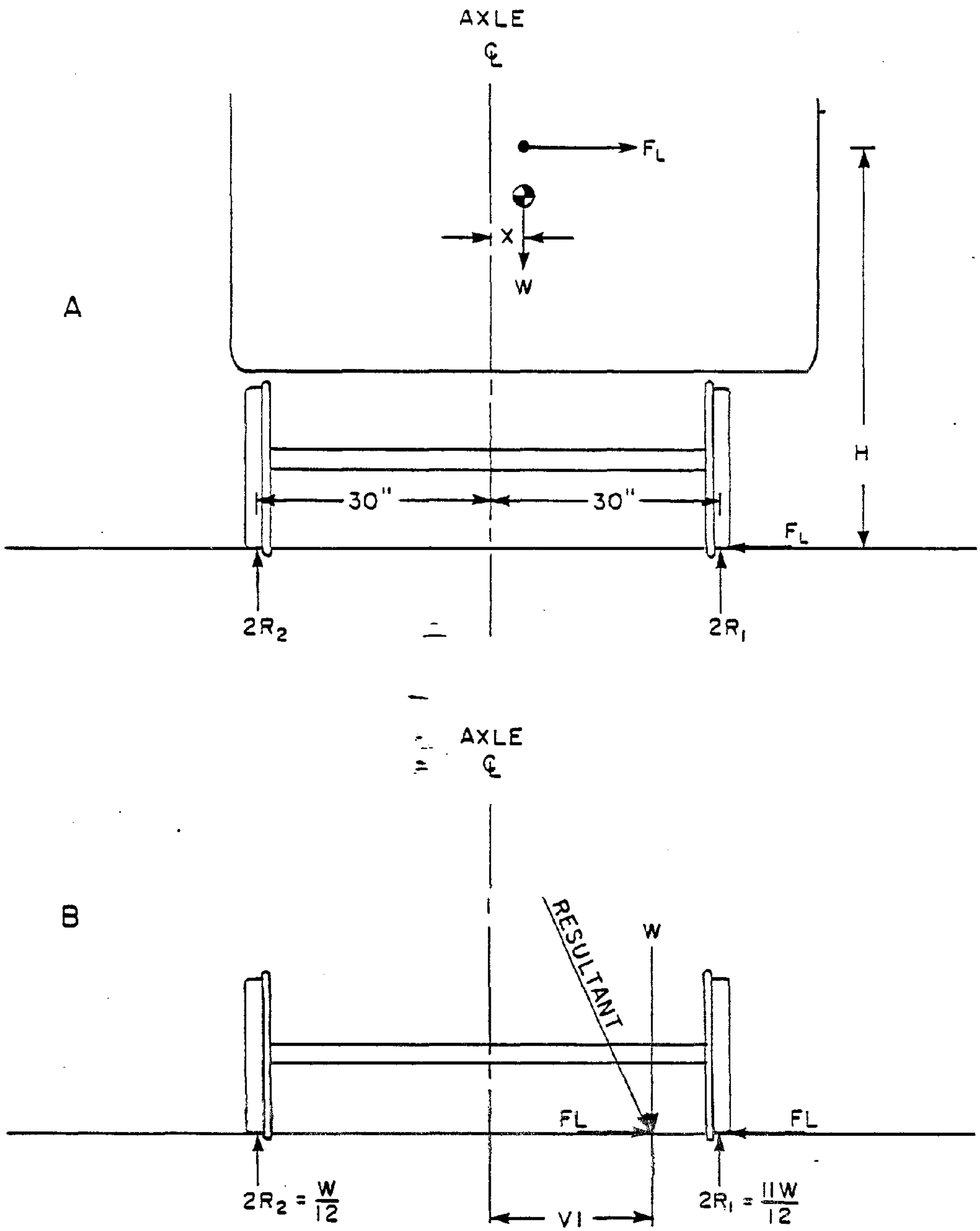


Figure B-2. Freebody Diagram, Overturning Moment Criteria

$$-(30 - X)W - HF_L + 60(2R_1) = 0$$

$$120R_1 = (1 + \frac{1}{1.2})\ (30 - X)W$$

Therefore:

$$\frac{R_2}{R_1} = \frac{0.2}{2.2} = \frac{1}{11}$$

$$W = 2R_1 + 2R_2 = 24R_2$$

$$2R_2 = W/12$$

$$2R_1 = 11W/12$$

Figure B-2B shows the resultant of W and F_L piercing the plane of the railheads at the vehicle balance point. The weight vector intercept is dimension VI. Taking moments about the balance point:

$$(30 - VI)2R_1 - (30 + VI)2R_2 = 0$$

$$(30 - VI)\frac{11W}{12} = (30 + VI)\frac{W}{12}$$

$$300 = 12VI$$

$$VI = 25"$$

The total weight transfer ratio expressed in terms of weight vector intercept for the overturning moment factor of safety criterion is 25 inches.

The amount of the 25 inch total dedicated to wind allowance will be calculated in the next section.

3. VERTICAL WHEEL LOAD REDUCTION RATIO (Japanese National Railway)

The overturning criteria used by JNR (Ref. 5) measures side to side weight transfer in terms of the percent reduction in the vertical load on the low rail wheels. The criteria specifies two levels of load transfer. A reduction in wheel load by 60% of the nominal (40% remaining) is permitted for steady state curving which includes the effect of wind speed and centrifugal acceleration (mV^2/r term), while an 80% reduction in low rail wheel load is appropriate for comparison to transient calculations or measurements. The most recent publication (6) of the Japan Railway Technical Service emphasizes the transient criteria, and the transient calculation is performed by adding to the steady state load transfer a factor to account for the effect of only the part of the maximum lateral acceleration in excess of the centrifugal acceleration in the curve body. The transient component in the JNR calculations is essentially the $-m\ddot{r}$ term in Figure B-1 computed for entry and exit spiral shapes. It is significant that the transient overturning computations do not include effects of alignment deviations (also manifested by the $-m\ddot{r}$ term) which can be a significant component of actual transient overturning measurements. Comparison of measured data, which included the effect of track perturbations, to the criteria is therefore more conservative than judgements based on the usual computation.

Wheel load reduction ratio may be expressed easily in terms of weight vector intercept to allow convenient comparisons to other overturning criteria and to the measurements made in this program. If R_1 is the high rail wheel load and R_2 the low rail wheel load as shown in Figure B-1, the wheel load reduction ratio, C_r is:

$$Cr = \frac{\Delta P}{P} \times 100\% = \left[\frac{\frac{R_1 + R_2}{2} - R_2}{\frac{R_1 + R_2}{2}}\right] \times 100\% = \frac{R_1 - R_2}{R_1 + R_2} \times 100\%$$

And the weight vector intercept, VI, is:

$$VI = 30 \text{ inches} \left(\frac{2R_1 - 2R_2}{2R_1 + 2R_2}\right) = 30\left(\frac{R_1 - R_2}{R_1 + R_2}\right)$$

$$VI = \frac{30\ C_r}{100\%} \text{ inches}$$

A load reduction ratio of 80% corresponds to 24 inches of weight vector intercept.

The effect of wind force can be computed in terms of load reduction ratio or weight vector crossing to determine the portion of the load transfer allowed by either the moment factor of safety or load reduction ratio criteria due to the specified maximum wind speed.

The wind force, Fw, for half vehicle model is:

$$F_w = \frac{S}{2} \frac{\rho V^2}{2} C_d$$

where:

S = the lateral surface area of the whole vehicle, ft^2

ρ = air density of .002378 $slug/ft^3$

V = speed in ft/sec

C_d = drag coefficient

Under the usual assumption that $C_d = 1$

$$F_w = 1.28 \times 10^{-3}\ SV^2 \text{ for V in mph}$$

The change in side to side load transfer that results from an overturning moment, M_o about the origin in Figure B-3 can be computed in general terms. Considering only M_o and summing moments about the origin.

$$0 = 2R_1\left(\frac{\ell}{2}\right) - 2R_2\left(\frac{\ell}{2}\right) - M_o$$

$$R_1 - R_2 = \frac{M_o}{\ell}$$

however the half vehicle weight $W = 2(R_1 + R_2)$

$$\frac{\Delta P}{P} = \frac{R_1 - R_2}{R_1 + R_2} = \frac{2M_o}{\ell W}$$

and

$$VI = \frac{60M_o}{\ell W}.$$

The overturning moment due to the wind load is

$$Mo = Fw(h_{cp})\cos\theta$$

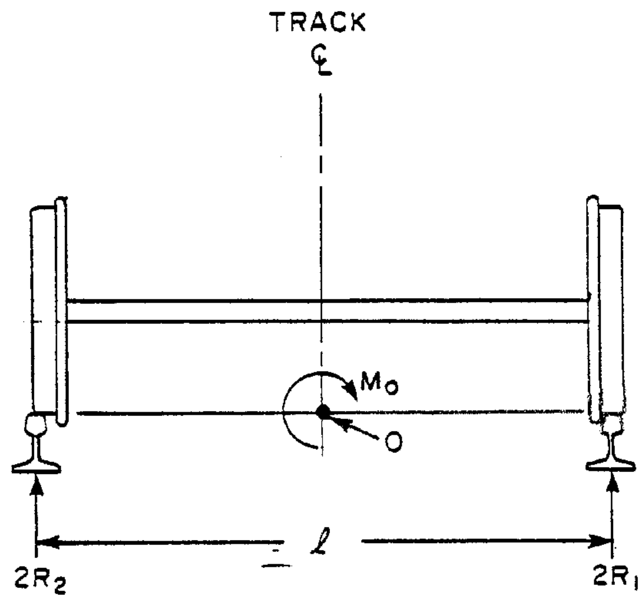


Figure B-3. Freebody Diagram for Weight Transfer Due to General Overturning Moment where h_{cp} is the height of the center of wind pressure in feet and θ is the crosslevel angle.

Since $\cos \theta \approx 1$ and $\ell = 5$ feet, the effect of wind in terms of load reduction ratio is:

$$Cr = \frac{2Fw(h_{cp})}{5W} = (0.51 \times 10^{-3} V^2 S(h_{cp})/W) \times 100\%$$

and in terms of weight vector intercept

$$VI = 0.0153\ V^2 S(h_{cp})/W \text{ with V in mph}$$

3a. Steady State Criteria

Reference 6 identifies the two sources of steady state load transfer as "excessive centrifugal force" and wind force. It gives the formula for "excessive centrifugal force", F_B as:

$$F_B = W_B \quad \frac{V^2}{127r} - \frac{s}{\ell}$$

where:

W_B = weight of half carbody

V = velocity of vehicle, KM/h

r = radius of curve, meters

s = superelevation

ℓ = effective tread gage

The JNR criteria apparently neglects the mass of the truck in the computation of the net steady state lateral force parallel to the railhead plane (centrifugal minus gravitational component). Although it is not stated explicitly in Reference 6, F_B causes a wheel load reduction by setting up two moments about the origin as in Figure B-4.

The direct lateral force moment is $F_B h_{GB}$, where h_{GB} is the height of the body c.g. above the railhead. The second moment, due to

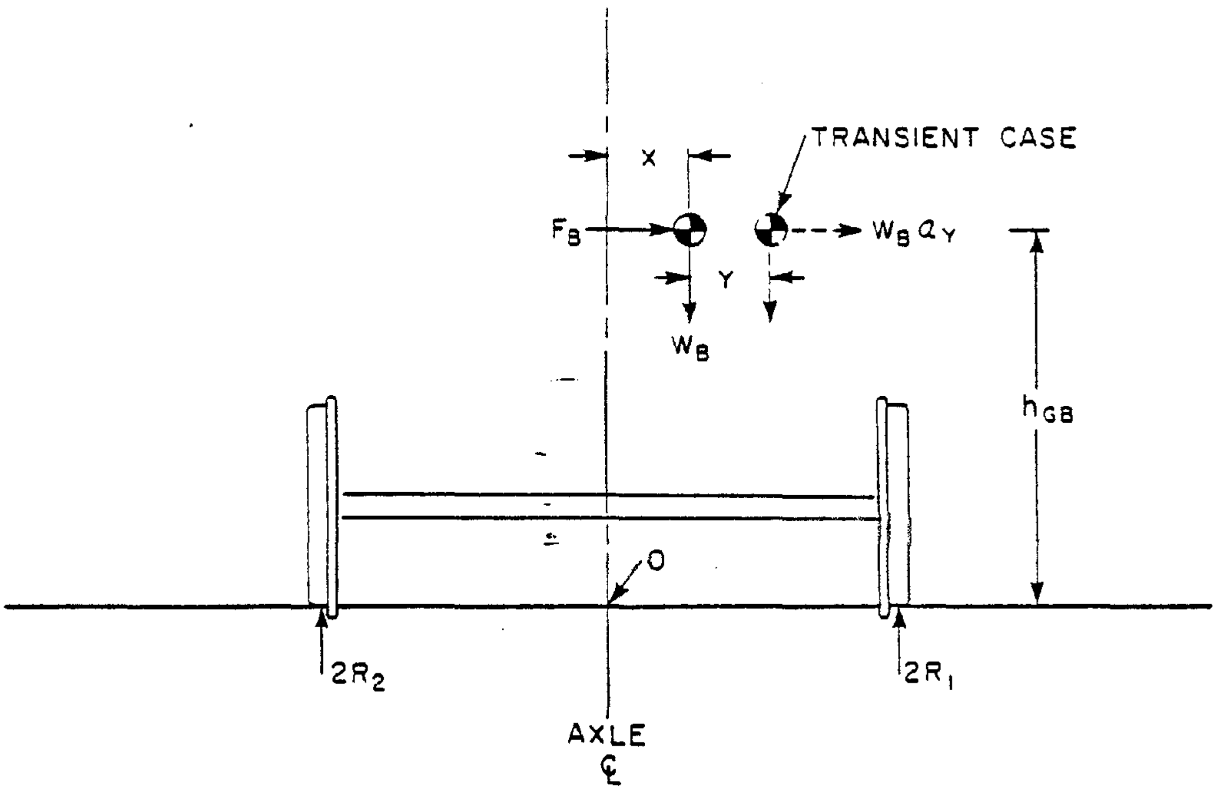


Figure B-4. Free Body Diagram, Vertical Load Reduction Ratio Criteria the lateral shift of the body c.g., is $X \cdot W_B$, where X is the lateral shift. Reference 6 uses a term C_y as the suspension lateral compliance in units of displacement per force. Presumably C_y accounts for lateral movement by both translation and rotation. Therefore, $X = C_yF_B$.

The wheel load reduction ratio, $\Delta P_1/P$, due to "excessive centrifugal force is:

$$\frac{\Delta P_1}{P} = \frac{2Mo}{\ell W} = \frac{2F_B(h_{GB} + C_yW_B)}{\ell W} = \frac{0.4F_B(h_{GB} + C_yW_B)}{W}$$

As derived previously the wheel load reduction ratio, $\Delta P_2/P$, due to a lateral wind of velocity, V:

$$\frac{\Delta P_2}{P} = 0.51 \times 10^{-3}V^2Sh_{cp}/W$$

The JNR steady state overturning safety criteria may be summarized:

$$.6 > \frac{\Delta P_1 + \Delta P_2}{P} = .4F_B(h_{GB} + C_yW_B) + .51 \times 10^{-3}V^2Sh_{cp}/W$$

where

F_B = net lateral centrifugal and gravitational force, lbs

h_{GB} = height of body c.g., ft

C_y = overall lateral compliance ft/lb

W_B = weight of half body, lbs

V = allowed wind velocity, mph

S = side area of whole carbody, ft^2

h_{cp} = height of center of wind pressure, ft

W = weight of half vehicle, lbs

It may be stated in terms of weight vector intercept as

$$18 \text{ inches} \geq [12F_B(h_{GB} + C_yW_B) + .0153V^2Sh_{cp}/W]$$

3b. Transient Criteria

Reference (6) adds a third source of load transfer to the steady state low rail wheel load reduction for an analytic model of transient wheel load reduction. The resulting wheel load reduction ratio is compared to the criteria maximum of 80% to assess operating safety regarding overturning.

The transient component of load transfer is attributed to "vibration of the carbody." The term apparently refers to the difference between the instantaneous maximum lateral acceleration in a spiral $(mV^2/r_s - m\ddot{r}_s - \sin\theta_s)$ and the steady state lateral acceleration in the curve body $(mV^2/r_c - \sin\theta_c)$. It is purely a function of spiral length, curve body radii, superelevation and speed under the assumption of perfect track geometry. The formula

$$\Delta P_3 = \frac{(h_{GB} + C_yW_B)W_Ba_y}{2\ell}$$

is given for the absolute load transfer in units of force.

This expression can be derived considering a_y as lateral acceleration in excess of the steady state value and superimposing its effects on the steady state equilibrium condition in Figure 5-4.

Summing moments about 0 for the transient effects

$$0 = 2\Delta R_1(\ell/2) - 2\Delta R_2(\ell/2) - W_Ba_yh_{GB} - yW_B$$

but

$$\Delta R_1 = \Delta P_3 \text{ and } \Delta R_2 = -\Delta P_3$$

and

$$y = C_yW_Ba_y$$

$$0 = \Delta P_3\ell - (-\Delta P_3)\ell - W_Ba_yh_{GB} - (C_yW_Ba_y)W_B$$

and

$$\Delta P_3 = \frac{(h_{GB} + C_y W_B) W_B a_y}{2\ell}$$

The additional load reduction ratio term in decimal form is:

$$\frac{\Delta P_3}{P} = \frac{\Delta P_3}{W/4} = \frac{2(h_{GB} + C_y W_B) W_B a_y}{\ell W}$$

since ℓ = 5 feet

$$\frac{\Delta P_3}{P} = \frac{.4(h_{GB} + C_y W_B) W_B a_y}{W}$$

The JNR transient overturning safety criteria may be summarized in vertical wheel load reduction ratio:

$$.8 \geq \frac{\Delta P_1 + \Delta P_2 + \Delta P_3}{P} = [.4(F_B + W_B a_y)(h_{GB} + C_y W_B) + .51 \times 10^{-3} V^2 S(hcp)]/W$$

or in terms of weight vector intercept:

$$24 \text{ inches} \geq [12(F_B + W_B a_y)(h_{GB} + C_y W_B) + .0153 V^2 S(hcp)]/W$$

B .1.2 COMPARISON OF OVERTURNING CRITERIA

The net weight vector intercept specified by each criteria for a particular lateral wind speed may be obtained by subtracting $.0153\ V^2 Sh_{cp}/W$ from the maximum weight vector intercept. The net weight vector intercept includes the effects of inertial and gravitational forces and is appropriate for comparison to test data or the results of mathematical modeling. Figures B-5 and B-6 compare the various overturning criteria as applied to the LRC coach and locomotive by plotting the net weight vector intercept as a function of lateral wind speed allowance. The characteristics of the vehicles effecting the wind speed allowance are shown below.

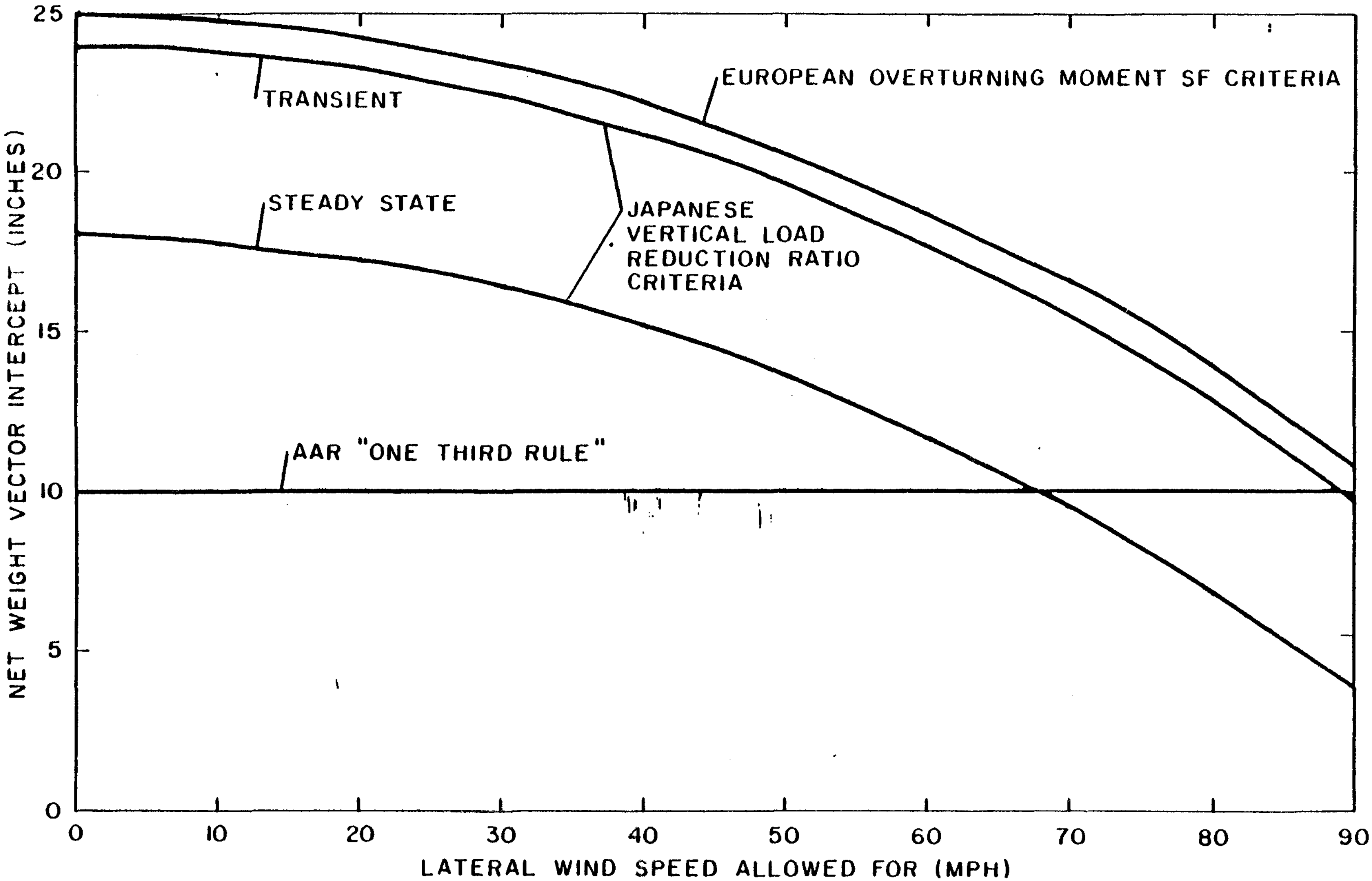


Figure B-5. Comparison of Various Overturning Safety Criteria for LRC Coach

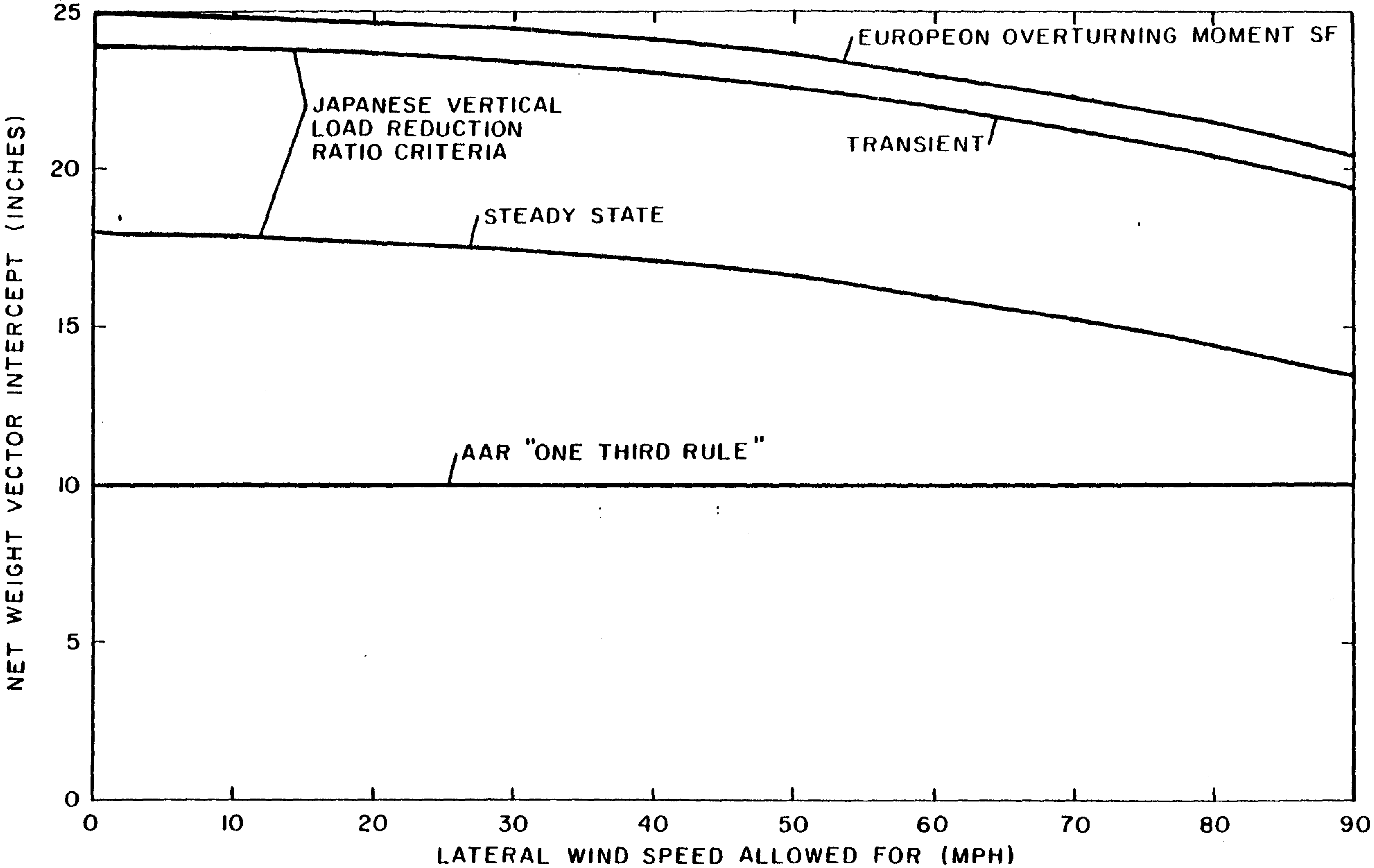


Figure B-6. Comparison of Various Overturning Safety Criteria for LRC Locomotive

	LRC Coach	LRC Locomotive
Body Length	85 ft	62 ft
Height of Body Side	11 ft	11 ft
Lateral Area, S	935 ft^2	682 ft^2
Height of Center of Pressure, h_{cp}	6-1/2 ft	6-1/2 ft
One-Half Vehicle Weight, W	52,750 lb	125,400 lb

Comparing Figures B-5 and B-6 reveals that the wind speed allowance can restrict the net weight vector intercept significantly for coaches without greatly limiting locomotives. A prudent choice of the allowance for wind speed is required because an overly conservative wind speed assumption wastefully reduces the normal operating criteria since other factors such as visibility and debris on the track limit track speed in high wind. Maximum operating wind speeds of 68 mph and 76 mph are assumed by German and British railroads, respectively whereas the Japanese compute the transient wheel load reduction ratio based on a 45 mph lateral wind. The maximum ten year mean recurrent wind speed at less than 15 feet altitude is 56 mph for cities along the Northeast Corridor. The assumption of normal operation in winds of 68 to 76 mph appears overly conservative.

The European overturning moment safety factor criteria is by far the least restrictive if it is interpreted as pertaining to quasistatic moments. It is slightly greater than the JNR transient criteria. The one third rule is much more restrictive than even the JNR steady state criteria for the LRC locomotive and for the LRC coach at wind speeds less than 67 mph. The one third rule would be more comparable to other criteria under moderate winds if unloaded box cars were under consideration.

B.2 WHEEL CLIMB CRITERIA

The classic characterization of wheel climb by Nadal in 1896 predicts that the critical ratio of lateral to vertical force for a single wheel is $L/V = (\tan \alpha \pm \mu)/(1 \pm \mu \tan \alpha)$ where α is angle of the wheel flange with respect to the horizontal at the point of contact with the side of the rail, and μ is the coefficient of friction between wheel and rail. Nadal's formula does not directly address several first order wheel climb factors including wheel/rail angle of attack critical time duration of the derailment quotient nor any of the reported second order factors such as absolute vertical load, vertical and lateral velocity at impact, wheelset mass, rail head contour or torsional and lateral track stiffness. The choice of μ is also controversial.

Many of the second order factors such as wheelset mass, lateral velocity and track stiffness should manifest themselves as components in the instantaneous L/V measurement. Railhead contour can be viewed as a modifier to the flange angle and forward velocity would seem to be related to the allowed time duration of the derailment quotient. Criteria which specify the critical L/V ratio as a function of time duration appear to address implicitly all the factors except absolute vertical load.

Yokose (Ref. 7) offers an interpretation of Nadal's formula as follows:

For positive angles of attack:

$$\text{Critical L/V} = \frac{\tan\alpha - \mu_e}{1 + \mu_e \tan\alpha}$$

For negative angles of attack:

$$\text{Critical L/V} = \frac{\tan\alpha + \mu_e}{1 - \mu_e \tan\alpha}$$

and for zero angle of attack:

$$\text{Critical L/V} = \tan\alpha$$

where μ_e is the effective coefficient of friction which converges to the static coefficient of friction μ as the angle of attack increases. This interpretation corresponds with the intuitive notion that flange friction promotes wheel climb at positive angles of attack and hinders it at negative angles. Reference 7 also presents laboratory test data obtained with scale model wheelsets which converge to the Nadal predictions ($\alpha = 61^{\circ}$, $\mu = .4$) for large angles of attack. In accordance with this study, the Japanese National Railway limits L/V to .8 for durations of 50 ms or greater. Another JNR researcher (Matsudaira, Ref. 8) recommends an L/V limit of 4 at 10 ms duration decreasing to .8 at 50 ms. JNR also recommends (Ref. 5) a maximum lateral impact speed of 1.6 ms at 6-ton static wheel loads in cases of hunting or severe alignment deviations.

Kaffman recommends (Ref. 9) a maximum L/V of 1.2 for British four-wheel "wagons" where the effect of track twist on long wheel base vehicles generates high L/V ratios by vertical force reduction as well as lateral force application. The recommendation apparently results from Nadal's formula also but with $\alpha = 68^{\circ}$ and $\mu = .33$. He cites test data in reference 10 of four-wheel cars sustaining L/V = 1.6 and bogie vehicles at L/V = 2.35 without wheel climb. The time duration is referred to as instantaneous without quantitative definition.

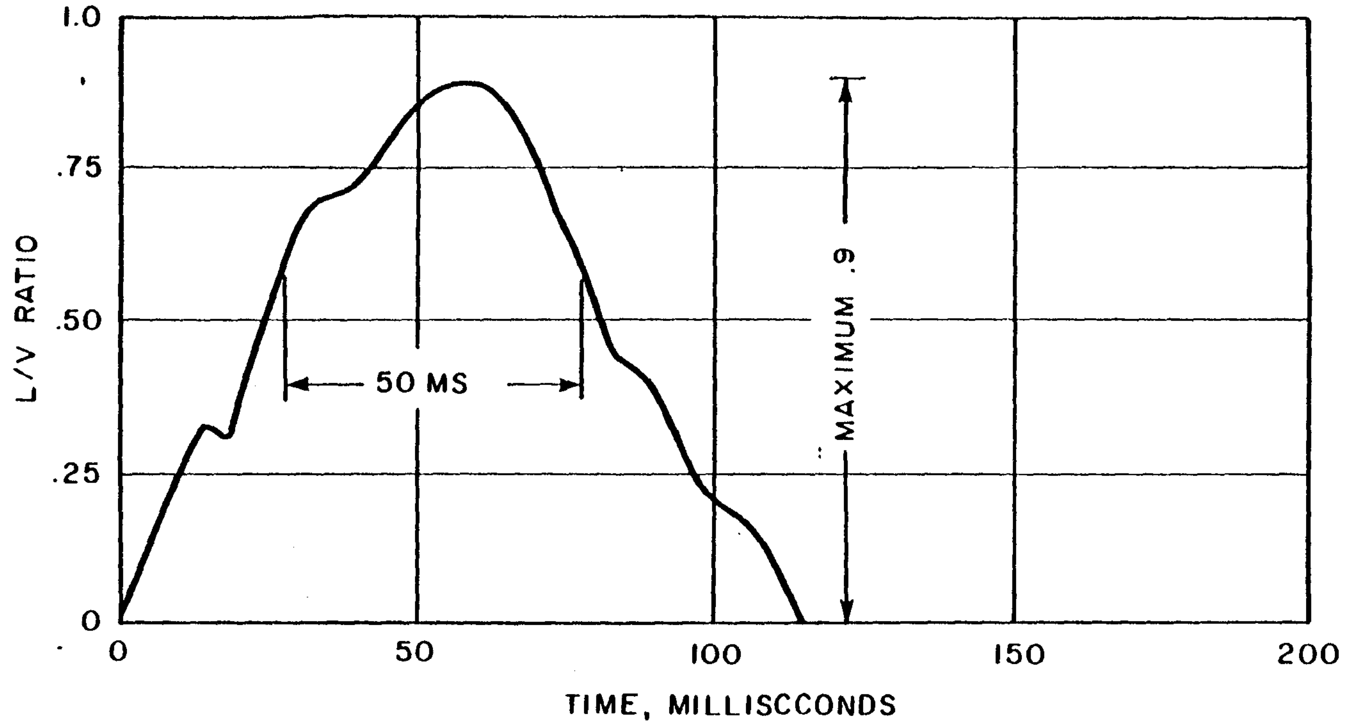
Amtrak has included in its specifications for the AEM-7 locomotive (AAR flange angle $66^{\circ} - 68^{\circ}$, increasing further with wear) an L/V criteria which clearly relates its permissable magnitude to pulse duration. The method of defining pulse duration in this specification has been described by EMD (Ref. 11). EMD interprets the pulse width as the time that L/V exceeds the criteria threshold rather than the time L/V exceed zero during a pulse peaking at the criteria. Figure B-7 illustrates the difference between the EMD and JNR interpretation of an L/V spike. The EMD definition is easier to apply to the usual test data pattern in which short duration spikes are superimposed over a steady state curving level. The maximum L/V ratio recommended by EMD is:

$$(L/V)\ \max < 0.056T^{-0.927}$$

with $(L/V)\ \max \leq .90$ at $T > 50$ ms

Dean and Ahlbeck (Ref. 12) recommend a maximum L/V of 1.0 for durations greater than 50 ms as a conservative limit supported by the results of tests by the European ORE Committee B55.

Figure B-8 compares the various recommended criteria. The JNR criteria is the most restrictive because of its interpretation of L/V measurements. All of the criteria represent judgements based in part on Nadal's formula. The judgement of μ greatly influences the predicted critical L/V. Rule of thumb estimates of μ have usually placed it between .2 and .3, but ORE Committee B10 has reported (Ref. 10) measurements of much higher wheel/rail friction coefficients. The effective lateral coefficient of friction μ_e converges on μ as the angle of attack becomes very



THE ABOVE L/V PULSE WOULD BE CLASSIFIED BY JNR AS A PULSE OF .9 FOR 115MS EXCEEDING THEIR CRITERIA. HOWEVER THE SAME PULSE WOULD BY CLASSIFIED BY EMD AS A PULSE EXCEEDING .6 FOR 50MS WELL WITHIN THEIR CRITERIA.

Figure B-7. Interpretation of Instantaneous L/V Measurements

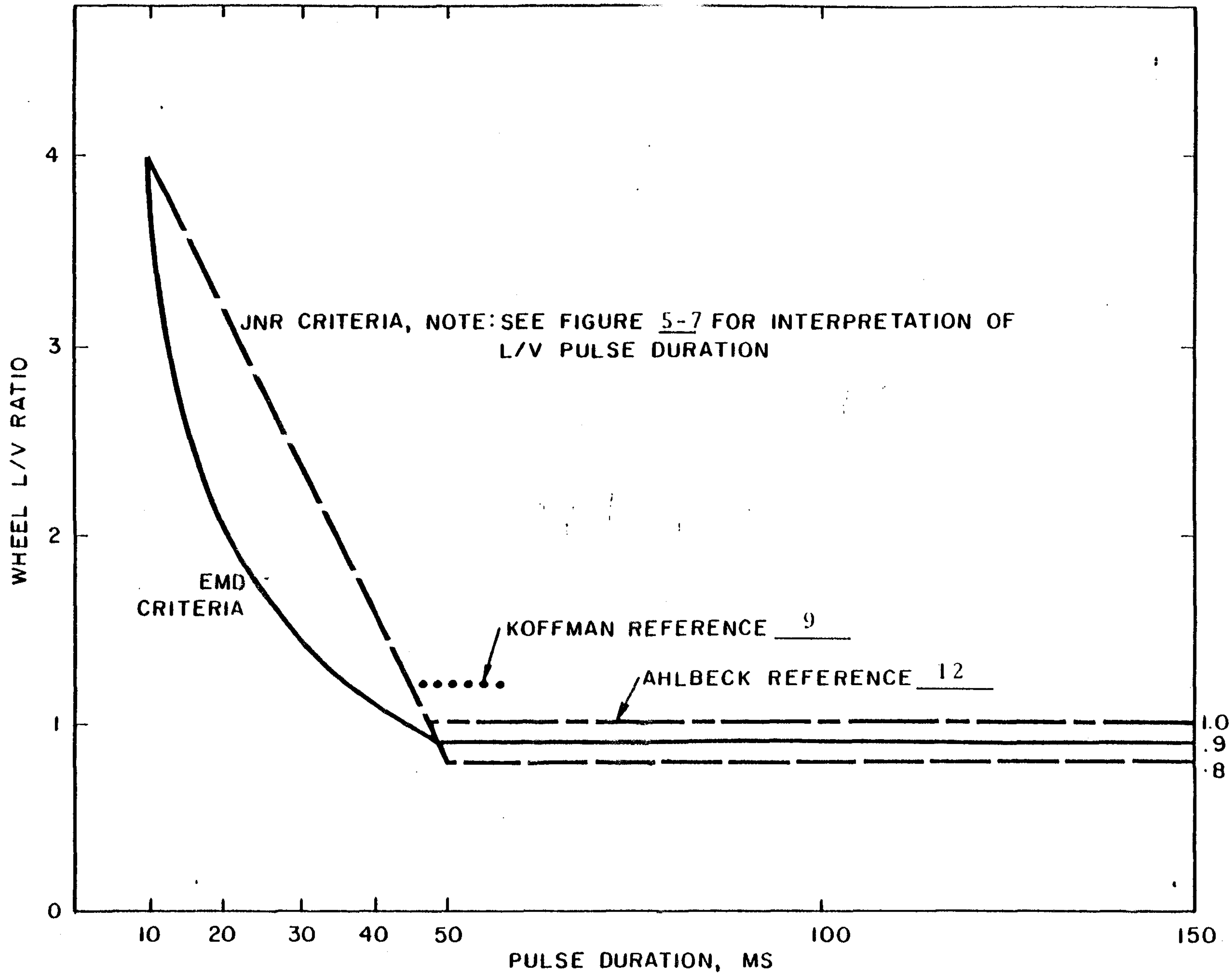


Figure B-8. Comparison of Various Wheel Climb Criteria large and the speed becomes very low but it should remain considerably less than the maximum reported values of μ of .55 for any conceivable high cant deficiency conditions. It also is in agreement with Nadal's formula that the most conservative L/V criteria was proposed by a railroad (JNR) using wheels with the lowest flange angle ($\alpha = 61^{o}$).

Experiments by BR using an instrumented full scale wheelset on a mobile test bed with controlled loads and angles of attack (Ref. 14) had indicated much higher L/V ratios at derailment than commonly expected. It has been hypothesized that very high longitudinal creep forces under derailment conditions reduce the lateral friction forces because the vector sum of longitudinal and lateral frictional forces is limited by μP. The concept of μ_e was an inexact way of describing the same phenomenon. Further evidence that the JNR criteria is overly conservative is recent testing of a low c.g. subway vehicle involving this author that indicates routine curving with peak L/V ratios exceeding 0.8. However, the time durations associated with critical L/V pulses have not been determined emperically at this time and even very specific transient L/V criteria such as that used by Amtrak is a product of judgement rather than testing.

B.3 RAIL ROLLOVER CRITERIA

Derailment is likely to occur more rapidly by rail rollover than by other hazards because the inertia of the rail opposing rotation is so slight. The knowledge of instantaneous conditions favoring rail rollover is especially important in assessing risk. Japanese and European papers covering other safety considerations (Ref. 5 and 14) appropriate to high speed curving do not offer rail rollover criteria, but a series of criteria based on various degrees of track structural integrity has been developed from AAR studies (Ref. 15 and 16). The instantaneous ratio of the sum of lateral forces to the sum of vertical forces of the wheels on the high rail side of a truck is used to quantify the likelyhood of rail rollover. It is known as the truck L/V ratio and is referred to in the data appendix by the more descriptive term of high rail side truck L/V ratio.

A totally unrestrained rail can sustain lateral forces without rollover as long as the resultant of the lateral and vertical wheel forces intersects a point within the base of the rail. The limit of the purely geometrical resistance to rollover as shown in Figure B-9 has been stated conservatively as L/V = 0.5. However even if the truck side L/V is less than 0.5, the lead wheel must be scrutinized separately if no fastener resistance is to be assumed. If the rail is held flat at the trailing wheel and the torsional rigidity of the rail is considered as rollover resistance at the front wheel, the front wheel lateral force cannot be greater by more than about 2,300 pounds over that permitted by

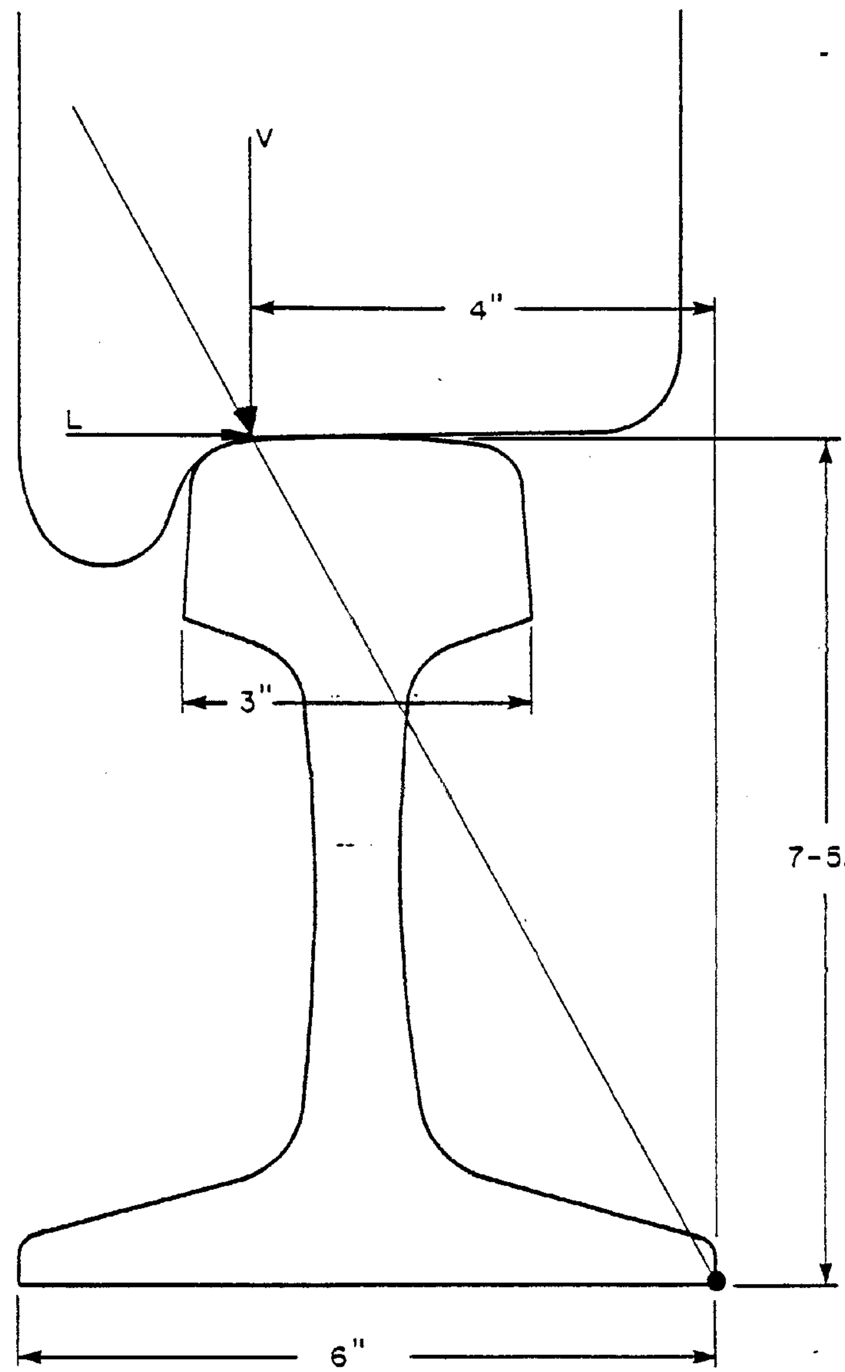


Figure B-9. Cross Section Geometric Resistence to Rail Rollover the rail cross section geometry alone without gage widening in excess of 1/4-inch (Ref. 16).

AAR reports that newly spiked wood tie fasteners can sustain a lateral force of 3,600 pounds and 8,000 pounds can be sustained at the railhead with concrete tie fasteners. This has been translated into a truck L/V limit of 0.5 + 3,600 lb/Pw (Ref. 1). Such a truck L/V limit appears to be erroneous because the instantaneous truck L/V will be calculated using a V greater than Pw because of load transfer while 3,600 pounds is actually an absolute number independent of the vertical load.

The torsional rigidity of the rail allows fasteners other than those at the wheels to contribute resistance to rail rollover. An additional 20,000 pounds of lateral force over the geometric limit can be sustained by newly spiked fasteners in a vicinity of up to seven ties with less than 1/4-inch of gage widening. This has been expressed as a truck L/V limit of 0.5 + 20,000 lb/Pw which also appears to overstate the value of the absolute lateral force.

The rail rollover criteria has several deficiencies. The geometric resistance to rollover appears to be overly conservative. Examination of the wheel and rail contact pattern during calibration of the instrumented wheelsets even at low lateral forces suggests the geometry pictured in Figure B-9 which results in an allowable L/V of 0.55. Flange contact would result in even more favorable geometry. Dean and Ahlbeck (Ref. 12) recommend 0.55 assuming no excessive wear.

It is well known that the spikes loosen quickly, and one is hesitant to base the rail rollover criteria on the additional lateral force sustainable by newly installed spikes (in addition to the inconsistent translation of lateral force to truck L/V between references). However, simulated revenue service test runs (Ref. 1) commonly exceed the truck L/V limited by cross section geometry and torsion alone. A realistic assessment of the rollover resistance to be expected from loosened fasteners as well as a clear definition of the time duration is necessary for a comprehensive rail rollover criteria. Empirical information concerning the critical pulse durations of truck L/V measurements and rollover resistance of loose fasteners were not found in the rail research literature.

A recent AEM-7 locomotive specification describes a rail rollover criteria which is specific in regard to pulse time duration and appears to be consistent with typical experience. The basis for its selection is experienced judgement rather than new data. Figure B-10 plots the criterion which may be stated:

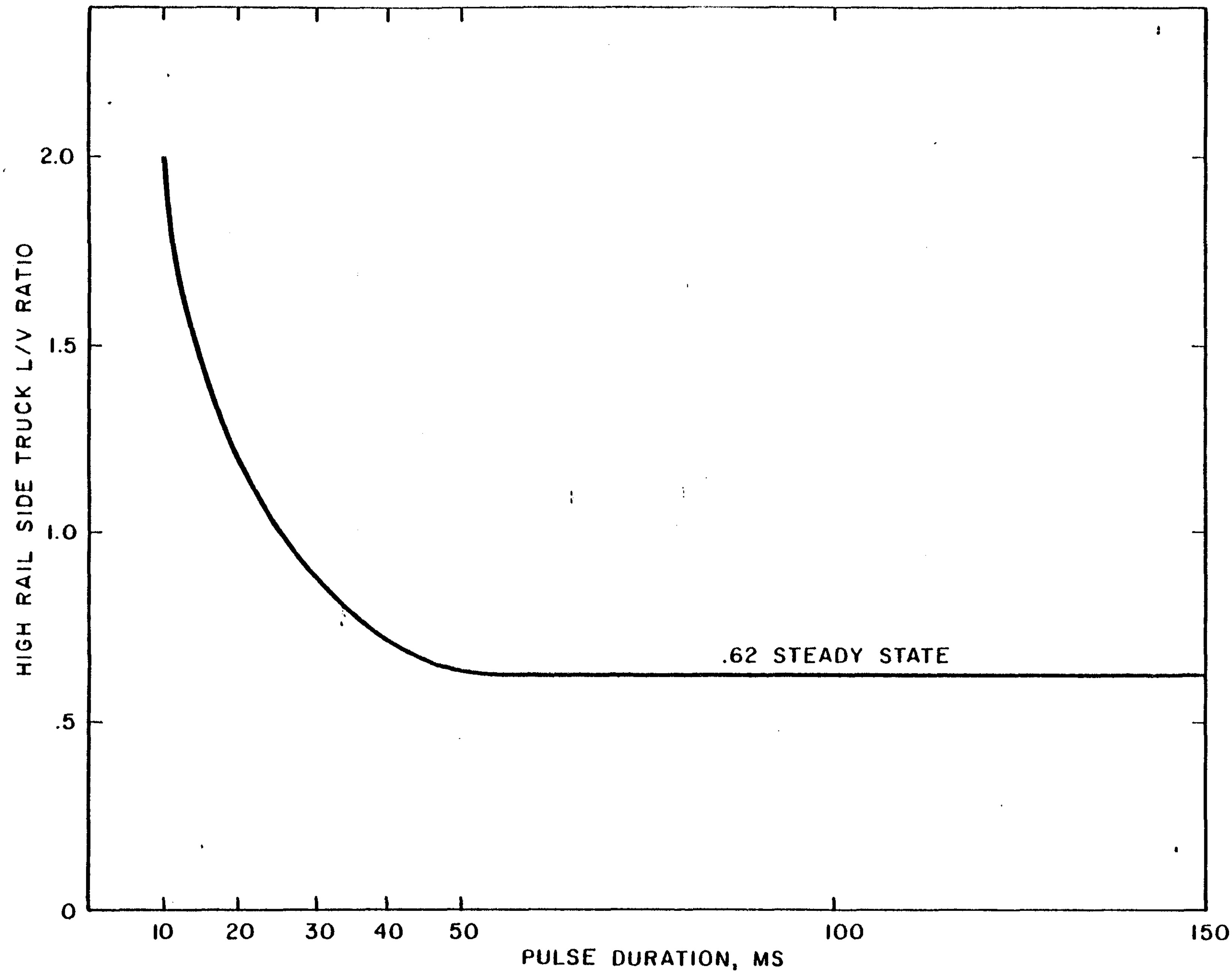


Figure B-10. Rail Rollover Criteria, AEM-7 Specifications

Truck $(L/V) < .070\ T^{-0.728}$

where T equals time duration exceeding limit, and Truck $(L/V) \leq .62$ for T = 50 ms or greater

B.4 LATERAL TRACK SHIFT CRITERIA

The steady state and transient inertial forces and the wind force acting on rail vehicles are transmitted to the ground through the track structure. Criteria have been proposed for limits on the lateral axle load to prevent the permenent lateral movement of the ties relative to the ground. The lateral translation of the rails relative to the ties is assumed to be prevented by friction and by the fasteners, and rail rotation has been considered in the previous set of criteria.

The restraint of the tie by the ballast and the number of ties sharing the burden determine the strength against lateral shift of the track. The interlocking ability of the ballast aggregate, its compaction, depth, width and gradation, the shape, weight and material of the tie and its vertical load determine the ultimate lateral tie resistance. Reference 16 lists test data from various sources that show a range 400 pounds to 1,550 pounds lateral resistance for various unloaded ties in uncompacted ballast and 1,170 pounds to 2,500 pounds in ballast compacted by two million gross tons of traffic. The differences due to the size, shape, and material of the ties appears to be of the same magnitude as the effect of compaction, although differences in the test methodology and ballasting may produce a deceivingly great range. Compaction causes a great increase in the lateral resistance of the unloaded ties and perhaps an even greater increase in the lateral resistance of loaded ties. Reference 16 cites a doubling of the lateral resistance of loaded ties after 100,000 gross tons (metric) of traffic and eventual stabilization at nearly three times the uncompacted resistance after about 1.5 million gross tons.

The distribution of the vehicle lateral forces among the ties depends on the tie spacing and the stiffness of the rail and fasteners. Experiments by SNCF suggest that about seven ties bear the load of a single wheelset with 40 to 60 percent taken by the tie under the wheelset. The advantage of stiff rails and fasteners in tie load distribution is outweighed by the internal track forces which result from tie restraint of continuous welded rails subjected to changes in temperature. These internal forces reduce the tie resistance available to oppose the vehicle forces. Reference 16 presents a reduction factor, Γ, to account for the maximum change in temperature from rail installation ($\Delta\theta$, F^{o}), the rail cross section area (A, in^2) and the curvature (D^{o}):

$$\Gamma = 1 - \frac{A\Delta\theta}{22320}(1 + .458D)$$

The lateral track shift criteria suggested by both Ahlbeck (1) and Lawson (16) are derived from measurements on French track using the "Wagon Derailleur" car (17). This tester features a third axle centrally located which is capable of applying various combinations of vertical and lateral loads while the car is in motion. Lateral loads causing actual permanent track shift were measured under realistic conditions and expressed as a function of vertical axle load for several track conditions. Although the following results were obtained with rail of about 92 lb/yard and tie spacing of 24 inches they apparently represent the most exact findings in the literature.

F_c = .33P + 2,245 pounds for uncompacted wood tie track

F_c = .33P + 4,400 pounds for uncompacted concrete tie track

F_c = .61P + 5,520 pounds for wood tie track compacted by nine million gross tons of traffic

where F_c is the net lateral axle load causing permanent deformation and P is the vertical axle load.

Alhbeck (1) has estimated for the more common 20-inch tie spacing:

F_c = .4P + 2,700 pounds for uncompacted ballast with wood ties

and

F_c = .7P + 6,600 pounds for compacted ballast with wood ties

similarly Lawson (Ref. 16) estimated:

F_c = .66P + 4,490 pounds for compacted ballast with wood ties

$Fc = .66P + 8,800$ pounds for compacted ballast with concrete ties

When these lateral axle forces are compared to zero wind measurements and applied to traffic on continuous welded rail, a wind force allowance and a reduction factor for thermally induced rail forces must be applied. Assuming temporary speed restrictions on new or newly worked track, measurements should be compared to the maximum axle lateral force for compacted ballast as calculated below, following Alhbeck's recommendation for wood ties:

$$Fmax = \left[1 - \frac{A\Delta\theta}{22320}(1 + .458D)\right]\left[.7P + 6,600\right] - (1.28 \times 10^{-3})sv^2$$

where:

A = rail cross section area, in^2

$\Delta\theta$ = max temperature change after rail installation, ^{o}F

D = track curvature, degrees

P = vertical axle load, pounds

S = lateral surface area of vehicle, ft^2

V = lateral wind speed, mph

and it is assumed that a single axle bears half the entire wind load. For typical NEC conditions of 140-pound rail ($A = 13.8$ in^2), $\Delta\theta$ max of $70^{o}F$ and D max of 4^{o}.

$$Fmax = .61P + 5,800 - 1.28 \times 10^{-3}SV^2$$

Figure B-11 shows the maximum lateral axle forces following Albeck's interpretation of the SNCF criteria for the LRC locomotive and LRC coach as a function of the wind speed allowance. The SNCF criteria for uncompacted ballast computed for the AEM-7 locomotive is also compared to the Amtrak procurement specification which assumes uncompacted ballast. Comparison with the high rail lateral wheel force is a conservative practice because a positive angle of attack results in a lateral creep force on the lower rail wheel which opposes the high rail flange force reducing the net lateral axle force.

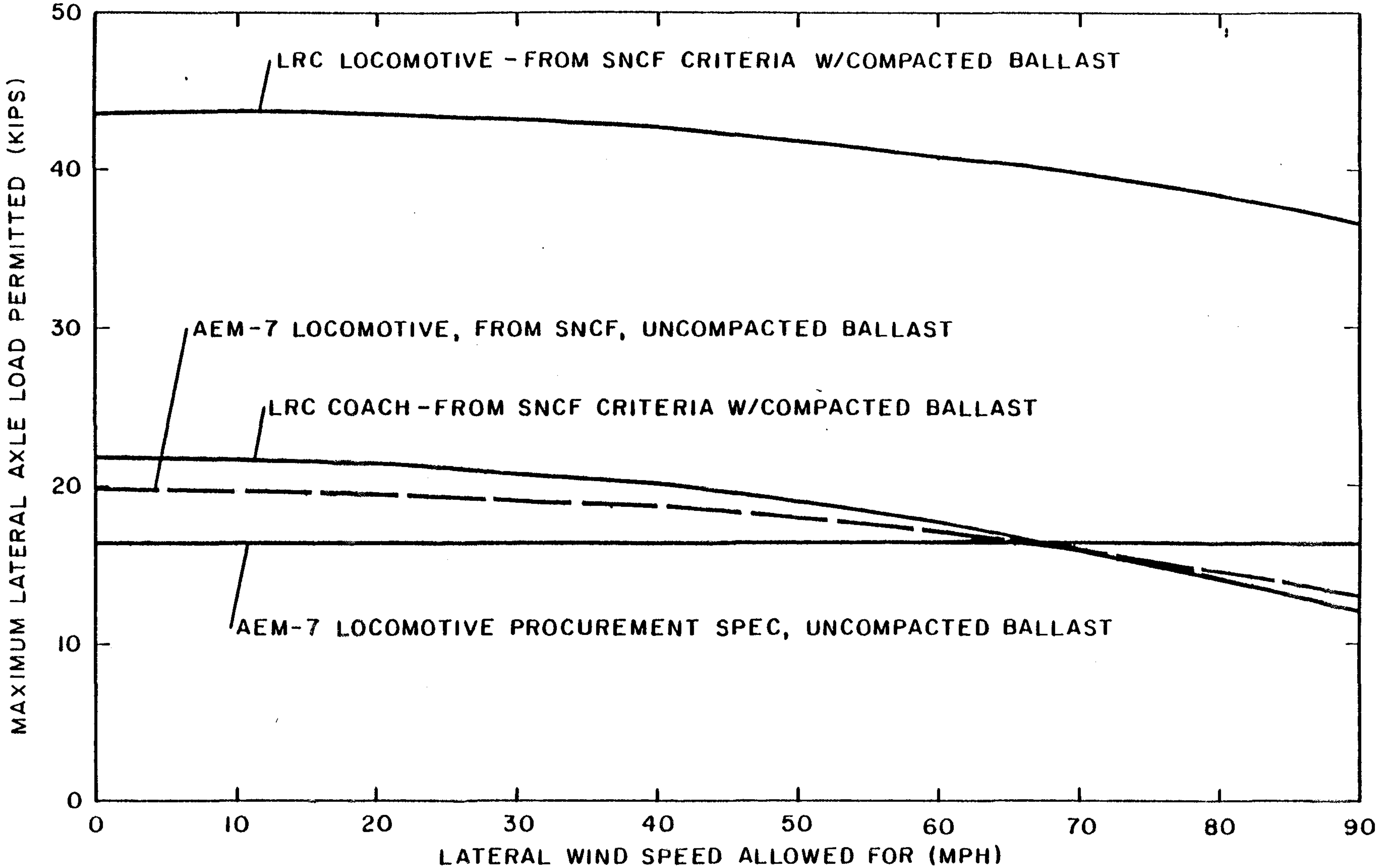


Figure B-11. Comparison of Various Lateral Track Shift Criteria

All of the track resistance measurements quoted were obtained from steady state experiments and presumably the resulting criteria should be compared to steady state rather than transient measurements. The only criteria, however, to specifically address the time duration of the measurement has been Amtrak's AEM-7 procurement specification which requires $F_{max} \leq .85\ (.33P + 2{,}200)$ for $T \geq 50$ms or $X \geq 6$ feet and includes CWR rail on uncompacted ballast. This criteria allows higher lateral axle loads for very short durations in recognition that considerable energy is required to deform the track permanently. However, 50 ms may be overly conservative in this respect because it is more typical of the time duration of a well filtered peak measurement rather than a steady state. This criteria would be more useful if it were defined for running on compacted ballast because high speed curving is normally prohibited on newly worked track.

Recommendations have been made by Battelle for considering the combined effect of several axles of one truck on shifting the track laterally. It has been proposed that: F_{max} (truck) = $.7nF_{max}$ (axle) where n equals the number of axles per truck. Reducing the axle force summation is reasonable because the seven tie influence zones of several axles will overlap.

The specific criteria proposed for U.S. service appears to be judgements by various investigators based mainly on the French experiments. Differences between the French test sites and typical NEC track are not known nor is the variation between places on the NEC. The relationship between time duration and amplitude of destructive lateral axle force pulses was not defined by the French experiments thus the topic has been treated conservatively or not at all.

B.5 RIDE QUALITY CRITERIA

The most obvious factor affecting passenger comfort in high cant deficiency curving is the steady state level of lateral acceleration. It can be determined mathematically by

$$A_y = \frac{V^2}{r} + g\ (\sin\phi - \frac{s}{\ell})$$

and expressing A_y in g's with ϕ a small angle:

$$A_y = (\frac{V^2}{rg} - \frac{s}{\ell}) + \phi = \frac{\text{cant deficiency}}{\ell} + \phi$$

where

V = running speed (ft/sec)

r = curve radius (ft)

g = gravitational acceleration (32.2 ft/sec^2)

s = crosslevel (inches)

ℓ = effective tread gage (60 in)

ϕ = body roll angle, radians

In the case of active or passive tilt body coaches, ϕ can be such that the acceleration of gravity cancels the effect of cant deficiency. The effectiveness of reducing the lateral acceleration by controlling the body roll angle depends on the range and control characteristics of the tilt system.

Figure B-12 presents the results of an often quoted AAR Study (Ref. 20) which related subjective assessment of comfort by observers to objective measurements. The observers were asked to disregard accelerations due to track irregularities and spiral transitions and to concentrate solely on steady state curving. A maximum steady state level of 0.1 g including the effect of body roll angle was recommended as a result of the test program. Lateral accelerations in successive opposite curves have been mentioned as especially harmful to ride comfort (Ref. 5). Table B-1 summarizes the lateral acceleration recommendations by several organizations (many of them quoted in Reference 21) for steady state lateral acceleration and other criteria.

Figure B-13 illustrates the characteristics of the lateral acceleration measured at the car floor during a typical curve negotiation. The average slope of lateral acceleration with respect to time in the spirals is known as "jerk" and it is a prime consideration in the design of transition spirals. AAR (Ref. 20) recommends that spiral lengths be set according to the formula L_{min} = 4.88 V which allows a minimum of 3.3 seconds travel time between tangent track and circular curve. This has been interpreted as a maximum "jerk" specification of .03 g/sec (to .1 g) but the actual rate could be higher due to body roll overshoot. The body roll overshoot is the subject of a comfort criteria used by JNR (Ref. 9). "Transient response diagrams" of "carbody vibration" are included in Reference 6 for ease in applying the $\pm$.08 g maximum specification. These diagrams (Figure 5-14 as an example) are essentially plots modeled throughout a curve of A_y (transient) = $(V^2/rg - s/\ell) + \phi - \ddot{r}/g$ where ϕ is assumed to reach a steady state value sufficient to cancel $(V^2/rg - s/\ell)$. The response time of ϕ is important to apparent "carbody vibration" under this application of the criteria. The concept of "carbody vibration" was applied to a tilt body coach in Reference 6, and

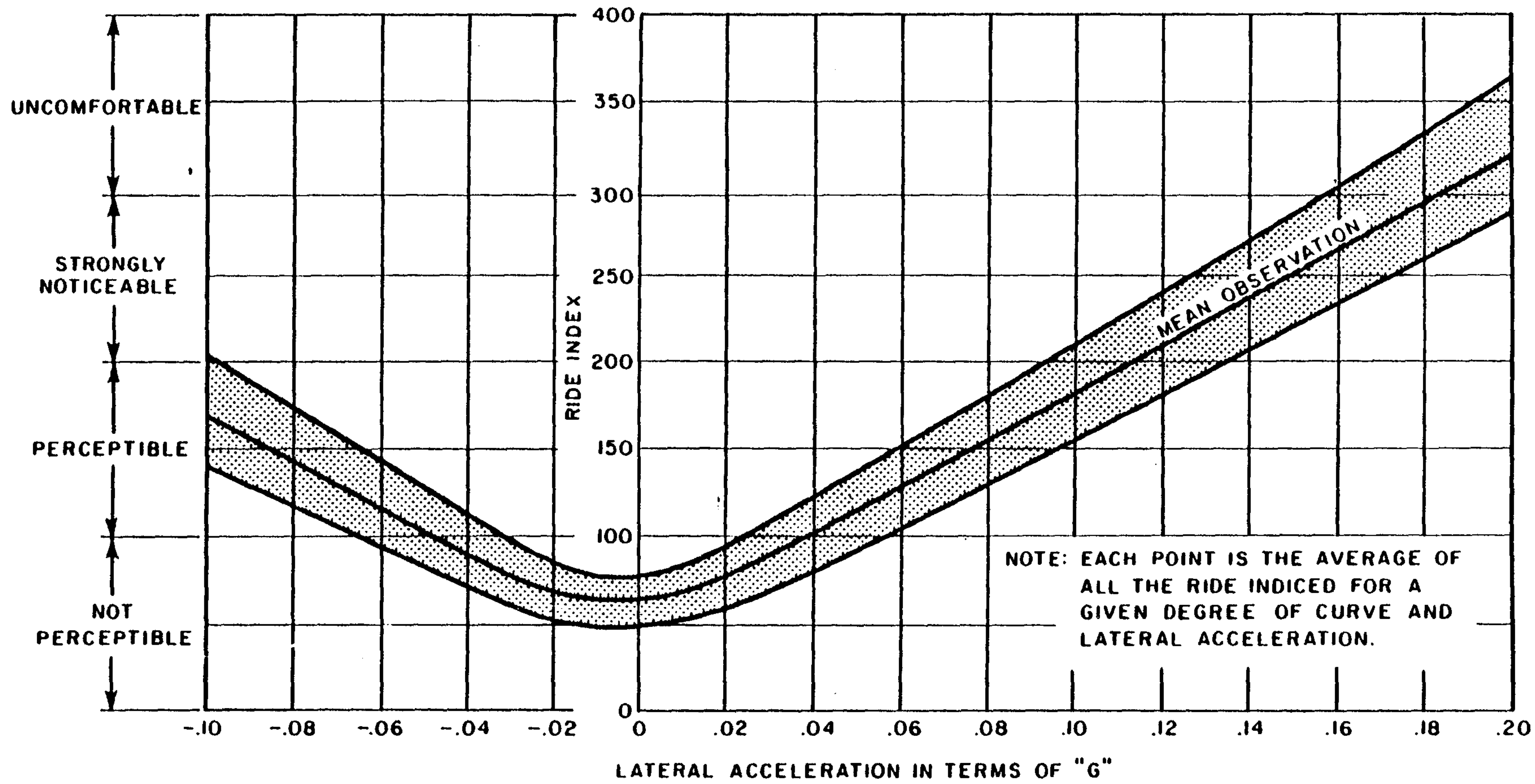


Figure B-12. Relation of Subjective Comfort Rating to Carbody Lateral Acceleration for AAR Experiment

TABLE B-1

LATERAL ACCELERATION RIDE COMFORT CRITERIA

Source		Maximum Steady State, g	Maximum "Jerk", g/sec	Maximum Carbody Vibration, g
BR		0.074	0.042	-
Koffman (ref 14)	General European practice	0.066 + body roll	-	-
	Newer European limits	0.087 + body roll	-	-
	Recommended max for locomotives	0.160 + body roll	-	-
AAR (ref 20)		0.1	0.03	
JNR (ref 5)		0.08	0.03	±.08
OHSGT		0.08	0.03	-
SNCF		0.15	0.1	-
DB		.066 (< 124 mph) .031 (< 186 mph)	- -	- -

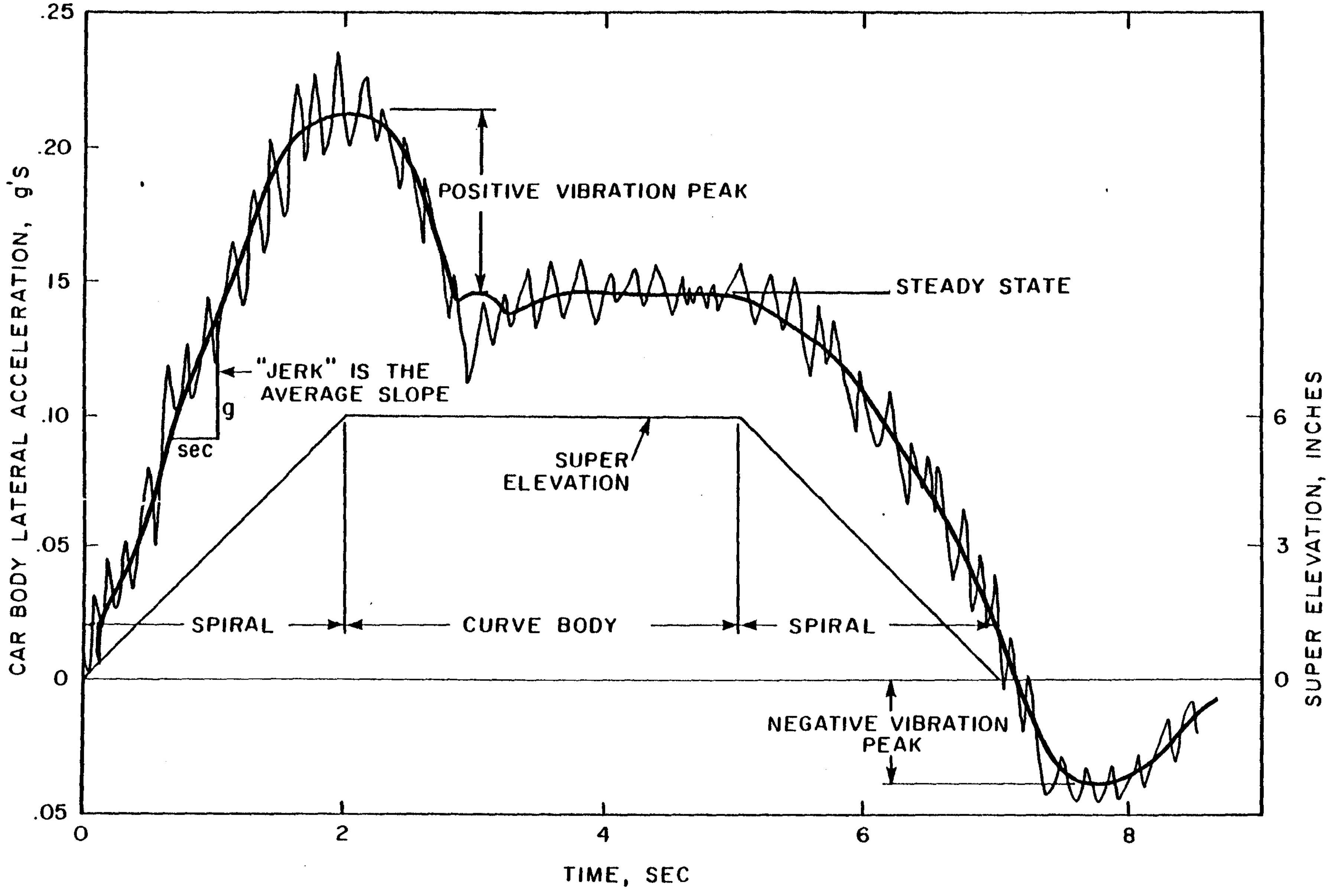


Figure B-13. Features of Carbody Lateral Acceleration Referenced to Location on Curve

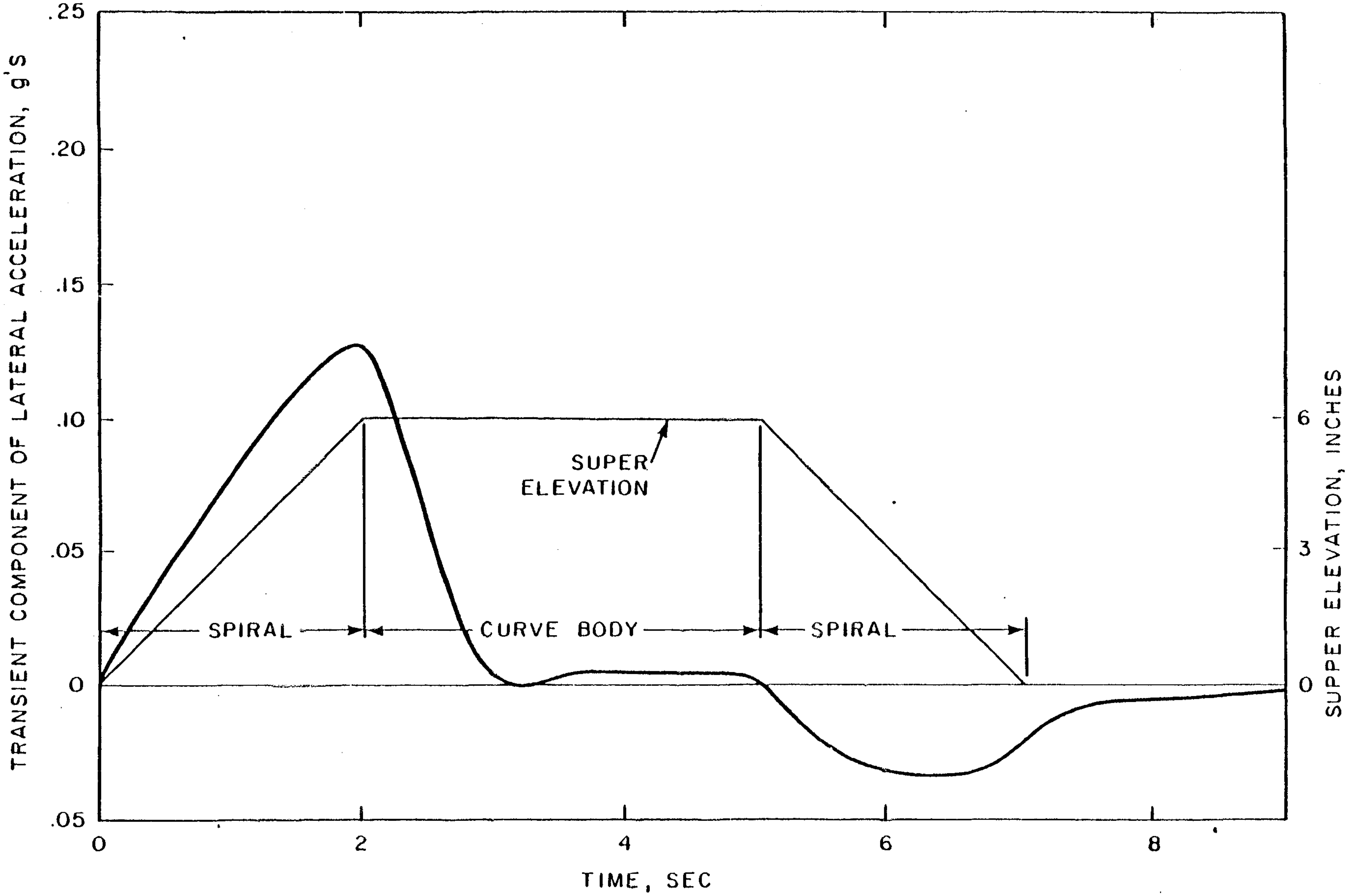


Figure B-14. Typical "Transient Response Diagram" of "Carbody Vibration" it may be more valuable than the other more common criteria of "jerk" and steady state lateral acceleration in assessing the effect on ride comfort of the extra lateral acceleration pulses that are a consequence of tilting motions. Steady state lateral acceleration, "jerk" and "carbody vibration" are all measured at 1 Hz or less. The higher frequency components are usually ignored in curving comfort criteria except for Reference 6 which uses Figure B-15 to judge the natural frequency response of proposed suspension systems to mathematically model steady sinusoidal track perturbations.

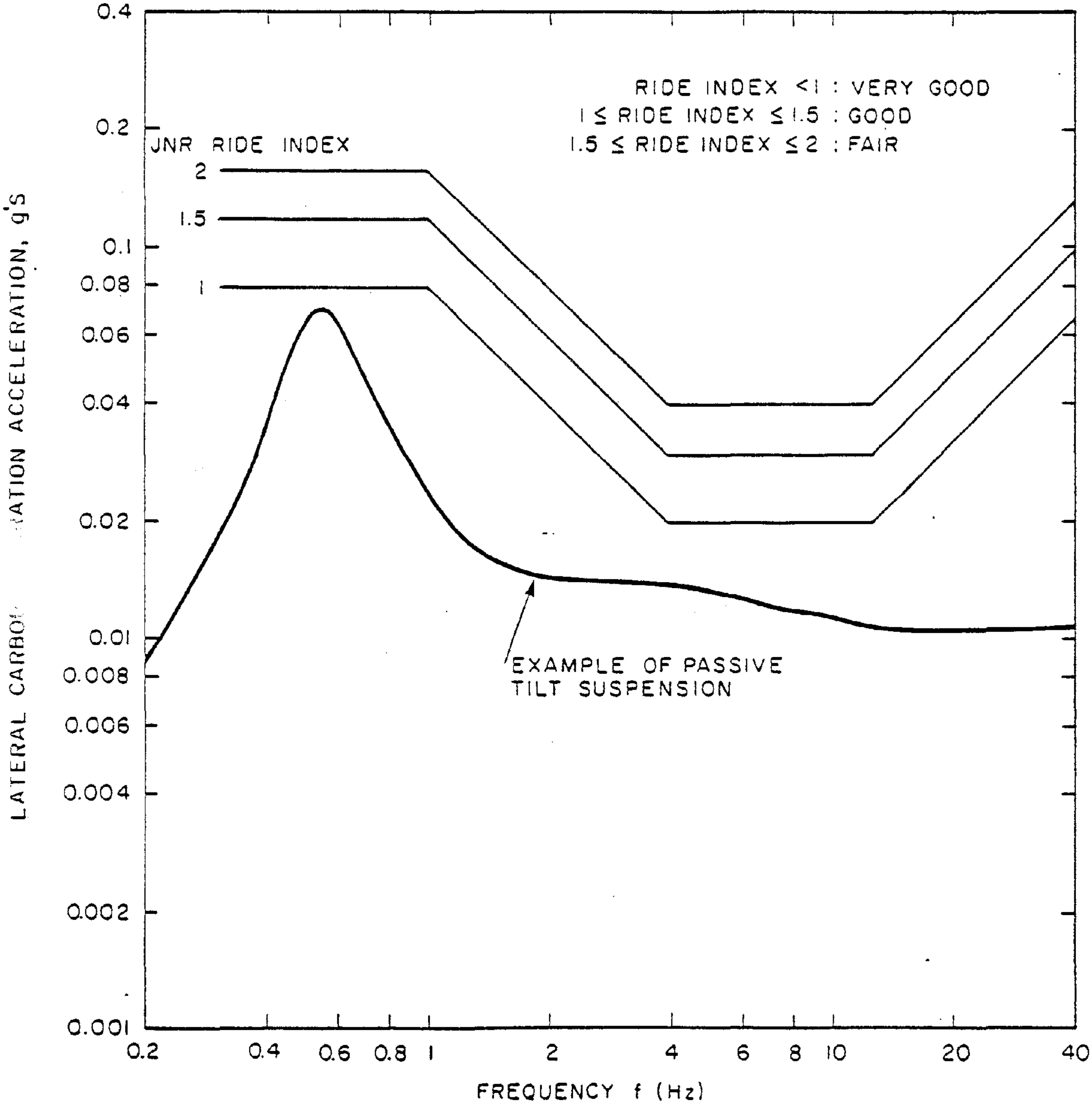


Figure B-15. Suspension Frequency Response to Steady Sinusoidal Lateral Perturbation

APPENDIX C
FORCE MEASUREMENTS WITH INSTRUMENTED WHEELSETS

This appendix discusses instrumented wheel technology using as examples the force sensing wheels produced by ENSCO, Inc. for several previous projects conducted by the Federal Railroad Administration. The Amcoach wheels used in the experiment were the same wheels mentioned below. The F40PH wheels were produced several years ago by ASEA/SJ for tests of the SDP40F. They use the same vertical force sensing bridges described for the Amcoach wheels, but they use a single proportinoal lateral bridge per wheel. Unfortunately the high pass filter technique for zero reference cannot be used on the ASEA lateral bridges, and continual reference measurements on tangent track were made to correct for zero shifts caused by change in speed and temperature.

C.1 WHEEL/RAIL FORCES

The instrumented wheelset is unsurpassed in obtaining accurate measurement of wheel/rail forces. The instrumented wheelset can provide accurate continuous measurements of lateral and vertical wheel/rail forces. It can measure frequencies up to 100 Hz or more, limited only by the fundamental resonant frequencies of the wheelset. Since the measurement is made in close proximity to the rail contact point (i.e., the wheelplate), the error introduced by inertial forces beyond the measurement point is negligible.

The objective of the design of force measuring wheels is to obtain adequate primary sensitivity for low signal/noise ratio and high resolution while controlling crosstalk, load point sensitivity, ripple, and the effects of heat, centrifugal force and longitudinal forces. The design philosophy is to choose strain gage bridge configurations which inherently minimize as many extraneous influences as possible and which are responsive to the general strain patterns expected in any rail wheel subjected to vertical and lateral forces. Such bridge configurations can be adapted to the standard production wheels of the desired test vehicles, eliminating problems of supply, mechanical compatibility, and possible alterations of vehicle behavior due to special wheels. The radial locations of the strain gages is optimized for each wheel size and shape while their angular locations are fixed by the chosen bridge configurations. The LRC locomotive, LRC coach, and Amcoach wheels have a large variation in tread diameter and wheelplate shape and yet were instrumetned successfully using the same general procedures. Cast freight car wheels (70 ton) instrumented similarly for another FRA project will be included in the following discussion to illustrate the general applicability of the techniques.

C.1.1 DESCRIPTION OF STRAIN GAGE BRIDGES

The vertical force measuring bridges follow a concept used by ASEA/SJ (Ref. 18). Each bridge consists of eight strain gages arranged in a wheatstone bridge having two gages per leg. Each leg of the bridge has one strain gage on the field side and one strain gage on the gage side of the wheel. The four legs are evenly spaced 90° apart on the wheel as shown in Figure C-1. The general strain distribution in a typical rail wheel plate due to a purely vertical load is characterized by maximum strains which are compressive and highly localized in the wheelplate above the point of rail contact. As the pair of gages in each leg of the bridge consecutively passes over the rail contact point, two negative and two positive peak bridge outputs occur per revolution. By correctly choosing the radial position of the gages, the bridge output as a function of rotational position of the wheel can be made to resemble a triangular waveform having two cycles per revolution. The purpose of having gages on both sides of the wheelplate in each leg is to cancel the effect of changes in the bending moments in the wheelplate due to lateral force and the change of axial tread/rail contact point.

When two triangular waveforms equal in amplitude and out of phase by one-fourth the wavelength, are rectified and added, the sum is a constant equal to the peak amplitude of the individual waveforms. In order to generate a strain signal proportional to vertical force and independent of wheel rotational position, the outputs of two identical vertical bridges out of phase by 45° of wheel arc are rectified and summed as shown in Figure C-2. Since the bridge outputs do not have the sharp peaks of true triangular waveforms, the sum of one bridge peak and one bridge null is lower than that of two concurrent intermediate bridge outputs. In order to reduce the ripple or variation in force channel output with wheel rotation, the bridge sum is scaled down between the dips coinciding with the rounded bridge peaks. By taking as the force channel output the greatest of either individual bridge output or the scaled down sum of both bridges, the scaling down is applied selectively to the part of the force channel output between the dips as shown in Figure C-2.

The general strain distribution of a typical rail wheelplate due to a purely lateral flange force is characterized by two components as shown in Figure C-3. One component is a function of radius only because the wheelplate acts as a symmetric diaphram in opposing the lateral force at the axle. The second component results from the moment about the hub caused by the flange force, and it tends to vary at a given radius with the cosine of the angular distance from the wheel/rail contact point. The strain distributions on the gage and field sides of the wheelplate are similar in magnitude but opposite in sign (compression or tension).

VERTICAL FORCE MEASUREMENT BRIDGE

"A + B" TRIANGULAR OUTPUT (ASEA/SJ)

- TWO BRIDGES
- GAGES ON BOTH SIDES OF WHEELPLATE
- TRIANGULAR WAVEFORMS-2-CYCLES PER REVOLUTION
- OUTPUT = MAX $\{|A|, |B|, K(|A| + |B|)\}$

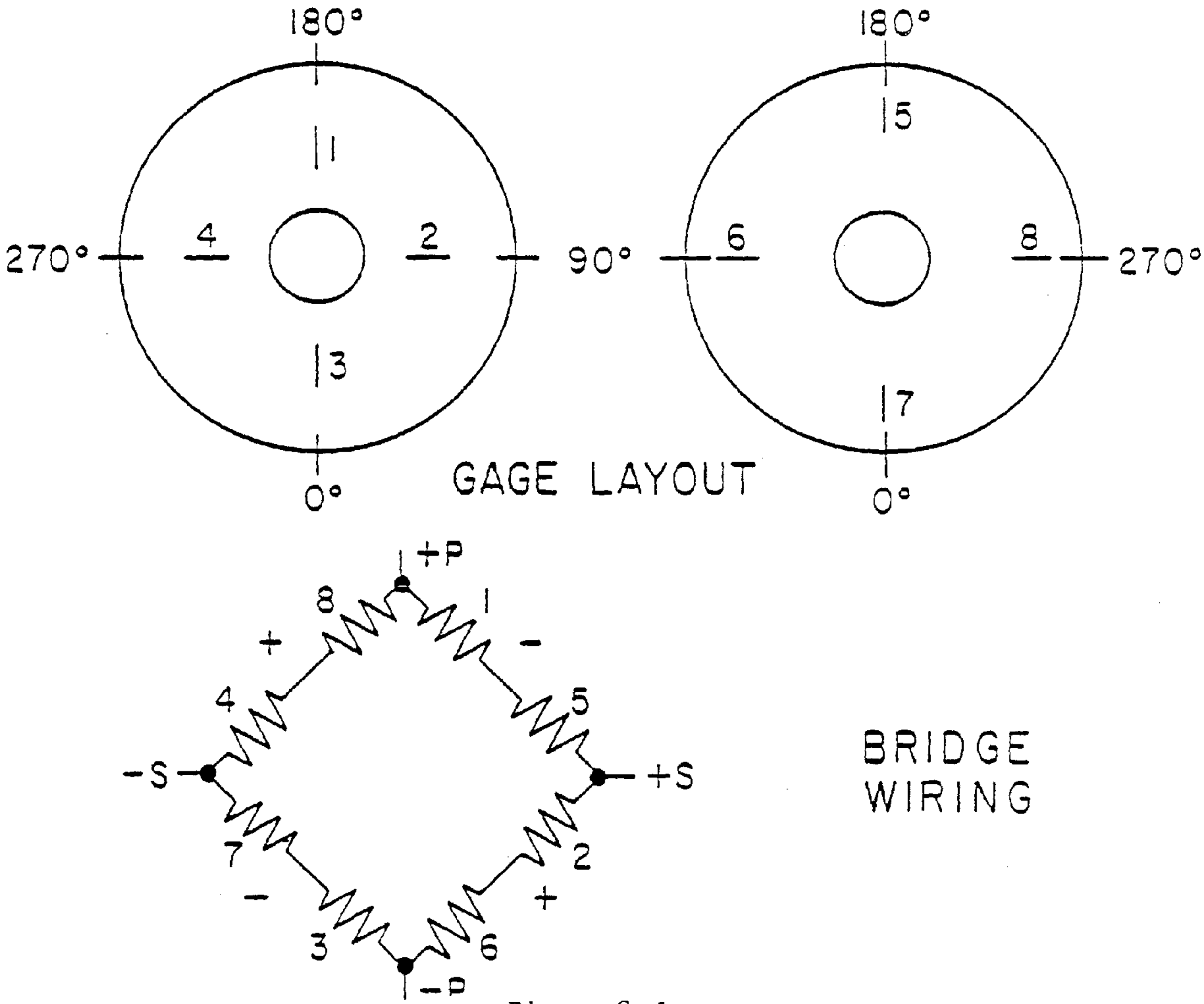


Figure C -1

TRIANGULAR OUTPUT AND "A + B

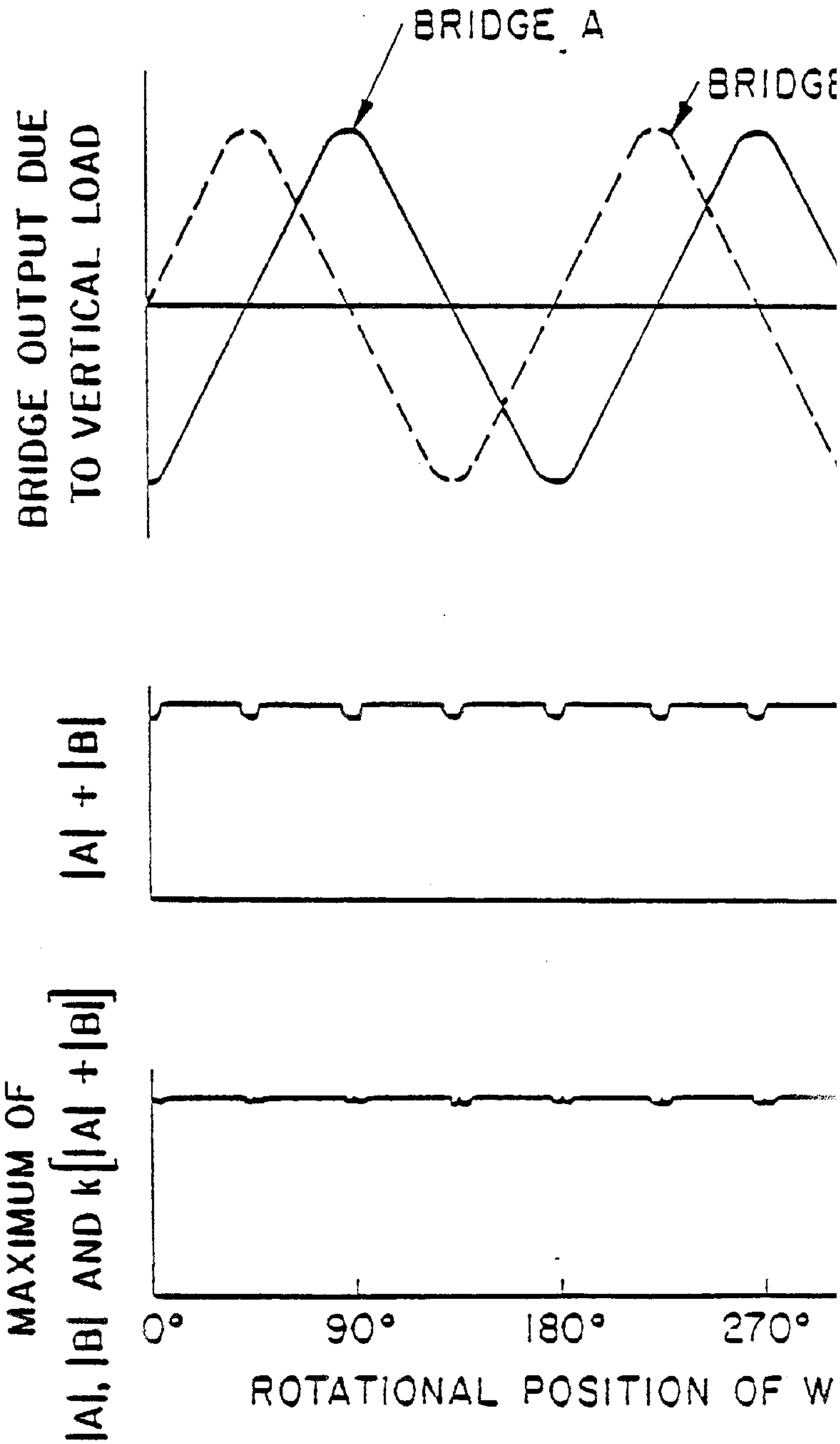


Figure C-2

LATERAL FORCE STRAIN DIST

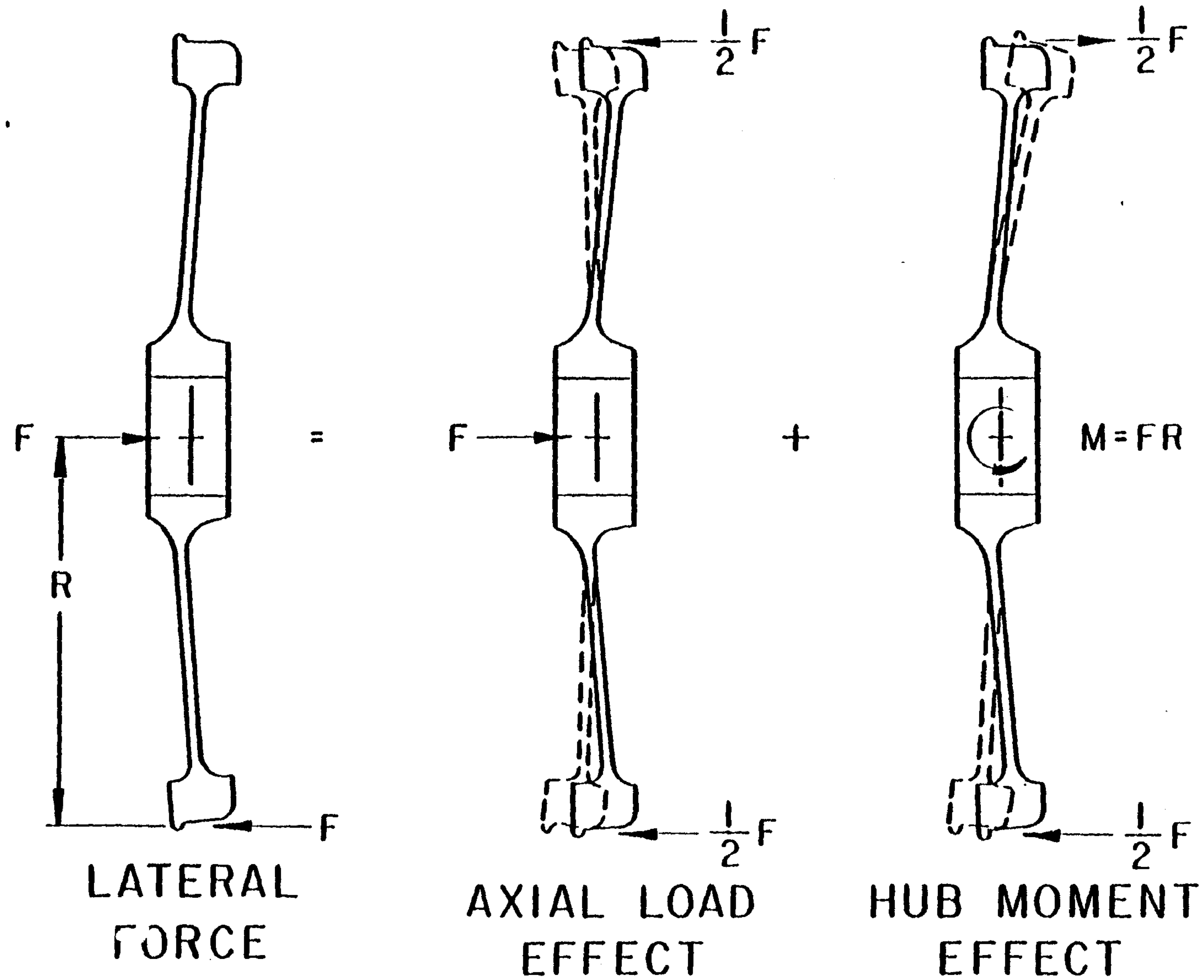


Figure C-3

Lateral force measuring bridges, which follow a concept advanced by EMD (Ref. 19), take advantage of the general strain distribution in a standard rail wheelplate. As shown in Figure C-4, each bridge is composed of eight gages evenly spaced around the field side of the wheelplate at the same radius. The first four adjacent gages are placed in legs of the bridge that cause a positive bridge output for tensile strain. The next four gages are placed in legs causing a negative bridge output for tensile strain. The resulting bridge cancels out the strain due to the axial load as all eight gages are at the same radius with four causing positive and four causing negative bridge outputs. However, the bridge is very sensitive to the sinusoidal strain component associated with the hub moment due to the flange force since the tensile strains and the compressive strains above and below the axle are fully additive in bridge output twice each revolution (once as a positive peak and once as a negative peak). Radial gage locations may be chosen such that the bridge output varies sinusoidally with one cycle per wheel revolution. Two identical bridges 90° out of phase are used to obtain a force channel output independent of wheel rotational position as a consequence of the geometric identity:

$$\sqrt{(L\sin\theta) + (L\sin\{\theta + 90^{\circ}\})} = L \quad \text{for any } \theta$$

C.1.2 PRIMARY SENSITIVITY AND CROSSTALK

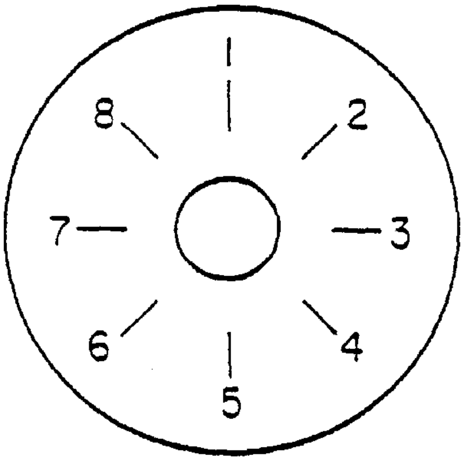
The first step in the production of instrumented wheels is the machining of all wheels in a production group to an identical contour. The contour is dictated by the minimum allowable wheelplate thickness and by the production variation of the available sample of wheels. The machining contour is usually close to the original design shape but at a minimum thickness. The thinning of the wheelplate is the easiest step in maximizing sensitivity because it does not involve compromise with the other measurement properties of the wheel.

The most powerful tool in selecting the radial locations of the strain gages for the best compromise between primary sensitivity, crosstalk, ripple, and sensitivity to axial load point variation is a detailed empirical survey of the strains induced in the given wheelplate by the expected service loads. The use of wheels machined to an identical profile makes the empirical approach to wheelset instrumentation practical since the results of the strain survey may be applied to all wheels in the group. The calibration loads and the reference lateral position of the wheel on the rail should reflect the type of experiment in which the wheels will be used.

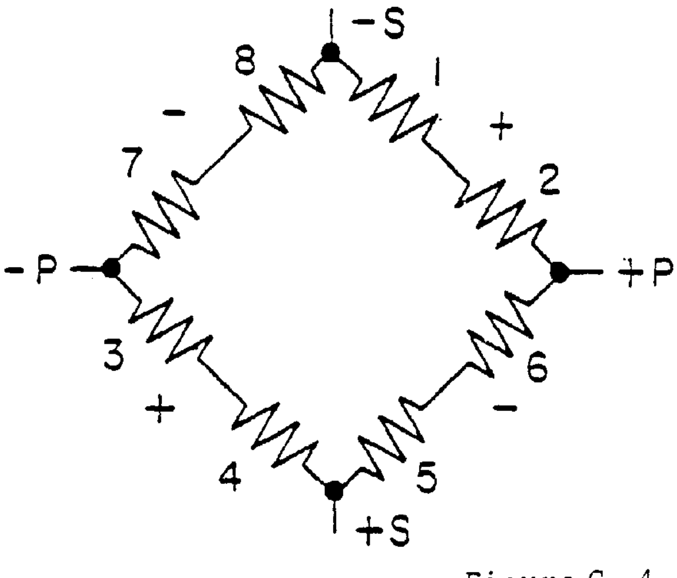
LATERAL FORCE MEASUREMENT BRIDGE

$\sqrt{\sin^2 + \cos^2}$ TECHNIQUE (EMD)

- TWO BRIDGES
- SINUSOIDAL OUTPUT
- 90° OUT OF-PHASE
- APPLIED AT SINGLE RADIUS TO ONE SIDE OF WHEELPLATE



GAGE LAYOUT



BRIDGE WIRING

Figure C -4

For example, wheels destined to measure high speed curving forces should be loaded to about 1-1/2 times the nominal wheel load (to simulate load transfer) with the rail adjacent to the flange to determine the primary vertical sensitivity. Primary lateral sensitivity should be determined from a high lateral load (corresponding to expected L/V ratios) applied with a device which bears against the gage sides of two wheels on an axle at the tread radius and spreads the wheel apart. Loads applied in this manner create strains of equal magnitude and opposite sign to those produced by the hub moment effect of a flange load, but they eliminate the extraneous effect of the vertical load hub moment (treated as crosstalk) from the determination of primary lateral sensitivity. A combined vertical and lateral loading at the expected service L/V ratio level accomplished by forcing the wheelset laterally against a rail while maintaining a vertical load is necessary to select strain gage locations for minimal crosstalk. Vertical loadings at several points across the tread should be taken to evaluate the sensitivity to axial load point.

In the strain survey conducted on the FRA wheels strain gages were applied at intervals of one inch or less on both field and gage sides of the wheelplate along two radial lines separated by 180° of wheel arc. The calibration loads were repeated at every 15° of wheel arc until the strain along 24 equally spaced radial lines on both gage and field side was mapped for each load. This data was used in a computer program to predict the output of a force channel as a function of the radial locations of the gages in the companion bridges.

The vertical force measuring bridges of the FRA wheels have strain gages on both sides of the wheelplate. The simulation program allows the rapid trial of many combinations of gage and field side radii as potential strain gage locations. The maximum sensitivity possible, for a purely vertical load on a given wheel of a bridge actually producing the triangular waveform, is rapidly revealed. The "triangularity" of the waveform of a candidate bridge can be tested by adding its output at each angular load position to that at a load position advanced by 45° of wheel arc. This test determines the ripple expected of a force channel composed of two out of phase candidate bridges.

A lateral force effects the vertical bridge both by directly changing the strain pattern in the wheelplate and by moving the point of vertical load contact with the rail toward the flange. By using as a measurement of crosstalk the difference in bridge output caused by adding a lateral load to an existing vertical load, correction factors may be chosen which compensate for net lateral force crosstalk which includes direct lateral force crosstalk and the effect of vertical load point movement. It is desirable to identify vertical bridges in which the direct lateral force crosstalk and the effect of load point changes are opposed and yield a minimum net crosstalk for flange forces in service. The accuracy of the highly loaded flanged wheel is enhanced by using a correction factor in processing based on the net lateral force crosstalk. Compromises in bridge selection are usually biased in favor of the flanged wheel because it generates the most vital data for vehicle dynamics or rail wear studies.

The primary sensitivities and crosstalk factors achieved for the cant deficiency test wheels and the freight car test wheels are shown in Figure C-5. The vertical bridges were chosen from a detailed simulation with radial position increments of 0.1 inches on a basis of maximum primary sensitivity while holding the simulated crosstalk and ripple below 5% and minimizing sensitivity to axial load point. The primary sensitivity was observed to be linear within about 1% because the strains at each gage are low and the wheelplate behaves elastically. Primary vertical force sensitivity appears to be inversely proportional to tread diameter and wheelplate thickness for the several wheelplate shapes.

The lateral force measuring bridges of the FRA wheels have gages on only one side of the wheelplate, and the trial simulation of bridges is used to determine the most advantagous side of the wheel and the radial gage position. The primary sensitivity was determined from pure lateral loads applied with a spreader bar. The absolute value difference in lateral force indication between a combined vertical and lateral load on a rail and the pure lateral load with the spreader bar at the same lateral load is attributed to vertical force crosstalk. This method of crosstalk determination takes into account the vertical load point at the L/V ratios of interest. While a correction factor based on the vertical force crosstalk perfectly compensates a lateral force at the optimized L/V ratio, it is usually still accurate to about 2% of the lower lateral force at one-half the optimized L/V ratio.

Figure C-5 gives the primary sensitivity and vertical force crosstalk actually achieved for several types of wheels. Lateral force measuring bridges of maximum sensitivity having less than 2% crosstalk and 5% ripple were sought in a simulation of possible bridges. Vertical load point sensitivity is not a great factor because the range of load points is narrow while lateral flange forces are being measured. The sensitivity of the sinusoidal lateral bridge is much greater than that of the triangular vertical bridge. Wheels of large tread diameter in general produce greater sensitivity.

C.1.3 RIPPLE

Ripple is caused by the failure of the bridges to produce the desired waveform and by deviation from the correct phase

TYPICAL WHEELSET CALIBRATION CONSTANTS

WHEEL DESCRIPTION	VERTICAL FORCE MEASUREMENT			LATERAL FORCE MEASUREMENT	
	SENSITIVITY	K	NET LATERAL FORCE CROSSTALK	SENSITIVITY	VERTICAL FORCE CROSSTALK
30" TREAD DIA., CONCAVE CONICAL WHEEL PLATE, 3/4" MIN. THICKNESS LRC COACH	$6 \frac{\mu\epsilon}{kip}$	.94	2 %	$18 \frac{\mu\epsilon}{kip}$	1 1/2 %
33" TREAD DIA., CONCAVE CURVED WHEEL PLATE, 3/4" MIN. THICKNESS 70 TON FREIGHT CAR	$5\ 1/2 \frac{\mu\epsilon}{kip}$	.94	4 %	$16\ 1/2 \frac{\mu\epsilon}{kip}$	3 %
36" TREAD DIA., CONVEX CONICAL WHEEL PLATE, 3/4" MIN. THICKNESS AMCOACH	$4\ 1/4 \frac{\mu\epsilon}{kip}$	.94	5 %	$17 \frac{\mu\epsilon}{kip}$	4 %
40" TREAD DIA., CONCAVE CONICAL WHEEL PLATE, 1" MIN. THICKNESS LRC LOCOMOTIVE	$3\ 1/2 \frac{\mu\epsilon}{kip}$	.92	1 1/2 %	$33 \frac{\mu\epsilon}{kip}$	1/2 %

Figure C-5

relationship between the companion bridges which are processed together as a force channel.

The wheelplates are machined for uniformity to reduce ripple and a grid of radial and circumferential lines is scribed on the wheelplate to aid accurate gage placement. The massive computer aided simulation of trial bridges was used to determine gage locations of minimum inherent ripple. The ripple of the vertical force channel is reduced by attenuating the high bridge sums occurring between the rounded bridge peaks as shown in Figure C-2. This method achieves a substantial reduction in ripple at a small cost in average sensitivity.

The lateral bridge output is inherently very sinusoidal. The requirement for two bridges at the same radius out of phase by 90° is in conflict with the 45° spacing between the gages in each bridge because theoretically both bridges should occupy the same space. Placing the gages side by side causes a deviation from the proper phase relationship which manifests itself as a ripple. Figure C-6 gives the maximum ripple for each set of four wheels of four types. Larger wheels which have less phase deviation between lateral bridges also have less ripple. Combined loads caused greater ripple for both vertical and lateral channels because crosstalk produced distortions of the waveforms.

Ripple does not create as much error as might be supposed. Even the peak wheel forces measured during vehicle dynamics testing are averaged for 50 to 100 milliseconds. A 36-inch wheel makes a full revolution in 100 milliseconds at 64 mph, totally negating ripple in a 100 millisecond average wheel force. A single instantaneous measurement is rarely sought and any filtering has a mitigating influence on ripple.

C.1.4 LOAD POINT SENSITIVITY

The lateral bridge is sensitive to a lateral movement of the vertical load contact patch because it changes the hub moment. However, as shown in Figure C-6, the vertical bridge is much more sensitive to load point than expected from the change in crosstalk due to the small change in lateral force measurement. The failure of the tread to transmit the moment due to load point offset uniformly into the wheelplate probably results in unusual changes to the local intense compressive strains in the wheelplate above the rail contact to which the vertical bridge is most sensitive. The high load point sensitivity of the 33-inch freight wheel having the thinnest tread supports this hypothesis.

The effect of load point sensitivity on measurements taken with the FRA wheels was minimized in two ways. Taking as the load

TYPICAL UNCORRECTED VARIABILITY

WHEEL DESCRIPTION	VERTICAL FORCE MEASUREMENT			LATERAL FORCE MEASUREMENT		
	SENSITIVITY TO AXIAL LOAD POINT	MAX. RIPPLE VERTICAL LOAD	MAX. RIPPLE COMBINED LOAD	SENSITIVITY TO AXIAL LOAD POINT	MAX. RIPPLE	MAX. RIPPLE COMBINED LOAD
30" TREAD DIA., CONCAVE CONICAL WHEEL PLATE, 3/4" MIN. THICKNESS LRC COACH	$+5.7 \frac{\%}{\text{inch}}$	±5 %	±8 %	$-2.4 \frac{\%}{\text{inch}}$	±7 %	±7 %
33" TREAD DIA., CONCAVE CURVED WHEEL PLATE, 3/4" MIN. THICKNESS FREIGHT WHEEL	$+9.5 \frac{\%}{\text{inch}}$	±6 %	±6 %	$-1 \frac{\%}{\text{inch}}$	±6 %	±7.5 %
36" TREAD DIA., CONVEX CONICAL WHEEL PLATE, 3/4" MIN. THICKNESS AMCOACH	$-10 \frac{\%}{\text{inch}}$	±7 %	±10 %	$-3.2 \frac{\%}{\text{inch}}$	±4 %	±6 %
40" TREAD DIA., CONCAVE CONICAL WHEEL PLATE, 1" MIN. THICKNESS LRC LOCOMOTIVE	$+4.7 \frac{\%}{\text{inch}}$	±5 %	±5 %	$-3 \frac{\%}{\text{inch}}$	±4 %	±4 %

Figure C-6

point for primary vertical sensitivity the wheel flange adjacent to the rail, causes the heavier loaded high rail wheel to deviate little from the calibrated load point. The additional movement of load point toward the flange under heavy lateral loading was accounted for in the net lateral force crosstalk correction factor. The lesser effect of vertical load point variation on lateral force was also accounted for in its crosstalk correction factor. The residual effect of load point variation is that load transfer from low rail wheel to high rail wheel in high cant deficiency curving is over estimated by about 5% because the low rail wheel is loaded at a less sensitive point on the tread.

C.1.5 THERMAL AND CENTRIFUGAL EFFECTS AND OTHER SOURCES OF DRIFT

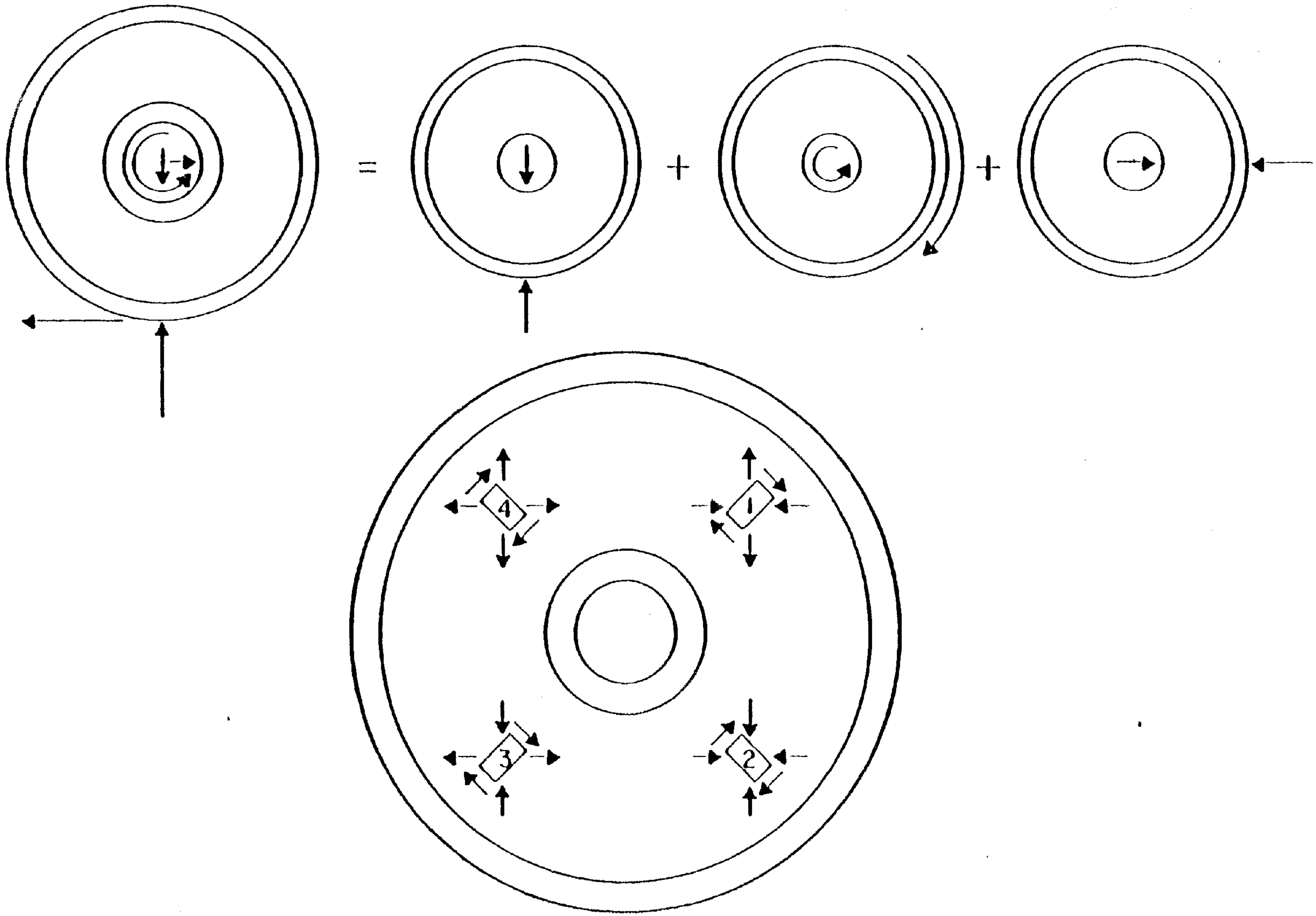
The vertical and lateral bridges used on the FRA wheelsets are particularly immune to drift by virtue of strain gage location and instrumentation technique. Strains induced by thermal change and centrifugal force are radially symmetric on each side of the wheelplate. The lateral bridge consists of eight gages at the same radius on the same side of the wheelplate positioned in the bridge so that four add and four subtract. A radially symmetric strain field is cancelled by the additions and subtractions. Similarly, the vertical bridges have four gages at the same radius on each side of the wheelplate. On each side two gages add and two subtract.

Each bridge generates a triangular or sinusoidal waveform as the wheel rotates under load. High pass filtering of the amplified bridge signals at 0.2 Hz does not attenuate the oscillating part of the signal but it forces the signal to oscillate about zero. High pass filtering eliminates gradual drift that could occur from thermal effects on the wheelset wiring and wheel to amplifier cabling and zero drift of the strain gage bridge amplifiers. It would also suppress thermal and centrifugal effects in bridges which do not self cancel them.

C.1.6 SENSITIVITY TO LONGITUDINAL FORCE

Longitudinal forces involved in braking and driving are extraneous influences on the vertical and lateral force measurement bridges. Brakes on instrumented wheelsets are usually disabled to avoid sensor damage caused by overheating and to avoid accidental flatspotting. However, instrumented wheelsets on self propelled vehicles must cope with driving forces. Figure C-7 shows the strain distribution in a driven wheel. The longitudinal force may be resolved into a torque about the axle and a horizontal force perpendicular to the axle. The similarity between the horizontal force component and the vertical force suggests an error source.

LONGITUDINAL FORCE STRAIN DISTRIBUTION



The vertical force measuring bridges on the FRA wheelsets are configured in such a way as to cancel the effect of longitudinal forces. Figure 3-7 shows the strain components at four gage positions on one side of the wheelplate due to vertical and driving forces. The bridge is shown in the vertical null output position. Gages at 180° spacing add together in their contribution to the bridge summation. The vertical, horizontal and shear components of strain are opposite in sense for gages spaced 180° apart and cancel each other out retaining the null bridge output. The longitudinal force does not create an intense local strain aligned with the sensitive axis of a strain gage which stimulates the vertical bridge in any rotational position. The insensitivity of the vertical bridges to longitudinal force has also been verified experimentally.

The lateral bridges used on the FRA wheelsets are also insensitive to longitudinal forces. The symmetric gage pattern limits the effect of the shear strains, and the horizontal force has the effect of adding vectorially to the vertical force to produce crosstalk. Since the longitudinal force is limited by friction to about 1/4 the vertical load, the vector sum of forces is only about 3% higher than the vertical force alone. An increase in crosstalk of 3% of 4% (0.12%) is insignificant. If the measurement of driving force is desired, torque sensing bridges can be added to the axle between each wheel and the drive gear.

APPENDIX D

D.1 CURVING MODEL

The simple quasistatic curving model shown in Figure D-1 was used to calculate weight vector intercept, vertical wheel force, total truck lateral force and lateral acceleration in the floor plane for comparison to the actual measurements. It takes into account the separate masses of the truck assembly (mass n) and the body (mass m) in a half vehicle representation. The suspension system is represented by an effective roll center which translates laterally with the body and has a roll stiffness, K_θ. The lateral stiffness considered at the secondary suspension is designated K_L. A computer program listed in this appendix along with a sample output was used for the convenient calculation of the above quantities as a function of cant deficiency. Vehicle constants obtained from manufacturers specifications and also from on site static experiments were used in the model. The constants for each vehicle from each source are listed in this Appendix.

The following terms are used in the quasistatic curving calculations:

- n = truck mass
- m = body mass
- V = speed in ft/sec
- S = crosslevel in inches
- D = curvature in degrees
- r = 5730/D is the curve radius in feet
- $\theta = \sin^{-1}(S/60)$ is crosslevel angle
- $\alpha = \tan^{-1}(V^2/rg)$ is the deviation of the resultant force vector for the vertical axis
- ϕ = roll angle of the body
- $(\alpha-\phi)$ = angular cant deficiency
- $u = 60 \sin(\alpha-\phi)$ is the cant deficiency in inches
- K_L = lateral suspension stiffness in lb/in
- K_ϕ = overall roll rate in ft-lb/degree

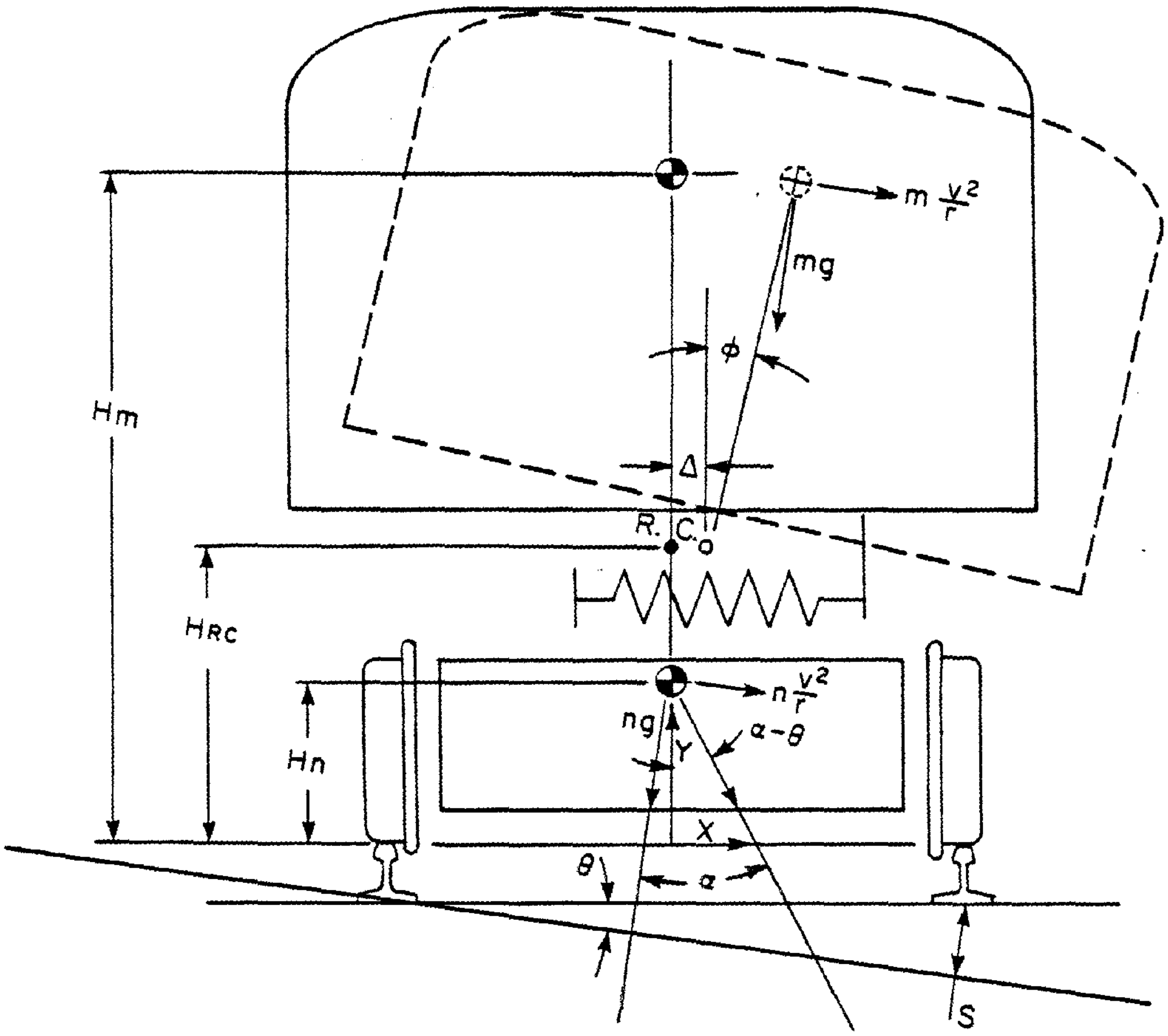


Figure D-1. Quasistatic Curving Model

The dimensions below are referenced to the track center line in the plane of the railheads:

H_n = height of truck c.g., vehicle at rest
X_n = lateral location of truck c.g.
Y_n = vertical location of truck c.g.
H_m = height of body c.g., vehicle at rest
X_m = lateral location of body c.g.
Y_m = vertical location of body c.g.
H_{RC} = height of roll center

Figure D-2 illustrates the computation of the height of the effective roll center given by:

$$H_{RC} = H_s - A = H_s - \left[\frac{\dfrac{H_m - H_s + B}{L_p^2 K_p}}{\dfrac{H_m - H_s + B}{L_p^2 K_p} + \dfrac{2(H_m - H_s)}{L_s^2 K_s}} \right] B$$

where

H_s = the height of the top of the secondary springs

B = the distance between the tops of the secondary and primary springs

K_p = the rate in lb/in of the primary suspension at one wheel

L_p = the lateral spacing of the primary springs

K_s = the rate in lb/in of the secondary suspension at one side

L_s = the lateral spacing of the secondary springs

The following equations are derived from the model and computed by the program:

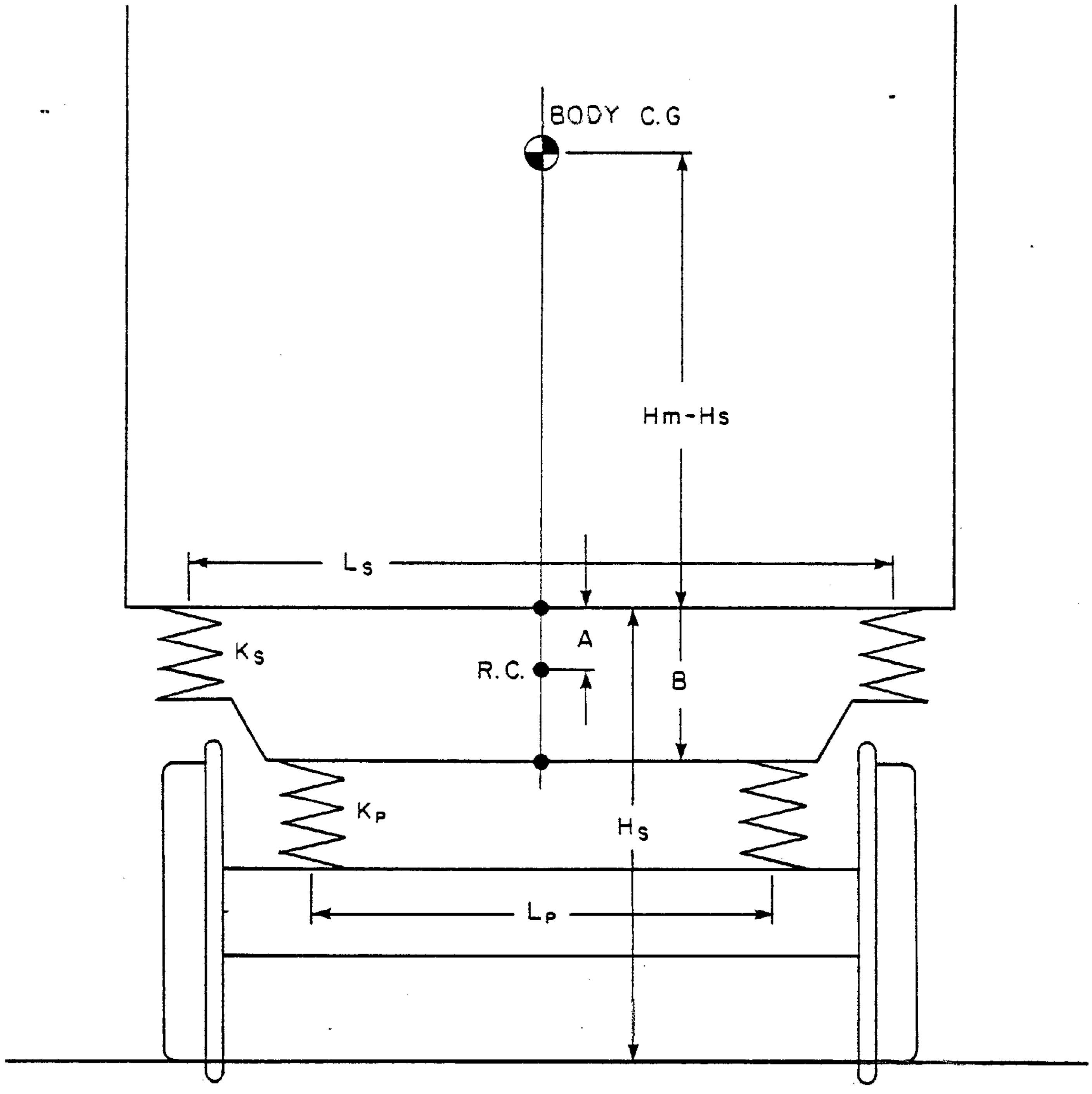


Figure D-2. Parameters for Effective Roll Center Height Calculation

LOCATION OF TRUCK C.G.

$$X_n = X_{on} \pm 1/2" \quad \begin{array}{l} + \text{ if } V^2/r > g \tan\theta \\ - \text{ if } V^2/r < g \tan\theta \end{array}$$

$$Y_n = H_n$$

LOCATION OF BODY C.G.

$$X_m = X_{om} \pm 1/2 + \frac{\frac{mV^2}{r}\cos\theta - mg\sin\theta}{K_L} + (H_m - H_{RC})\sin\phi$$

where

$$\phi = \frac{(H_m - H_{RC})\left(\frac{mV^2}{r}\cos\theta - mg\sin\theta\right)}{K_\phi}$$

but

$$\tan\alpha = \frac{V^2}{rg}$$

$$\therefore \frac{mV^2}{r}\cos\theta - mg\sin\theta = mg\,(\cos\theta\,\tan\alpha - \sin\theta)$$

and

$$\sin(\alpha - \theta) = \sin\alpha\cos\theta - \cos\alpha\sin\theta$$

$$\therefore \frac{mV^2}{r}\cos\theta - mg\sin\theta = \frac{mg\sin(\alpha-\theta)}{\cos\alpha} = \frac{mgu}{60\cos\alpha}$$

So that

$$X_m = X_{om} \pm 1/2 + \frac{mgu}{60 K_L \cos\alpha} + (H_m - H_{RC})\sin\phi$$

and

$$\phi = \frac{(H_m - H_{RC})\,mgu}{60\cos\alpha\, K_\phi}$$

$$Y_m = H_{RC} + (H_m - H_{RC}) \cos \phi$$

WEIGHT VECTOR INTERCEPT

$$VI = \frac{n[X_n + (Y_n) \tan (\alpha - \theta)] + m[X_m + (Y_m) \tan (\alpha - \theta)]}{n + m}$$

AVERAGE HIGH RAIL WHEEL LOAD

$$R_1 = 1/2 \left(\frac{30 + VI}{60}\right)(n + m) \left(g \cos \theta + \frac{V^2}{r} \sin \theta\right)$$

but

$$\tan \alpha = \frac{V^2}{rg}$$

$$\therefore R_1 = 1/2 \left(\frac{30 + VI}{60}\right)(n + m) \; g \cos \theta \,(1 + \tan \alpha \tan \theta)$$

AVERAGE LOW RAIL WHEEL LOAD

Similarly,

$$R_2 = 1/2 \left(\frac{30 + VI}{60}\right) (n + m) \; g \cos \theta \; (1 + \tan \alpha \; \tan \theta)$$

TRUCK LATERAL FORCE

$$F_{LT} = (n + m) \left(\frac{V^2}{r} \cos \theta - g \sin \theta\right) = \frac{(n + m)gu}{60 \cos \alpha}$$

ACCELEROMETER READING AT FLOOR PLANE

Floor angle = $\phi - \theta$

$$a_L = \frac{V^2}{r} \cos (\phi - \theta) + g \sin (\phi - \theta)$$

$$a_L = g \cos(\phi - \theta)\ (\tan\alpha + \tan(\phi - \theta)$$

for banking coach $\theta' = \theta + \theta_{bank}$

$$\therefore\ a_L = g \cos(\phi - \theta')\ (\tan\alpha + \tan(\phi - \theta')$$

The following program in Basic performs the above calculations. A sample output is included.

```
LIST
  10  REM***QUASISTATIC CURVING MODEL TO PREDICT VECTOR INTERCEPT,
  11  REM***WHEEL FORCES,AND LATERAL ACCELERATION IN THE FLOOR PLANE
  12  REM***FORCES ARE EXPRESSED IN POUNDS,DISTANCE IN INCHES,
  13  REM***ACCELERATION IN g'S,AND ANGLES IN DEGREES
  20  PRINT "CROSSLEVEL";
  21  INPUT S
  30  PRINT "CURVATURE";
  31  INPUT D
  35  DIM V$[20]
  40  PRINT "ENTER VEHICLE IN QUOTES";
  41  INPUT V$
  50  PRINT "K sub phi ,ROLL RATE IN FT-LB/DEGREE";
  51  INPUT K1
  58  PRINT "FOR SINGLE STAGE SECONDARY LATERAL SUSPENSION"
  59  PRINT "ENTER ZEROS FOR STAGE 1 SPRING RATE AND COMPLIANCE !"
  60  PRINT "K sub L, LATERAL SPRING RATE IN LB/IN ,ENTER STG 1,STG 2";
  61  INPUT K2,K3
  65  PRINT "MAXIMUM LATERAL COMPLIANCE ,ENTER STG 1,STG 2";
  66  INPUT L1,L2
  70  PRINT "INPUT TRUCK WEIGHT COMA 1/2 BODY WEIGHT";
  71  INPUT W1,W2
  75  PRINT "WEIGHT OFFSET (POSITIVE TOWARD HIGH RAIL)";
 _76  INPUT O1
```

```
80  PRINT "HEIGHTS OF TRUCK C.G.,BODY C.G.,AND EFFECTIVE ROLL CENTER";
81  INPUT H1,H2,H3
90  LET T=S/60
91  LET T1=T*57.3
95  LET R=5730/D
100  PRINT
102  PRINT
110  PRINT "THE CURVE BEING MODELED HAS:"
120  PRINT "X-LEVEL,IN     ","X-LEVEL,DEG.","CURVATURE,DEG.","RADIUS,FT"
130  PRINT S,T1,D,R
140  PRINT
141  PRINT
150  PRINT "THE VEHICLE BEING MODELED IS THE ";V$;" WITH THE CONSTANTS:"
160  PRINT "Ksub phi","Ksub L#1","Ksub L#2","TRUCK WT.","1/2 BODY WT."
165  PRINT K1,K2,K3,W1,W2
166  PRINT
170  PRINT "TRUCK C.G.","BODY C.G.","ROLL CNTR","LAT. COMP.","WT. OFST"
175  PRINT H1,H2,H3,L1;";";L2,O1
180  LET G=32.2
185  PRINT
186  PRINT
187  PRINT
190  PRINT "CANT  ";"SPEED  ";"VECTOR  ";"H VERT  ";"L VERT  ";
191  PRINT "TK LAT  ";"ACCEL  ";"THETA  ";"ALPHA  ";"PHI"
```

```
195  PRINT
200    FOR U=1 TO 15
210    LET V1=SQR((S+U)/(.0007*D))
220    LET V2=V1*88/60
230    LET A=ATN((V2*V2)/(R*G))
240    LET A1=A*57.3
250    REM*****COORDINATES OF TRUCK C.G.**********************
260    LET O2=1/2
270    LET X1=O1+O2
280    LET Y1=H1
300    REM*****COORDINATES OF BODY C.G.***********************
310    LET O3=W2*U/(K2*60*COS(A))
312    IF O3<L1 THEN 330
314    LET O3=L1+(((W2*U)/(60*COS(A)))-K2*L1)/K3
320    IF O3>L2 LET O3=L2
330    LET P1=(H2-H3)*W2*U/(720*K1*COS(A))
340    LET P=P1/57.3
350    LET O4=(H2-H3)*SIN(P)
360    LET X2=O1+O2+O3+O4
370    LET Y2=H3+(H2-H3)*COS(P)
400    REM*******VECTOR INTERCEPT, V**********
410    LET V=(W1*(X1+Y1*TAN(A-T))+W2*(X2+Y2*TAN(A-T)))/(W1+W2)
420    REM*****AVG HIGH RAIL WHEEL VERTICAL LOAD,F1**********
430    LET F0=.5*(W1+W2)*COS(T)*(1+TAN(A)*TAN(T))
```

```
440   LET F1=F0*(30+V)/60
450    REM*****AVG LOW RAIL VERTICAL LOAD,F2******************
460    LET F2=F0*(30-V)/60
470    REM*****NET TRUCK LATERAL FORCE,F3**********************
480    LET F3=(W1+W2)*U/(60*COS(A))
490    REM*******LAT. ACCELERATION IN FLOOR PLANE******************
500    LET A2=COS(P-T)*(TAN(A)+TAN(P-T))
510    PRINT  USING 520;U,V1,V,F1,F2,F3,A2,T1,A1,P1
520    IMAGE 2D.D,2X,3D.D,2X,2D.D,4X,3(5D,3X),D.3D,3(2X,2D.2D)
530    NEXT U
540  END
```

```
ENTER VEHICLE IN QUOTES?"F40PH SPECS"
K sub phi ,ROLL RATE IN FT-LB/DEGREE?48873
FOR SINGLE STAGE SECONDARY LATERAL SUSPENSION
ENTER ZEROS FOR STAGE 1 SPRING RATE AND COMPLIANCE !
K sub L, LATERAL SPRING RATE IN LB/IN ,ENTER STG 1,STG 2?0,3970
MAXIMUM LATERAL COMPLIANCE ,ENTER STG 1,STG 2?0,2.25
INPUT TRUCK WEIGHT COMA 1/2 BODY WEIGHT?34235,93510
WEIGHT OFFSET (POSITIVE TOWARD HIGH RAIL)?0
HEIGHTS OF TRUCK C.G.,BODY C.G.,AND EFFECTIVE ROLL CENTER?28.6,85.8,35.5


THE CURVE BEING MODELED HAS:
X-LEVEL,IN      X-LEVEL,DEG.    CURVATURE,DEG. RADIUS,FT
5.25            5.01375         2.6            2203.85


THE VEHICLE BEING MODELED IS THE F40PH SPECS WITH THE CONSTANTS:
Ksub phi        Ksub L#1        Ksub L#2       TRUCK WT.       1/2 BODY WT.
48873.          0               3970           34235.          93510.

TRUCK C.G.      BODY C.G.       ROLL CNTR      LAT. COMP.      WT. OFST
28.6            85.8            35.5           0     ;2.25               0
```

CANT	SPEED	VECTOR	H VERT	L VERT	TK LAT	ACCEL	THETA	ALPHA	PHI
1.0	58.6	2.0 +.04	34265	29944	2141	0.019	5.01	5.94	0.13
2.0	63.1	3.6 +.08	35961	28341	4289	0.038	5.01	6.89	0.27
3.0	67.3	5.1 +.12	37660	26735	6447	0.057	5.01	7.82	0.40
4.0	71.3	6.6 +.17	39363	25126	8617	0.076	5.01	8.76	0.54
5.0	75.0	8.2 +.2	41069	23512	10799	0.095	5.01	9.69	0.68
6.0	78.6	9.6 +.25	42663	22011	12997	0.114	5.01	10.61	0.82
7.0	82.0	10.8 +.29	44054	20713	15211	0.133	5.01	11.53	0.95
8.0	85.3	12.0 +.33	45447	19413	17442	0.152	5.01	12.45	1.10
9.0	88.5	13.3 +.38	46840	18113	19694	0.171	5.01	13.35	1.24
10.0	91.5	14.5 +.42	48235	16811	21967	0.190	5.01	14.25	1.38
11.0	94.5	15.7 +.46	49631	15508	24263	0.209	5.01	15.15	1.52
12.0	97.4	16.9 +.5	51028	14204	26583	0.228	5.01	16.03	1.67
13.0	100.1	18.2 +.5	52426	12899	28928	0.248	5.01	16.91	1.82
14.0	102.8	19.4 +.5	53825	11592	31302	0.267	5.01	17.78	1.97
15.0	105.5	20.6 +.5	55226	10284	33704	0.286	5.01	18.64	2.12

BASIC READY

>

Contribution of primary lateral compliance

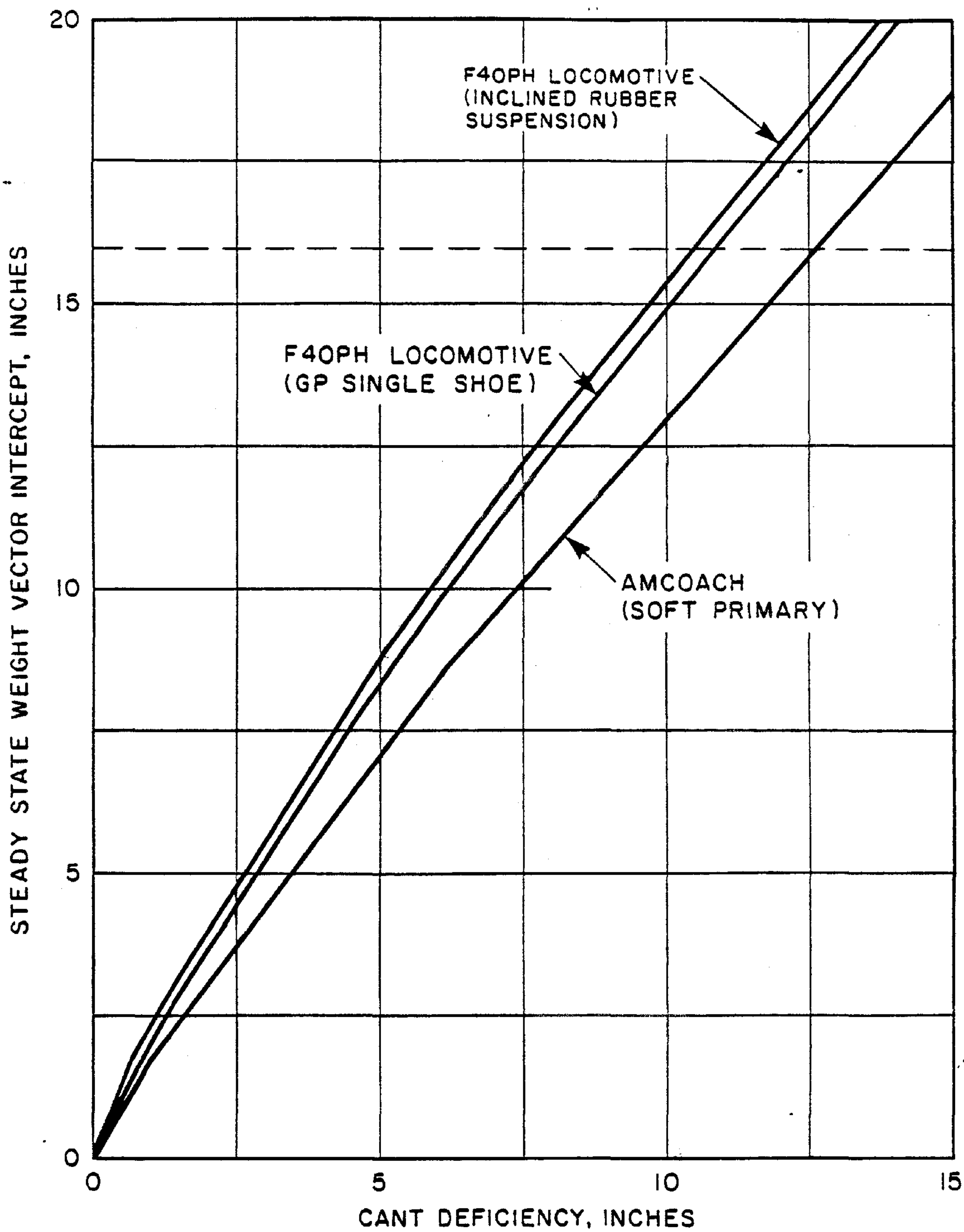


Figure D-3. Prediction of Quasistatic Weight Transfer

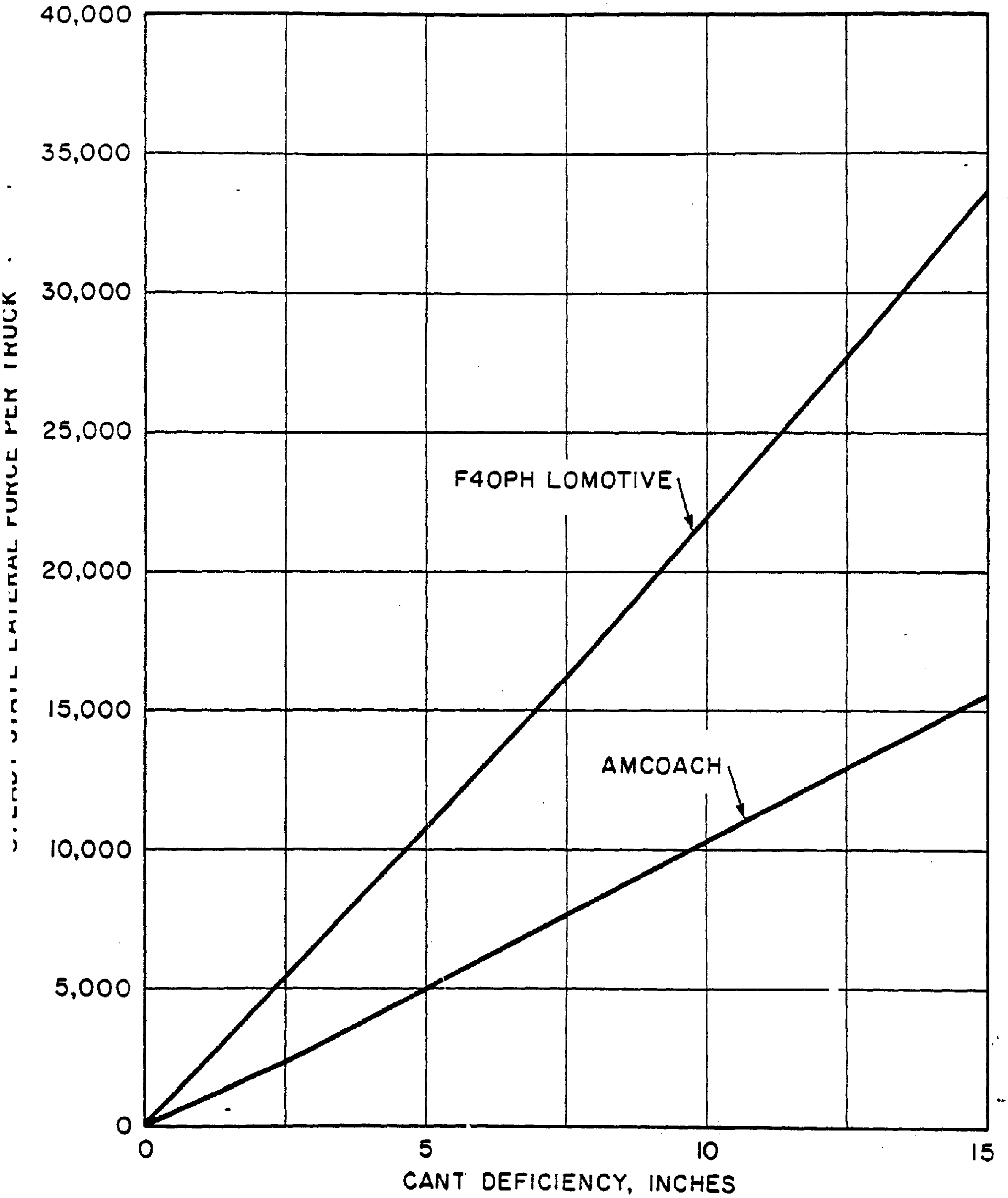


Figure D-4. Prediction of Quasistatic Lateral Truck Force

D.2 VEHICLE CHARACTERISTICS

The vehicle characteristics used in the quasistatic curving model in this Appendix and other characteristics of general interest are listed for each vehicle. The constants for the model are listed first. The value listed in or derived from the manufacturer's specifications is given first and a second value for some constants which was measured experimentally is also included:

The effective roll center was derived from the manufactuer's specifications by the following relation (see Figure D-2):

$$H_{RC} = H_s - A = H_s - \left[\frac{\dfrac{H_m - H_s + B}{L^2_p K_p}}{\dfrac{H_m - H_s + B}{L^2_p k_p} + \dfrac{2(H_m - H_s)}{L^2_s K_s}} \right] B$$

where

H_s = the height of the top of the secondary springs

B = the distance between the tops of the secondary and primary springs

K_p = the rate in lb/in of the primary suspension at one wheel

L_p = the lateral spacing of the primary springs

K_s = the rate in lb/in of the secondary suspension at one side

L_s = the lateral spacing of the secondary springs

It was determined experimentally by:

$$H_{RC} = H_s - A = H_s - \left[\frac{\phi_p}{\phi_p + \phi_s} \right] B$$

where ϕ_p is the roll angle of the primary suspension measured with the vehicle parted on track having about 6 inches of cross-level, and ϕ_s is the secondary suspension roll angle measured under the same condition.

The overall roll rate K_ϕ (ft-lb/degree) was derived from the vehicle specifications as follows:

$$\frac{1}{K_\phi} = \frac{1}{K_{\phi p}} + \frac{1}{K_{\phi s}}$$

where the primary suspension roll rate,

$$K_{\phi p} = \frac{2L_p^2\ K_p}{1375}$$

where:

L_p = the lateral spacing of the primary springs in inches

K_p = the rate in lb/in of primary suspension at one wheel

and the secondary suspension roll rate,

$$K_{\phi s} = \frac{L_s^2\ K_s}{1375}$$

where

L_s = the lateral spacing of the secondary springs in inches

K_s = the rate in lb/in of the secondary suspension at one side of the truck.

K_ϕ was determined experimentally by parking the vehicle on track having a crosslevel angle θ and measuring the body roll angle ϕ. K_ϕ can be computed as:

$$K_{\phi} = \frac{(H_m - H_{rc})\ mg \sin \theta}{12_{\phi}}$$

where H_m is the body c.g. height and m is the body mass.

The weight offset at the instrumented truck was determined by measuring the weight vector intercept and averaging over several tangent sections. The weight offset is considered as a vehicle c.g. offset in a half vehicle model but it is possible that an opposite offset would be measured at the rear truck and the vehicle c.g. is actually on the centerline.

TABLE D-1

AMCOACH II (Soft Primary)

	SPECIFICATION
Truck Weight, ng or W1	14,500 lb
Half Body Weight, mg or W2	38,475 lb+6000 lb Load
Truck c.g. Height, H_n or H1	22.2 in
Body c.g. Height, H_m or H2	75.3 in
Vehicle c.g. Height	62.8 in
Effective Roll Center Height, H_{rc} or H3	36 in. until air spring bottoms 32 in. after bottoming
Overall Roll Rate, K_ϕ	~8,000 ft-lb/degree until air spring bottoms ~15,000 ft-lb/degree after bottoming
Secondary Lateral Spring Rate, K_L	5,000 lb/in
Lateral Compliance	1-1/2 in.
Weight Offset	0
Primary Suspension:	
Wheel Rate, K_p	25,900 lb/in
Lateral Wheel Rate	100,000 lb/in
Longitudinal Wheel Rate	60,000 lb/in
Lateral Spacing, L_p	46 in
Spring Top Height, H_p	18 in
Secondary Suspension:	
Vertical Spring Rate per side, K_s	1,859 lb/in until air spring buttoms 4,780 lb/in after bottoming
Vertical Spring Spacing, L_s	90 in
Vertical Spring Top Height, H_s	40 in
Truck Wheelbase	102 in
Wheel Diameter	36 in
Truck Spacing (Vehicle Wheelbase)	714 in
Body Length	85 ft
Body Height	9 ft
Approximate Lateral Surface Area	765 ft^2
Approximate Height of Center of Wind Pressure	7-1/2 ft

TABLE D-2

F40PH (Inclined Rubber Suspension)

	SPECIFICATION
Truck Weight, ng or W1	34,235 lb
Half Body Weight, mg or W2	93,510 lb
Truck c.g. Height, H_n or H1	28.6 in
Body c.g. Height, H_m or H2	85.8 in
Vehicle c.g. Height	70.5 in
Effective Roll Center Height, H_{rc} or H3	31.2 in.
Overall Roll Rate, K_ϕ	35,432 ft-lb/deg
Secondary Lateral Spring Rate, K_L	3,500 lb/in
Lateral Compliance	2.25 in
Weight Offset	
Primary Suspension:	
Wheel Rate, K_p	6,750 lb/in
Lateral Wheel Rate	Avg. 26,000 lb/in to 1/2" stop
Lateral Spacing, L_p	79 in
Spring Top Height, H_p	41 in
Secondary Suspension:	
Vertical Spring Rate per side, K_s	20,000 lb/in
Vertical Spring Spacing, L_s	76 in
Vertical Spring Top Height, H_s	22.5 in
Truck Wheelbase	108 in
Wheel Diameter	40 in
Truck Spacing (Vehicle Wheelbase)	396 in
Body Length	54 ft
Body Height	12 ft
Approximate Lateral Surface Area	730 ft^2
Approximate Height of Center of Wind Pressure	8-1/4 ft

TABLE D-3

F40PH (GP Standard Suspension)

	SPECIFICATION
Truck Weight, ng or W1	34,235 lb
Half Body Weight, mg or W2	93,510 lb
Truck c.g. Height, H_n or H1	28.6 in
Body c.g. Height, H_m or H2	85.8 in
Vehicle c.g. Height	70.5 in
Effective Roll Center Height, H_{rc} or H3	35.5 in
Overall Roll Rate, K_ϕ	48,873 ft-lb/deg
Secondary Lateral Spring Rate, K_L	3,970 lb/in
Lateral Compliance	2.25 in
Weight Offset	
Primary Suspension:	
Wheel Rate, K_p	6,750 lb/in
Lateral Wheel Rate	Avg. 26,000 lb/in to 1/2" stop
Lateral Spacing, L_p	79 in
Spring Top Height, H_p	41 in
Secondary Suspension:	
Vertical Spring Rate per side, K_s	57,500 lb/in
Vertical Spring Spacing, L_s	76 in
Vertical Spring Top Height, H_s	22.5 in
Truck Wheelbase	108 in
Wheel Diameter	40 in
Truck Spacing (Vehicle Wheelbase)	396 in
Body Length	54 ft
Body Height	12 ft
Approximate Lateral Surface Area	730 ft^2
Approximate Height of Center of Wind Pressure	8-1/4 ft

APPENDIX E

DUAL UNIT LOCOMOTIVE CONFIGURATION

A train consisting of Amcoachs pulled by a single F40PH locomotive satifies the operating safety criteria at 8 inches cant deficiency on the Northeast Corridor at all but a very few perturbed curves. The limiting factor is the vertical load transfer (overtuning) of the coach. The transfer of vertical load from the inside wheels to the outside wheels at a curve is the result of lateral acceleration, body c.g. movement from suspension deflections, and the potential force of high cross wind. The general cant deficiency limit of the single F40PH locomotive at curves without severe perturbations was 9 inches. The cant deficiency was limited by vertical load transfer rather than by lateral forces (track shift) or L/V ratios (rail rollover and wheel climb). The cant deficiency limit of the locomotive was greater than that of the Amcoach because the allowance for high cross winds had less influence on the short, heavy locomotive. It is not anticipated that the standard Amcoach, because of ride quality considerations, will be operated at cant deficiencies greater than 6 inches. Permissible cant deficiency for the single F40PH locomotive was greater than 6 inches even at the most severely perturbed curves in the New Haven - Boston test zone.

The independence of vertical load transfer between coupled vehicles, even with great differences in roll angle, was demonstrated by running the banking Amcoach behind the instrumented F40PH. Repeated runs with and without banking operation revealed that the couplers could not develop enough torque to influence vertical load transfer.

Lateral forces of 4 axle locomotives also appear to be largely equivalent between leading and trailing locomotives. Tests have been performed using rails instrumented by Battelle Columbus

Laboratories to compare the lateral forces at various axles of a train. Wayside measurements of dual unit locomotives at intentionally perturbed test curves have been performed (ref. 26), and a few dual unit Amtrak trains with trailing F40PH locomotives were measured at a curve on the Northeast Corridor (Ref. 25). The available data indicate that the maximum axle lateral forces of the E8, GP40-2 and F40PH do not differ significantly between leading and trailing units. However, leading and trailing SDP40F 6 axle units have shown significant differences, and the greatest transient axle force of the dual unit SDP40F was as much as 30 percent higher than the greatest axle force of a single unit at the worst case perturbed curve at the highest speed. The highest transient axle lateral force of the F40PH tested on the NEC (with continuous measurements from instrumented wheels on the front truck) was less than 21,000 lb at 9.7 inches cant deficiency (curve 109 track 1); the critical level indicated by the lateral track shift criterion (section 5.3) is 41,900 lb. The available measurements of the trailing F40PH locomotive axle force suggest that the trailing unit axle forces are not different than single unit axle forces. Even if the F40PH trailing unit were to experience transient lateral axle forces 30 percent higher than the single unit at some perturbed curves (as in the case of the 6-axle SPD40F), axle forces of 27,000 lb would still be very low compared to the 41,900 lb critical level. If the maximum wheel and truck side L/V ratios were amplified by 30 percent at a trailing unit, they would also remain below critical values by a considerable margin. A dual configuration of F40PH units towing Amcoaches could certainly operate safely up to the 8 inch cant deficiency limit imposed by the Amcoaches.

Figures E-1 to E-7 were taken from reference 26 to summarize the tests of dual units at the specially perturbed curves at the Transportation Test Center. Section 4.4 refers to section of the perturbed track of nominal 1-1/2^{o} curvature and 3 inch crosslevel with a piecewise linear alignment perturbation; section 5.23 of the perturbed track has the same nominal specifications but includes both a rectified sinusoidal alignment perturbation and a profile perturbation. About ten rail spots at intervals of about 4 feet in the perturbed area of each curve were instrumented with strain gages and calibrated to measure the instanteous lateral force of a passing wheel. The lateral wheel force plotted in the figures was the greatest of the ten instanteous measurements of a particular axle as it passed over all of the instrumented spots.

Figures E-1 and E-2 compare the high rail wheel forces versus speed for the 12 axles of a dual unit combination of E8 locomotives. Configuration "A" had both locomotive units facing forward. The axle forces of trucks 1 and 3 are similar at both curves, and the same is true for trucks 2 and 4. The wheel forces are independent of the position of the locomotive unit in the consist, and presumably the wheel forces of the lead locomotive unit are independent of the existance of the trailing locomotive unit. In general, the lead axle of the lead trucks of the E-8 units experienced the greatest transient lateral forces.

Figures E-3 and E-4 show corresponding data for dual SDP40F locomotives. Curve section 4.4 was rebuilt after the E8 test to increase the perturbation, and it induced extremely high transient lateral forces in the SDP40F. The force at analogous axles of trucks 1 and 3 are not similar, and those of truck 2 and 4 also differ considerably. The SDP40F locomotive unit appears to be sensitive to its position in the consist with higher lateral forces at the trailing unit. Figure E-5 shows the corresonding lateral forces of a single unit SDP40F (with the same freight cars) at the severely perturbed curve 4.4. A maximum lateral force of 37,000 lb occurred at the trailing axle of the lead truck of the single unit (Figure E-5). A maximum lateral force of 48,000 lb was recorded for the corresponding axle of the trailing unit of dual configuration "A" (Figure E-4). The maximum lateral wheel force of the dual unit was about 30 percent greater than that of the single unit at a curve perturbed enough to create an extreme situation for the SDP40F. The lateral wheel

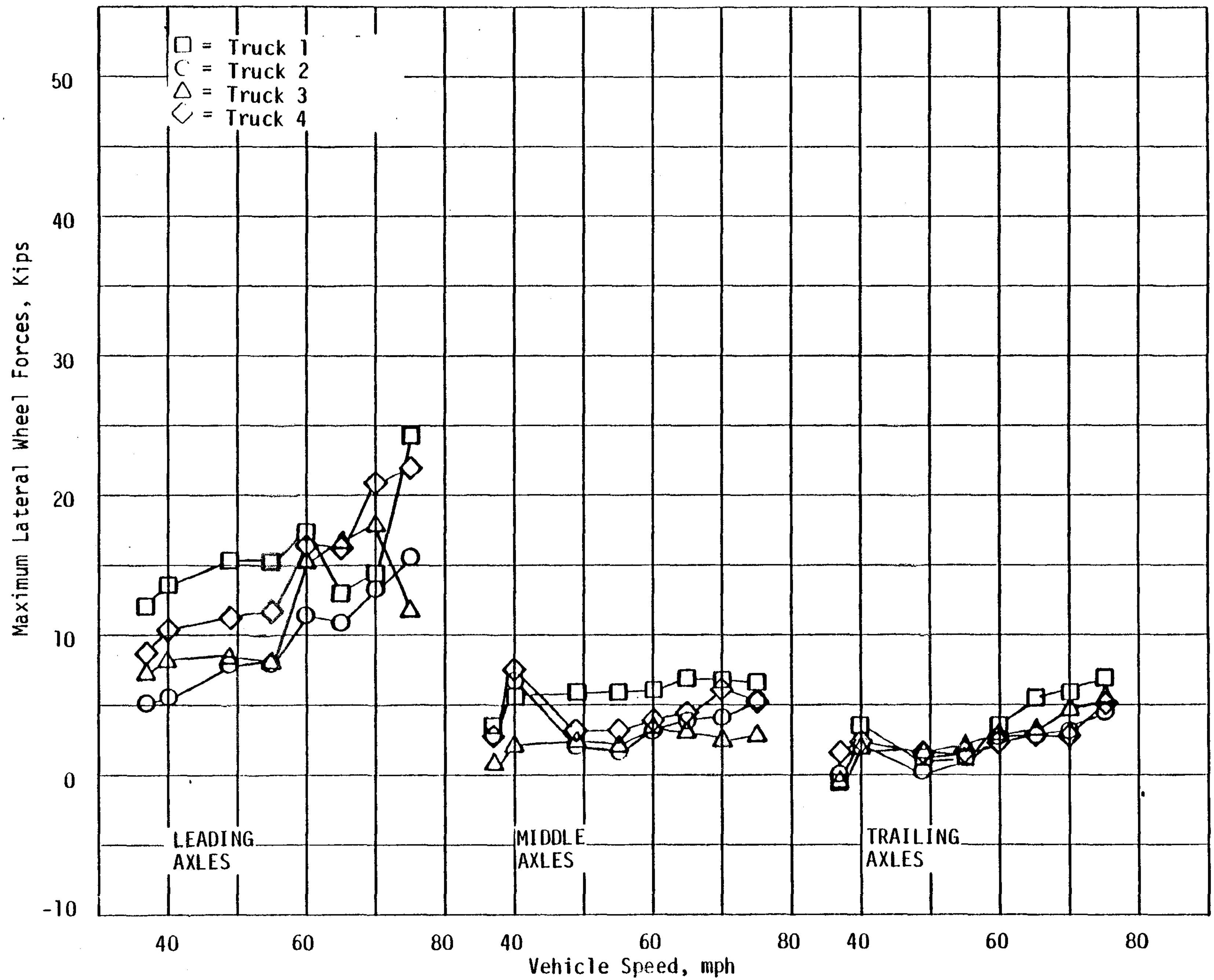


FIGURE E-1 MAXIMUM LATERAL WHEEL FORCES FOR THE LEADING, MIDDLE, AND TRAILING AXLES OF E8,

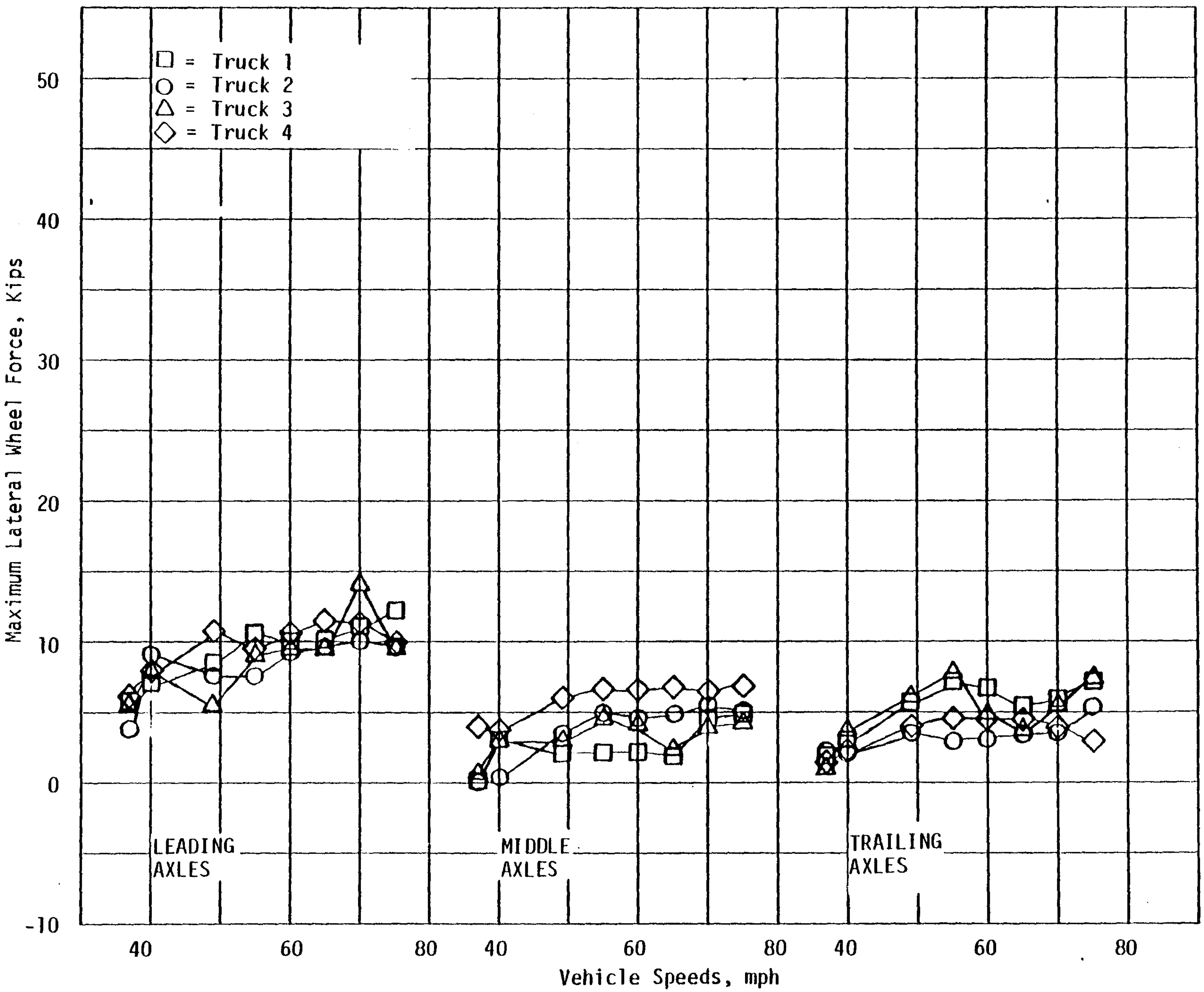


FIGURE E-2. MAXIMUM LATERAL WHEEL FORCES FOR THE LEADING, MIDDLE, AND TRAILING AXLES OF E8, CONFIGURATION "A", TRAVERSING SECTION 4.4 OF THE PERTURBED TRACK

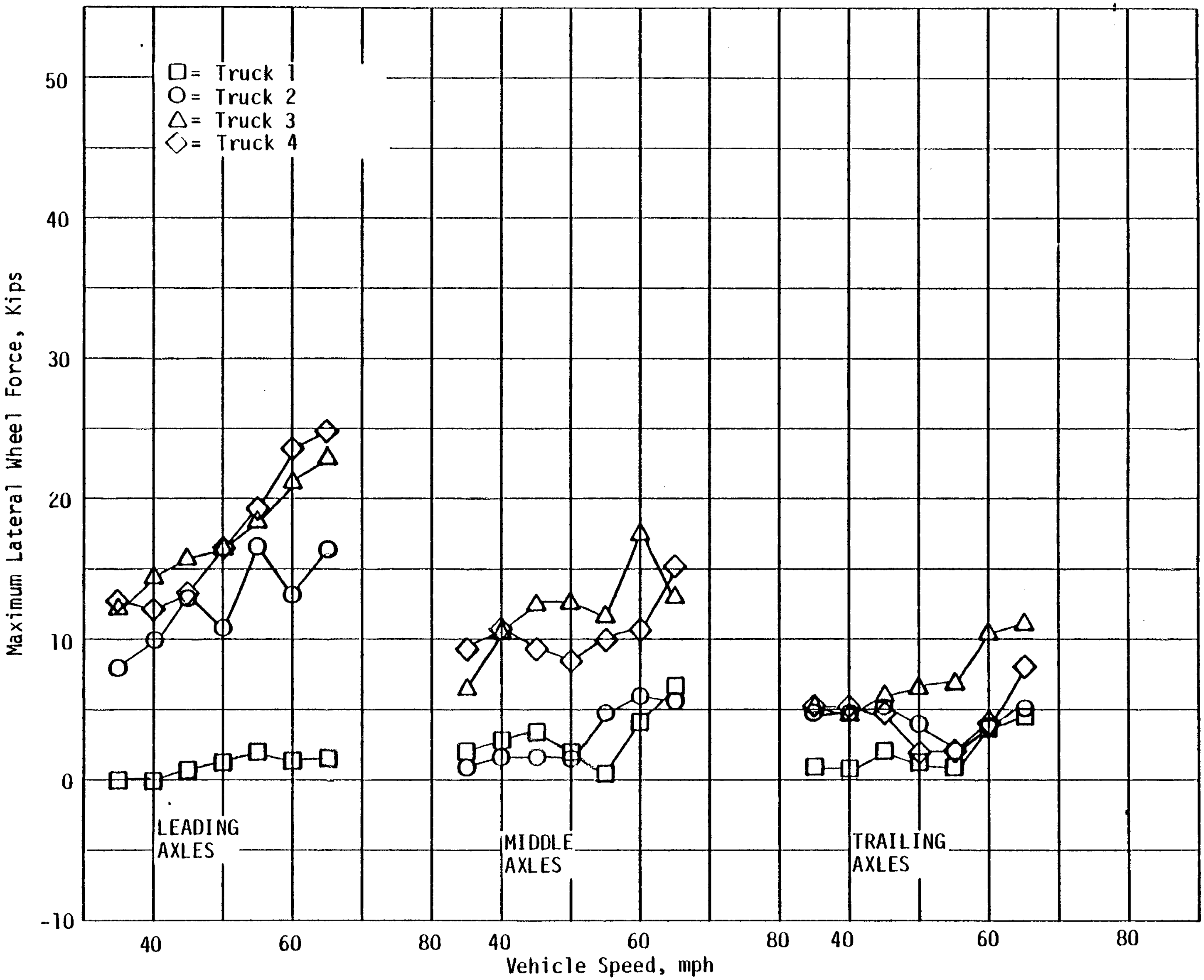


FIGURE E-3. MAXIMUM LATERAL WHEEL FORCES FOR THE LEADING, MIDDLE, AND TRAILING AXLES OF SDP40F, CONFIGURATION "A" TRAVERSING SECTION 5.23 OF THE PERTURBED TRACK

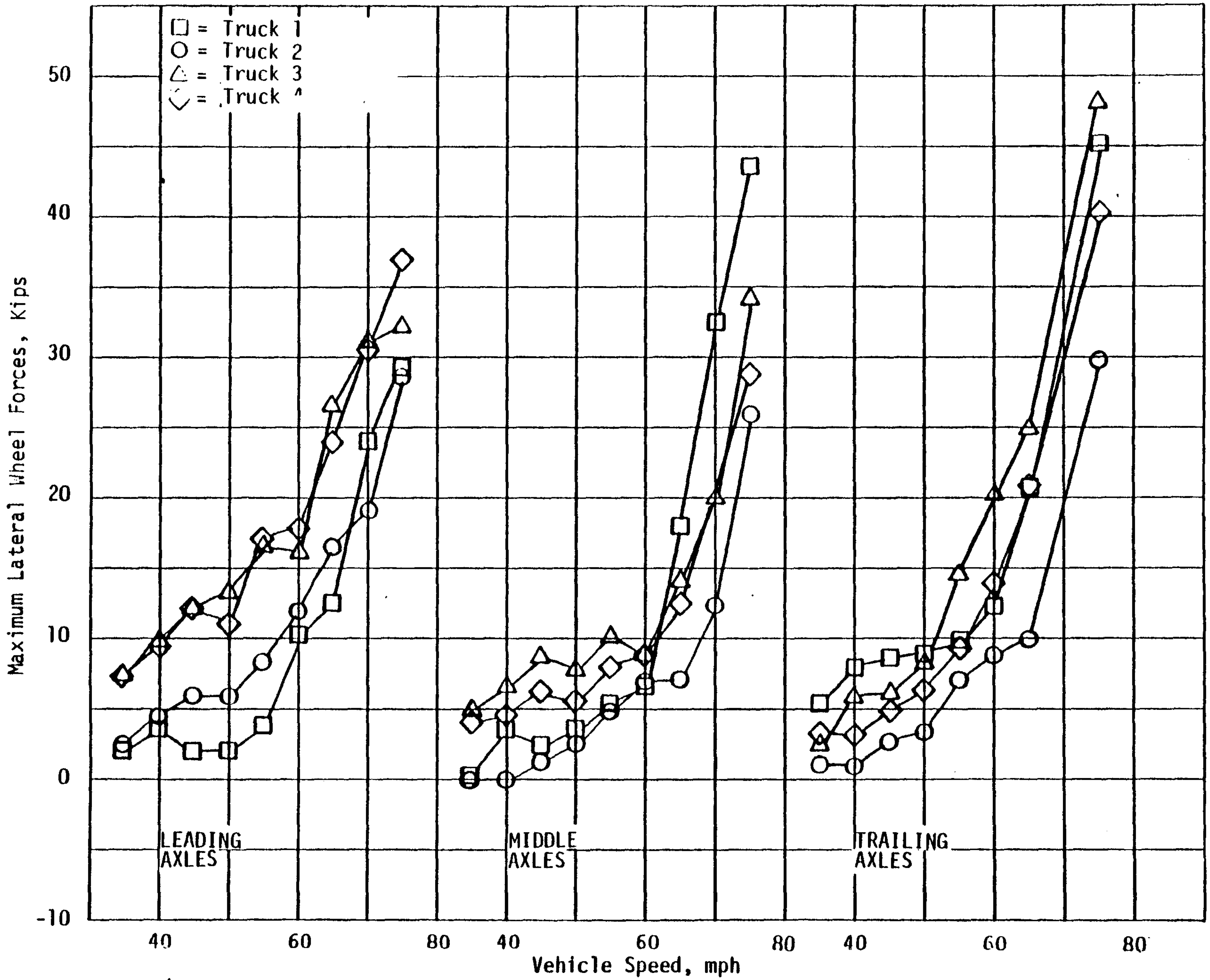


FIGURE E-4. MAXIMUM LATERAL WHEEL FORCES FOR THE LEADING, MIDDLE, AND TRAILING AXLES OF SDP40F, CONFIGURATION "A", TRAVERSING SECTION 4.4 OF THE PERTURBED TRACK

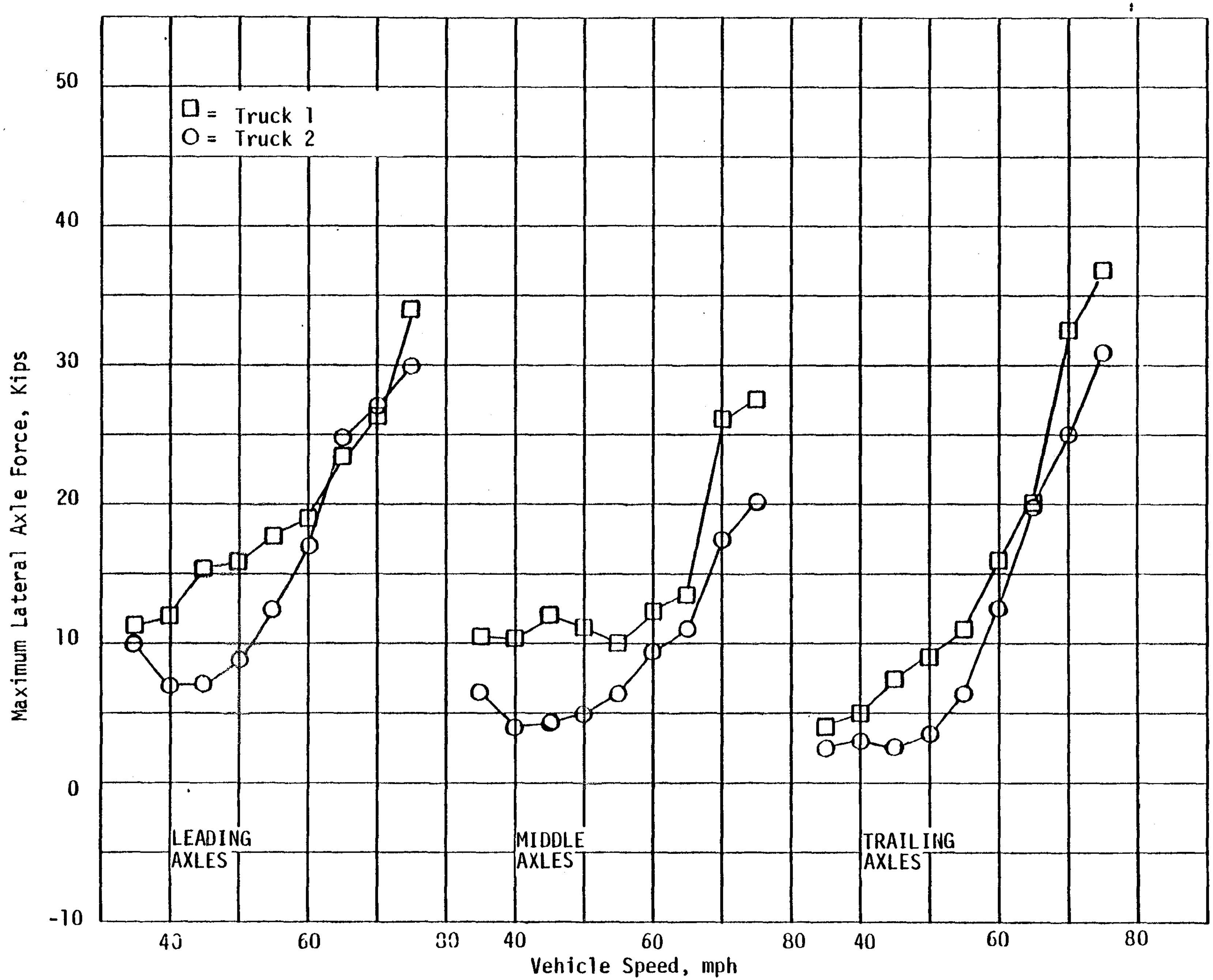


FIGURE E-5. MAXIMUM LATERAL AXLE FORCES FOR THE LEADING, MIDDLE AND TRAILING AXLES OF SDP40F, forces of the single F40PH unit (having the same vertical wheel load) tested on the Northeast Corridor never exceeded 21,000 lb even at almost 12 inches cant deficiency. The critical lateral force of the SDP40F was also about 42,000 lb with regard to safety against lateral track shift. Figure E-4 shows forces exceeding the safety criteria, and it is possible that even higher forces occurred between locations of rail instrumentation. Lateral track shift occurred during the testing of the SDP40F.

Figures E-6 and E-7 show the maximum lateral wheel forces at the lead axles of each truck of a dual configuration of GP40-2 units superimposed on previous data for the dual E8 and SDP40F locomotives at the same curves. The GP40-2 is a four axle freight locomotive similar in many respects to the F40PH. The maximum transient lateral wheel forces were similar for the leading and trailing units unlike the SDP40F, and curve section 4.4 did not produce the extreme lateral forces observed for the SDP40F. Figure E-8 shows the moderate axle forces of the dual GP40-2 units as functions of distance throughout the perturbed curve section 4.4. The average force levels as well as the maximum transients were uniform between the two 4 axle locomotive units at the same curve where great differences were observed between two 6 axle SDP40F locomotive units.

Table E-1 lists measurements of lateral wheel forces of the F40PH locomotive at curve 67 track 2 on the Northeast Corridor at Bradford, RI. The first two sections of the table are the maximum instantaneous measurements of revenue trains passing over 6 instrumented spots on the high rail. The measurements at corresponding axle positions were averaged to eliminate considerable differences between locomotives due in large part to wheel wear; a wide range of speeds was not possible because all revenue traffic was attempting the same speed. The average wheel force for corresponding axles of leading and trailing F40PH locomotive units does not appear to differ significantly although the data

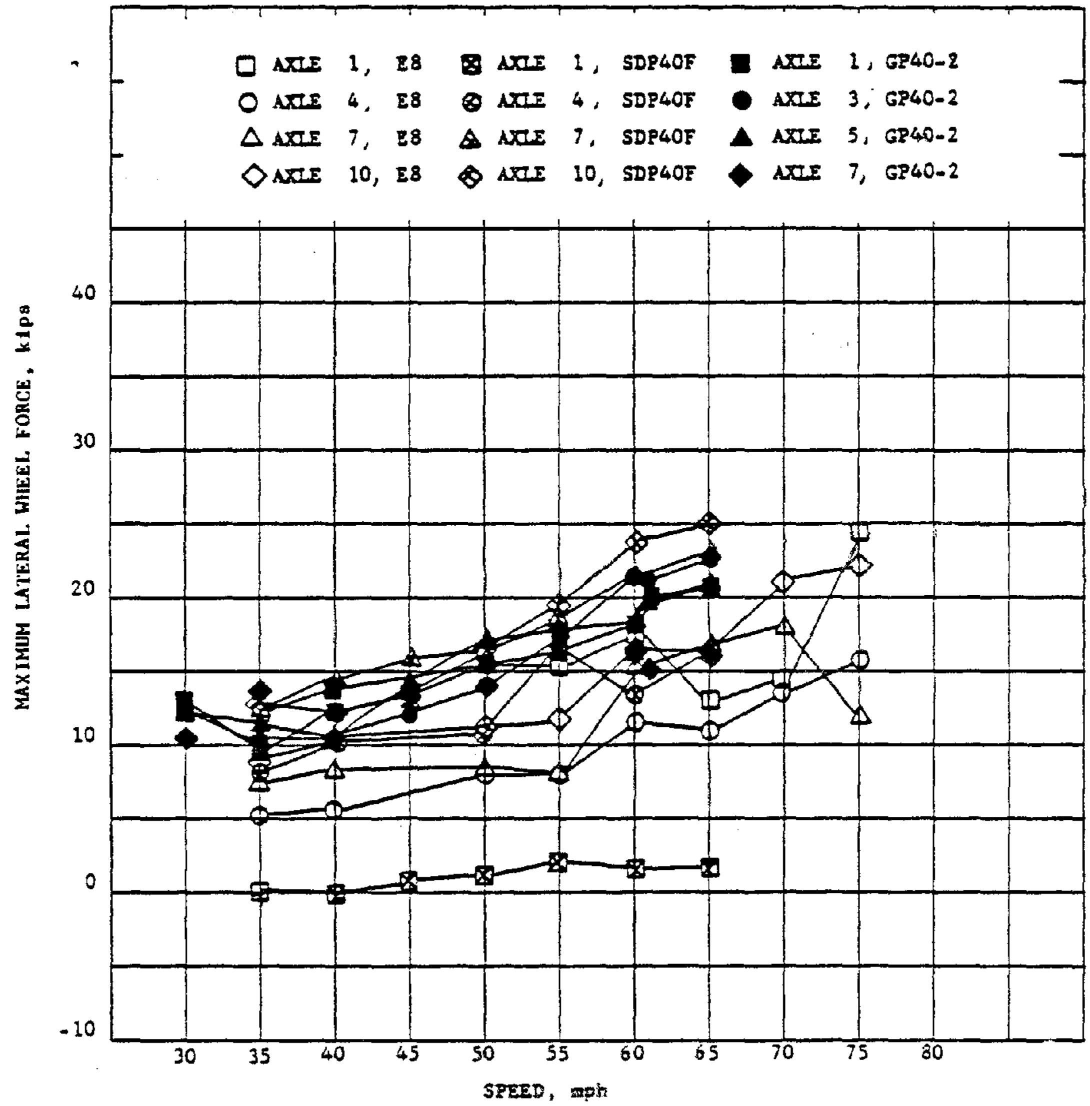


FIGURE E-6. MAXIMUM LATERAL WHEEL FORCE LOCOMOTIVE LEAD AXLES IN SECTION 5.23

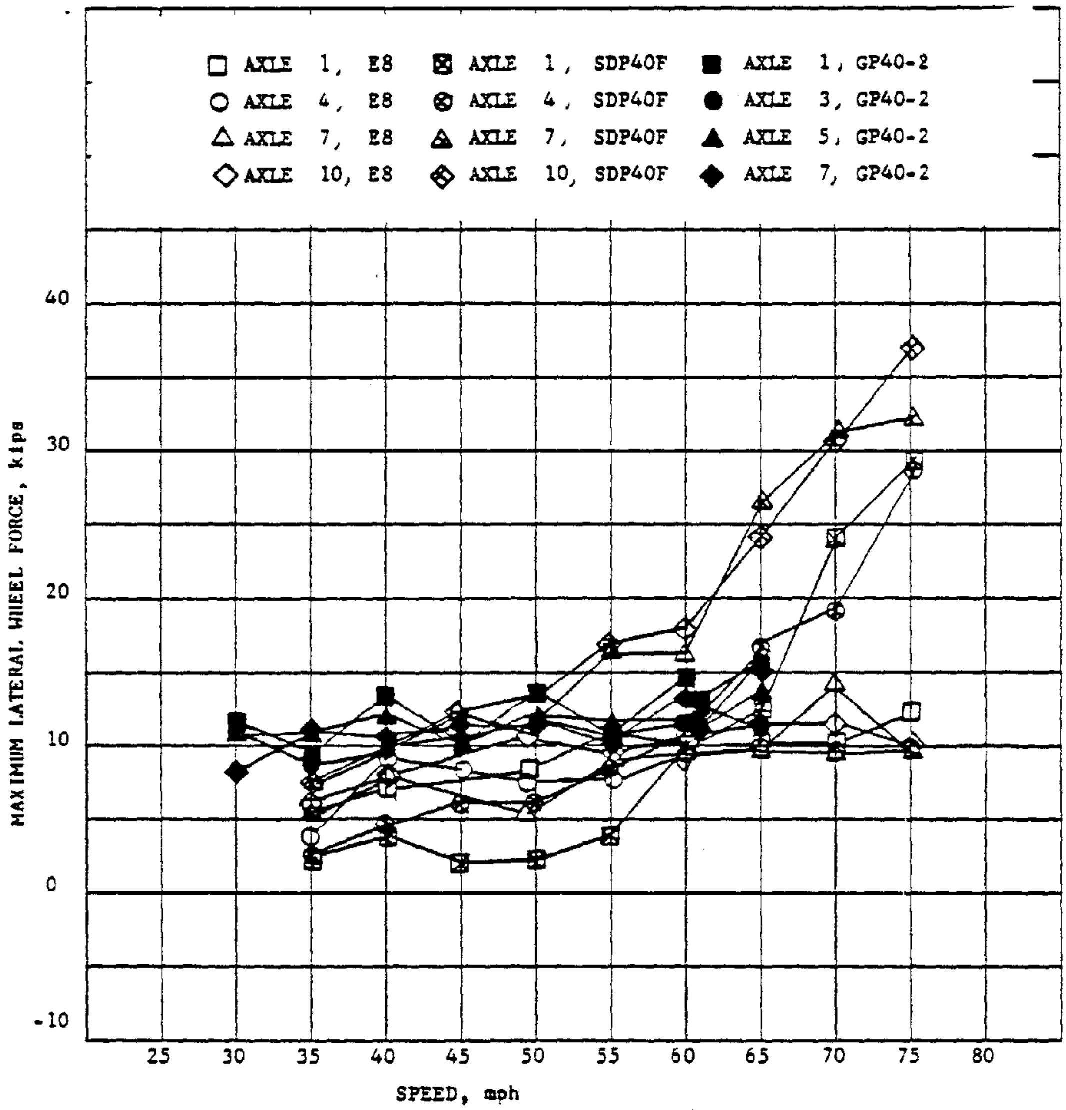


FIGURE E-7. MAXIMUM LATERAL WHEEL FORCE FOR LOCOMOTIVE LEAD AXLES IN SECTION 4.3-4.4

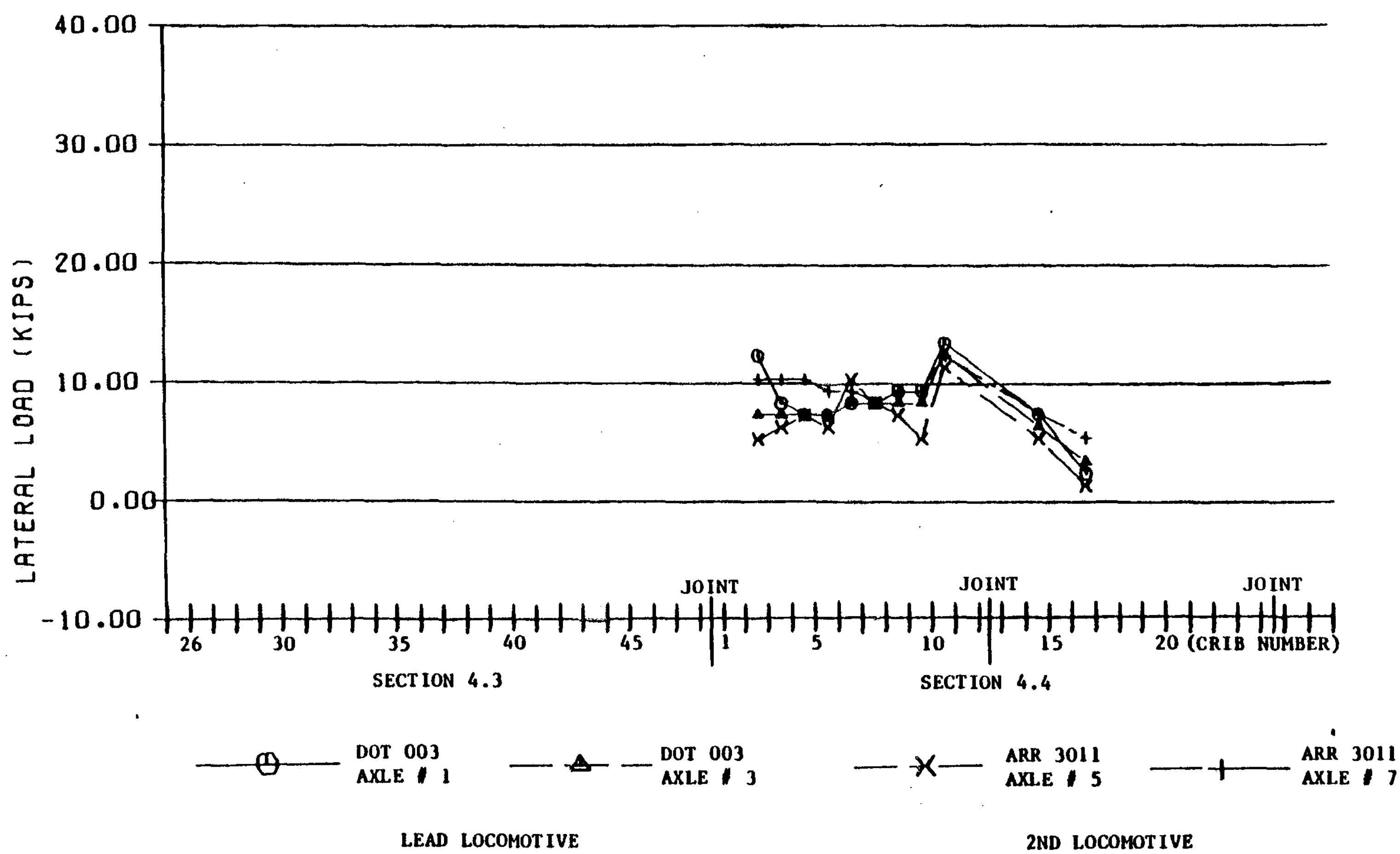


FIGURE E-8. 11-CAR FREIGHT CONSIST (2/3/79) FROM BCL WAYSIDE INSTRUMENTATION, SECTION 4 - LEAD AXLES OF BOTH TRUCKS FOR DUAL GP40-2 - SPEED 60 MPH

TABLE E-1

WAYSIDE DATA SUMMARY CURVE 67 TRACK 2
MAXIMUM LATERAL FORCES - KIPS

TRAILING F40PH (1980 TEST)

Run	Speed	Axle 1	Axle 2	Axle 3	Axle 4	Truck
26-5	65	2.7	2.4	3.9	2.1	3.9 (T)
27-12	67	6.3	2.2	2.4	5.0	7.0 (L)
22-3	72	6.8	4.8	4.6	5.5	10.6 (L)
Avg	68	5.3	3.1	3.6	4.2	7.2
Sd		2.2	1.4	1.1	1.8	3.4

LEADING F40PH (1980 TEST)

Run	Speed	Axle 1	Axle 2	Axle 3	Axle 4	Truck
27-12	67	3.0	4.2	3.9	2.4	4.8
2-6	67	8.7	4.2	5.1	3.3	8.7
30-1	68	1.8	3.0	-2.1	4.2	4.2
31-5	69	6.9	3.9	5.4	2.4	9.0 (L)
31-10	69	2.1	5.1	4.8	3.0	6.6
1-12	69	2.4	3.3	2.7	3.0.	4.5
26-4	70	8.7	5.1	5.1	3.0	12.1
27-4	70	5.4	3.6	3.3	4.2	6.0
30-6	71	-3.0	4.8	4.2	3.9	7.5
30-8	71	8.7	3.9	8.1	3.0	10.8
31-8	71	4.5	5.1	3.6	6.6	9.6 (T)
1-9	71	6.0	5.1	-1.5	6.6	11.1
29-9	72	2.4	4.2	2.4	3.9	6.3
2-11	72	3.6	5.7	2.1	5.4	8.4
26-9	72	5.1	6.0	6.3	3.6	8.4 (L)
1-13	75	8.1	5.1	8.4	4.2	10.8
30-4	76	6.3	5.4	5.4	4.8	11.7 (L)
31-12	76	7.8	5.1	7.2	4.5	12.3 (L)
Avg	71	4.9	4.6	4.1	4.0	8.5
Sd		3.1	0.8	2.8	1.2	2.7

LEADING F40PH WITH INSTRUMENTED WHEELS (1982 TEST)

Run	Speed	Axle 1	Axle 2	Axle 3	Axle 4	Truck
1E	68	9.4	4.6	-	-	10.2 (L)
2E	70	9.4	4.6	-	-	10.8 (L)
3E	70	8.3	4.0	-	-	9.6 (L)
4E	78	11.5	4.4	-	-	14.1 (L)
6E	78	9.0	3.1	-	-	11.8 (L)
Avg	73	9.5	4.1			11.3
Sd		1.1	0.6			3.4

LEADING E-8 (1980 TEST)

Run	Speed	Axle 1	Axle 2	Axle 3	Axle 4	Axle 5	Axle 6	Truck
26-5	65	5.4	1.2	4.4	7.2	5.1	3.0	13.6 (T)

is limited. The maximum forces occurred at the leading axle for both units.

Measurements at the same curve with instrumented wheelsets of the F40PH tested in this program are included for reference in the third section of the table. The lead axle of this locomotive with freshly turned wheels seemed to bear greater lateral forces relative to the second axle than many revenue trains. The lateral force measurements of the test locomotive were consistent with the revenue train measurements since maximum transient measurements with instrumented wheels usually exceed wayside measurements because they can include peaks not coinciding with distinct instrumented rail spots.

The maximum truck forces were estimated by summing measurements of leading and trailing wheels occurring at the same instant. Fewer instrumented track locations were available than for wheel forces because the whole truck must be within the instrumented rail zone. Equivalence between leading and trailing locomotive units was observed, and consistency with the instrumented test locomotive was evident. The greatest force normally occurred at the lead truck.

The wayside truck force measurement of an E8 unit listed in the forth part of the table was greater than the wayside measurements of any F40PH.

In view of the low lateral wheel forces of the F40PH and the apparent equivalence between leading and trailing units it is extremely unlikely that limiting values of transient lateral axle force could occur at less than the 8 inch safe cant deficiency of the towed Amcoaches.