7641

OPERATIONAL TEST PLAN
F40PH/BANKING AMCOACH
CANT DEFICIENCY TEST

ENSCO, INC.®

REVISION A - 5/13/82

OPERATIONAL TEST PLAN
F40PH/BANKING AMCOACH
CANT DEFICIENCY TEST

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Task 207

Prepared for:

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Federal Railroad Administration
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Washington, DC 20590

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1.0 INTRODUCTION

The objectives of testing the F40PH and the prototype banking Amcoach at high cant deficiency are:

- o To evaluate the maximum safe curving speeds of the F40PH diesel locomotive and Amcoach specially modified to tilt in the curves.
- o To measure the level of passenger comfort in high speed curving and the effectiveness of the banking system.
- o To make direct comparisons of the curving performance of these vheicles to other advanced and conventional vehicles.

The principal sensors used for this test are wheelsets instrumented with strain gage bridges to sense vertical and lateral wheel/rail forces. The wheel forces and summations and ratios derived from them are used to indicate the danger of various modes of derailment. A data acquisition system is used to provide real-time computations and display of safety information and to record up to 64 data channels. The same equipment also accomplishes statistical post processing and graphics.

The test is designed to probe the curving safety limits by a series of orderly steps. The first step is a computation of steady state weight transfer, lateral acceleration, and lateral truck force as a function of cant deficiency based on manufacturer's specifications. This step establishes the first estimate of speed limitations for test planning. The second step is a static lean test (on approximately 6 inches of crosslevel) to estimate the "as installed" suspension constants of the particular test vehicle. The results of the static lean tests are used to refine the computation in step 1 so that the test runs can be monitored better and the results quickly interpretated. Appendix B gives the steady state computations for the F40PH locomotive and the banking Amcoach.

The third step is the steady state testing. Repeated runs at increasing speed over two well known curves (one right and one left) establish the steady state limits and give an indication of the gradient of peak forces with increases in cant deficiency. The forth step is the over-the-road testing which includes many curves and a typical array of track perturbations. The transient wheel forces which occur below the steady state cant deficiency limit are of major interest. This step is used to determine whether the ultimate limit on safe curving speed is set by steady state or by transient curving considerations.

The techniques and equipment developed under the Office of Passenger and Freight Systems/Amtrak for high cant deficiency testing of the LRC train, Standard Amcoach and AEM-7 Locomotive are fundamental to this program. The test method and safety criteria used for the previous high speed curving test of the LRC Train will be used as the basis for this test program.

2.0 TESTING

2.1 GENERAL

The F40PH/Banking Amcoach High Cant Deficinecy Test has been divided into three parts. In the first part the locmotive and banking coach will be run repetatively through right and left test curves. The speed will be increased incrementally to well beyond the currently speed to investigate steady state performance. In the second part, over-the-road tests will be performed to determine the worst case transient reaction for a wide variety of curves. And in the third part, test runs will be made at high curving speeds to demonstrate the ride and performance achieved by operating at the higher curving speeds.

2.2 CONSIST ASSEMBLY

The consist for all phases of the this test will be configured as shown in Section 3.0. The test cars will be balanced as described in Section 3.0.

As described in detail in Section 5.0 the instrumentation and data recording system will be installed in the tilt body Amcoach which will be the first car in the train. Instrumented wheels will be installed on the locomotive and coach being tested. After the equipment is installed cabling will run between the locomotive and test car and between the test car and first Amcoach behind the test car.

Shop services will be required for installation of the instrumented wheels and the other instrumentation. During final check out of the instrumentation and data collection system, three phase power at the coach power cabinet will be required.

2.3 STATIC LEAN TEST AND DYNAMIC PRE-TEST

The consist configured as descrbed in Section 3.0 will be run from New Haven to Groton and back. Instrumentation checks, static lean tests on the locomotive and coach, and final adjustments to the coach banking controls will be made. The pre-test runs should be scheduled to allow for making static lean measurements on the test curve or other high cross level sections of track. To make the lean measurement the consist will need to be parked on the track for approixmately 45 minutes. These measurements need to be made on both right- and left-hand curves.

The static lean and dynamic check out runs will be used to check both the operation of the test instrumentation and data collection system and the banking coach system. Data will be collected and processed from these test runs to confirm that the systems are operating properly but no data will be recorded or distributed.

2.4 PART A - REPEATED STEADY STATE CURVING TESTS

The locomotive and banking Amcoach will be instrumented as described in Section 5.0 and the consist arranged as described in Section 3.0. The repeated steady state curving test will consist of repeated runs at increasing speeds through the eastbound curve 67 on track 2 and similar westbound runs through curve 67 on track 1. For these tests the consist containing the instrumented locomotive and coach equipment will be run through the test curve at speeds which correspond to elevated cant deficiencies listed in Table 2.1.

2.5 PART B - OVER-THE-ROAD TESTS

The same consist as used for the repeated static curving tests will be run from New Haven to Groton at speeds which correspond to elevated cant deficiencies as shown in Table 2.2. For the over-the-road test, the consist speed will be controlled so that

TABLE 2-1
TEST SCHEDULE

PART A - REPEATED STEADY STATE

Day	Run	Cant	SPD	Zone	Special Notes
1	1	3"	71	Track #2 MP 138-153 East	Data taken from MP 144-146
1	2	3"	71	Track #2 MP 138-153 East	Data taken from MP 144-146
1	3	5"	79	Track #2 MP 138-153 East	Data taken from MP 144-146
1	4	5"	79	Track #2 MP 138-153 East	Data taken from MP 144-146
1	5	7"	85	Track #2 MP 138-153 East	Data taken from MP 144-146
1	6	7"	85	Track #2 MP 138-153 East	Data taken from MP 144-146
2	7	8"	88	Track #2 MP 138-153 East	Data taken from MP 144-146
2	8	8"	88	Track #2 MP 138-153 East	Data taken from MP 144-146
2	9	9"	92	Track #2 MP 138-153 East	Data taken from MP 144-146
2	10	9"	92	Track #2 MP 138-153 East	Data taken from MP 144-146
2	11	10"	94	Track #2 MP 138-153 East	Data taken from MP 144-146
2	12	10"	94	Track #2 MP 138-153 East	Data taken from MP 144-146
2	13	11"	97	Track #2 MP 138-153 East	Data taken from MP 144-146
2	14	11"	97	Track #2 MP 138-153 East	Data taken from MP 144-146

TABLE 2-1 (Cont'd)

TEST SCHEDULE

PART A - REPEATED STEADY STATE

Day	Run	Cant	SPD	Zone	Special Notes
3	1	3"	71	Track #1 MP 138-153 West	Data taken from MP 144-146
3	2	3"	71	Track #1 MP 138-153 West	Data taken from MP 144-146
3	3	5"	79	Track #1 MP 138-153 West	Data taken from MP 144-146
3	4	5"	79	Track #1 MP 138-153 West	Data taken from MP 144-146
3	5	7"	85	Track #1 MP 138-153 West	Data taken from MP 144-146
3	6	7"	85	Track #1 MP 138-153 West	Data taken from MP 144-146
3	7	8"	88	Track #1 MP 138-153 West	Data taken from MP 144-146
4	8	8"	88	Track #1 MP 138-153 West	Data taken from MP 144-146
4	9	9"	92	Track #1 MP 138-153 West	Data taken from MP 144-146
4	10	9"	92	Track #1 MP 138-153 West	Data taken from MP 144-146
4	11	10"	94	Track #1 MP 138-153 West	Data taken from MP 144-146
4	12	10"	94	Track #1 MP 138-153 West	Data taken from MP 144-146
4	13	11"	97	Track #1 MP 138-153 West	Data taken from MP 146-144
4	14	11"	97	Track #1 MP 138-153 West	Data taken from MP 146-144

the curves in the test zone (New Haven to Groten) will be negotiated at the elevated cant deficiency. The speed profile for these test runs will be predetermined to allow adequate distance for braking and accelerating. The result from each of these high speed runs will be examined and the targeted speeds will be modified to maintain safe operations.

The over-the-road test will be repeated at increasing levels of cant deficiency as indicated by Table 2.2. The results from each test will be reviewed and curves producing high results will be monitored carefully and restricted when the limits established in Section 9.0 would be exceeded by higher speed operation.

Allowance for wind speed is not required for test safety because actual wheel forces are displayed constantly. However, tests should not be conducted when wind gusts over 40 mph are forecasted in order to avoid data scatter.

Speed profiles for each of the over-the-road tests listed in Table 2.2 will be developed prior to the test and will be included as an appendix of this test plan. The detailed speed profile will be distributed well in advance of the test so that special track clearances and operating procedures can be worked out.

2.6 PART C - DEMONSTRATION TEST

The demonstration runs will be round trips from New Haven to Boston. The test will be made at an elevated speed corresponding to a cant deficiency which can be safely achieved with the F40 locomotive, the banking Amcoach, and standard Amcoach cars. A target profile will be selected on the basis of the repeated steady state test and the over-the-road tests. The first round trip to Boston will be made at speeds which are less than the target speed so that the curves beyond the zone used for the over-the-road test can be examined. If the results of the first round trip to Boston confirm that the target speed profile can be

TABLE 2-2

TEST SCHEDULE

PART B - OVER-THE-ROAD TEST

Day	Run	Cant	SPD	Zone	Special Notes
1	1	5"	PRO*	Track #2 MP 72-125	Data taken from MP 80-120
2	3	7"	PRO*	Track #2 MP 72-125	Data taken from MP 80-120
2	4	7"	PRO*	Track #1 MP 125-72	Data taken from MP 120-80
2	5	7"	PRO*	Track #2 MP 72-125	Data taken from MP 80-120
3	6	7"	PRO*	Track #1 MP 125-72	Data taken from MP 120-80
4	7	9"	PRO*	Track #2 MP 72-125	Data taken from MP 80-120
4	8	9"	PRO*	Track #1 MP 125-72	Data taken from MP 120-80
4	9	9"	PRO*	Track #2 MP 72-125	Data taken from MP 80-120
4	10	9"	PRO*	Track #1 MP 125-72	Data taken from MP 120-80

*PRO - speed profile developed from track class and test cant.

NOTE: All above runs will be restricted if wind gusts exceed 50 mph.

TABLE 2-2 (Cont'd)

TEST SCHEDULE

PART C - DEMONSTRATION RUNS, NEW HAVEN TO BOSTON

Day	Run	Cant	SPD	Zone	Special Notes
1	1	9/5	PRO*	Track #2 MP 70-228 East	Data taken from MP 80-220
1	2	5/9	PRO*	Track #1 MP 228-70 West	Data taken from MP 220-80
3	3	9/7	PRO*	Track #2 MP 70-228 East	Data taken from MP 80-220
3	4	7/9	PRO*	Track #1 MP 228-70 West	Data taken from MP 220-80
4	5	TBD**	PRO*	Track #2 MP 70-228 East	Data taken from MP 80-220
4	6	TBD**	PRO*	Track #1 MP 228-70 West	Data taken from MP 220-80

*PRO - speed profile developed from track class and test cant.
**TBD - to be determined from series 1, 2, & 3 data analysis.
***Cant from New Haven-Groton/Groton-Boston

TABLE 2-3

F40/AMCOACH BANKING COACH

Original: ☐ Date: ____________

Reproduction: ☐ By: ____________

STRIP CHART NO.: ______

RUN NO.: ____________

CANT (IN): ____________

ZONE: ____________

CHART SPEED (MM/SEC): ____________

Brush Channel	Data Channel	Channel Name	Range	S/F (V/DIV)
1	237	Locomotive Force Vect. Int.	±20"	TBD
2	200	Locomotive Axle #1 Vertical Force Right Rail	±50 Kip	TBD
3	202	Locomotive Axle #1 Vertical Force Left Rail		TBD
4	201	Locomotive Axle #1 Lateral Force Right Rail	±50 Kip	TBD
5	203	Locomotive Axle #1 Lateral Force Left Rail		TBD
6	33	ALD	Event	TBD
Event #1	N/A	Milepost	Event	TBD
Event #2	N/A	Message	Event	TBD

TABLE 2-3 (Cont'd)

F40/AMCOACH BANKING COACH

Original: ☐ Date: ______________

Reproduction: ☐ By: ______________

STRIP CHART NO.: ______

RUN NO.: ______________________

CANT (IN): ______________________

ZONE: ______________________

CHART SPEED (MM/SEC): __________

Brush Channel	Data Channel	Channel Name	Range	S/F (V/DIV)
1	216/217	Locomotive Axle #1 L/V High Rail	0-2	TBD
2	218/219	Locomotive Axle #2 L/V High Rail	0-2	TBD
3	224/225	Locomotive Truck L/V High Rail	0-2	TBD
4	52	Speed	0-150 mph	TBD
5	45	Locomotive Carbody Lateral Acceleration	-1g	TBD
6	33	ALD	Event	TBD
Event #1	N/A	Milepost	Event	TBD
Event #2	N/A	Message	Event	TBD

NOTES: *High rail data channel selected by computer operator.

TABLE 2-3 (Cont'd)

F40/AMCOACH BANKING COACH

Original: ☐ Date: ____________

Reproduction: ☐ By: ____________

STRIP CHART NO.: ______

RUN NO.: ____________

CANT (IN): ____________

ZONE: ____________

CHART SPEED (MM/SEC): ____________

Brush Channel	Data Channel	Channel Name	Range	S/F (V/DIV)
1	*220/221	Coach Axle #1 L/V High Rail	0-2	TBD
2	*222/223	Coach Axle #1 L/V High Rail	0-2	TBD
3	*226/227	Coach Truck L/V High Rail	0-2	TBD
4				
*5	50	Coach Carbody Lateral Acceleration	$\pm$1g	TBD
6	33	ALD	Event	TBD
Event #1	N/A	Milepost	Event	TBD
Event #2	N/A	Message	Event	TBD

*Channel = 55 coach tilt - Phase II only.

TABLE 2-3 (Cont'd)

F40/AMCOACH BANKING COACH

Original: ☐ Date: ______________

Reproduction: ☐ By: ______________

STRIP CHART NO.: ______

RUN NO.: ______________

CANT (IN): ______________

ZONE: ______________

CHART SPEED (MM/SEC): ______________

Brush Channel	Data Channel	Channel Name	Range	S/F (V/DIV)
1	238	Coach Vertical Force Int.	±20"	TBD
2	208	Locomotive Axle #1 Vertical Force Right Rail	±50 Kip	TBD
3	210	Locomotive Axle #1 Vertical Force Left Rail		TBD
4	209/211	Coach Axle #1 Lateral Force High Rail	±50 Kip	TBD
5				
6	33	ALD	Event	TBD
Event #1	N/A	Milepost	Event	TBD
Event #2	N/A	Message	Event	TBD

safely fulfilled, the second and third round trips willl be made at higher cant deficiencies.

The speed profile for the demonstration runs will be adjsuted if the weather predictions call for wind gusts exceeding 50 mph. Any curves or other track features which would produce results exceeding the test safety criteria in Section 9.0 will be negotiated at slower speeds.

3.0 TEST CONSIST

The consist for this test is shown in Figure 3-1. This consist will include the following vehicles.

- F40 PH Locomotive
- Banking Amcoach
- Standard Amcoach
- Standard Amcoach or Amcafe car

The special Amcoach car equipped with banking will be located directly behind the locomotive. The Data Acquisition System (DAS) will be located in the banking Amcoach car so that once the consist is made up the locomotive and banking coach cannot be separated until the intercar cabling is removed.

As described in Section 5.0, the DAS will be installed in the banking coach. All of the cabling runs from the DAS to instrumentation on the locomotive or the Banking Amcoach. Therefore, the other two cars can be separated from the test vehicles.

To install the DAS system in the coach many of the seats must be removed. The DAS weight is equivalent to the removed seats, but a minimum amount of ballast should be added to restore the side to side balance. The locomotive will not need to be ballasted, but it should be run with a full load of sand and about 1/2 full fuel tanks. The instrumented wheels can be damaged by heat; therefore, the brake equipment must be removed or disabled from the lead trucks of the locomotive and coach.

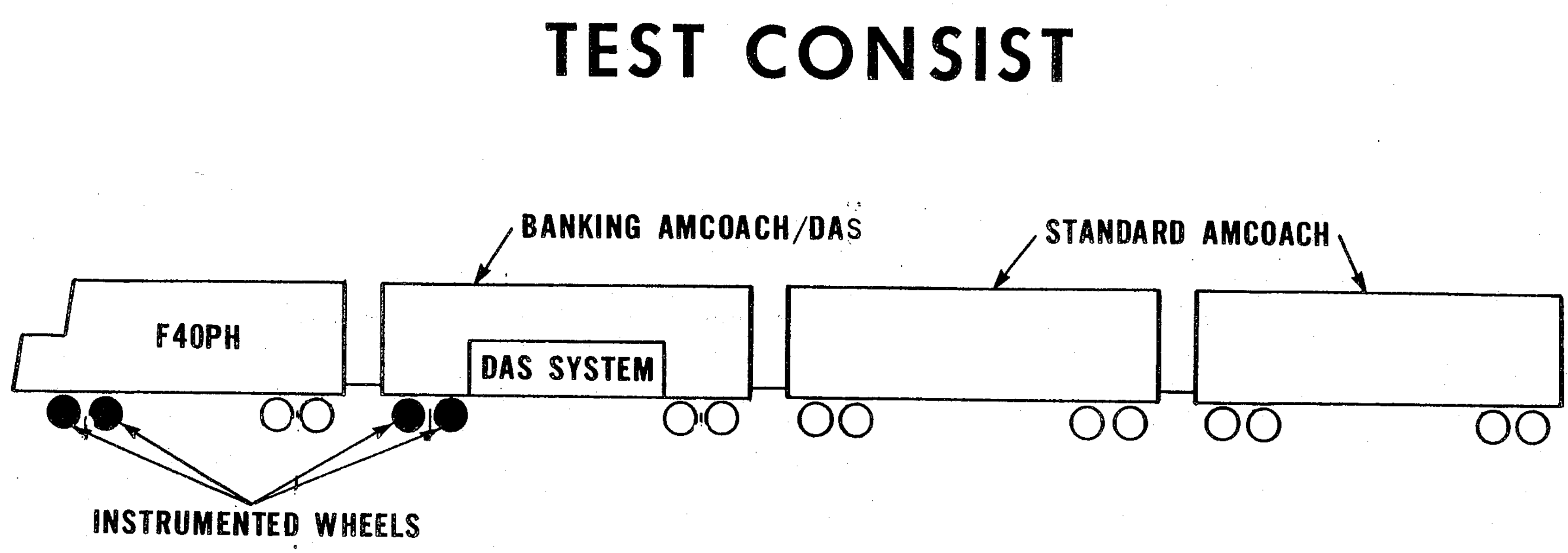


Figure 3-1. Test Consist

4.0 TEST ZONES

4.1 TEST ZONE SELECTION

The test zone was selected from the NEC trackage located between New Haven, CT and Providence, RI (Figure 4-1). This trackage was selected because between MP 73 (New Haven) and MP 182 (Providence) there are approximately 102 curves with a large variation in elevation and curvature. Curve 67 between MP 145 and 146 was chosen for the instrumented cuve site. The photos in Figure 4-2 show the instrumented site.

1. Shows each approach, looking east
2. Shows middle of curve, looking west
3. Shows west approach, looking east

Curve 67:

- $2^{o}36'$ curve with 6.25 superelevation
- Remote location
- No. 2 track has concrete ties
- No. 1 track has wooden ties
- For 11 inch cant deficiency the speed requirement would be approximately 97 mph

The test consist will be based at the New Haven, CT, Amtrack facility, so that the locomotive and coaches can receive required inspection and support.

The test zones are divided into four areas to support the four test series described in Section 2.1.

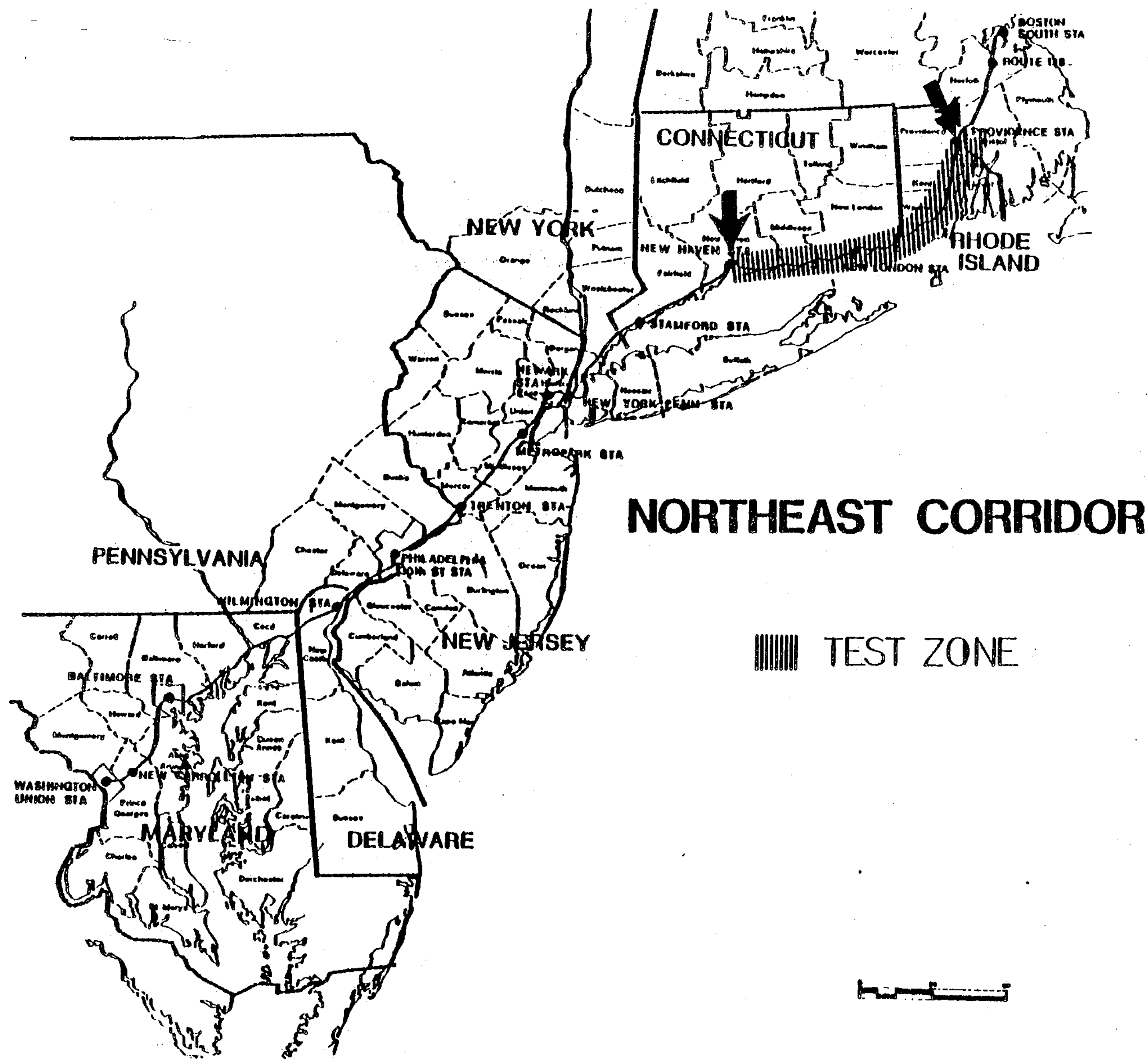


FIGURE 4-1 NEC TEST ZONE

4.2 TEST ZONE FOR STEADY STATE CURVING TEST

The repeated steady state test zone will be beween MP 138 and 153 on track No. 2 including the curve no. 67 (concrete ties) between MP 145 and 146. The planned speeds are shown in Table 2.1.

Repeated steady state tests will also be performed between MP 153 and 138 on track No. 1 including the curve no. 67 (wood ties) between MP 145 and 146.

4.3 TEST ZONE FOR OVER-THE-ROAD TESTS

This test series will be run between New Haven and Groton (see track charts in Appendix A). The test consist will run east on track No. 2 to Groton and west on track No. 1 to New Haven. Maximum speed anticipated in this test zone would be 103 mph.

4.4 TEST ZONE FOR DEMONSTRACTION RUNS

The demonstration test will be run on Northeast Corridor track from New Haven to Boston. The route from New Haven to Groton will cover the same track used for the over-the-road test.

5.0 INSTRUMENTATION

5.1 GENERAL

The instrumentation for the test is classified into three major categories:

- o Data Acquisition System
- o Transducers
- o Cabling/Junction Boxes

During both phases of the test, the lateral and vertical wheel-to-rail forces generated by the locomotive and the banking Amcoach shall be measured. The lead truck of the locomotive and of the coach being tested shall have both wheelsets instrumented. The magnitude and location of the resultant weight vector in the plane of the railhead will be computed from the wheel loads measured for both the locomotive and the coach.

Table 5-1 lists the parameters to be recorded during the steady state and over-the-road tests.

5.2 DATA ACQUISITION SYSTEM (DAS)

The DAS will be installed onboard the banking Amcoach provided by Amtrak for this purpose. Figure 5-1 shows the coach layout. This will require two-thirds of the seats to be removed. The DAS will be attached to a false floor installed in the coach. The coach will be configured for B-end leading during testing.

A block diagram of the DAS configuration is shown in Figure 5-2.

The instrumentation amplifiers will be ENSCO models 0531 and 0529. The test configuration will be ten high gain and 20 low gain amplifiers. The strain gage signal conditioning system will be a Natel model 2088 carrier system, used in conjunctin with an

TABLE 5-1

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	2
Wheel L/V	Computed	4
Weight Vector Intercept	Computed	1
Primary Vertical Displacement (1 Axle)	LVDT	2
Secondary Vertical Displacement	Sring Pot	2
Secondary Lateral Displacement (& Truck Yaw)	String Pot	2
Cab Lateral Acceleration	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	2
Wheel L/V	Computed	4
Weight Vector intercept	Computed	1
Speed	Decelostat	1
Location	ALD	1
Primary Vertical Displacement (1 Axle)	LVDT	2
Secondary Vertical Displacement	String Pot	2
Secondary Lateral Displacement (& Truck Yaw)	String Pot	2
Banking Motions	String Pot	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; 6.8v/g	1
Tilt Comand Signal	Tilt System Test Point; +2V,0, or -2V	1
Primary Lateral Displacement	LVDT	1
Torsion Bar Stress*	Budd Strain Gage Bridge to Natch	1
Cylinder Pressure*	Budd Sensor to Natels	2
	Coach Data Channels	35

*Optional

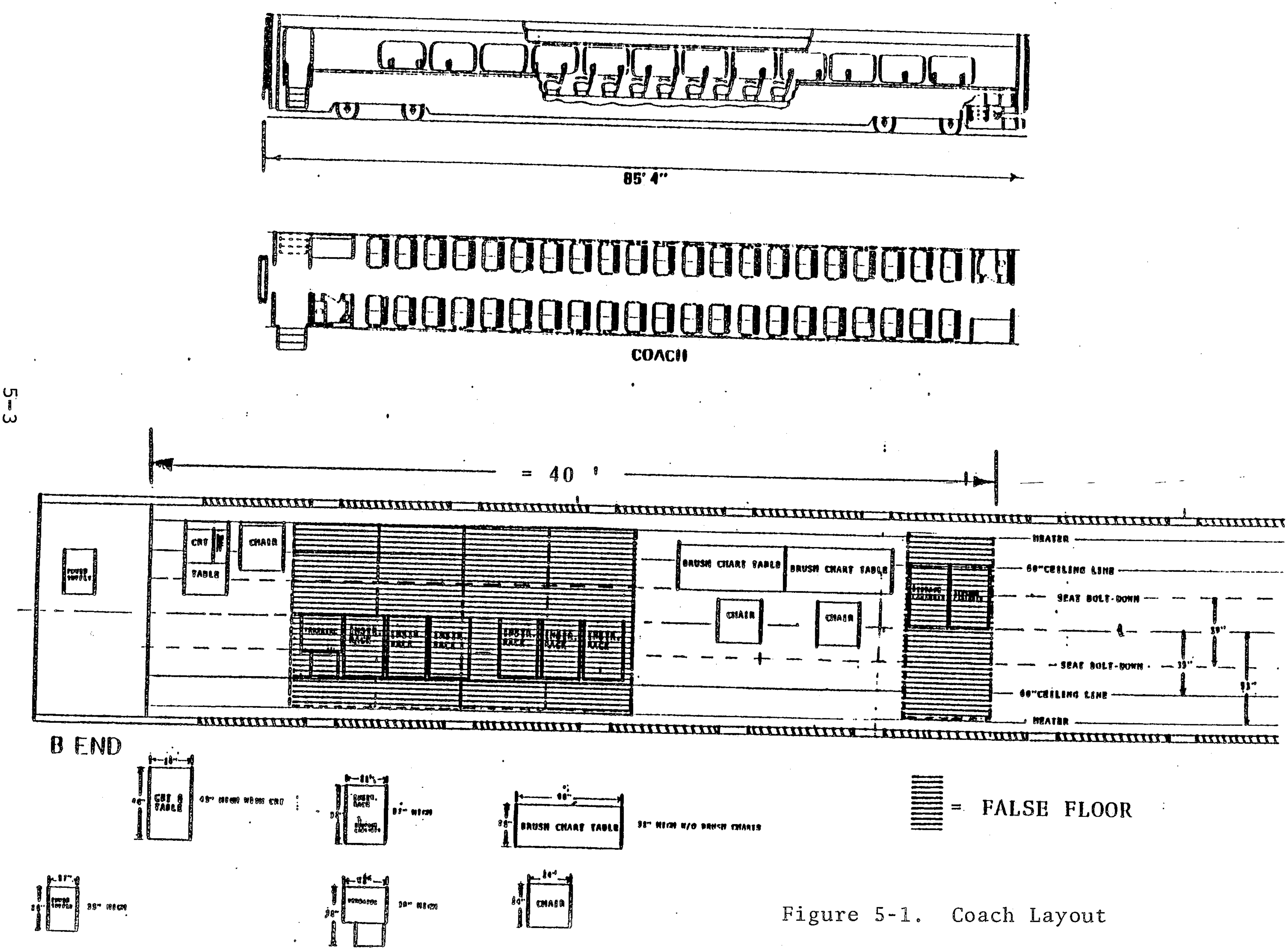


Figure 5-1. Coach Layout

ENSCO model 475 Wheel Simulator and Filter Unit. The ALD signal conditioning system is covered under instrumentation in Section 5.3. The tachometer system will convert the wheel tach output into logic level signals, which will be fed to the programmable clock and used by the DAS for a distance base and recorded on the digital tape.

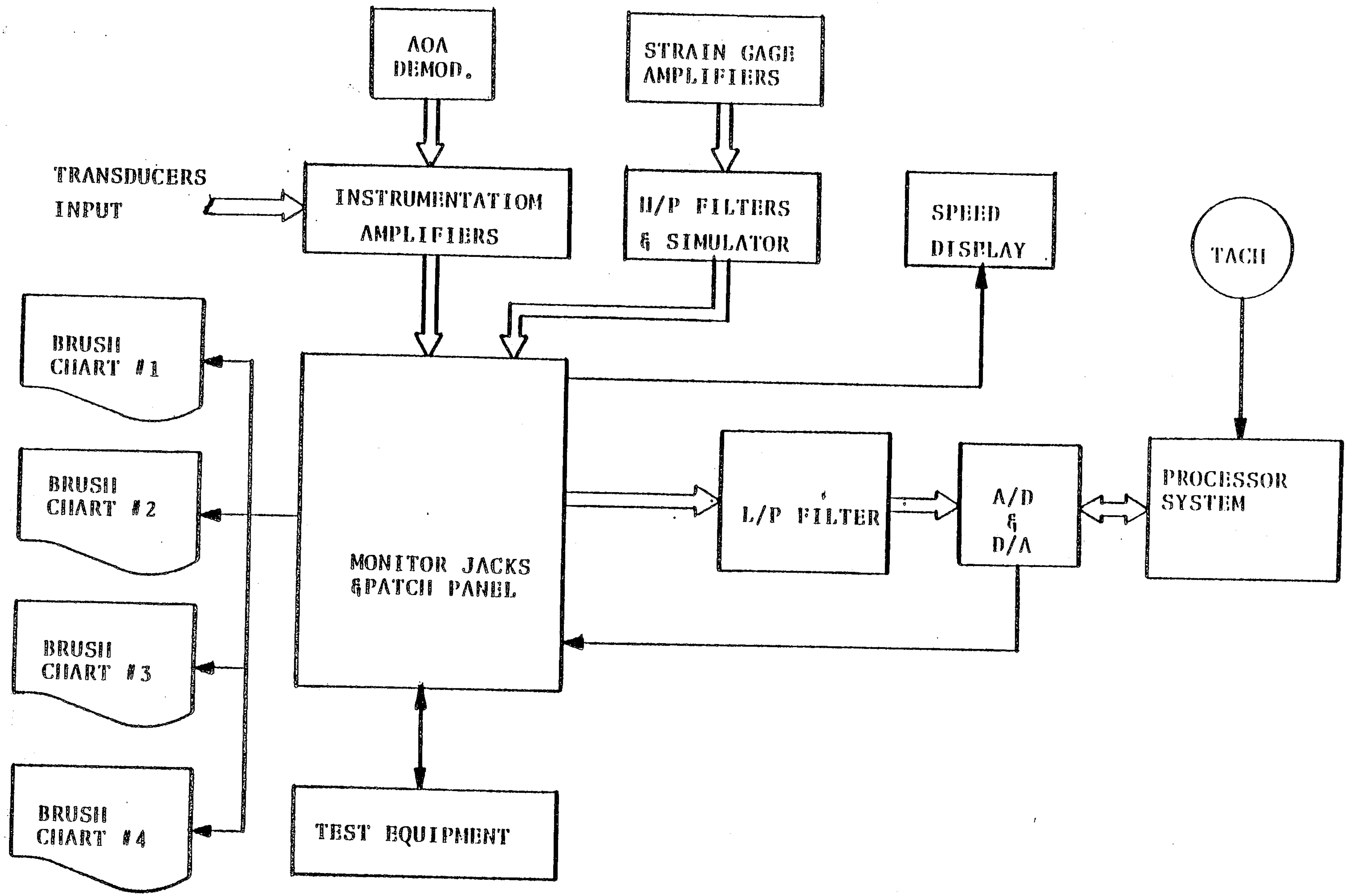
Anti-aliasing filters are programmable, low pass, four pole bessel, active filters. The 3db setting for each filter is shown in Table 5-2 under CTF FRQ column.

The brush charts will be Gould model 260 6-channel ink recorder with Z-fold paper. They will be configured as stated in Table 5-2.

Figure 5-3 is a system configuration diagram of the CPU system and peripherals. For this test the maximum A/D configuration for a sample rate of 256 Hz is 96 A/D channels. The hardware limit of 64 channels of D/A channel is the anticipated test configuration limit. Additional hardware would have to be acquired for any additional channels over the 64 channel configuration.

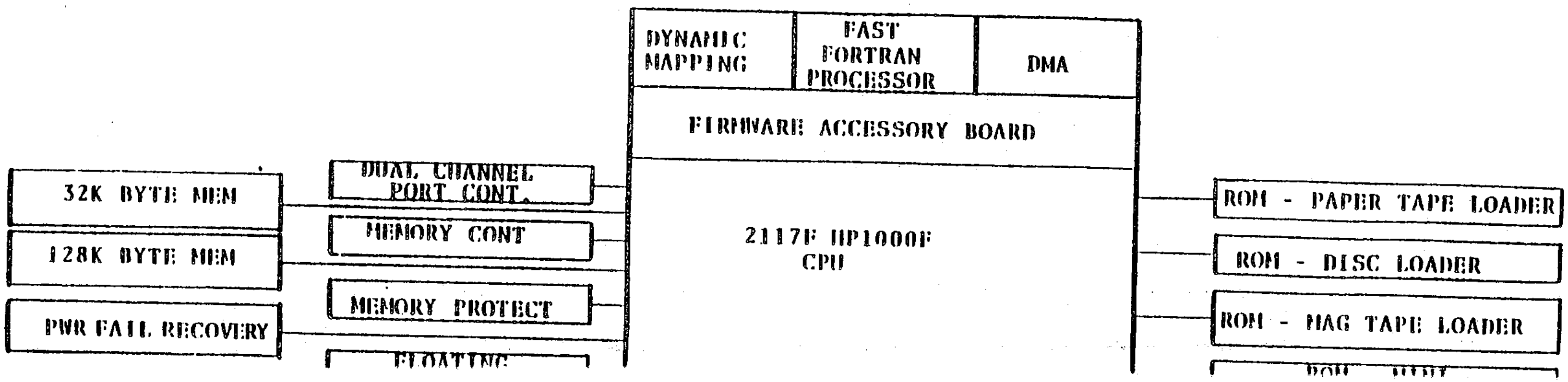
The data tapes will be digital recordings NRZ1, 9-track, 1600 cpi, 45 ips. Format for the tapes is detailed in Appendix E.

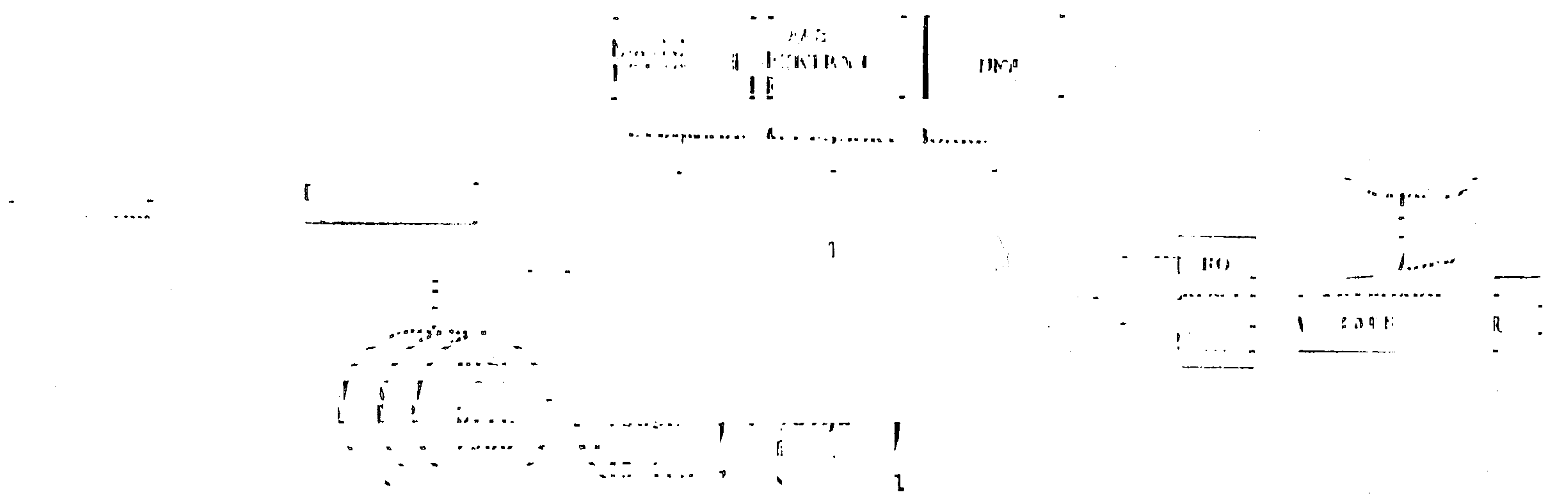
Power for the DAS will be supplied by a regulator converter that will be connected to the 480V 3 phase train line by a 480 to 220 3 phase transformer. Figure 5-4 shows the power configuration for the DAS coach. For grounding/safety all racks will be bonded together. The racks will be isolated from ground by the false floor, therefore, all ground wires will be run to the power regulator. The power regulator will be grounded to the carbody and ground integrity checked daily.



DAS CONFIGERATION

FIGURE 5-2


DYNAMIC MAPPING
FAST FORTRAN PROCESSOR
DMA
FIRMWARE ACCESSORY BOARD
2117F HP1000F CPU
32K BYTE MEM
128K BYTE MEM
PWR FAIL RECOVERY
DUAL CHANNEL PORT CONT.
MEMORY CONT
MEMORY PROTECT
ROM - PAPER TAPE LOADER
ROM - DISC LOADER
ROM - MAG TAPE LOADER



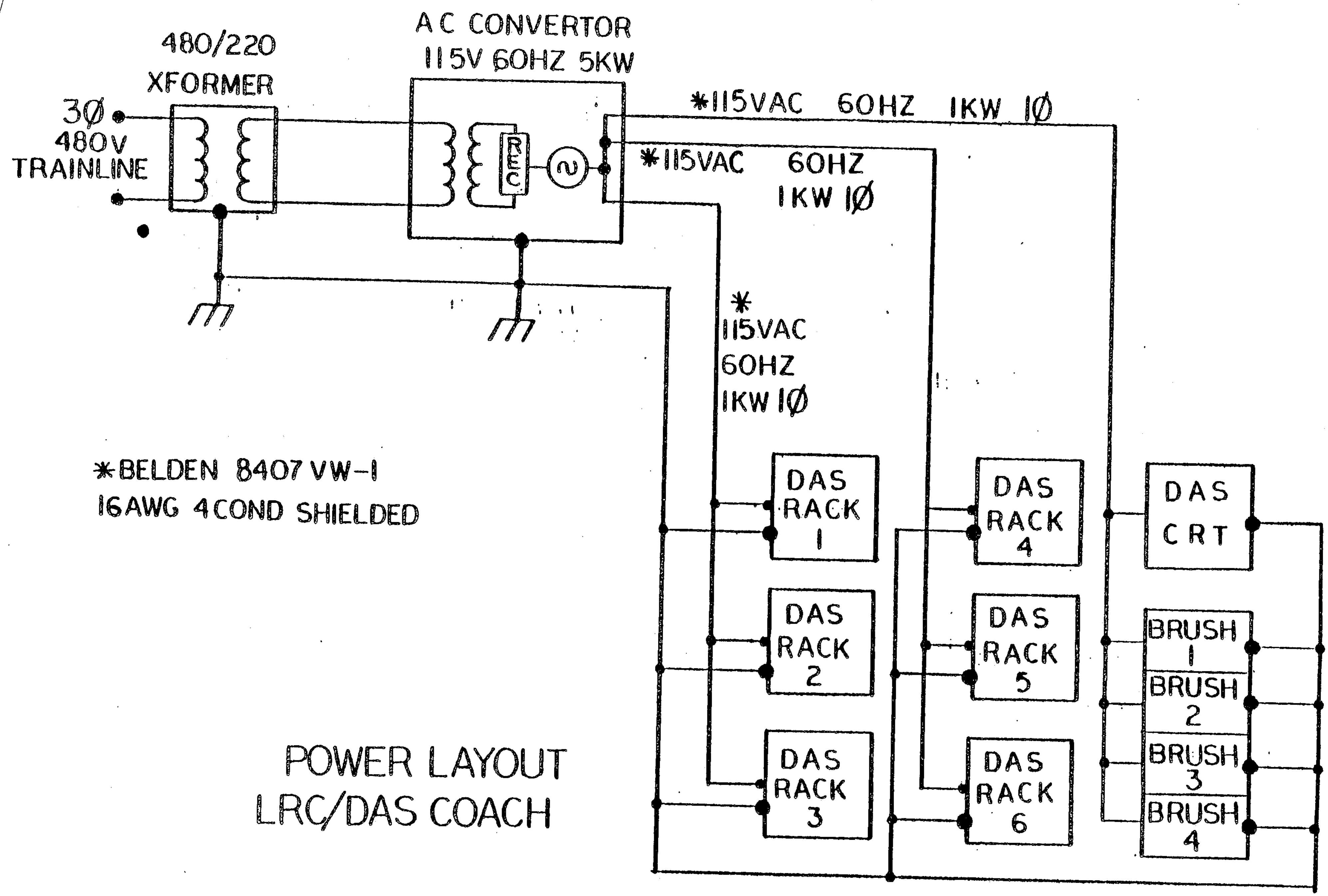

480/220
XFORMER
3Ø
480V
TRAINLINE
A C CONVERTOR
115V 60HZ 5KW
REC
*115VAC 60HZ 1KW 1Ø
*115VAC 60HZ
1KW 1Ø
*
115VAC
60HZ
1KW 1Ø
DAS RACK 1
DAS RACK 2
DAS RACK 3
DAS RACK 4
DAS RACK 5
DAS RACK 6
DAS CRT
BRUSH 1
BRUSH 2
BRUSH 3
BRUSH 4
*BELDEN 8407 VW-1
16AWG 4COND SHIELDED
POWER LAYOUT
LRC/DAS COACH

FIGURE 5-4

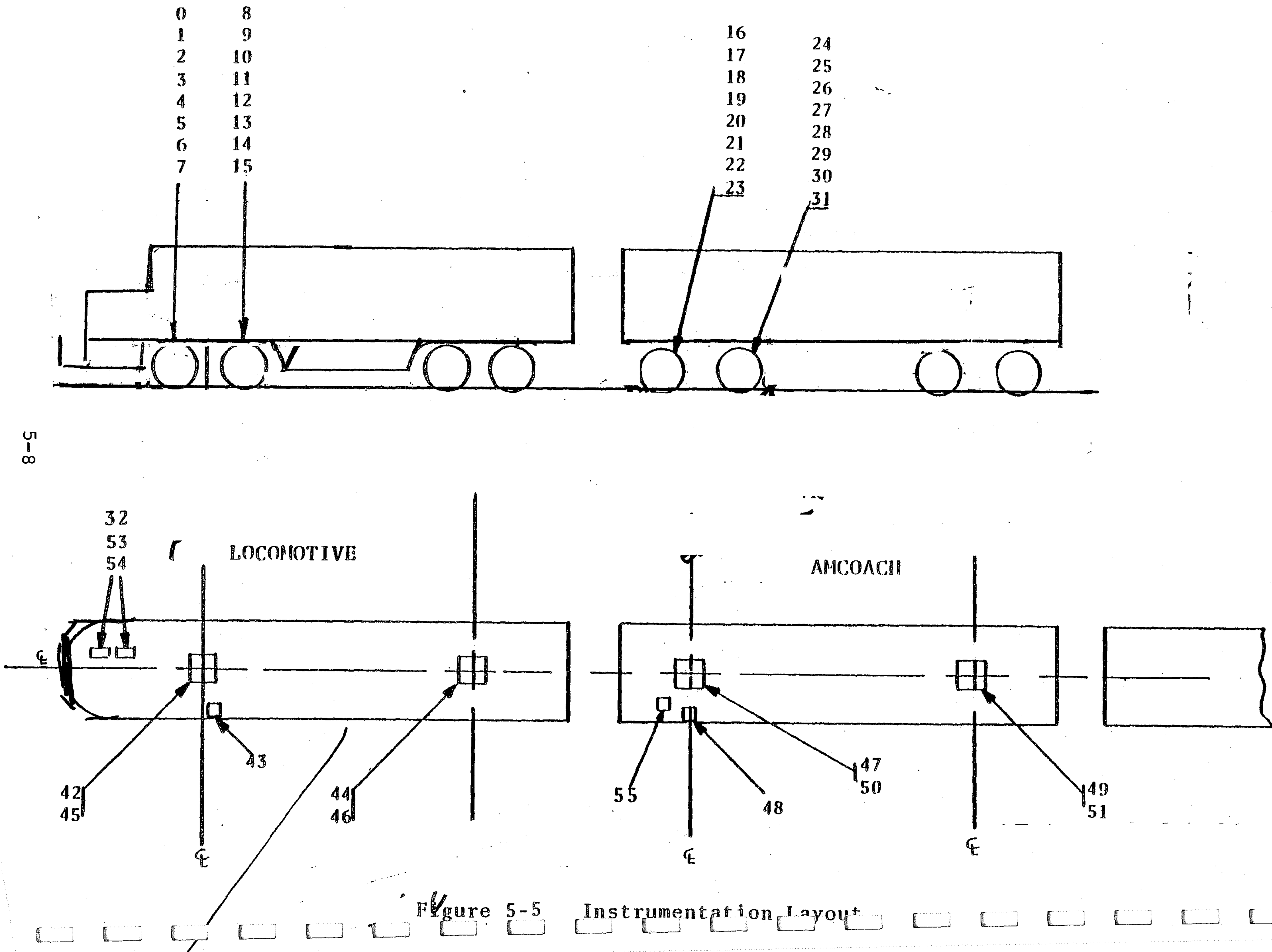


Figure 5-5 Instrumentation Layout

5.3 TRANSDUCERS

Table 5-2 lists the instrumentation for both Part A and Part B. All 200 series (CHNL under column one) are real time processed signals computed from the 56 raw data channels. Figure 5-5 shows the location of the instrumentation on the test consists.

5.3.1 WHEELSETS

The wheelsets developed by ENSCO for FRA will be used on the banking coach. Similar instrumented wheelsets developed by ASSEA for an Amtrack/FRA joint project will be used on the F40PH locomotive. Figures 5-6 and 5-7 show the data flow for these wheelsets while Figure 6-1, indicates the processing used to provide continuous vertical and lateral signals.

The wheels on the lead truck of the locomotive and the first coah in the consist will be instrumented with strain gages located on the wheel plate. The wheel plates have been conformally machined. A slip-ring is located on each end of the instrumented axle. The gaging technique provides both continuous vertical and lateral force measurements. The exact gage locations were determined by an exploratory gaging survey designed to determine optimum gage locations to produce maximum output with a minimum interaction between the lateral and vertical forces.

The gages are welded and bonded with adhesive capable of withstanding temperatures from -20 to 180 degrees Fahrenheit. The gage circuits and instrumentation wiring are covered with silicone sealant and then shielded with strainless steel, spot welded to the wheel plate over the gages.

5.3.2 ALD (AUTOMATIC LOCATION DETECTOR)

The ALD will be mounted on the leading truck of the DAS/LRC coach. The ALD will be a capacitive displacement system. The sensor will be mounted on a bracket, six inches from the top of the railhead and in approximately the center of the truck. The

Vertical Force Measurement

"A + B" TRIANGULAR OUTPUT

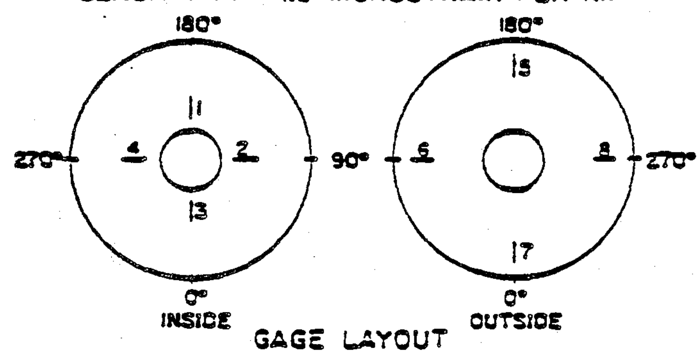
TWO BRIDGES

GAGES ON BOTH SIDES OF WHEELPLATE

TRIANGULAR WAVEFORMS - 2 - CYCLES PER
PER REVOLUTION

OUTPUT = MAX{|A|, |B|, K(|A| + |B|)}

SENSITIVITY ≈ 4.0 MICROSTRAIN PER KIP



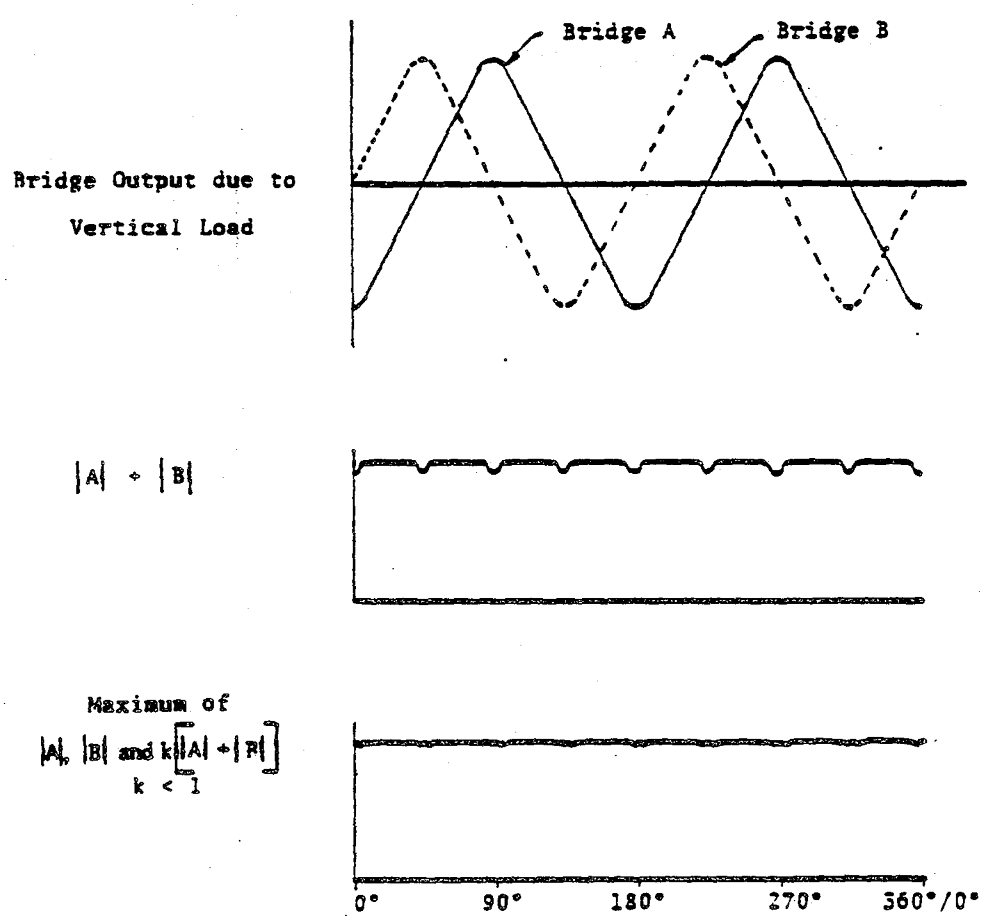


Figure 5-6 Vertical Force Measurement

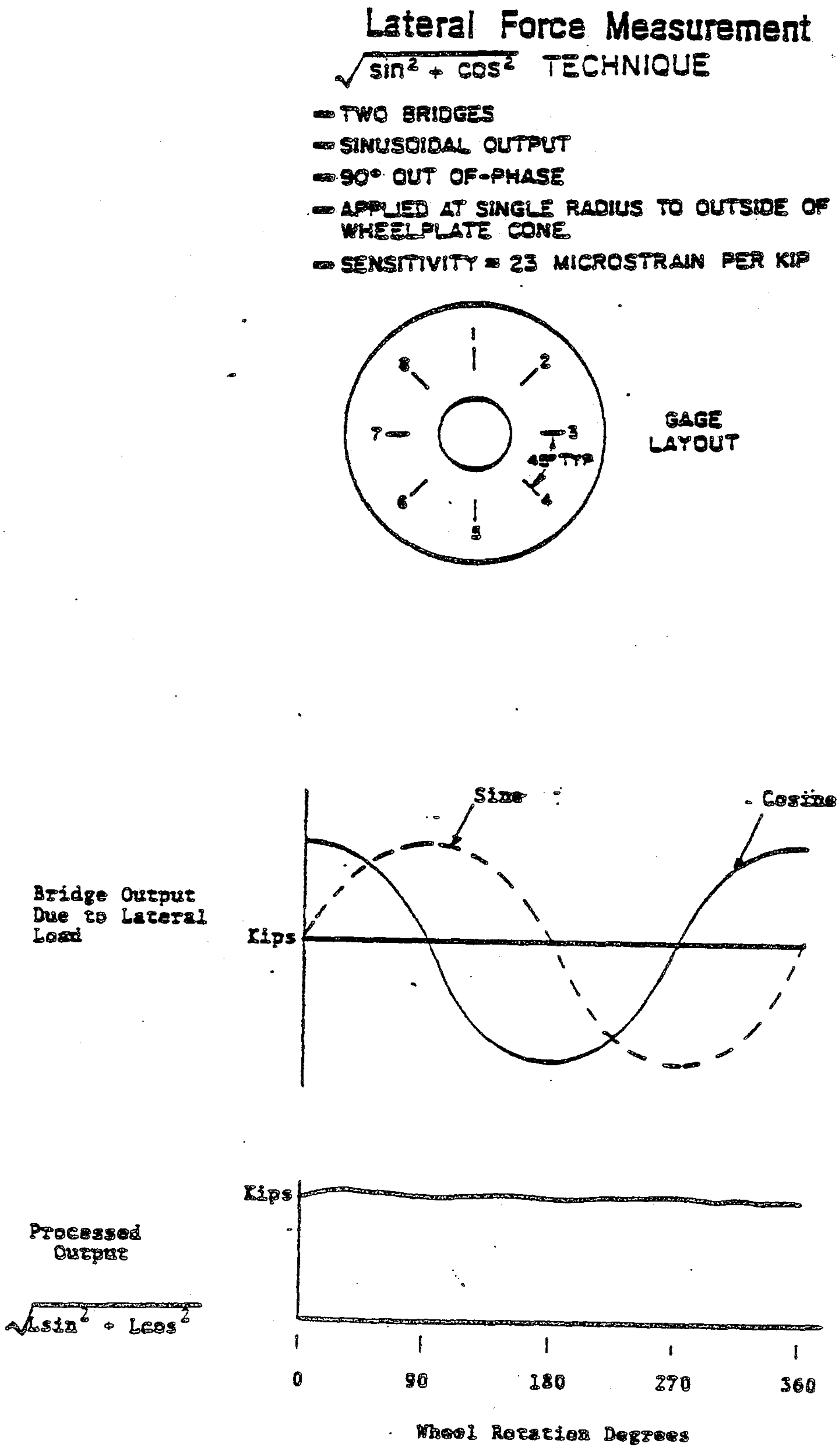


Figure 5-7 Lateral Force Measurement

distance from the center of the sensor to the center of instrumented axle shall be noted.

5.3.3 ACCELEROMETERS

Servo-accelerometers will be mounted to both the locomotive and the coach carbody during testing to provide ride-quality data. Figure 5-11 shows the signal flow for these accelerometers.

5.3.4 SPECIAL INSTRUMENTATION

5.4 CABLING AND J-BOXES

Figure 5-12 shows cabling and junction-box layout. 28 VDC power will be sent to all J-boxes (by use of DC/DC converters) and converted to required instrument excitation.

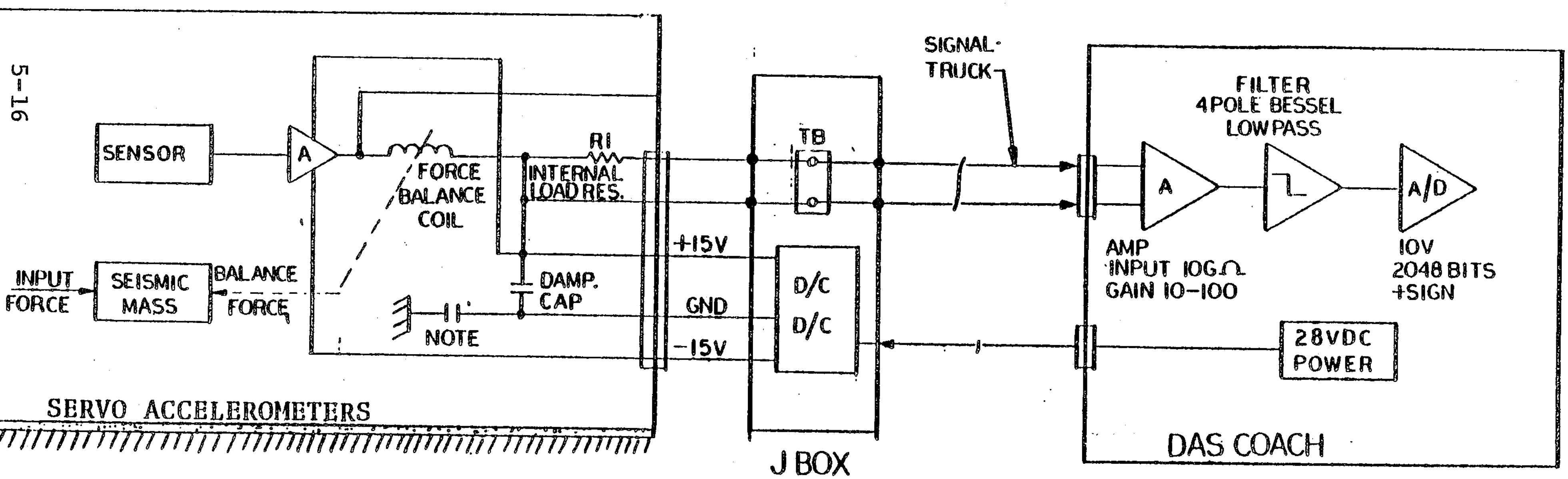
The input cabling for the DAS coach requires a plywood panel to replace one-half of the siding door on the B-end. The cabling systems will be designed for quick configuration changes and trunk cable replacement.

All outside connections will be waterproof. A sufficient service loop will be left in cabling between cars and between the carbody and the truck.

Junction boxes shall be located inside in an accessible area, mid-body of the locomotive and A-end of coach.

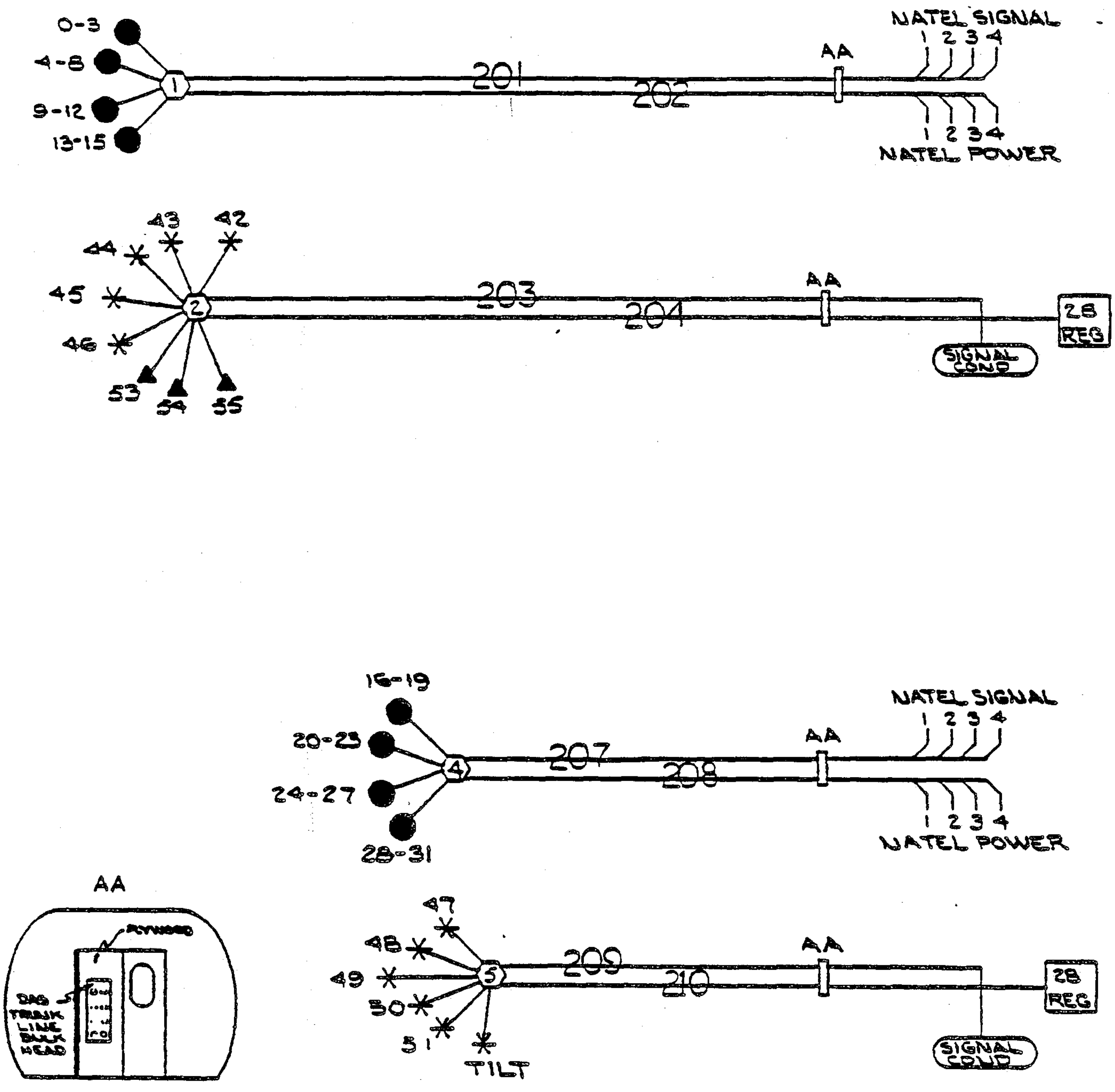
All personnel working in the instrumented areas will be made aware of the fragile nature of the interior cabling, i.e., the cabling will not tolerate stepping-on or kicking.

Before moving any of the test consist, coupler lock between test vehicles must be verified after inter-car cabling is installed.



ACCELEROMETER CONFIGERATION

FIGURE 5-11



CABLE CONFIGURATION

FIGURE 5-12

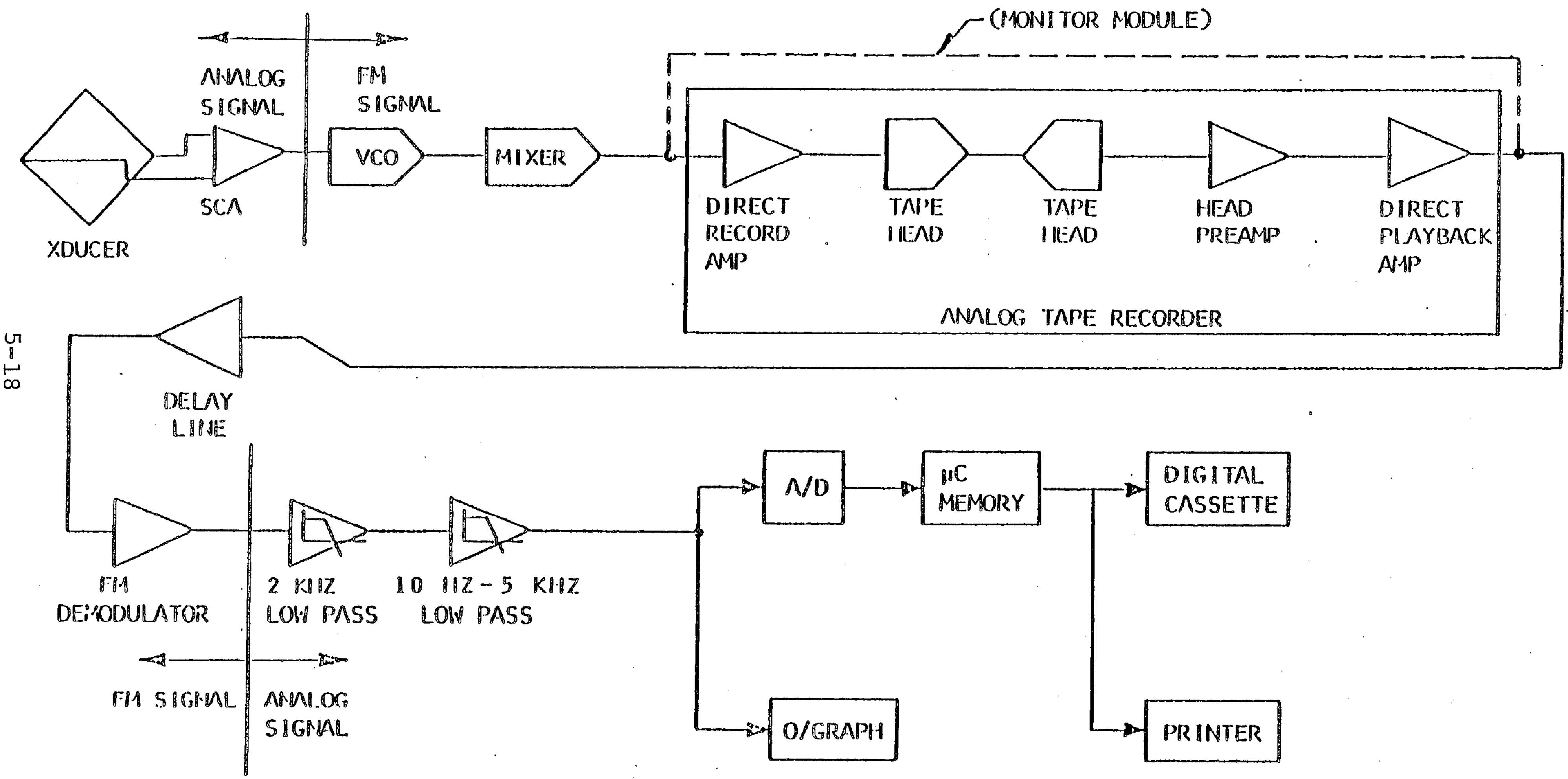


FIGURE 5-13 SIGNAL PATH FOR ONE DATA CHANNEL

6.0 CALIBRATION

6.1 INSTRUMENTED WHEELSETS

Figure 6-1 shows the onboard processing to compute wheel force from wheel bridge strains. It differs from the manual processing used in laboratory calbiration in two respects, filtering and amplifier gain. The manual processing of bridge output to obtain force channel output was done in terms of strain, and the channel output was then converted to force by dividing by the channel sensitivity. It is preferred to convert the individual bridge strains into force units before force channel processing with the onboard instruments. Therefore, the gain of the amplifiers will be used to set a ±10v range for each bridge equivalent to ±25,000 pounds at the strain per unit force sensitivity of the appropriate force channel (two combined bridges). Operating in force units at the bridge level yields valid force channel output because multiplying each bridge output by a constant changes the channel output by the same constant. Table 6-1 gives the original calibration characteristics of the Amcoach wheels. This table will be revised to include the F40PH wheels.

6.1.1 BRIDGE RESISTANCE MEASUREMENTS

Check for bridge damage by making resistance checks of bridge arms, bridge to ground, bridge to shield, and shield to ground. Check with the ditigal meter at the slip ring connectin before and after installation on the vehicle. and record on QC Resistance Form (Table 6-2).

6.1.2 SHUNT CALIBRATIONS

Shunt calibrations will be used to evaluate and correct for the effect of cable resistance from the wheels to the amplifers and to adjust the amplifer gain so that the onboard instrumentation agrees with the Vishay/Ellis 20 strain indicator used in the original laboratory force calibration. If the indicator cannot

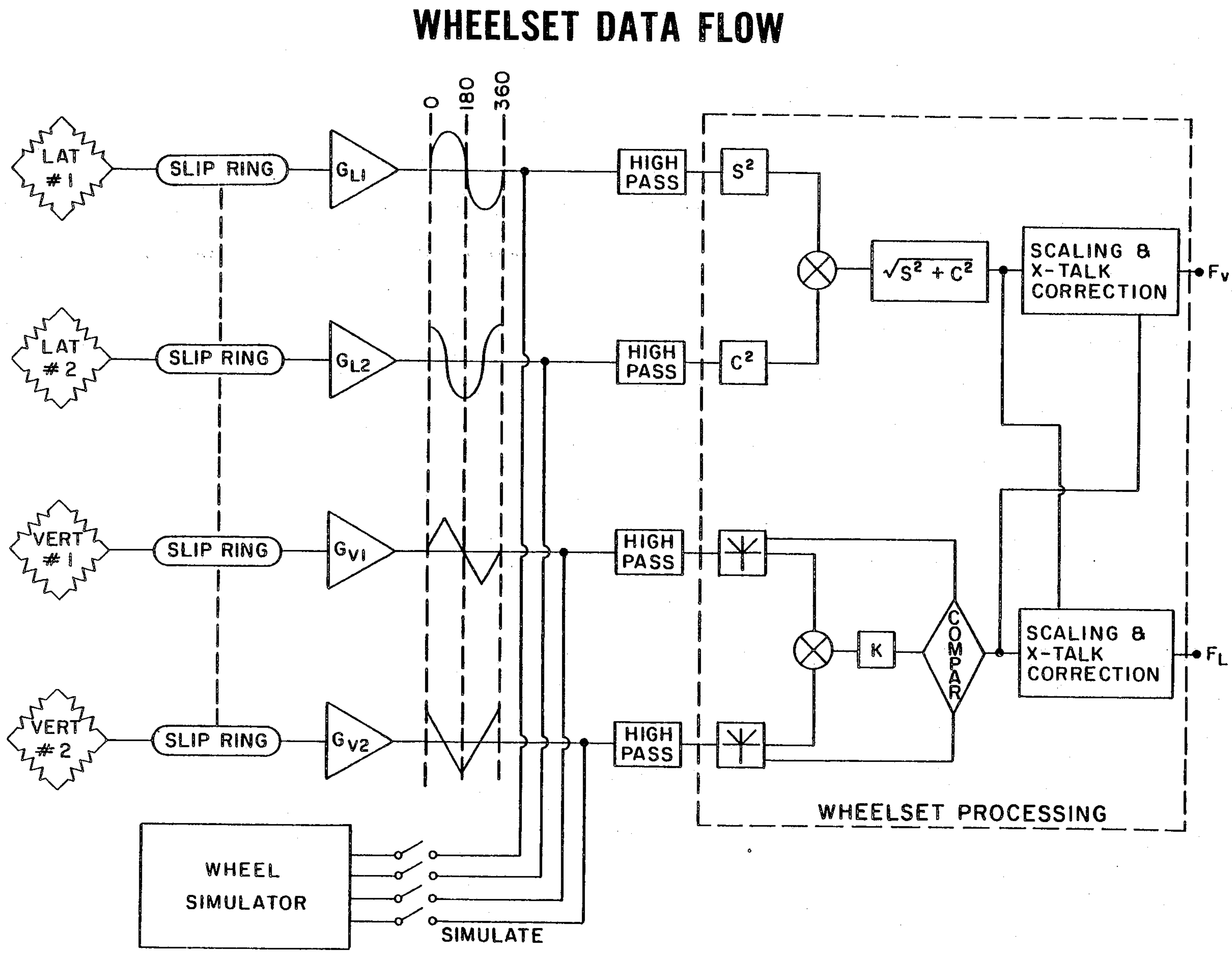


Figure 6-1. Wheelset Data Flow

TABLE 6-1A

F40PH LOCOMOTIVE INSTRUMENTED WHEEL CHARACERISTICS

1. Processing Constants

Wheel	K	G_V $\mu\varepsilon$/kip	G_L $\mu\varepsilon$/kip	H_V lb/lb	H_L lb/lb
1A	.926	3.878	14.348	.126	-.023
1B	.922	3.765	14.144	.098	-.007
3A	.917	3.881	14.316	.109	-.000
3B	.914	3.938	14.568	.074	-.000

where:

uncorrected forces, $F'_V = \frac{1}{G_V}\left[\text{greatest of} \begin{cases} |VA| \\ |VB| \\ K(|VA| + |VB|) \end{cases}\right]$

$$F'_L = \frac{1}{G_L}\left(\sqrt{LA^2 + LB^2}\right)$$

and corrected forces,

$$F_V = F'_V + (H_V)F'_L$$

$$F_L = F'_L + (H_L)F'_V$$

2. Average Vertical Load Point Effect for 1 inch Movement from Flanged Position:

G_V	G_L
1.6%/in increasing toward flange	2.3%/in decreasing toward flange

3. Ripple

Load	Maximum Vert	Maximum Lat
(25-00)	±5%	-
(35-20)	±6%	±1%
(00-20)	-	±1%

TABLE 6-1B

AMCOACH INSTRUMENTED WHEEL CHARACERISTICS

1. Processing Constants

Wheel	K	G_V με/kip	G_L με/kip	H_V lb/lb	H_L lb/lb
1X	.943	4.280	16.647	+.149	-.047
1Y	.938	4.378	16.558	+.156	-.051
2X	.937	4.314	16.160	+.122	-.056
2Y	.958	4.401	16.360	+.144	-.050

where:

uncorrected forces, $F'_V = \frac{1}{G_V}\left[\text{greatest of}\begin{cases}|VA|\\|VB|\\K(|VA| + |VB|)\end{cases}\right]$

$$F'_L = \frac{1}{G_L}\left(\sqrt{LA^2 + LB^2}\right)$$

and corrected forces,

$$F_V = F'_V + (H_V)F'_L$$

$$F_L = F'_L + (H_L)F'_V$$

2. Average Vertical Load Point Effect for 1 inch Movement from Flanged Position:

G_V	G_L
9.1%/in increasing toward flange	3.8%/in decreasing toward flange

3. Ripple

Load	Maximum Vert	Maximum Lat
(15-00)	±7%	-
(25-15)	±7%	±6%
(00-15)	-	±4%

TABLE 6-2. QC RESISTANCE

DATE ________ WHEEL ________ BRIDGE ______

ARM	Ω	MΩ/SD	MΩ/r_h	SD/r_h
______	______	______	______	______
______	______	______	______	______
______	______	______	______	______
______	______	______	______	______

BRIDGE	TB	TB#
+P/-P	______	______
+S/-S	______	______

balance the vertical bridge under the static load of the vehicle, either move the vehicle to a null position of the bridge of interest or start with a strain indication of the expected magnitude at the present wheel position and reocrd the change in magnitude reuslting from the shunt resistance. A three step shunt calibration is used to identify and correct for the resistance of the cabling between the wheelset slip ring and the strain gage bridge amplifer onboard the vehicle.

Step 1: Use the V/E 20 strain indicator and the shunt resistor switching unit at each slip ring connector. Tale 6-3 lists the shunt resistor and the corresponding simulated strain for each wheel bridge. Place the shunt resistor in the -P, +S and -P, -S arms of the bridge consecutively and record the results on the shunt calibration form (Table 6-4).

Step 2: Repeat the shunt calibration with the shunt resistor switching unit at the slip ring but with the V/E 20 indicator separated by the wheel to amplifier cable, and record. This configuration best simulates the actual service of the wheel instrumentation system.

Step 3: Repeat the shunt calibration with the wheel to amplifier cable between the slip ring connector and both the shunt resistor switching unit and the V/E 20 indicator and record the results.

6.1.3 AMPLIFIER GAIN CALIBRATION

The shunt calibration at the slip ring with the V/E 20 strain indiator (Step 1) should verify the simulated strains listed in Table 6-3. The laboratory calibration established the relationships between known rail/wheel forces and the strain indication as measured with the V/E 20 without long cables; these relationships are the sensitivities G_V and G_L expressed in me/kip. The use of the shunt resistor at the slip ring is equivalent to the application of a known load equal to the simulated strain divided by the bridge sensitivity. The simulated load for each wheel is

TABLE 6-3 SHUNT CALIBRATION RESISTORS AND CORRESPONDING SIMULATED STRAINS AND LOADS

	Vertical Bridges			Lateral Bridges		
F40PH Wheel	Shunt Resistor	Simulated Load	Simulated Strain	Shunt Resistor	Simulated Load	Simulated Strain
1A	687,030Ω	44,352 lb.	172 με	349,500Ω	35,336 lb.	507 με
1B	(599,880Ω + 87,150Ω)	45,153 lb.	170με	349,500Ω	35,421 lb.	501 με
3A	"	43,803 lb.	170 με	349,500Ω	34,996 lb.	501 με
3B	"	43,169 lb.	170 με	349,500Ω	34,390 lb.	501 με
Amcoach Wheel						
1X	1,118,880Ω	24,509 lb.	105 με	283,650Ω	24,833 lb.	413 με
1Y	(999,000Ω + 119,800Ω)	23,961 lb.	105 με	(249,000Ω + 34,650Ω)	24,967 lb.	413 με
2X	"	24,316 lb	105 με	"	25,582 lb.	413 με
2Y	"	23,835 lb.	105 με	"	25,269 lb.	413 με

TABLE 6-4. SHUNT CALIBRATION FORM
(Example from another test)

	Bridge Shunted Arm	$V_A, \mu\varepsilon$		$V_B, \mu\varepsilon$		$L_A, \mu\varepsilon$		$L_B, \mu\varepsilon$	
		-P/+S	-P/-S	-P/+S	-P/-S	P/+s	- P/-S	P/+S	- P/-S
1.	Resistor at slip ring, meter at slip ring	-152	151	-151	151	-1342	1342	-1344	1343
2.	Resistor at slip ring, meter remote	-152	152	-151	151	-1342	1340	-1342	1342
3.	Resistor remote, meter remote	-156	155	-156	154	-1372	1370	-1372	1371

listed in Table 6-3. The load simulated by the shunt resistor at the slip ring (Step 2) remains the same despite the presence of a cable between the shunt resistor and the indicator (or amplifier), and the amplifier gain should be adjusted to produce a force indication equal to the tabulated load. However, the strain indication of the V/E 20 changes when the cable is placed between the slip ring connector and the shunt resistor (Step 3) because the force simulated by the shunt resistor has been changed by the cable resistance. When the shunt resistor is wired to a switch in the amplifier (internal resistor), the simulated load in Table 6-3 must be adjusted by approximately 2-10% because the bridge resistance is altered by the long wheel-to-amplifier cables.

The force equivalent of the internal shunt resistor can be determined by first adjusting the amplifier gain to the tabulated load with the shunt resistor at the slip ring, then switching in the internal shunt resistor with the shunt resistor at the slip ring removed. The force indicated by the amplifier output is equivalent to the effect of the remote shunt resistor. The new simulated loads based on the internal shunt resistor should be tabulated for daily calibration.

6.1.4 DAILY CALIBRATION

In practice the forces simulated by the shunt calibrations either at the slip ring or at the amplifier will be measured as voltages at the amplifier output. A Daily Calibration form such as Table 6-5 should be used to set the amplifier gain before each test day. The calibration methods in sections 6.1.4.1 and 6.1.4.2 are the most rigorous, but it is normally acceptable to use the following simplified method. Unbalance the bridge to set zero at any random wheel position. Set the gain using the internal calibration resistor. Switch out the high pass filter and measure the amplifier output with an analog meter as the vehicle is moving slowly. Re-adjust the balance so that the meter deflects symmetrically about zero. Switch in the high pass filter before testing.

TABLE 6-5

DAILY CALIBRATION FORM, SHEET 1

DATE: ____________________

OPERATOR: ________________

Channel Number	Signal Description	Sensitivity lb/Volt (10v = full scale)	Natel Shunt Cal.Voltage (TOL±.05V)	✓	✓	Cal.Voltage (Tol±.05V)	Simulated Force (lb.)	Simulated Strain με
0	VA-LocoLF,1-1(3A)	10,000				4.380	43,803	170
1	VB " " "	"				4.380	"	"
2	LA " " "	5,000				6.999	34,996	501
3								
4	VA-LocoRF,1-2(3B)	10,000				4.317	43,169	170
5	VB " " "	"				4.317	"	"
6	LA " " "	5,000				6.878	34.390	501
7						"		
8	VA-LocoLR,2-1(1B)	10,000				4.515	45,153	170
9	VB " " "	"				4.515	"	"
10	LA " " "	5,000				7.084	35,421	501
11								
12	VA-CoachRR,2-2(1A)	10,000				4.435	44,352	172
13	VB " " "	"				4.435	"	"
14	LA " " "	5,000				7.067	35,336	507
15								
16	VA-CoachLF,3-1(1X	5,000				4.902	24,509	105
17	VA " " "	"				4.902	"	"
18	LA " " "	"				4.967	24,833	413
19	LB " " "	"				4.967	"	"
20	VA-CoachRF,3-2(1Y)	"				4.792	23,961	105
21	VB " " "	"				4.792	"	"
22	LA " " "	"				4.993	24,967	413
23	LB " " "	"				4.993	"	"
24	VA-CoachLR,4-1(2Y)	"				4.767	23,835	105
25	VB " " "	"				4.767	"	"
26	LA " " "	"				5.054	25,269	413
27	LB " " "	"				5.054	"	"
28	VA-CoachRR,4-2(2X)	"				4.863	24,316	105
29	VB " " "	"				4.863	"	"
30	LA " " "	"				5.116	25,502	413
31	LB " " "	"				5.116	"	"

This procedure assures that the amplifier is still operating in its linear range by removing the initial unbalance which could cause saturation. The purpose of the high pass filter (approximately .2 Hz) is to remove slow zero drift and thermal effects which could be superimposed on the relatively high frequency wheel bridge signals.

6.1.4.1 Lateral Bridges

Run the vehicle on smooth level tangent track at 5 to 10 mph. Eliminate the high pass filter and use the calibration switch on the amplifier. Set the balance control so that the amplifier output oscillates about zero and switch to the run position. Switch in the internal shunt resistor and adjust the amplifier output to the simulated load (adjusted from C-3 for internalcal resistor). Iterate between zero and gain adjustment if necessary. Switch out the shunt resistor and switch in the high pass filter.

6.1.4.2 Vertical Bridges

The static load on each instrumented wheel must be determined by weighing in order to make the zero level adjustment. The single peak point of the oscillating signal of an individual vertical bridge is not necessarily equal to the true vertical force because of the "K" factor used in the channel processing and because of ripple. Refer to the original calibration data to determine the ripple at the desired wheel position. Scale to the actual wheel load to account for ripple at the particular load point.

With the vehicle stationary, set the gain for the wheel load including ripple. Switch the amplifier to calibrate and set the gain so that the internal shunt resistor causes a change in amplifier output equal to the simulated load (adjusted from Table 6-3). Do not adjust the balance at this time, adjust the gain on the basis of a change in putput. The balance should be reasonably close from the previous day (except for the first adjustment which may require a tedious iteration).

With the vehicle in motion as in the lateral bridge zeroing, adjust the balance so that the amplified bridge output oscillates between the expected peaks. If its peak to peak amplitude is not correct adjust the balance so that the negative and positive peaks are approximately equal in absolute value and then scale them with the gain adjustment.

6.1.5 WHEELSET PROCESSING CHECKOUT

The wheelset processor combines the bridge signals to form continuous force indications using the algorithms and constants listed in Table C-1. Its functions are hard wired and no adjustment is necessary. The only switches on the unit are for the selection of AC or DC coupling. If separate high pass filters are used select DC coupling; AC coupling provides a built-in high pass filter.

Confirm the proper operation of processing algorithms and the proper setting of constants by using input signals from the wheel signal simulation. Set the simulator for ±5volts triangular waves for vertical channels and ±5 volts sinusoidal waves for lateral channels. Sample both the corrected and uncorrected force channel outputs of the processor. The uncorrected vertical should be about 1.03 x 5 volts.

Confirm the crosstalk correction factors as follows:

$$H_V = \frac{\text{Vertical Uncorrected} - \text{Vertical Corrected}}{\text{Lateral Uncorrected}}$$

$$H_L = \frac{\text{Lateral Uncorrected} - \text{Lateral Corrected}}{\text{Vertical Uncorrected}}$$

6.1.6 INITIAL VERTIFICATION

If the shakedown test wheel forces do not seem reasonable in comparison with published specs or previous test data, the vehicle vertical wheel loads should be weighed and the following onboard calibration performed for verification.

6.1.6.1 Lateral Force Channel

When the wheels are lifted from the rails set the lateral and vertical amplifier balance adjustments to zero output (with the amplifier in the calibrate position), and set the lateral gain using the internal shunt resistor. Apply a known load with the spreader bar at several positions around the wheel tread. Compare the known loads to the onboard force indication to see that the agreement is within the bounds of the ripple.

6.1.6.2 Vertical Force Channels

Set the zero and gain with the wheels off the rails. Lower the wheels to the rails and compare the onboard force indication to the static wheel loads computed during weighing. Roll the vehicle to several positions to evaluate ripple.

6.2 ACCELEROMETERS

Prior to any testing, each individual accelerometer will be calibrated to determine its static scale factor and output polarity, and to verify interconnection wiring between the transducer and data acquisition system. This calibration is performed by measuring the accelerometer output at a rest position and then by rotating it through 180° to impose +1, 0 and -1g on the accelerometer. The change in output voltage as the accelerometer is rotated at the output of the signal conditioning circuitry divided by a change of 2 g's (+1g - (-1g)) produced the scale factor in volts/g for each accelerometer.

6.3 DISPLACEMENT TRANSDUCERS

The string pot displacement transducers will be calibrated using the cylinder displacement method. This method provides a convenient way of applying a physical calibration after the string pot has been installed. The following procedure is used.

1. With the string pot installed and the test vehicle on tangent track and in a static condition, the channels are electrically zeroed.

2. A calibration cylinder of known circumference is applied to the string by wrapping one revolution of the string around the cylinder. While holding it steady, a reading is taken corresponding to the known displacement.

The LVDT displacement transducers will be calibrated similarly using a stop on the rod and a block of known thickness to determine the movement of the stop from the zero output position. The amplifier gain will be set for the desired displacement/voltage scale.

6.4 INSTRUMENTATION AND CALIBRATION CHECKLIST

A. Power-up DC amplifiers, carrier amplifiers and wheelset processor for 1/2 hour warm-up.

B. Check tape recorders and strip chart recorders (can be performed during warm-up)

1. Set wheel simulators for ±5 volts
2. Record short sample from channels 0-7 at patch panel monitor. 0, 1, 4, 5 are triangular; 2, 3, 6, 7, are sinusoidal.
3. Repeat 2 for all tape recorder channels.
4. Set strip chart zeros and gain to desired scale using a DC power supply.
5. Playback tape into strip chart recorder 14 channels at a time, staggering assignments between recorders to test all 16 strip chart channels.
6. Return tape recorder and strip chart recorder input assignments to original patch panel positions.

C. Calibrate wheel force bridge amplifiers

1. With the car stationary and amplifiers warmed-up balance each bridge.
2. Switch bridge to run and zero if necessary
3. Apply internal cal resistor.
4. Adjust gain to the Amplifier Shunt Calibration Voltage listed in Table 6-5 reading the output voltage at the Natel Amp.
5. Adjust other amplifiers similarly.
6. When the car is moving on straight level track adjust zero for the center of each built-in meter to avoid over ranging at high amplitude.

D. Calibrate accelerometers

1. Remove accelerometer and place in cal fixture.
2. Hold in zero position and zero amplifier while measuring at the patch panel.

3. Hold in +1g position and adjust gain of amplifier for 2v/g with exterior accelerometers and 10v/g with interior accelerometers.

4. Hold in -1g position and check linearity.

5. Fasten accelerometer to bracket and rezero for level track if necessary.

E. Calibrate displacement transducers

1. With train on level shop track over the pit all displacements are at the nominal zero position. Adjust connection line so that the string pot line has been extended approximately 5½" and set amplifier to zero. Adjust an LVDT body so that the core rod is centered in the body and adjust the amplifier to zero.

2. Apply a known displacement and make the appropriate gain adjustment for 2 volts/inch with the string pots and 20 volts/inch with the LVDT's.

   a) The known displacement is applied to the string pots by wrapping the string around a known cylindrical object.

   b) The known displacement is applied to the LVDT's by clamping a collar to the core rod using a calibration block to set the spacing from the body and then loosening the body clamp so that the body may be moved into contact with the collar.

3. Return sensors to the zero position.

F. Final Checks

1. Make sure that all patch cords are firmly seated in the plugs for the desired measurement.

2. Check that the strip charts and tape recorders have sufficient paper and tape.

3. Make sure that all power supplies and equipment are still powered up.

4. The instrumentation is ready for testing.

G. Record Cal Pulses

1. Immediately before each test run inject plus and minus 10 volt levels into each channel to be recorded on tape and record on tape.

2. Record and label full scale levels on strip charts.

3. Code test series and run number on tape by making two spaced groups of +10v pulses on the ALD channel, the first for the sequential test series number and the second for the run number, and describe the test variables in the log book.

7.0 TEST DOCUMENTATION

7.1 GENERAL

All test logs shall be checked and initialed by the Test Supervisor or his designee on a daily basis.

7.2 EVENTS LOG

It shall be the responsibility of the Test Supervisor to maintain a daily log which will serve as a record of all pertinent test information, and as a record of any special events. The Event Log shall be maintained on a daily basis beginning with the first day in the field.

7.3 DAS SYSTEM LOG

It shall be the responsibility of the Computer Operator to maintain a log of DAS status and digital tape records.

7.4 INSTRUMENTATION AND CALIBRATION LOG

It shall be the responsibility of the Lead Engineer to maintain a daily log beginning with the first day in the field, of instrumentation operational status, instrumentation calibration, and any special events or occurrences directly related to the instrumentation.

7.5 BRUSH CHART LOG

It shall be the responsibility of the person assigned to operation of the Brush chart recorder to maintain a daily log, beginning with the first day in the field, of Brush chart operational status, channel assignments, channel sensitivities, chart speeds, test runs, and all pertinent information necessary for the proper identification of all brush charts generated during the test.

7.6 SAFETY MONITOR LOG

It shall be the responsibility of the Safety Monitor or his designee to maintain a log of critical curves in the test zone.

7.7 WHEELSET STATUS LOG

It shall be the responsibility of the Wheelset Engineer to maintain a daily log on location status and calibration record of all unstrumented wheelsets. He will, on a daily basis and before the test consist moves, verify and enter into the log that brakes on all instrumented wheelsets are disabled.

7.9 RUN ID NUMBERS

All data shall be annotated as described in Section 8.1.

8.0 DATA MANAGEMENT

8.1 DATA TAPES

All data will be recorded on digital tapes. (See Appendix E for format.) The sampling rate for all signals shall be 256 Hz. All signals shall be filtered as shown in Table 5-2. All tapes will be logged with an eight digit run ID number.

12-34-5/6-78

12 - 1st two digits are month
34 - 2nd two digits are day
5/6 - 3rd two digits are phase and series number
78 - 4th two digits are run number

i.e., 06-12-1/3-01 =

June, 12, Phase 1, Series 3, Run 01

8.2 BRUSH CHARTS

All brush charts will be annotated with run numbers as described in 8.1. A brush chart log (Figure 8-1) will be filled in and attached to each brush chart.

8.3 REPRODUCTIONS

A duplicate copy of selected digital tapes and brush charts shall be made available as required. Original tapes and charts will not be available for distribution and will be under control of the Test Supervisor. Request for copies must come from the Test Director.

Original: ☐ Date: ______________

Reproduction: ☐ By: ______________

LRC/CANT DEFIENCY TEST

STRIP CHART NO. ____

RUN NO. ______________

CANT (IN) ______________

ZONE ______________

CHART SPEED (MM/SEC) ________

BRUSH CHANNEL	DATA CHANNEL	CHANNEL NAME	RANGE	S/F (V/DIV)
1				
2				
3				
4				
5				
6				
EVENT #1				
EVENT #2				

NOTES:

Figure 8-1. Brush Chart Form

8.4 DATA EVALUATION

Analysis of data will be limited for this test to real-time parameters as noted below and in Appendix E. RQ data will be available approximately 60 days after completion of the test.

8.4.1 REAL TIME

Using the following parameters wheel-to-rail forces and vehicles dynamic response will be evaluated and displayed during testing:

- Peak vertical loads (high rail)
- Peak lateral loads (high rail)
- Maximum single wheel L/V (high rail)
- Maximum truck L/V (high rail)
- Carbody resultant force vector
- Time duration of events

8.4.2 POST TEST

Statistical processing of key parameters will be performed from digital data tapes so that the maximum safe operational cant deficiency of each vehicle may be determined using the criteria described in Appendix E. Appropriate indices of ride comfort will be applied to determine the effectiveness of the coach banking system.

9.0 SAFETY MONITORING

The safety criteria recommended by world-wide sources for determining operating cant deficiency limits are listed in Appendix E. It is a detailed discussion exerpted from the LRC test report. The criteria addresses four principal hazzards 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 compute safe curving speeds for revenue trains by including maximum wind forces and making conservative "worst case" interpretations. The test conditions are designed to allow for safety at higher cant deficiency than appropriate for revenue trains. The wheel forces will be monitored continuously while cant deficiency is increased incrementally. Explicit foreknowledge provides the margin of safety that allows elevated test speeds. The decision to make a higher speed pass over a particular curve will be based on a linear extrapolation of measurements from previous runs. The trends of wheel forces and force ratios versus cant deficiency meaxured during the LRC cant deficiency test were quite linear.

The test termination criteria regarding vehicle overturning and wheel climb will be less restrictive than that governing recommendations for revenue operating cant deficiency. The more restrictive operating safety criteria, however, will be applied when considering rail rollover and track panel shift to prevent the possibility of track damage during the test runs. The wind force allowance factors will not be required because the actual wheel forces will be monitored in real time, but tests will not be conducted when gusts greater than 40 mph are forecasted, to avoid extraneous influences on the measurements.

9.1 VEHICLE OVERTURNING

The minimum vetical force on the low rail wheels will be monitored to prevent wheel lift. When the vertical force of a low rail wheel of the coach or locomotive has remained below 2000 pounds for 50 milliseconds or more, the speed at that curve will be reduced in any subsequent runs. All force and force ratio displays will be filtered at 25 Hz with a 4 pole filter.

9.2 WHEEL CLIMB

The lateral to vertical force ratio of the high rail lead wheel of the locomotive and of the coach will be monitored to indicate the danger of wheel climb. An L/V ratio of 1.0 or greater with a duration of 50 milliseconds or greater will signal a reduction in speed for any subsequent runs on that curve.

9.3 RAIL ROLLOVER

The high rail side truck (L/V) ratios of the locomotive and coach will be monitored to indicate danger of rail rollover. A truck side (L/V) ratio greater than 0.5 + 2300 lb/P_W with a duration of 50 milliseconds or greater will signal a reductin in speed for any subsequent runs at that curve. P_W is the nominal vertical wheel load of the particular vehicle.

9.4 TRACK PANEL SHIFT

The sum of the lateral wheel forces on the high rail at the lead trucks of the locomotive and coach will be monitored to indicate danger of track panel shift. A truck side lateral force greater than .85P + 8100 lb with a duration of 50 milliseconds or greater will signal a reduction in speed for any subsequent runs at that curve. P is the nominal vertical axle load of the particular vehicle. The above track panel shift criteria has been adjusted to account for the maximum unfavorable effect of likely rail temperatures on the continuously welded rail typical of the Northeast Corridor.

Table 9-1 summarizes the exact wheel/rail force levels that will require the termination of test speed increases at any particular curve.

TABLE 9-1
TEST SPEED LIMITING CRITERIA

Hazard	Measurement Monitored	Locomotive Limit	Coach Limit
Vehicle Overturning	Minimum Vertical Wheel Load; t-50 ms	2000 lb.	2000 lb.
Wheel Climb	Wheel (L/V) ratio; t - 50 ms	1.0	1.0
Rail Rollover	Truck Side (L/V) ratio; t - 50 ms	.57	0.65
Track Panel Shift	Truck Side Lateral t - 50 ms	63,350 lb.	30,500 lb.

10.0 SCHEDULE

The following schedule covers the preparation, installation and removal of test equipment and a test program requiring about 15 days of track time.

<u>April 1-30</u>

1. Wheelsets

   - repair of Amcoach wheel 1-Y
   - calibration of Amcoach wheels
   - modification and installation of slip ring units
   - repair of F40 wheel 3-A
   - rewiring of F40 axle 3
   - calibration of F40 wheelsets

2. Mechanical

   - design of Amcoach sensor brackets
   - fabrication of sensor brackets

<u>May 1-21</u>

1. Mechanical

   - design of locomotive sensor brackets
   - redesign of slip ring units for NEC clearance
   - fabrication of components

2. Electrical

   - preparation of input cabling, connectors, junction boxes, power supplies and amplifiers.
   - reconfiguration of output patch panel
   - refurbishment of FRA strip chart recorders

3. Software

   - sensor channel assignments and scaling
   - real time computations for body roll angle and filtered "jerk"
   - post-processing computations for body roll angle and filtered "jerk"
   - strip chart repro channel for scaled analog speed

May 22-28

- laboratory checkout of complete assembled system - from sensor to statistical computation

June 1

- disassembly and packing of system and wheelsets

June 2

- delivery to Wilmington Amtrak Shop

June 3-17

- installation of wheelsets, sensors & computer on Amcoach
- improvement of speed profile program for acceleration and braking rates
- balance coach weight

June 18-20

- movement of coach to New Haven Amtrak Shop
- delivery of wheelsets and sensors to New Haven

June 21-30

- installation of wheelsets, slip ring units, and sensors on F40PH locomotive
- inter-vehicle cabling

July 6-9

- shake down runs for instrumentation
- measurement of braking and accelerating rates
- shake down of coach adjustments
- static lean test
- ALD markers, curve 67
- final instrument checkout

July 12-13

- steady state tests - curve 67E

July 14

- examination of data

July 15-16

- steady state tests - curve 67W

July 19-23

- over-the-road testing: New Haven to Groton

July 26-30

- Demonstration runs: New Haven to Boston

Aug 2-8

- Removal of equipment from test vehicles

APPENDIX A

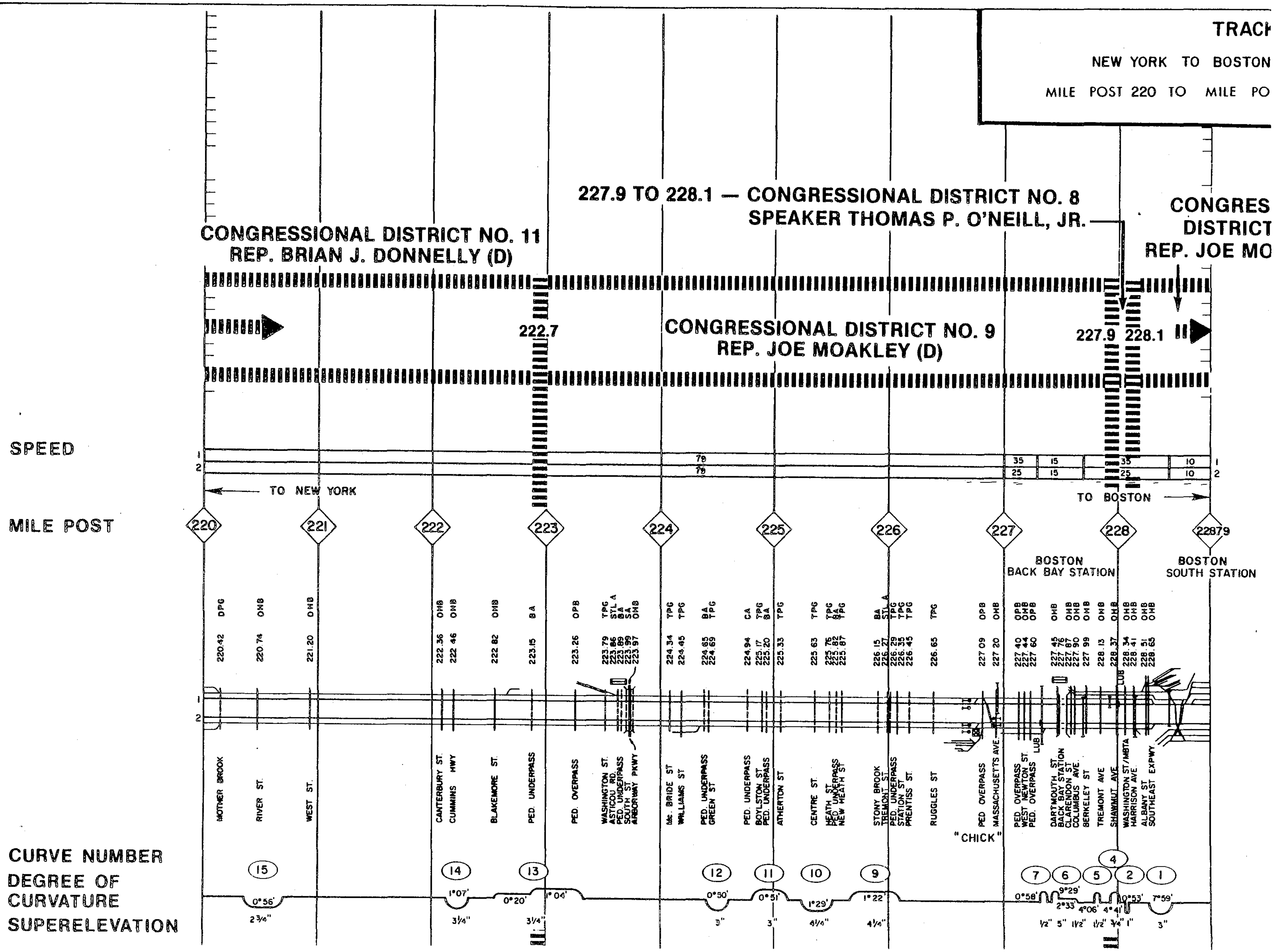
TRACK CHARTS

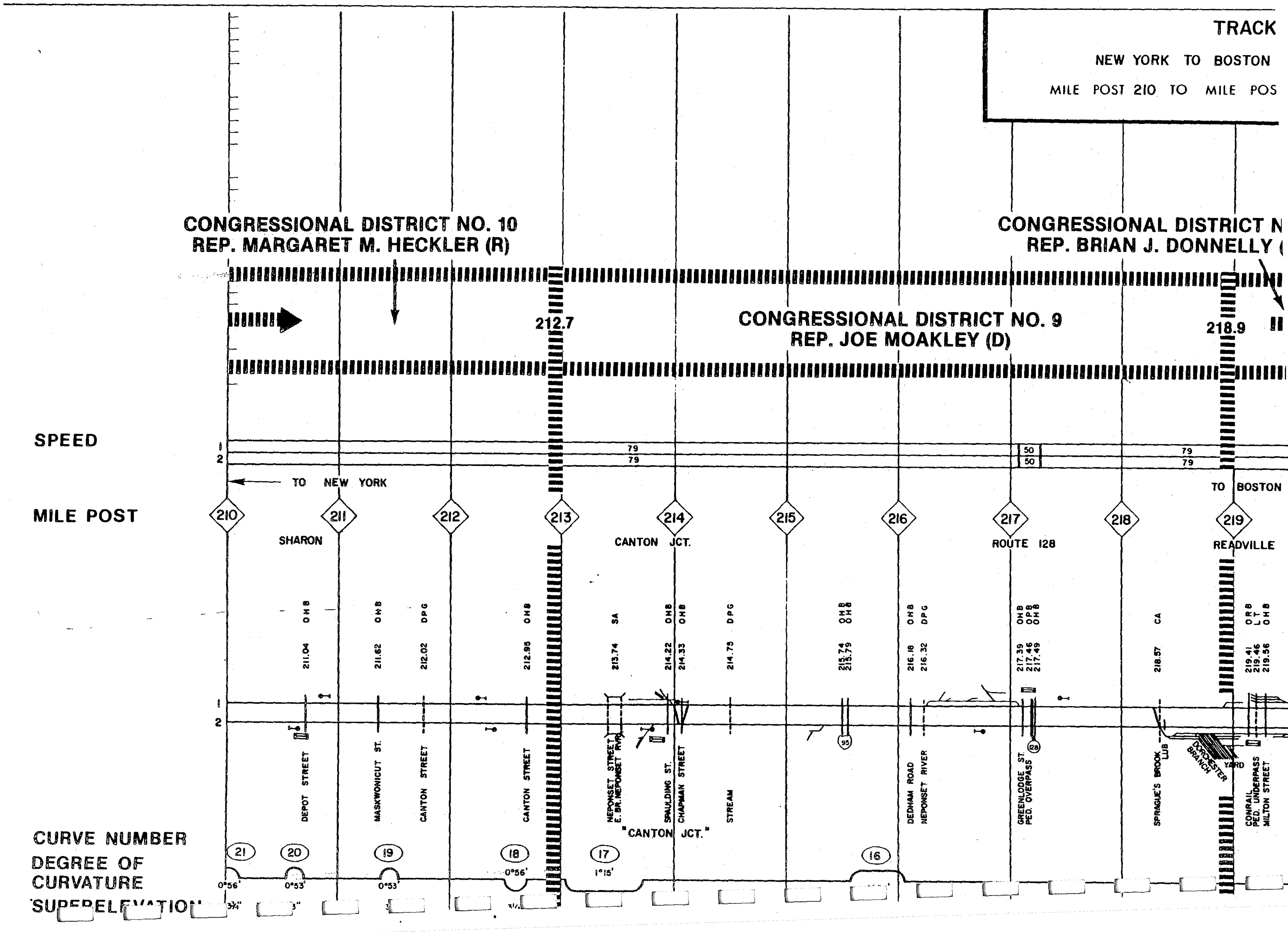
AMCOACH II (Soft Primary)

	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

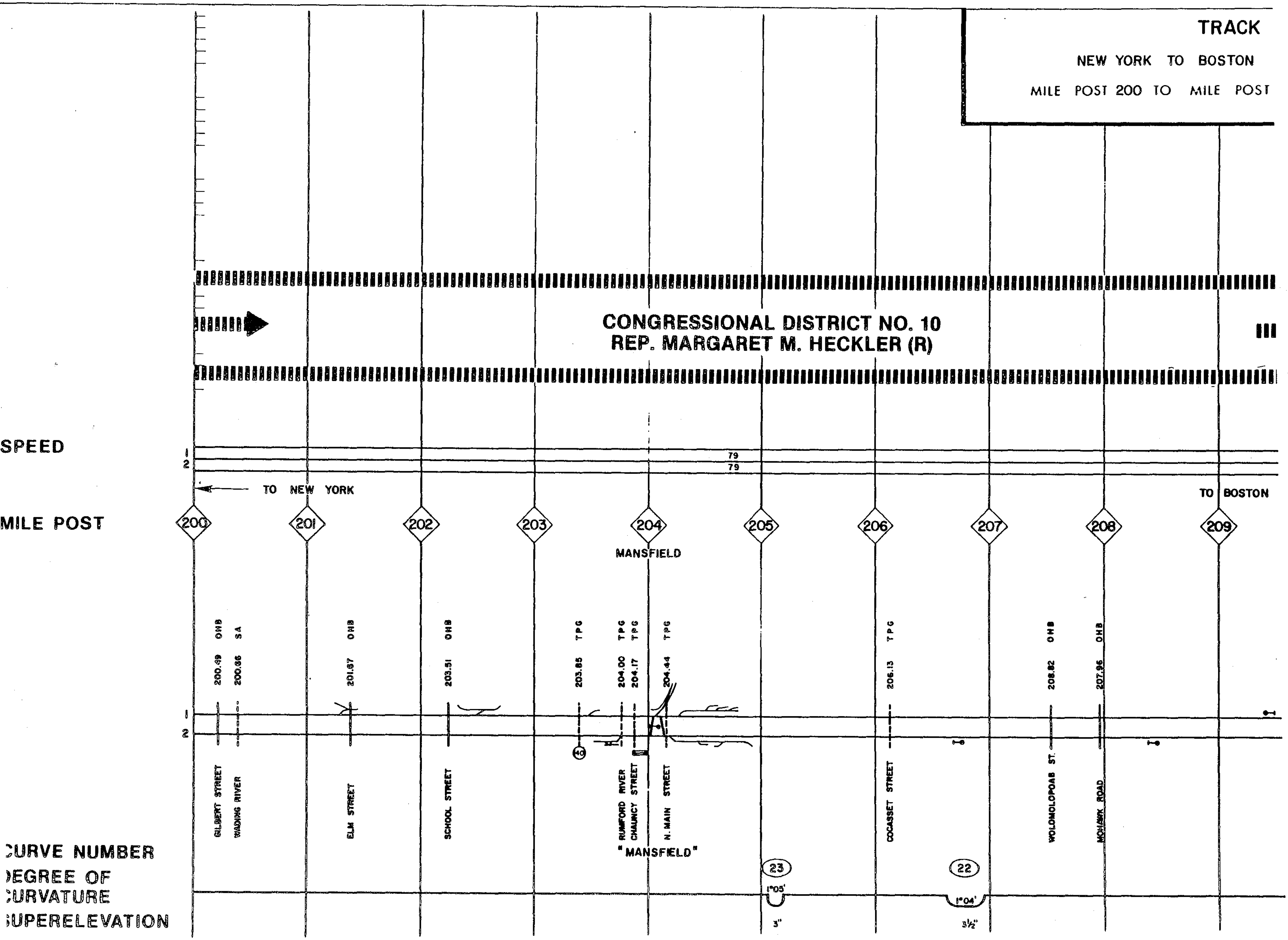
F40PH (Inclined Rubber Suspension)

	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


TRACK
NEW YORK TO BOSTON
MILE POST 220 TO MILE PO
227.9 TO 228.1 — CONGRESSIONAL DISTRICT NO. 8
SPEAKER THOMAS P. O'NEILL, JR.
CONGRES
DISTRICT
REP. JOE MO
CONGRESSIONAL DISTRICT NO. 11
REP. BRIAN J. DONNELLY (D)
222.7
CONGRESSIONAL DISTRICT NO. 9
REP. JOE MOAKLEY (D)
227.9
228.1
SPEED
79
35
25
15
10
TO NEW YORK
TO BOSTON
MILE POST
220
221
222
223
224
225
226
227
228
228.79
BOSTON BACK BAY STATION
BOSTON SOUTH STATION
220.42 DPG MOTHER BROOK
220.74 OHB RIVER ST.
221.20 OHB WEST ST.
222.36 OHB CANTERBURY ST.
222.46 OHB CUMMINS HWY
222.82 OHB BLAKEMORE ST.
223.15 BA PED. UNDERPASS
223.26 OPB PED. OVERPASS
223.79 TPG WASHINGTON ST.
223.86 STL A ASTICOU RD.
223.89 BA PED. UNDERPASS
223.99 SA SOUTH ST.
223.97 OHB ARBORWAY PKWY
224.34 TPG Mc. BRIDE ST
224.45 TPG WILLIAMS ST
224.65 BA PED. UNDERPASS
224.69 TPG GREEN ST
224.94 CA PED. UNDERPASS
225.17 TPG BOYLSTON ST.
225.20 BA PED. UNDERPASS
225.33 TPG ATHERTON ST
225.63 TPG CENTRE ST.
225.76 TPG HEATH ST
225.82 BA PED. UNDERPASS
225.87 TPG NEW HEATH ST
226.15 BA STONY BROOK
226.27 STL A TREMONT ST
226.29 TPG PED. UNDERPASS
226.35 TPG STATION ST
226.45 TPG PRENTISS ST
226.65 TPG RUGGLES ST
227.09 OPB PED OVERPASS
227.20 OHB MASSACHUSETTS AVE.
"CHICK"
227.40 OPB PED. OVERPASS
227.44 OPB WEST NEWTON ST
227.60 OPB PED. OVERPASS
LUB
227.45 OHB DARTMOUTH ST.
227.76 BACK BAY STATION
227.87 OHB CLARENDON ST
227.90 OHB COLUMBUS AVE.
227.99 OHB BERKELEY ST
228.13 OHB TREMONT AVE
228.37 OHB SHAWMUT AVE
LUB
228.34 OHB WASHINGTON ST/MBTA
228.41 OHB HARRISON AVE.
228.51 OHB ALBANY ST
228.65 OHB SOUTHEAST EXPWY
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
15 0°56' 2¾"
14 1°07' 3¼"
13 0°20' 1°04' 3¼"
12 0°50' 3"
11 0°51' 3"
10 1°29' 4¼"
9 1°22' 4¼"
7 0°58' ½"
6 9°29' 2°33' 5"
5 4°06' 1½"
4 4°41' ½"
2 0°53' ¾" 1"
1 7°59' 3"



TRACK
NEW YORK TO BOSTON
MILE POST 210 TO MILE POS
CONGRESSIONAL DISTRICT NO. 10
REP. MARGARET M. HECKLER (R)
212.7
CONGRESSIONAL DISTRICT NO. 9
REP. JOE MOAKLEY (D)
218.9
CONGRESSIONAL DISTRICT N
REP. BRIAN J. DONNELLY
SPEED
79
79
50
50
79
79
TO NEW YORK
TO BOSTON
MILE POST
210
211
212
213
214
215
216
217
218
219
SHARON
CANTON JCT.
ROUTE 128
READVILLE
211.04 OHB
211.62 OHB
212.02 DPG
212.95 OHB
213.74 SA
214.22 OHB
214.33 OHB
214.75 DPG
215.74 OHB
215.79 OHB
216.18 OHB
216.32 DPG
217.39 OHB
217.46 OPB
217.49 OHB
218.57 CA
219.41 ORB
219.46 LT
219.56 OHB
DEPOT STREET
MASKWONICUT ST.
CANTON STREET
CANTON STREET
NEPONSET STREET
E. BR. NEPONSET RVR.
SPAULDING ST.
CHAPMAN STREET
STREAM
DEDHAM ROAD
NEPONSET RIVER
GREENLODGE ST.
PED. OVERPASS
SPRAGUE'S BROOK
LUB
DORCHESTER BRANCH
YARD
CONRAIL
PED. UNDERPASS
MILTON STREET
"CANTON JCT."
95
128
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
21
20
19
18
17
16
0°56'
0°53'
0°53'
0°56'
1°15'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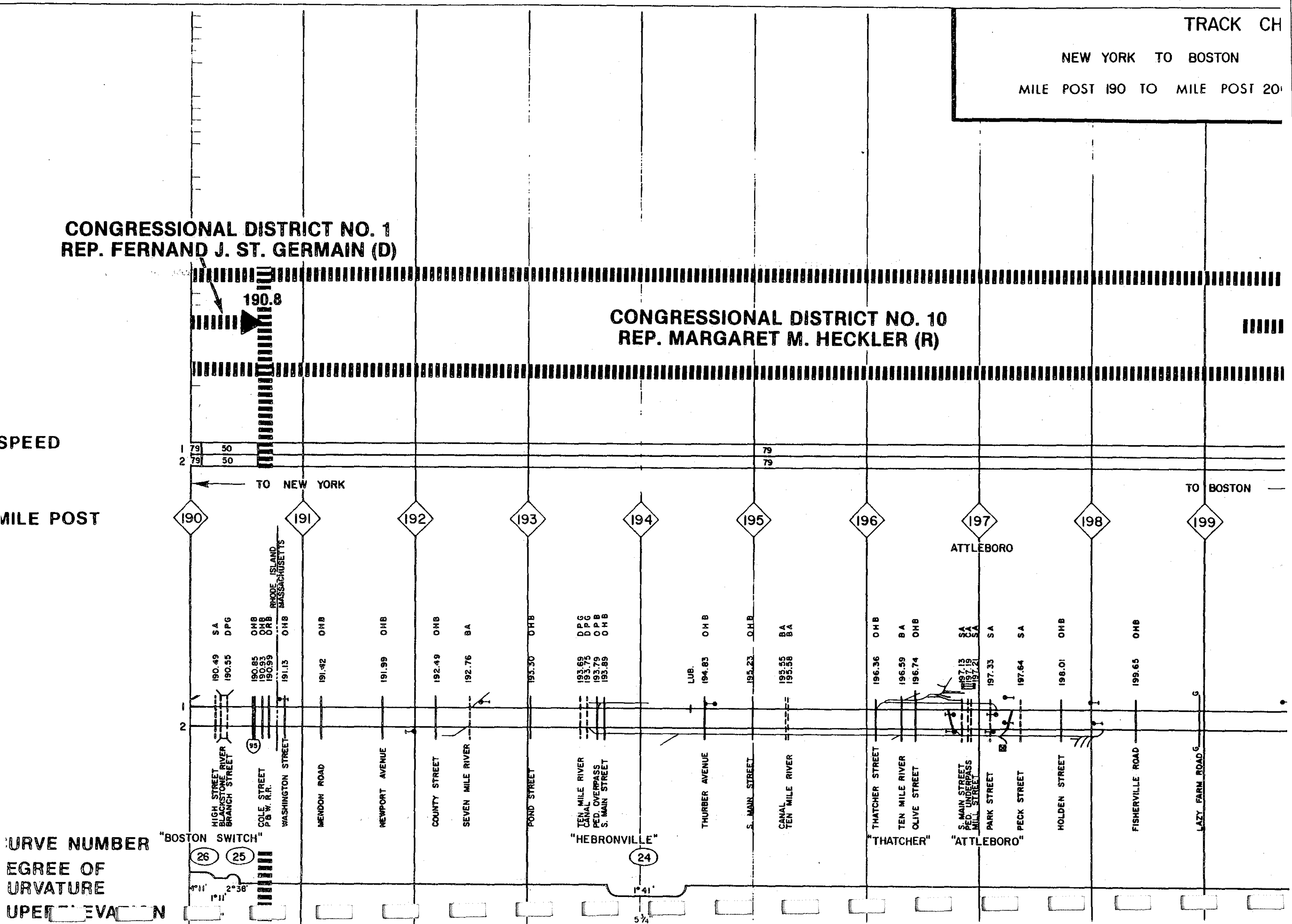
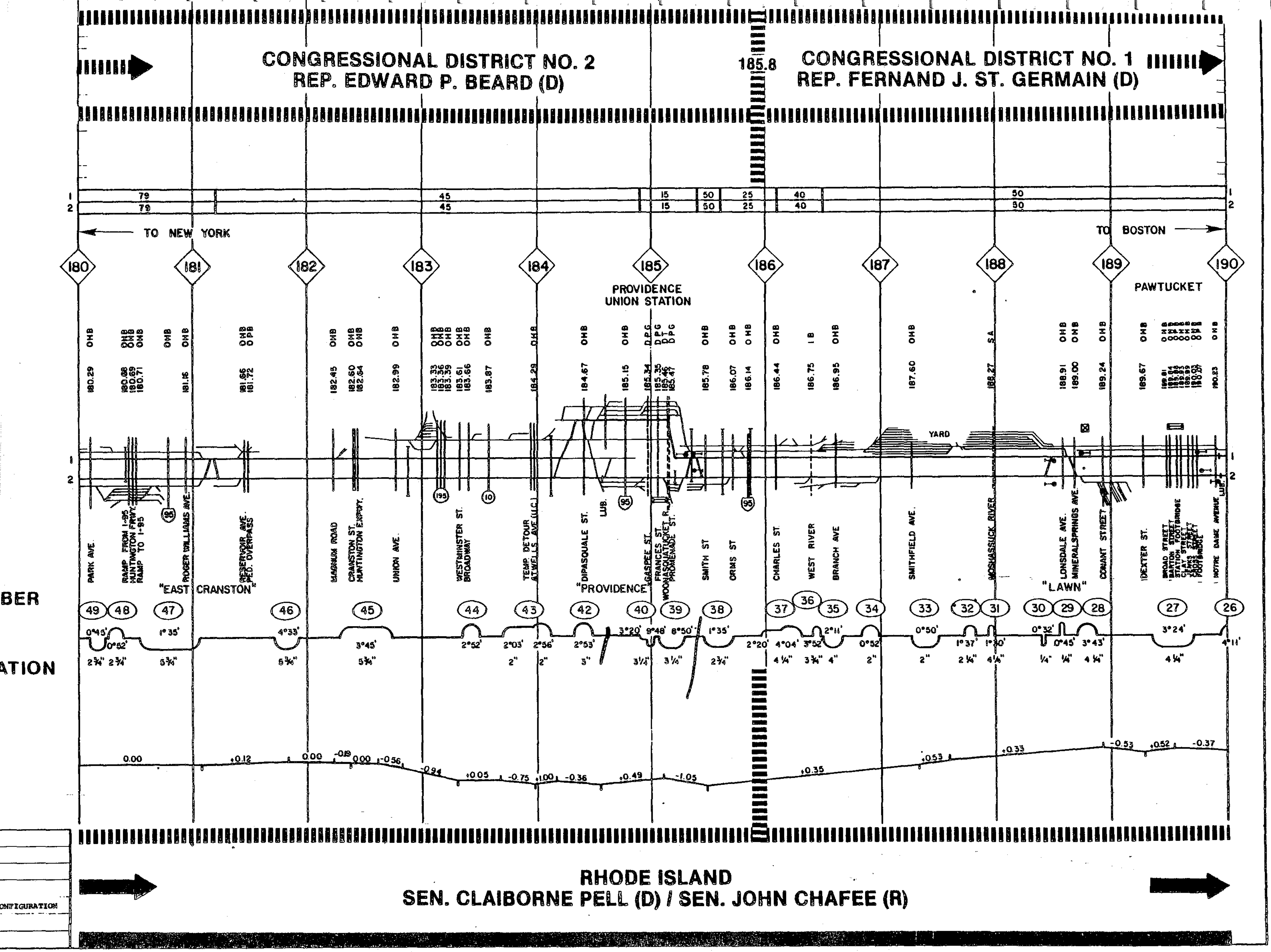
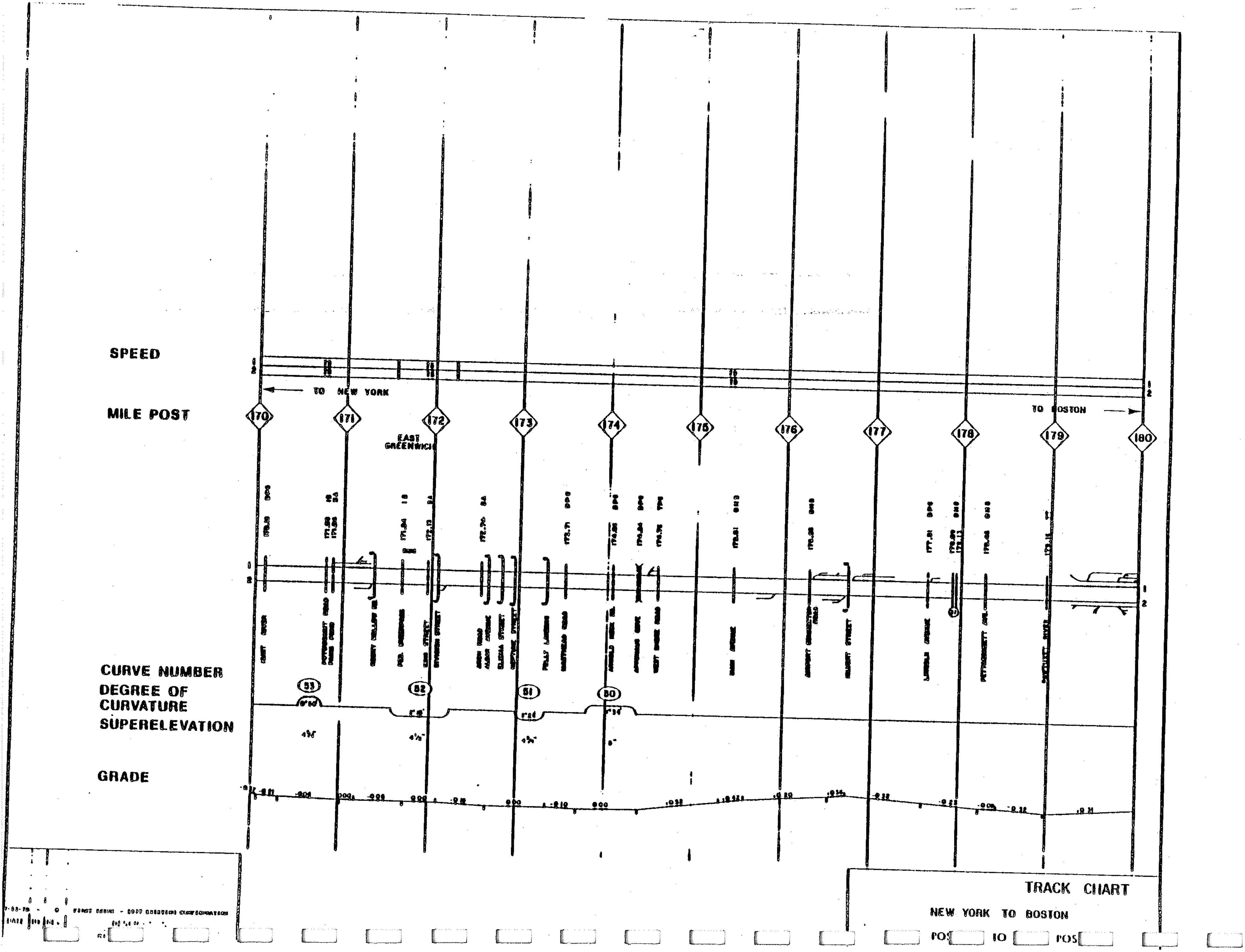
NEW YORK TO BOSTON
MILE POST 200 TO MILE POST
CONGRESSIONAL DISTRICT NO. 10
REP. MARGARET M. HECKLER (R)
SPEED
1
2
79
79
TO NEW YORK
TO BOSTON
MILE POST
200
201
202
203
204
205
206
207
208
209
MANSFIELD
200.49 OHB
200.66 SA
201.67 OHB
203.51 OHB
203.85 TPG
204.00 TPG
204.17 TPG
204.44 TPG
206.13 TPG
208.82 OHB
207.96 OHB
40
GILBERT STREET
WADING RIVER
ELM STREET
SCHOOL STREET
RUMFORD RIVER
CHAUNCY STREET
N. MAIN STREET
"MANSFIELD"
COCASSET STREET
WOLOMOLOPOAB ST.
MOHAWK ROAD
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
23
1°05'
3"
22
1°04'
3½"

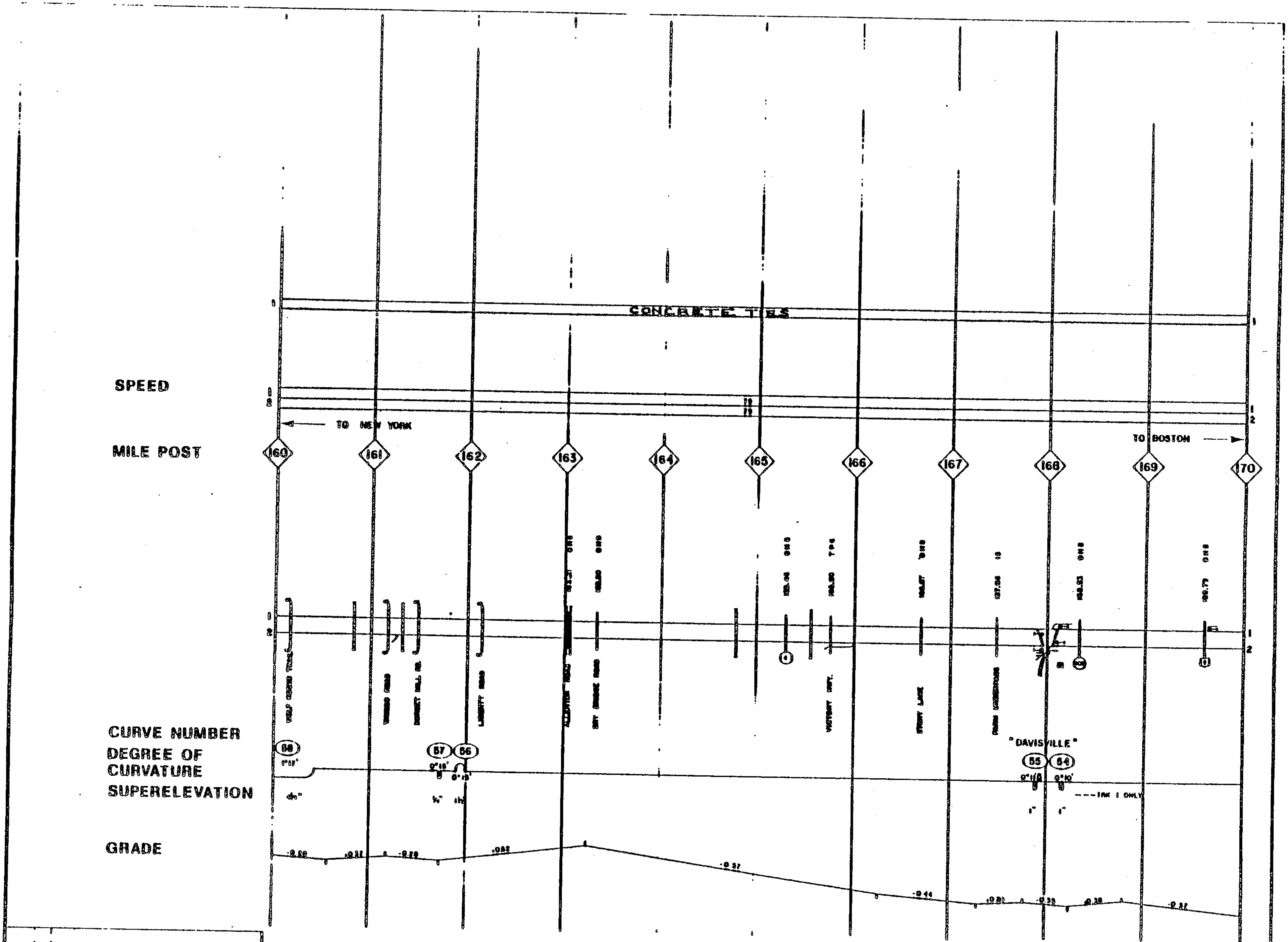


TRACK CH
NEW YORK TO BOSTON
MILE POST 190 TO MILE POST 20
CONGRESSIONAL DISTRICT NO. 1
REP. FERNAND J. ST. GERMAIN (D)
190.8
CONGRESSIONAL DISTRICT NO. 10
REP. MARGARET M. HECKLER (R)
SPEED
79
50
TO NEW YORK
TO BOSTON
MILE POST
190
191
192
193
194
195
196
197
198
199
ATTLEBORO
RHODE ISLAND
MASSACHUSETTS
190.49 SA HIGH STREET
190.55 DPG BLACKSTONE RIVER BRANCH STREET
190.85 OHB COLE STREET
190.93 OHB P&W R.R.
190.99 ORB
191.13 OHB WASHINGTON STREET
191.42 OHB MENDON ROAD
191.99 OHB NEWPORT AVENUE
192.49 OHB COUNTY STREET
192.76 BA SEVEN MILE RIVER
193.30 OHB POND STREET
193.69 DPG TEN MILE RIVER
193.75 DPG CANAL
193.79 OPB PED. OVERPASS
193.89 OHB S. MAIN STREET
LUB.
194.83 OHB THURBER AVENUE
195.23 OHB S. MAIN STREET
195.55 BA CANAL
195.58 BA TEN MILE RIVER
196.36 OHB THATCHER STREET
196.59 BA TEN MILE RIVER
196.74 OHB OLIVE STREET
197.13 SA S. MAIN STREET
197.19 CA PED. UNDERPASS
197.21 SA MILL STREET
197.33 SA PARK STREET
197.64 SA PECK STREET
198.01 OHB HOLDEN STREET
199.65 OHB FISHERVILLE ROAD
LAZY FARM ROAD
95
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
"BOSTON SWITCH"
26
25
4°11'
1°11'
2°38'
"HEBRONVILLE"
24
1°41'
5¾
"THATCHER"
"ATTLEBORO"

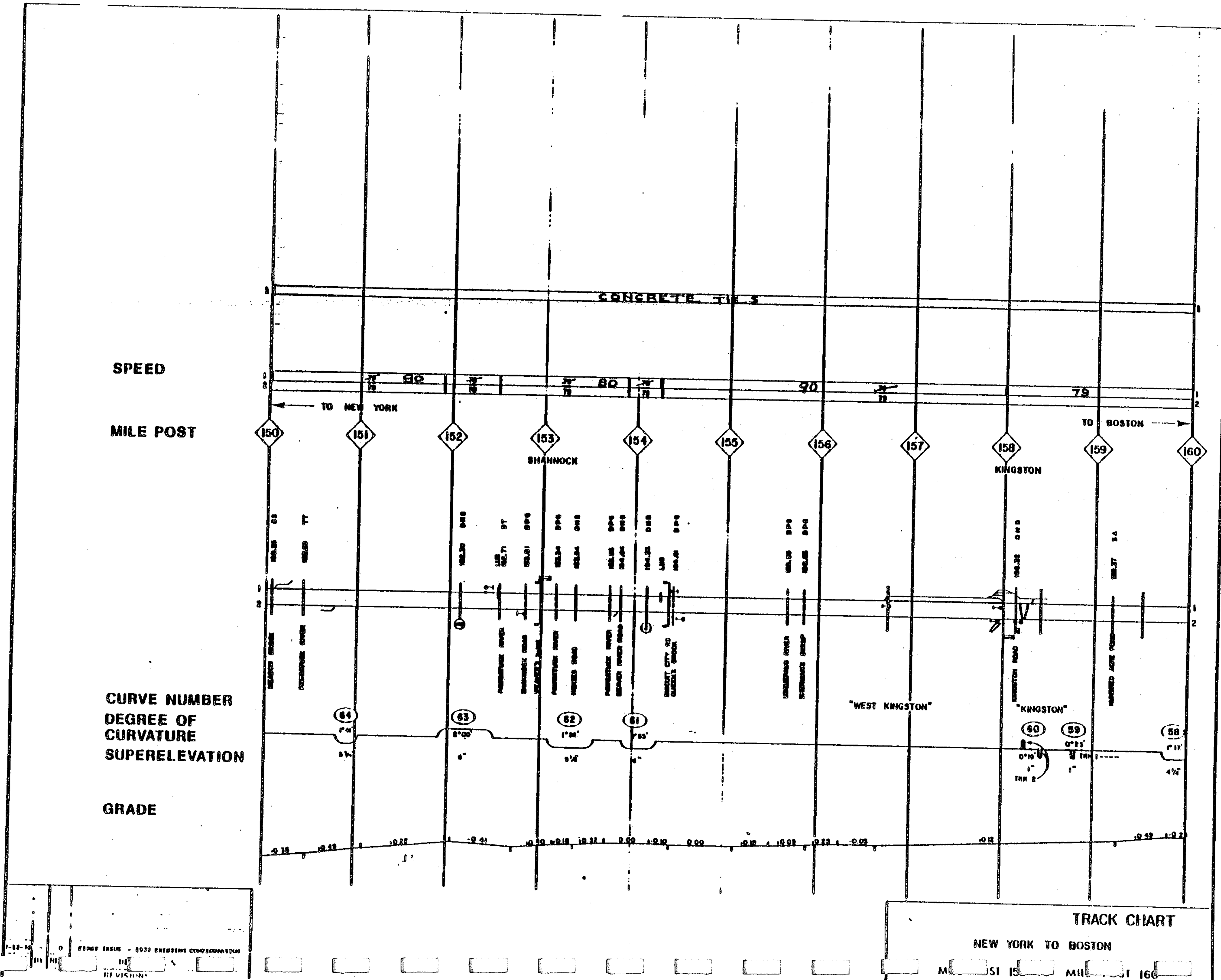

CONGRESSIONAL DISTRICT NO. 2
REP. EDWARD P. BEARD (D)
185.8
CONGRESSIONAL DISTRICT NO. 1
REP. FERNAND J. ST. GERMAIN (D)
TO NEW YORK
TO BOSTON
PROVIDENCE UNION STATION
PAWTUCKET
YARD
PARK AVE.
RAMP FROM I-95
HUNTINGTON FRWY.
RAMP TO I-95
ROGER WILLIAMS AVE.
"EAST CRANSTON"
RESERVOIR AVE.
PED. OVERPASS
MAGNUM ROAD
CRANSTON ST.
HUNTINGTON EXPWY.
UNION AVE.
WESTMINSTER ST.
BROADWAY
TEMP. DETOUR
ATWELLS AVE (U.C.)
DIPASQUALE ST.
LUB.
"PROVIDENCE"
GASPEE ST.
FRANCES ST.
WOONASQUATUCKET R.
PROMENADE ST.
SMITH ST
ORMS ST
CHARLES ST
WEST RIVER
BRANCH AVE
SMITHFIELD AVE.
MOSHASSUCK RIVER
LONSDALE AVE.
MINERALSPRINGS AVE.
"LAWN"
CONANT STREET
DEXTER ST.
BROAD STREET
BARTON STREET
STATION FOOTBRIDGE
CLAY STREET
JENKS STREET
CROSS STREET
FOOTBRIDGE
NOTRE DAME AVENUE
RHODE ISLAND
SEN. CLAIBORNE PELL (D) / SEN. JOHN CHAFEE (R)
BER
ATION
CONFIGURATION



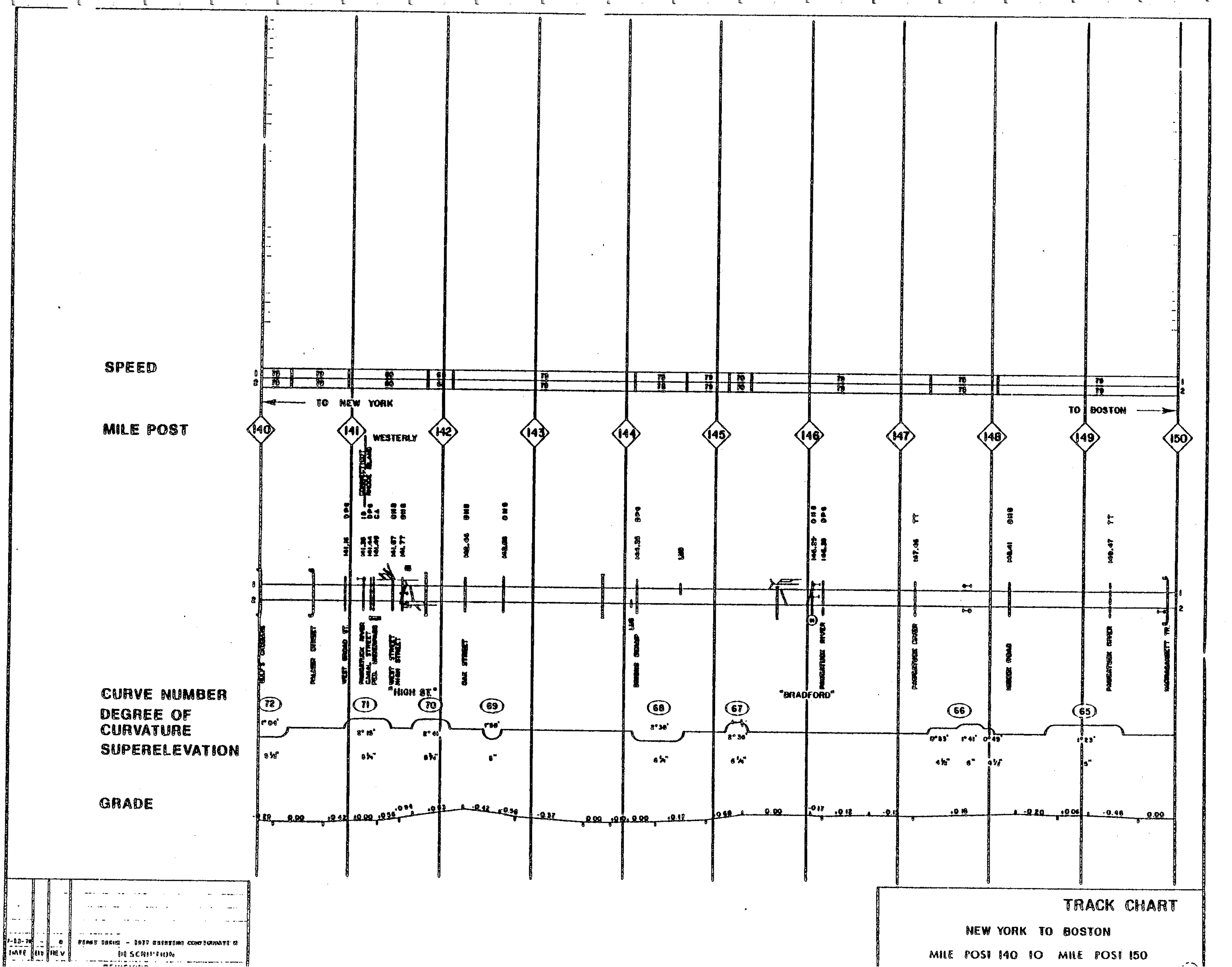
SPEED
MILE POST
TO NEW YORK
TO BOSTON
170
171
172
173
174
175
176
177
178
179
180
EAST GREENWICH
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
GRADE
53
52
51
50
TRACK CHART
NEW YORK TO BOSTON

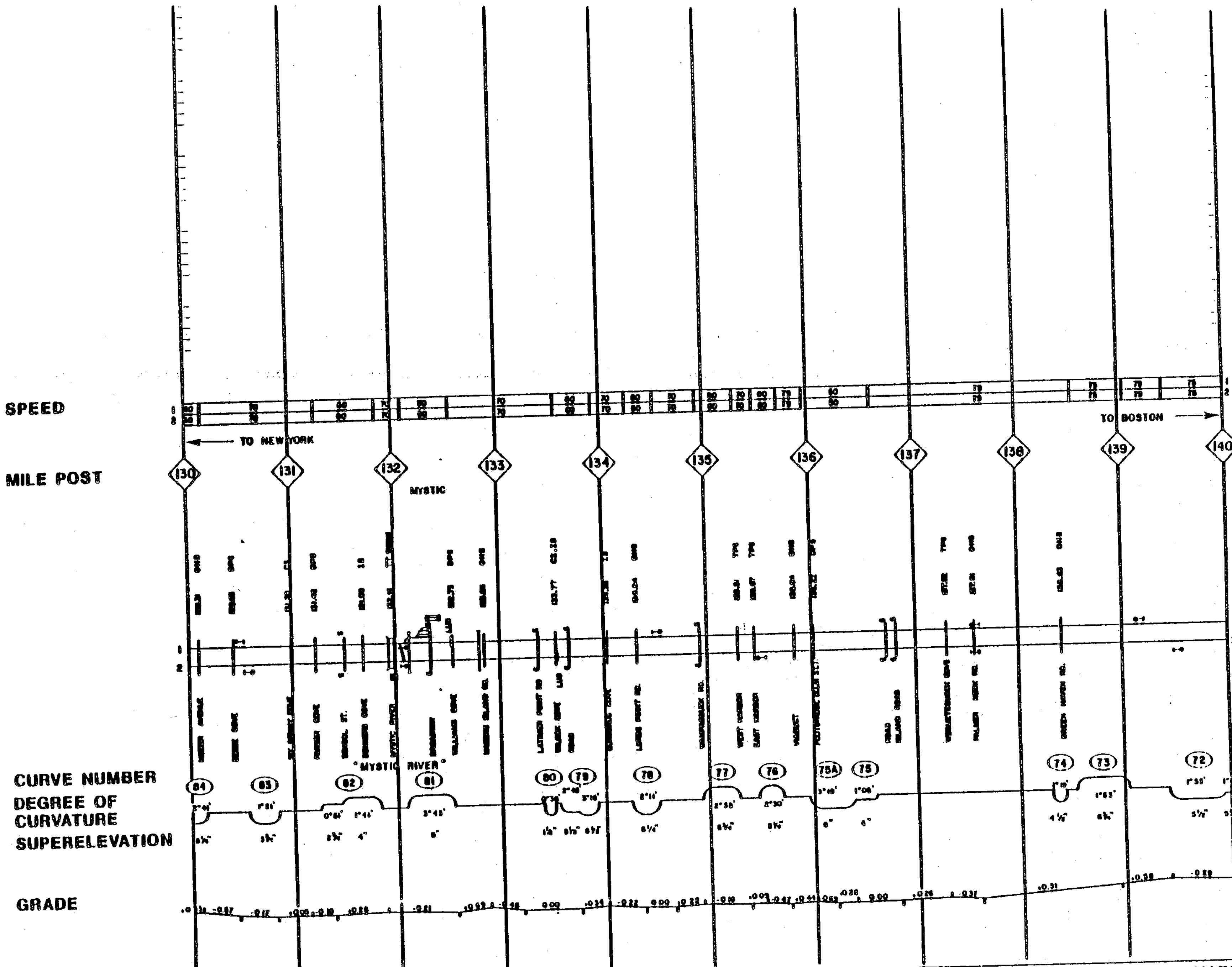


TRACK CHART
NEW YORK TO BOSTON
MILE POST 160 TO MILE POST 170



SPEED
MILE POST
CURVE NUMBER
DEGREE OF
CURVATURE
SUPERELEVATION
GRADE
CONCRETE TIES
80
80
90
79
TO NEW YORK
TO BOSTON
150
151
152
153
154
155
156
157
158
159
160
SHANNOCK
KINGSTON
"WEST KINGSTON"
"KINGSTON"
64
63
62
61
60
59
58
8°00'
0°23'
TRK 2
TRACK CHART
NEW YORK TO BOSTON
REVISION


SPEED
MILE POST
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
GRADE
TO NEW YORK
TO BOSTON
140
141
WESTERLY
142
143
144
145
146
147
148
149
150
PALMER STREET
WEST BROAD ST.
PAWCATUCK RIVER
CANAL STREET
WEST STREET
HIGH STREET
"HIGH ST."
OAK STREET
"BRADFORD"
PAWCATUCK RIVER
72
71
70
69
68
67
66
65
TRACK CHART
NEW YORK TO BOSTON
MILE POST 140 TO MILE POST 150



TRACK CHART

NEW YORK TO BOSTON

MILE POST 130 TO MILE POST 140

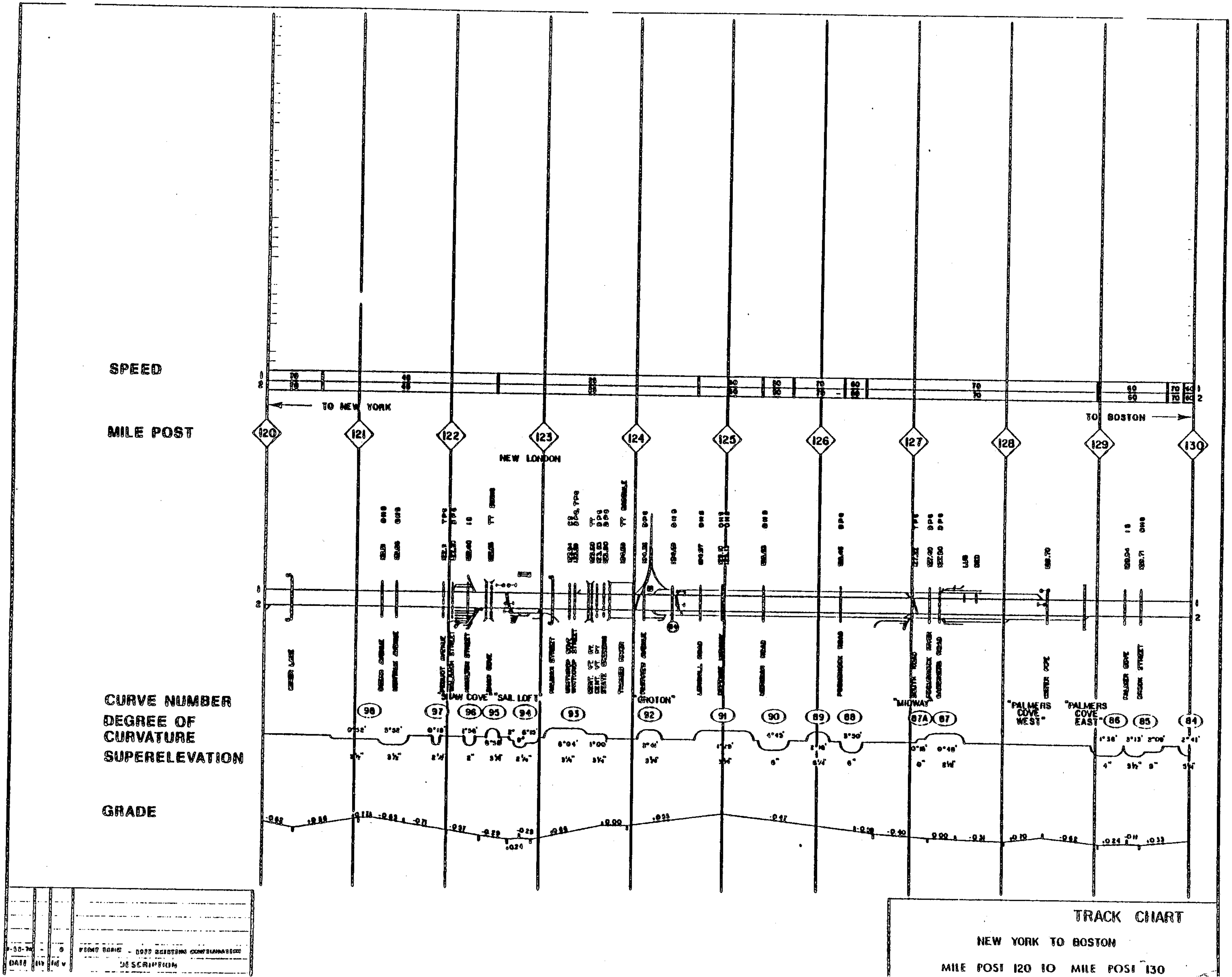
(38)

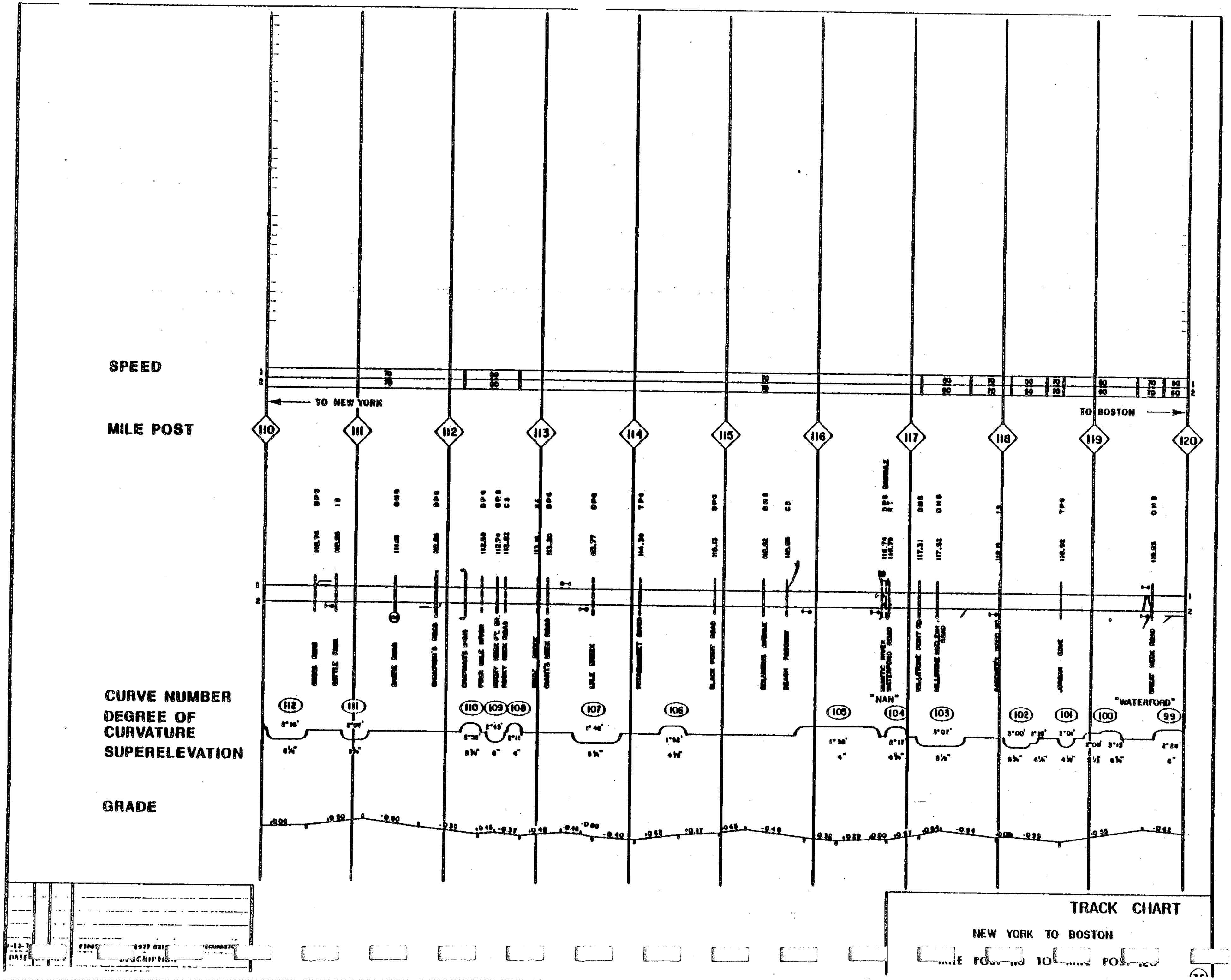
DATE	BY	REV	DESCRIPTION
7-28-78	-	0	FIRST ISSUE - 1977 EXISTING CONFIGURATION

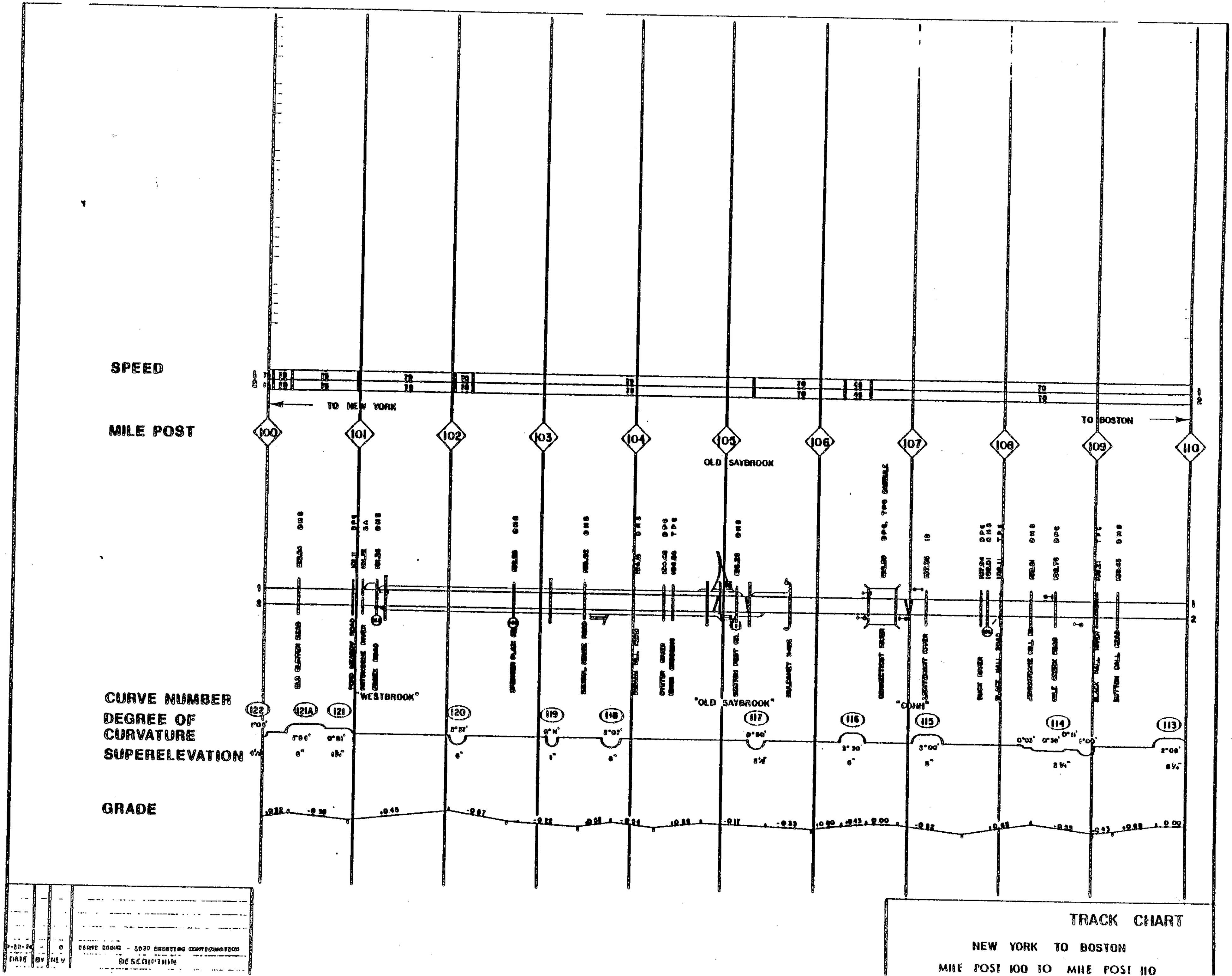
REVISIONS

NEW YORK TO BOSTON

MILE ... 140 ... MILE ... 150


SPEED
MILE POST
TO NEW YORK
TO BOSTON
120
121
122
123
124
125
126
127
128
129
130
NEW LONDON
"SHAW COVE"
"SAIL LOFT"
"GROTON"
"MIDWAY"
"PALMERS COVE WEST"
"PALMERS COVE EAST"
98
97
96
95
94
93
92
91
90
89
88
87A
87
86
85
84
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
GRADE
TRACK CHART
NEW YORK TO BOSTON
MILE POST 120 TO MILE POST 130
DATE
DESCRIPTION

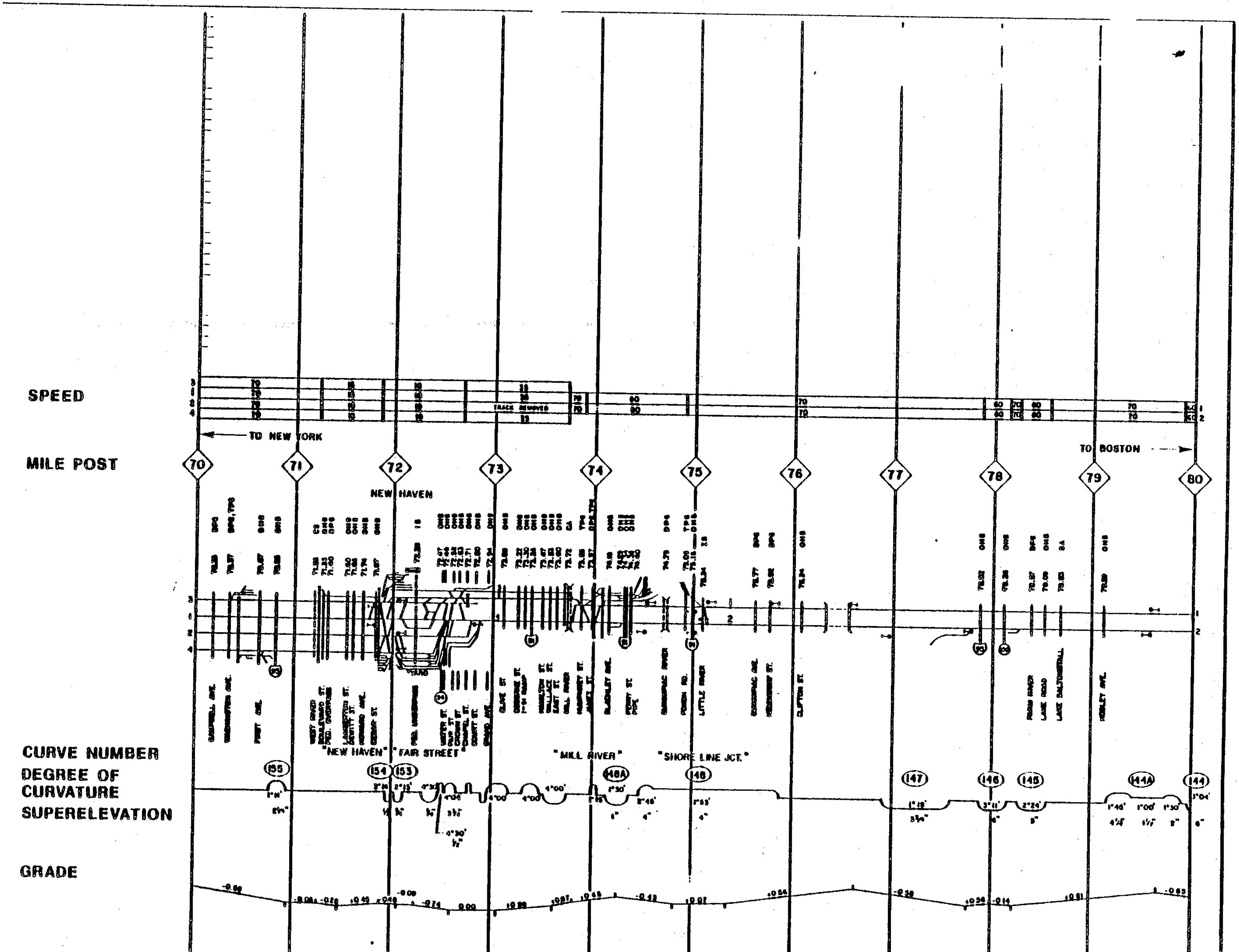

SPEED
MILE POST
CURVE NUMBER
DEGREE OF CURVATURE
SUPERELEVATION
GRADE
TO NEW YORK
TO BOSTON
110
111
112
113
114
115
116
117
118
119
120
"NAN"
"WATERFORD"
TRACK CHART
NEW YORK TO BOSTON



TRACK CHART

NEW YORK TO BOSTON

MILE POST 100 TO MILE POST 110

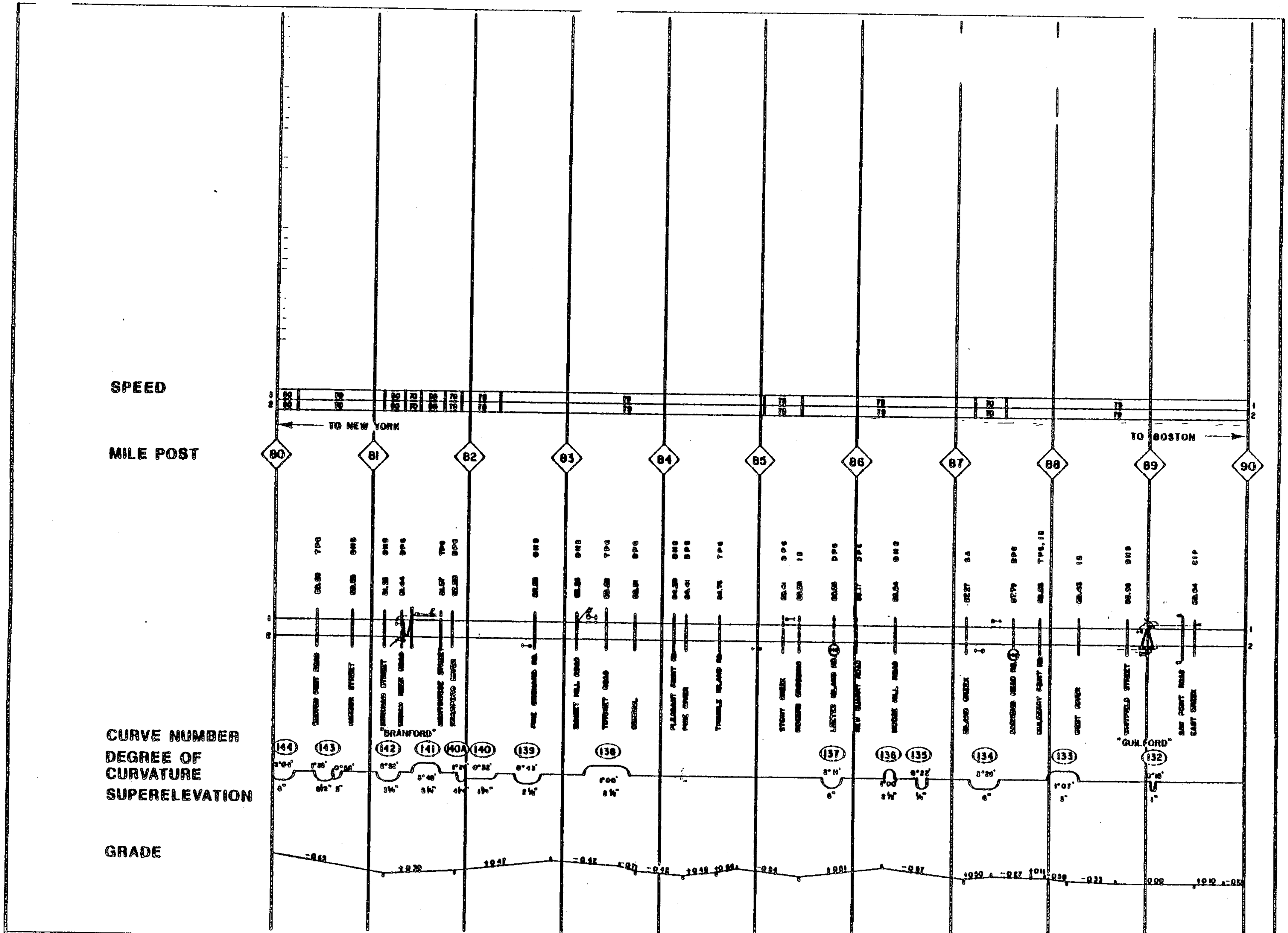


TRACK CHART

NEW YORK TO BOSTON

POST [illegible] O' M[illegible] OST [illegible]

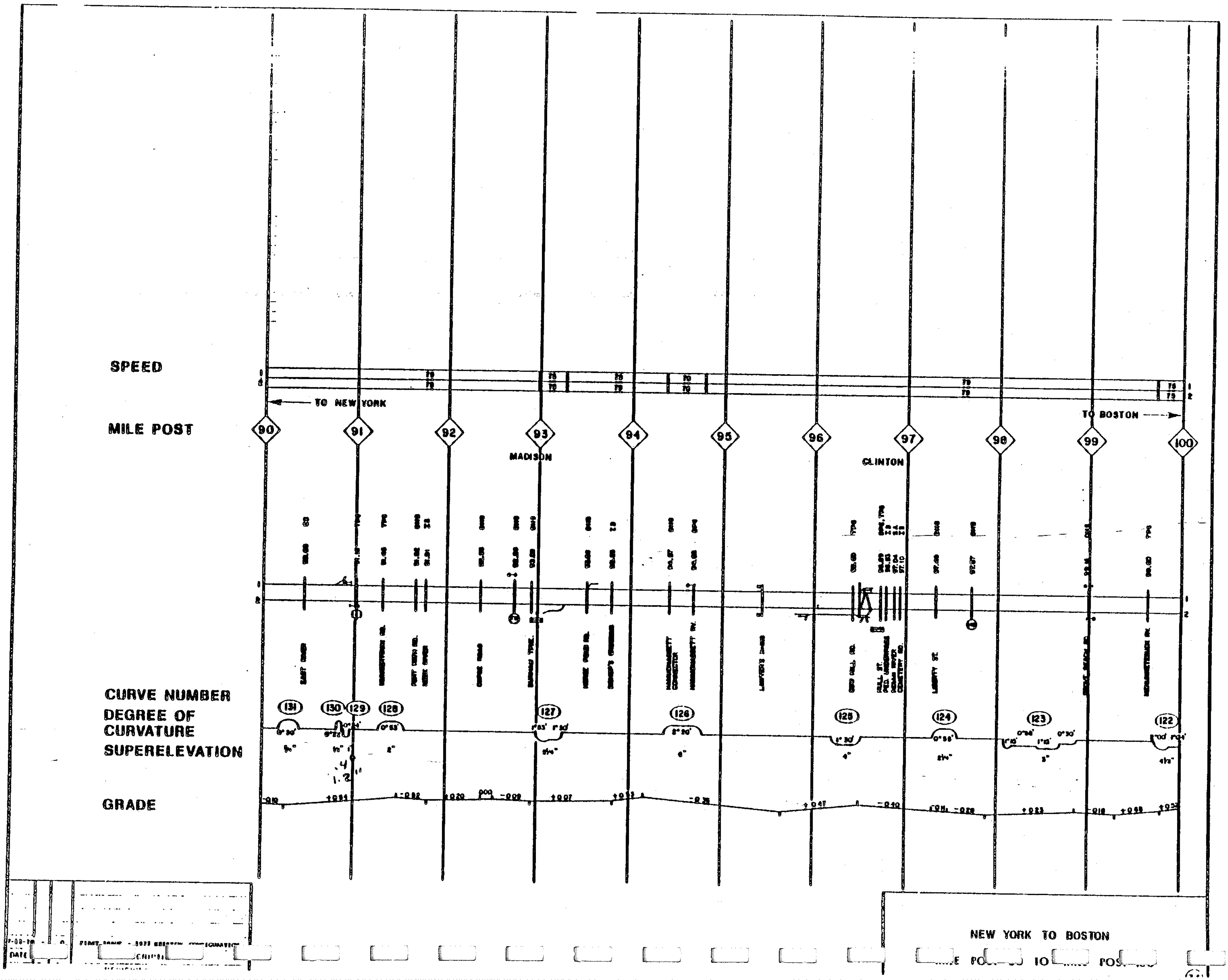
REVISIONS



TRACK CHART

NEW YORK TO BOSTON

MILE POST 80 TO MILE POST 90

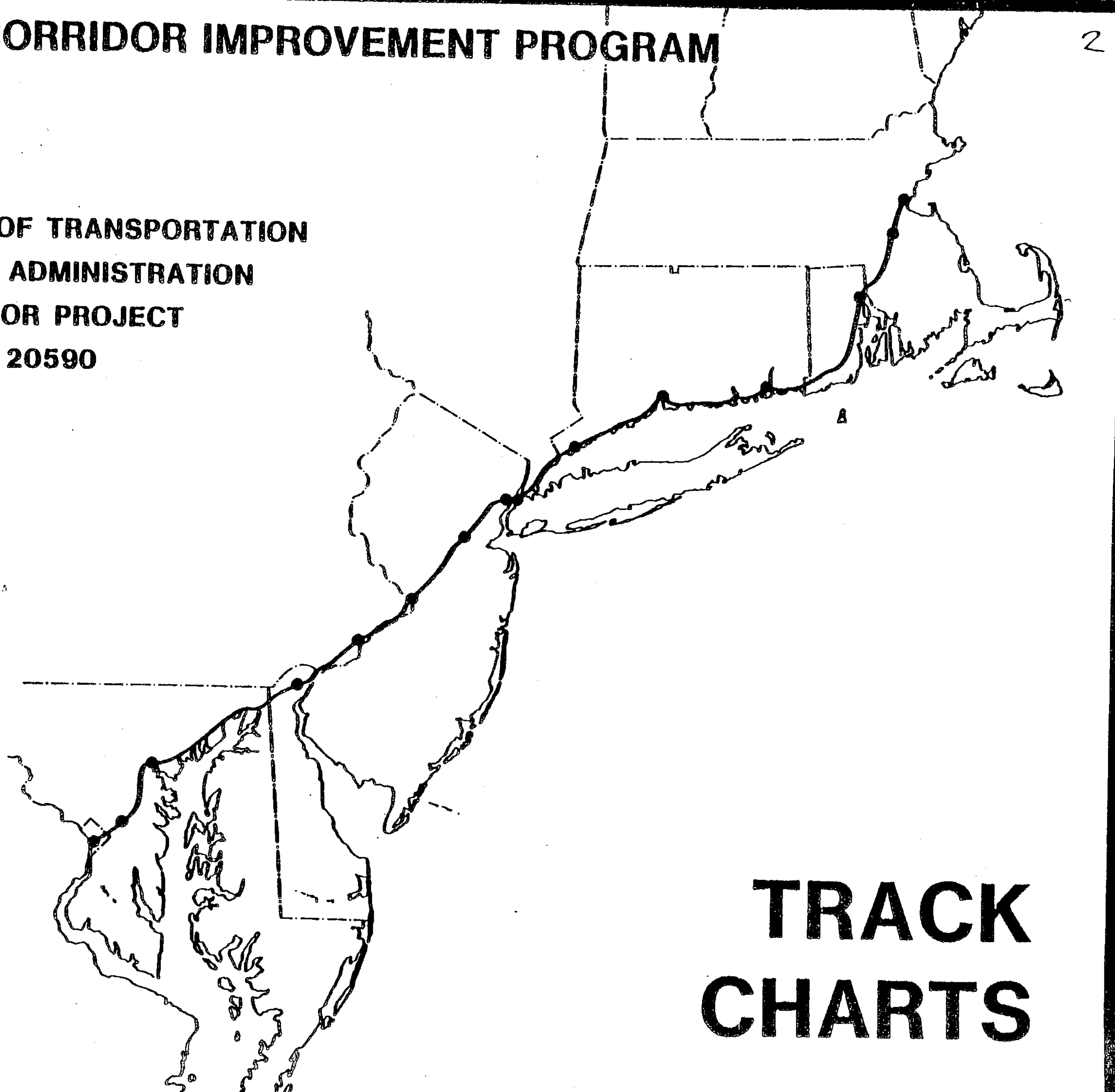


SPEED
MILE POST
CURVE NUMBER
DEGREE OF
CURVATURE
SUPERELEVATION
GRADE
TO NEW YORK
TO BOSTON
90
91
92
93
94
95
96
97
98
99
100
MADISON
CLINTON
131
130
129
128
127
126
125
124
123
122
NEW YORK TO BOSTON

NORTHEAST CORRIDOR IMPROVEMENT PROGRAM

2

U. S. DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION
NORTHEAST CORRIDOR PROJECT
WASHINGTON, D. C. 20590



TRACK CHARTS

APPENDIX B

APPENDIX B

The simple quasistatic curving model shown in Figure B-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 Appendix C.

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

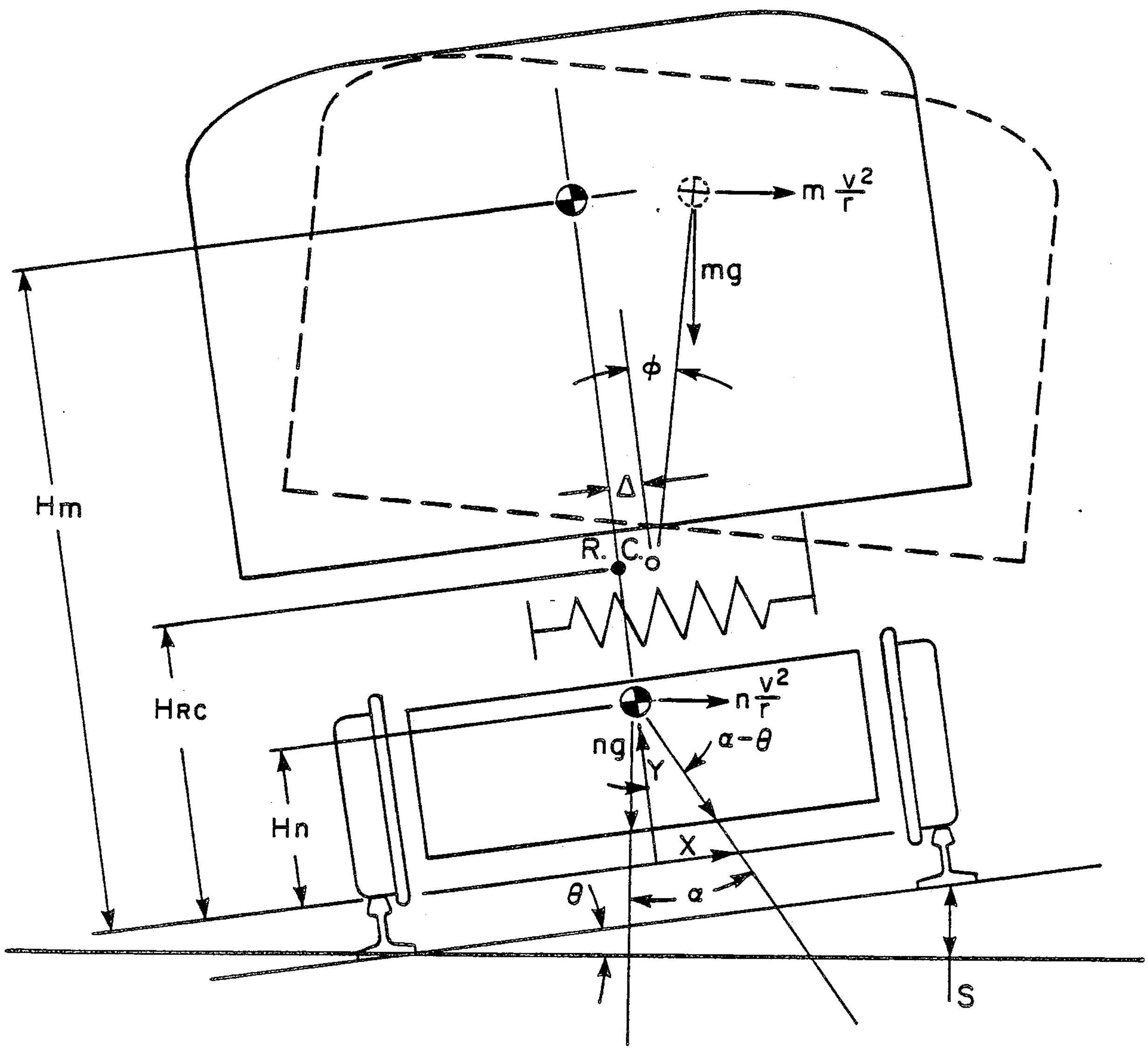


Figure B-1. Quasistatic Curving Model

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

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

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

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

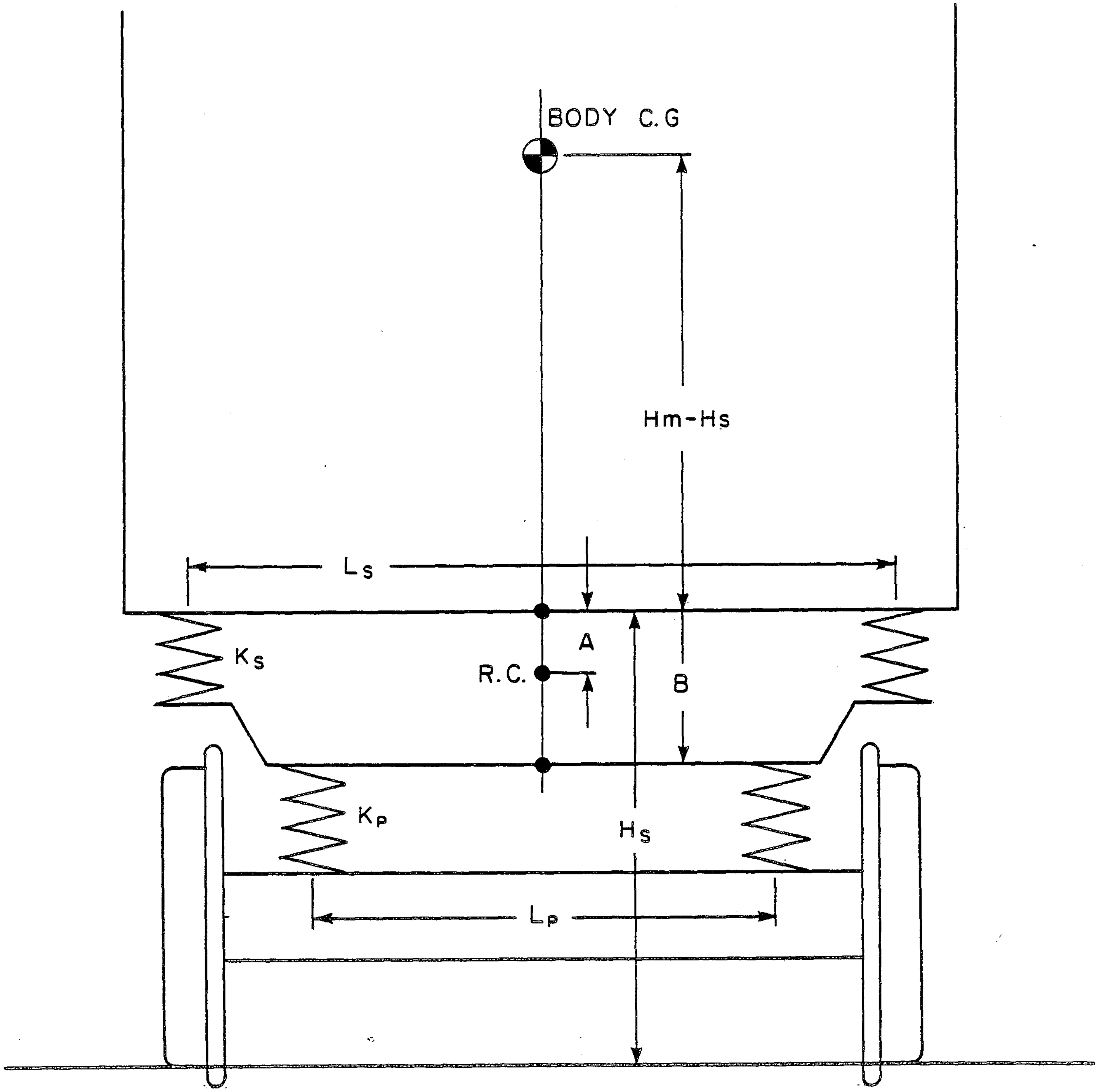


Figure B-2. Parameters for Effective Roll Center Height Calculation

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.

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

where

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

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

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

$$\therefore \frac{\sin(\alpha-\theta)}{\cos\alpha}(\cos\theta)\, mg\sin\theta = \frac{mg\sin(\alpha-\theta)}{\cos\alpha} = \frac{mgu}{60\cos\alpha}$$

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=D1+D2
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=D1+D2+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

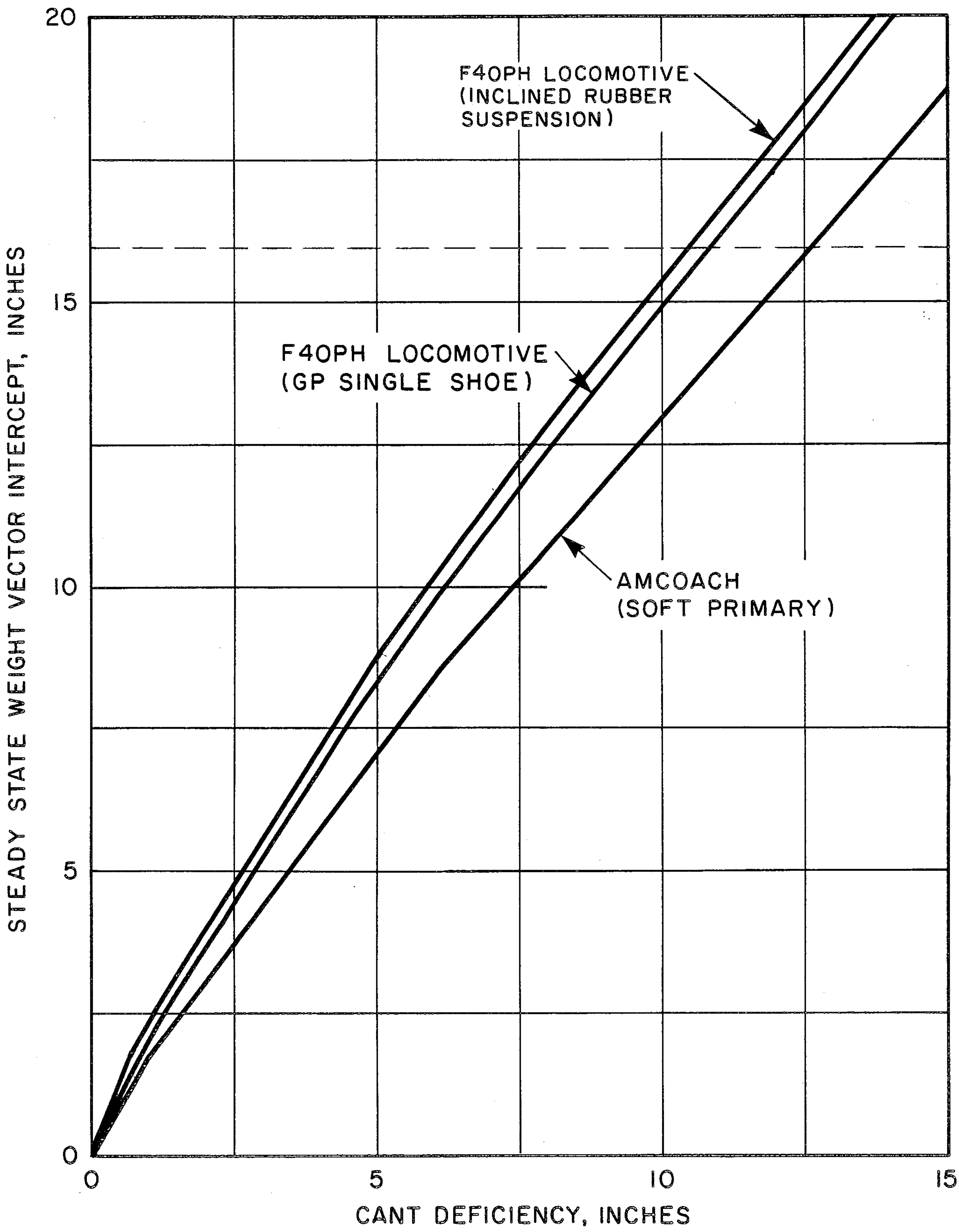


Figure B-3. Prediction of Quasistatic Weight Transfer

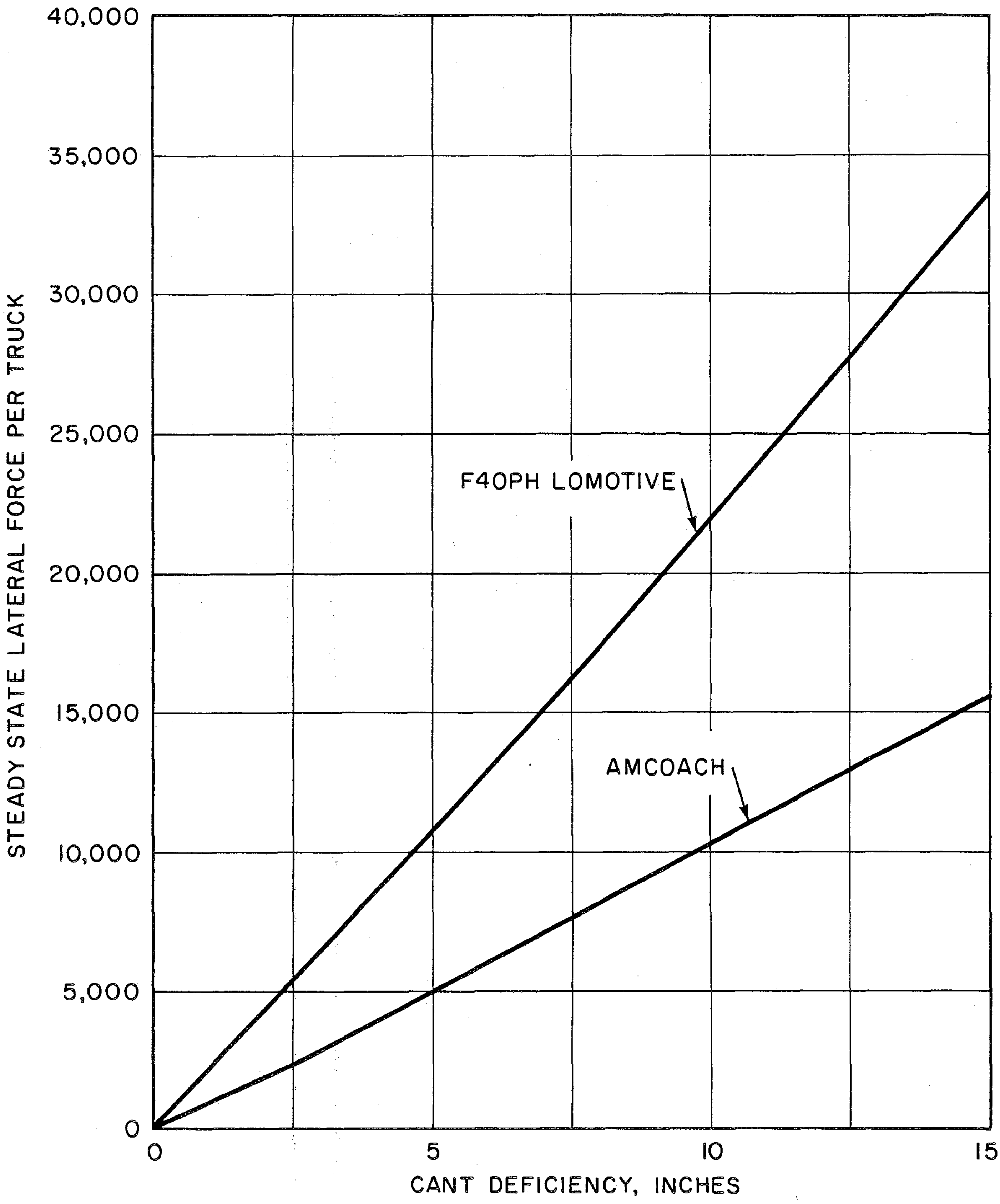


Figure B-4. Prediction of Quasistatic Lateral Truck Force

VEHICLE CHARACTERISTICS

The vehicle characteristics used in the quasistatic curving model in Appendix B 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 as described in Appendix B was derived from the manufactuer's specifications by the following relation (see Figure B-2):

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

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

It was determined experimentally by:

$$H_{RC} = H_s - A = H_s - \frac{p}{p + s} B$$

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:

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

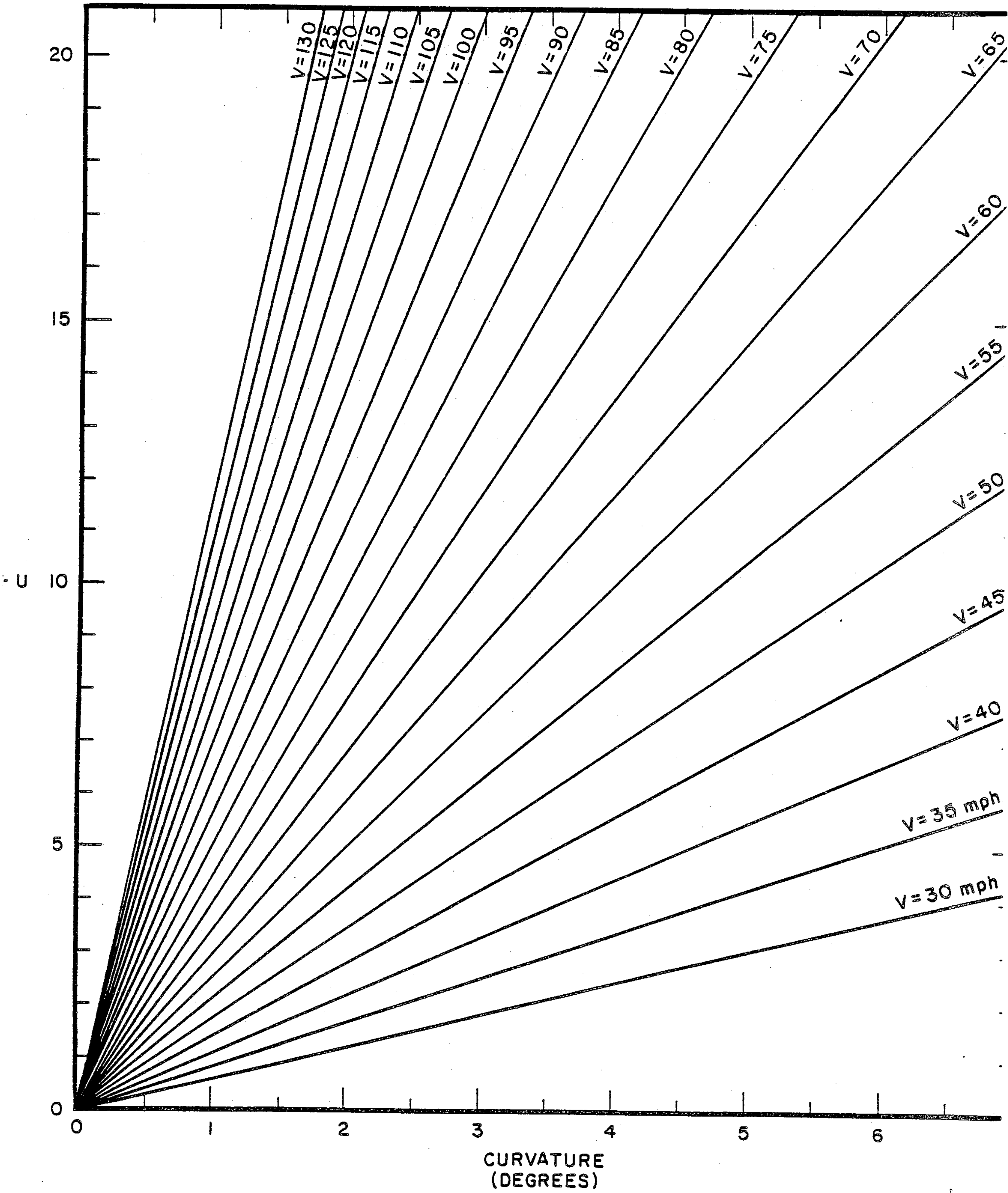
where H_m is the body c.g. height and m is the body mass.

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.

APPENDIX C

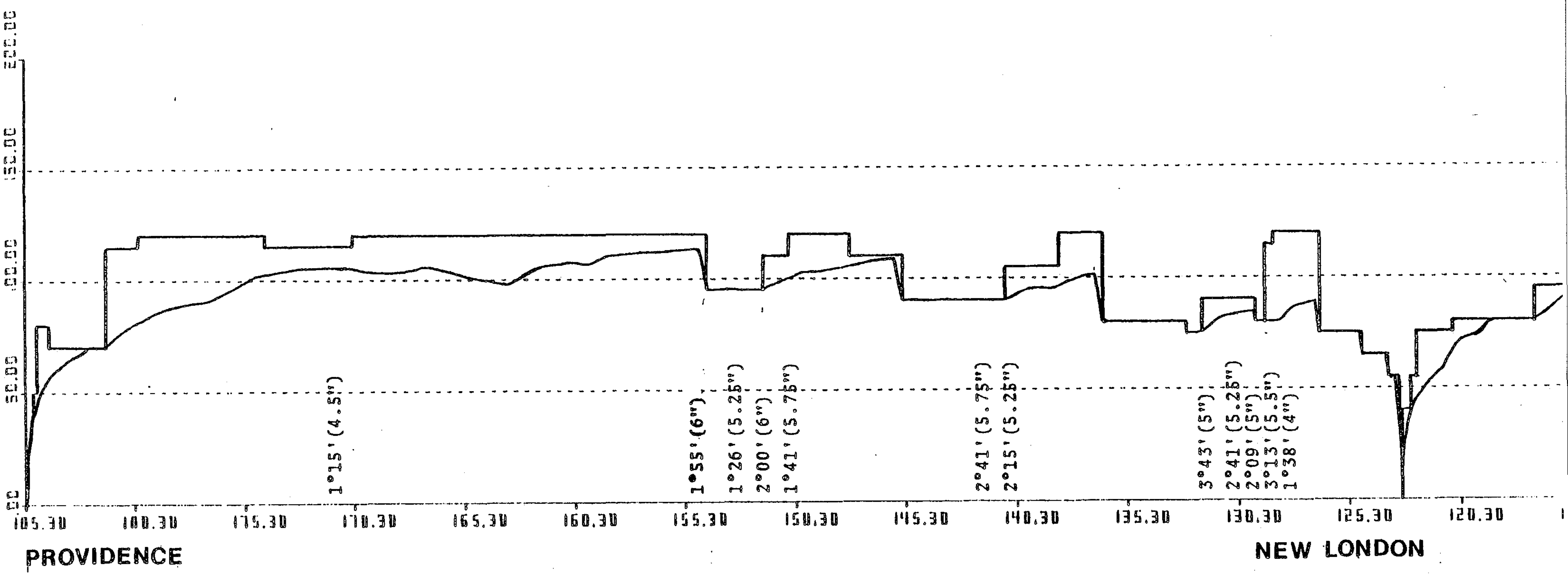
HIGH SPEED CURVING SPEED CHART

HIGH SPEED CURVING CANT DEFICIENCY



APPENDIX D

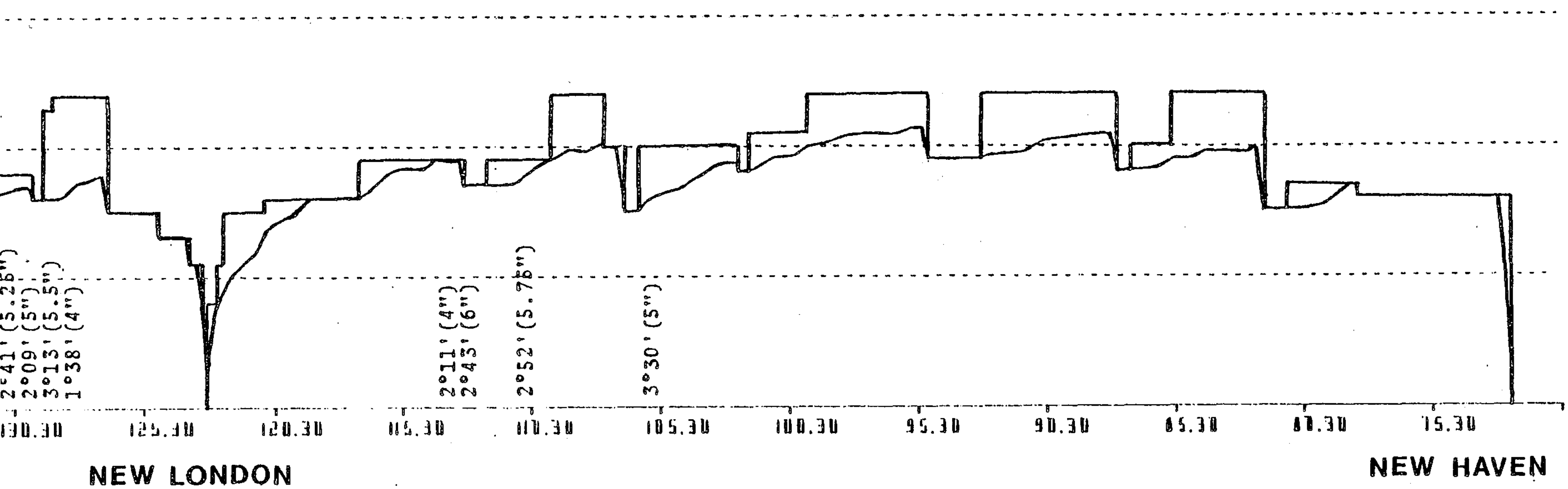
SIMULATED RUN NEW HAVEN - PROVIDENCE, RI



SIMULATED RUN

PROVIDENCE TO NEW HAVEN

EU=9" 120 mph LRC, 1 + 5 1-STOP



SIMULATED RUN

PROVIDENCE TO NEW HAVEN

EU:9"   120 mph   LRC, 1 + 5   1-STOP

APPENDIX E

5.0 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 here. 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.

5.1 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 5-1 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 5-1. 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.

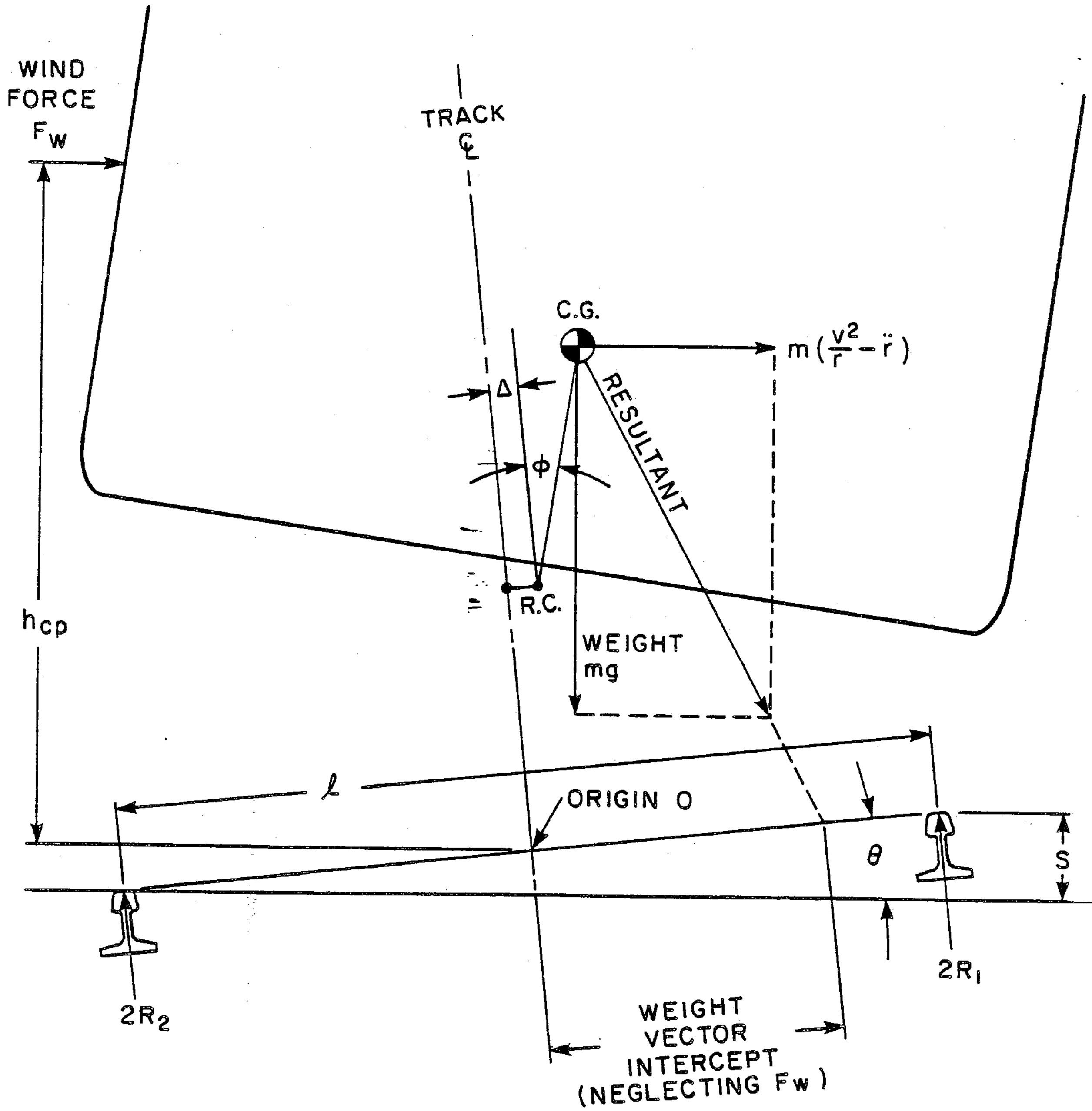


Figure 5-1. General Vehicle Overturning Model

The concept of weight vector intercept is used frequently to express the risk of vehicle rollover. Momentarily neglecting the wind force, Figure 5-1 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 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 5-1 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 because 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.

5.1.1 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 ($\pm$ 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 5-2:

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:

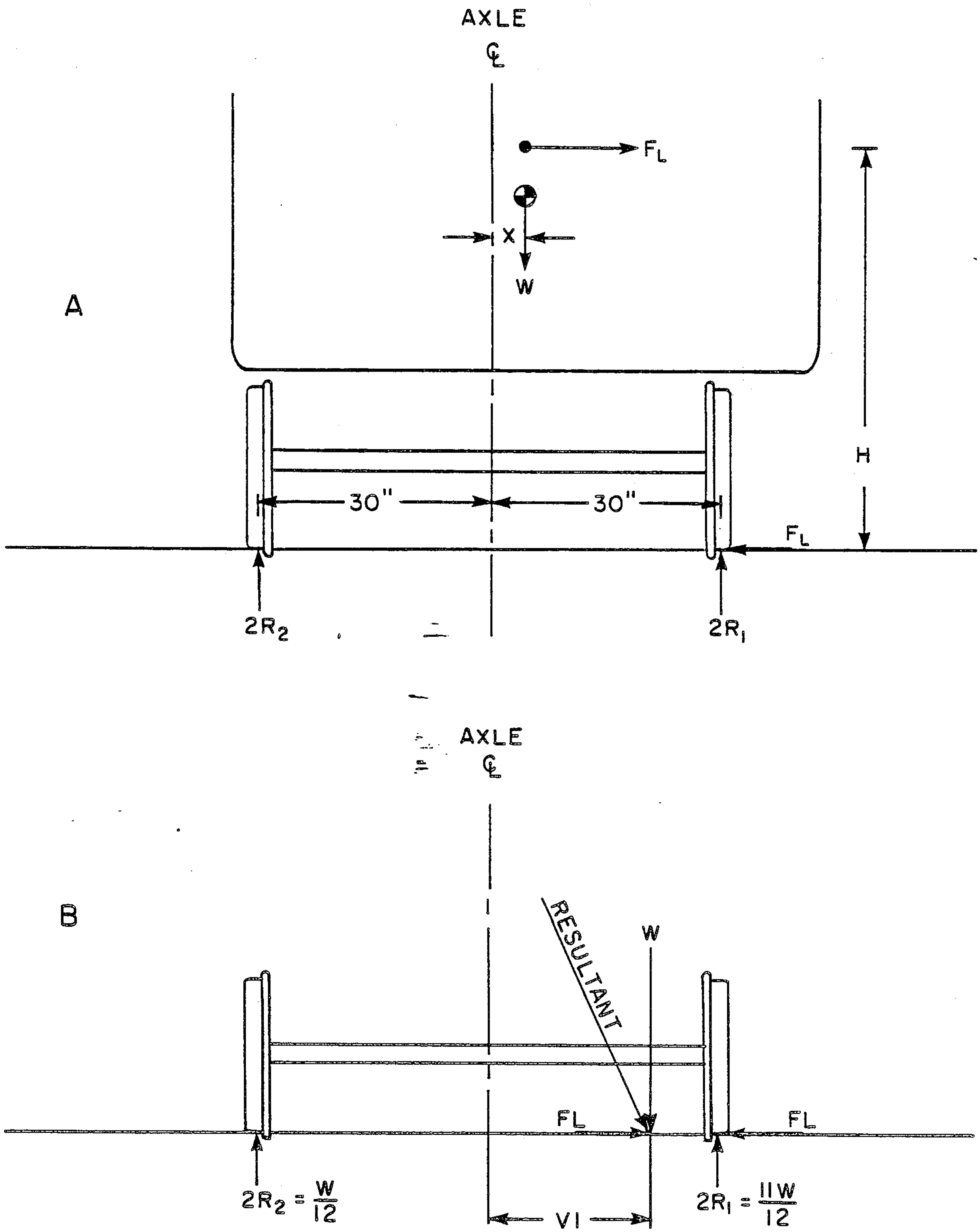


Figure 5-2. Freebody Diagram, Overturning Moment Criteria

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

Taking moments about the outer rail:

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

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

Taking moments about the inner rail

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

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

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 5-2B 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 5-1 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 5-1, the wheel load reduction ratio, C_r is:

$$C_r = \frac{P}{P} \times 100\% = \left[\frac{\left(\frac{R_1 + R_2}{2}\right) - R_2}{\left(\frac{R_1 + R_2}{2}\right)} \right] \times 100\% = \frac{R_1 - R_2}{R_1 + R_2} \times 100\%$$

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 <u>whole</u> 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 5-3 can be computed in general terms. Considering only M_o and summing moments about the origin.

$$0 = 2R_1 \frac{\ell}{2} - 2R_2 \frac{\ell}{2} - M_o$$

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

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

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

where h_{cp} is the height of the center of wind pressure in feet and is the crosslevel angle.

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

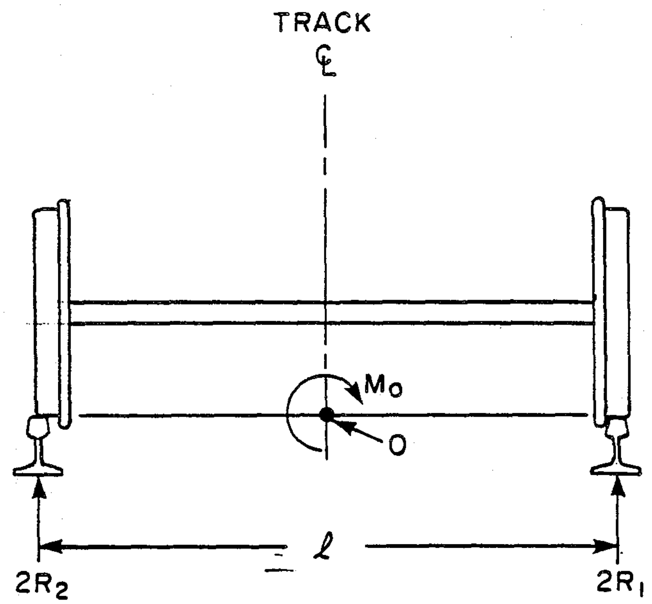


Figure 5-3. Freebody Diagram for Weight Transfer Due to General Overturning Moment

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

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 5-4.

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 the lateral shift of the body c.g., is $X \cdot W_B$, where X is the

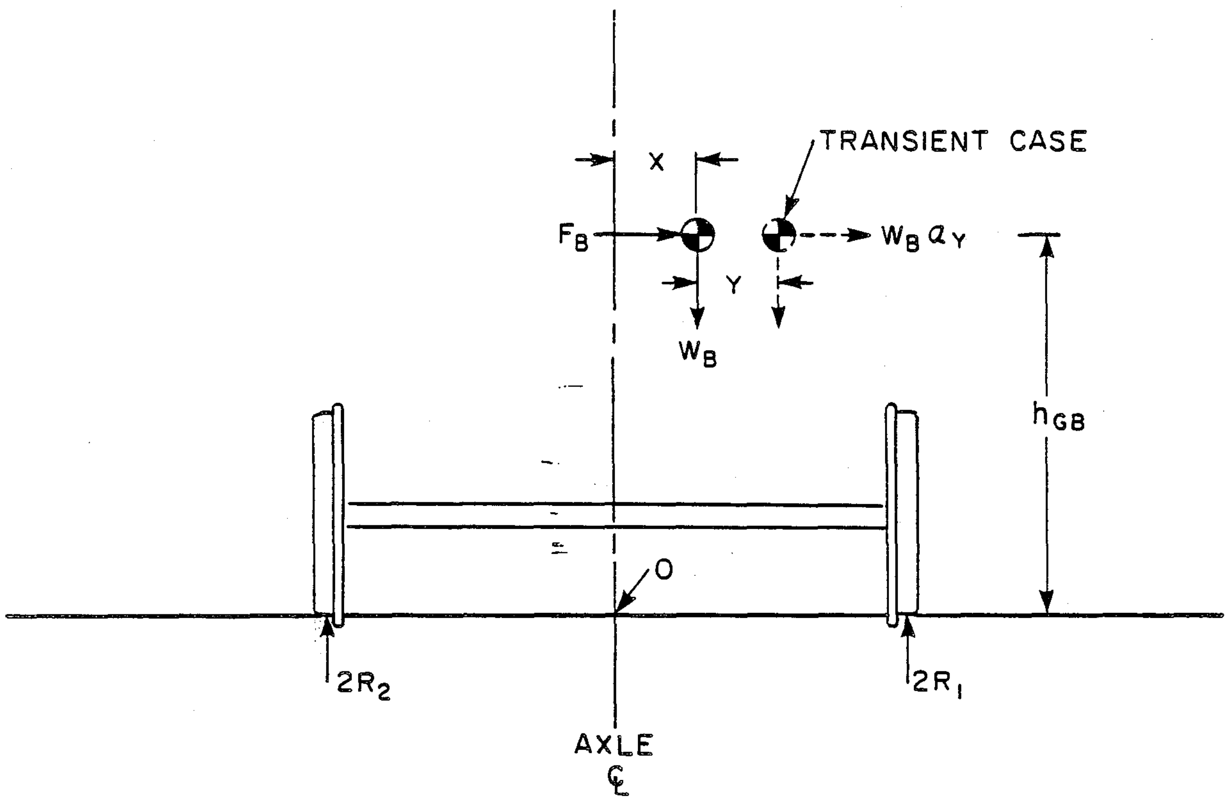


Figure 5-4. Free Body Diagram, Vertical Load Reduction Ratio Criteria

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, $X = C_y F_B$.

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

$$\frac{\Delta P_1}{P} = \frac{2Mo}{\ell W} = \frac{2F_B \; h_{GB} + C_y W_B}{\ell W} = \frac{0.4F_B \; h_{GB} + C_y W_B}{W}$$

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

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

The JNR steady state overturning safety criteria may be summarized:

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

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

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

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

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

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

$$0 = 2 R_1(\ell/2) - 2\Delta R_2(\ell/2) - W_B a_y h_{GB} - y W_B$$

but

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

and

$$y = C_y W_B a_y$$

$$0 = \Delta P_3 \ell - (-\Delta P_3)\ell - W_B a_y h_{GB} - (C_y W_B a_y) W_B$$

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 $\ell = 5$ 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:

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

5.1.2 COMPARISON OF OVERTURNING CRITERIA

The net weight vector intercept specified by each criteria for a particular lateral wind speed may be obtained by subtracting $.0153\ V^2Sh_{cp}/W$ 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 5-5 and 5-6 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.

	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 5-5 and 5-6 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 allowed for wind speed is required because an overly conservative wind speed assumption wastefully reduces the normal operating criteria since other factors such as visability and debris on the track already 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

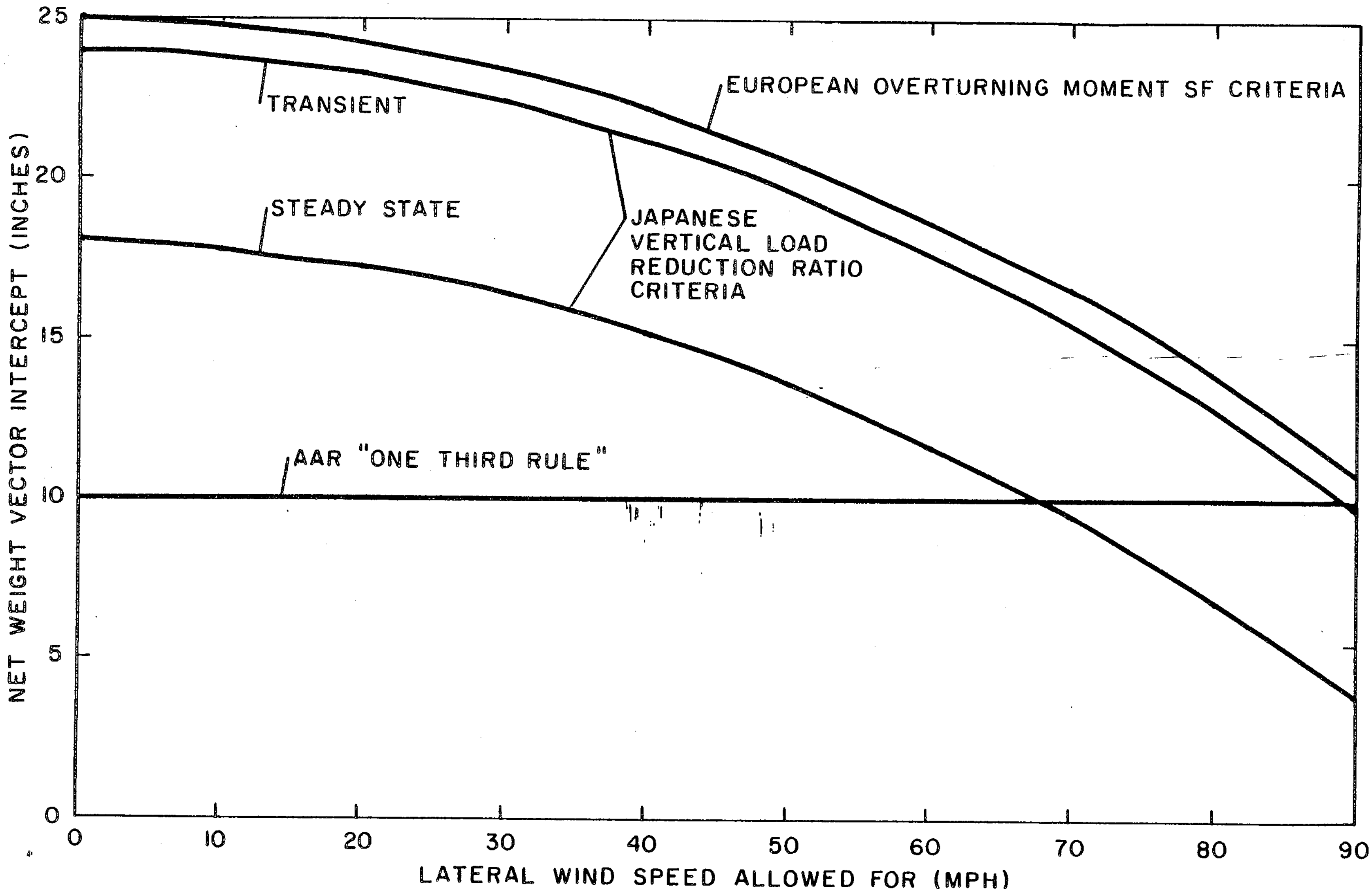


Figure 5-5. Comparison of Various Overturning Safety Criteria for LRC Coach

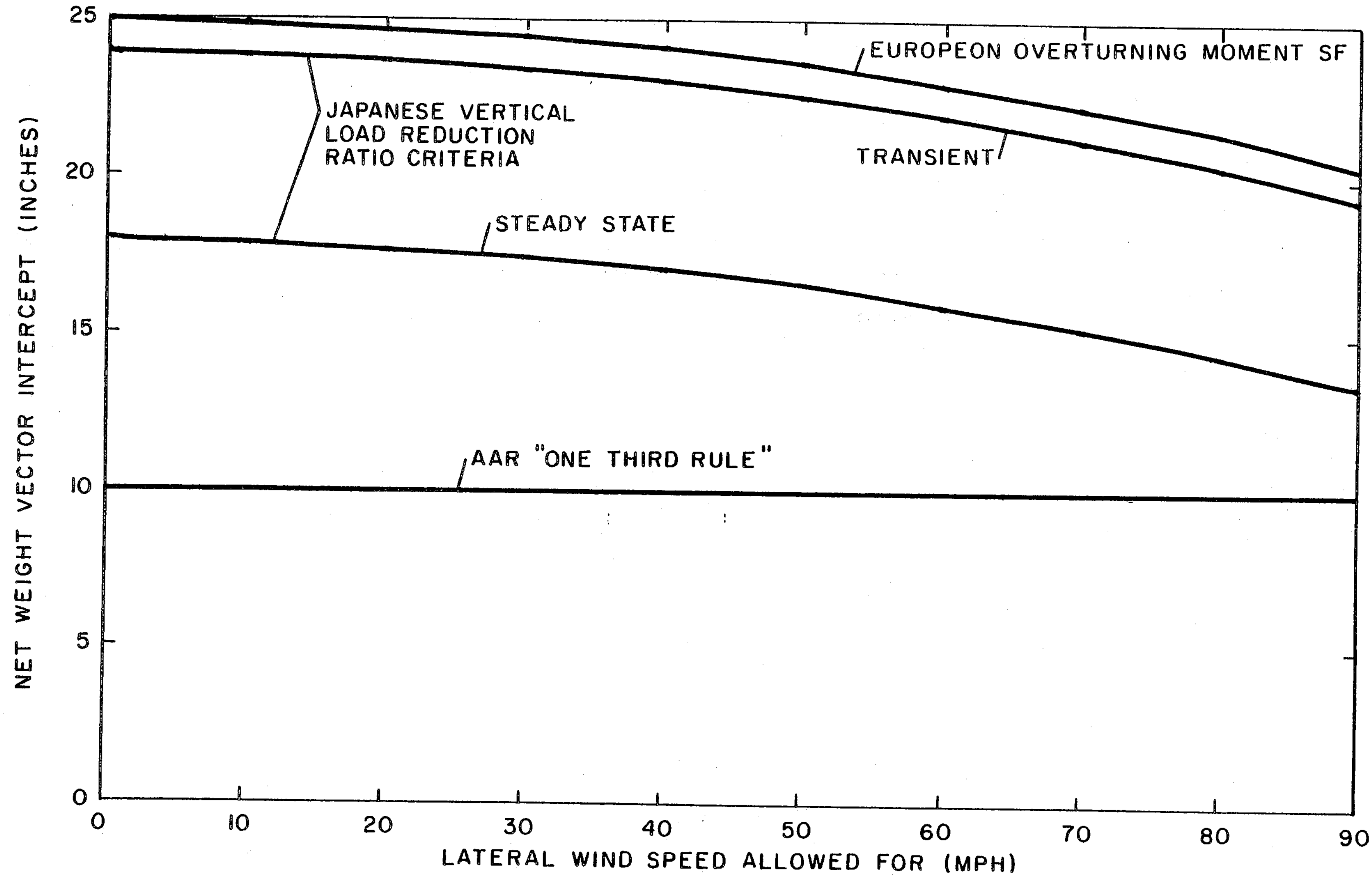


Figure 5-6. Comparison of Various Overturning Safety Criteria for LRC Locomotive

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.

5.2 <u>WHEEL CLIMB CRITERIA</u>

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 $L/V = (\tan \alpha \pm \mu)/(1 \pm \mu \tan \;)$ where 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 and 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 is also controversial.

Many of the second order factors such as weelset 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:

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

For negative angles of attack:

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

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 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 ($\alpha = 61^o$, $\mu = .4$) 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.

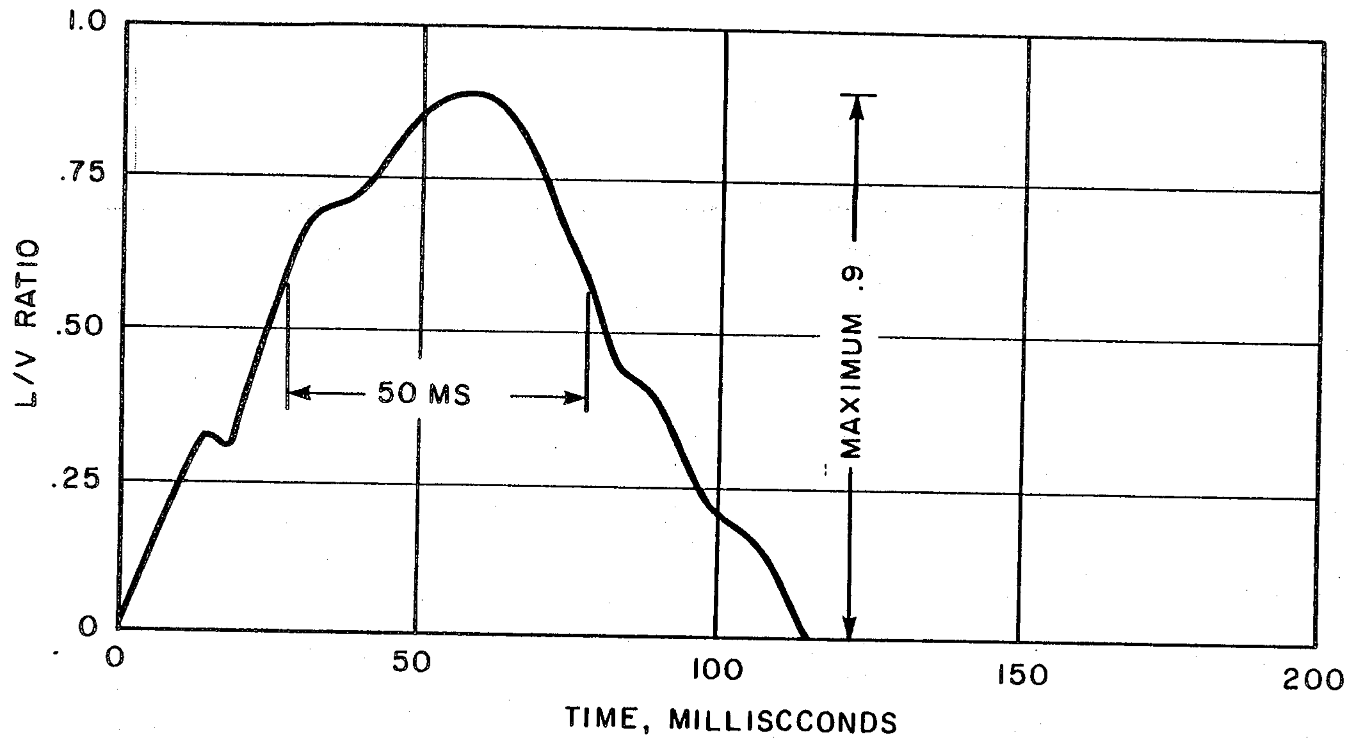
Amtrak has included in its specifications for the AEM-7 locomotive (AAR flange angle $66^{\circ} - 68^{\circ}$, increasing further with wear) an L/V criteria which clearly relates its permissable magnitude to pulse duration. The method of defining pulse duration in this specification has been described 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 5-7 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 5-8 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



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 5-7. Interpretation of Instantaneous L/V Measurements

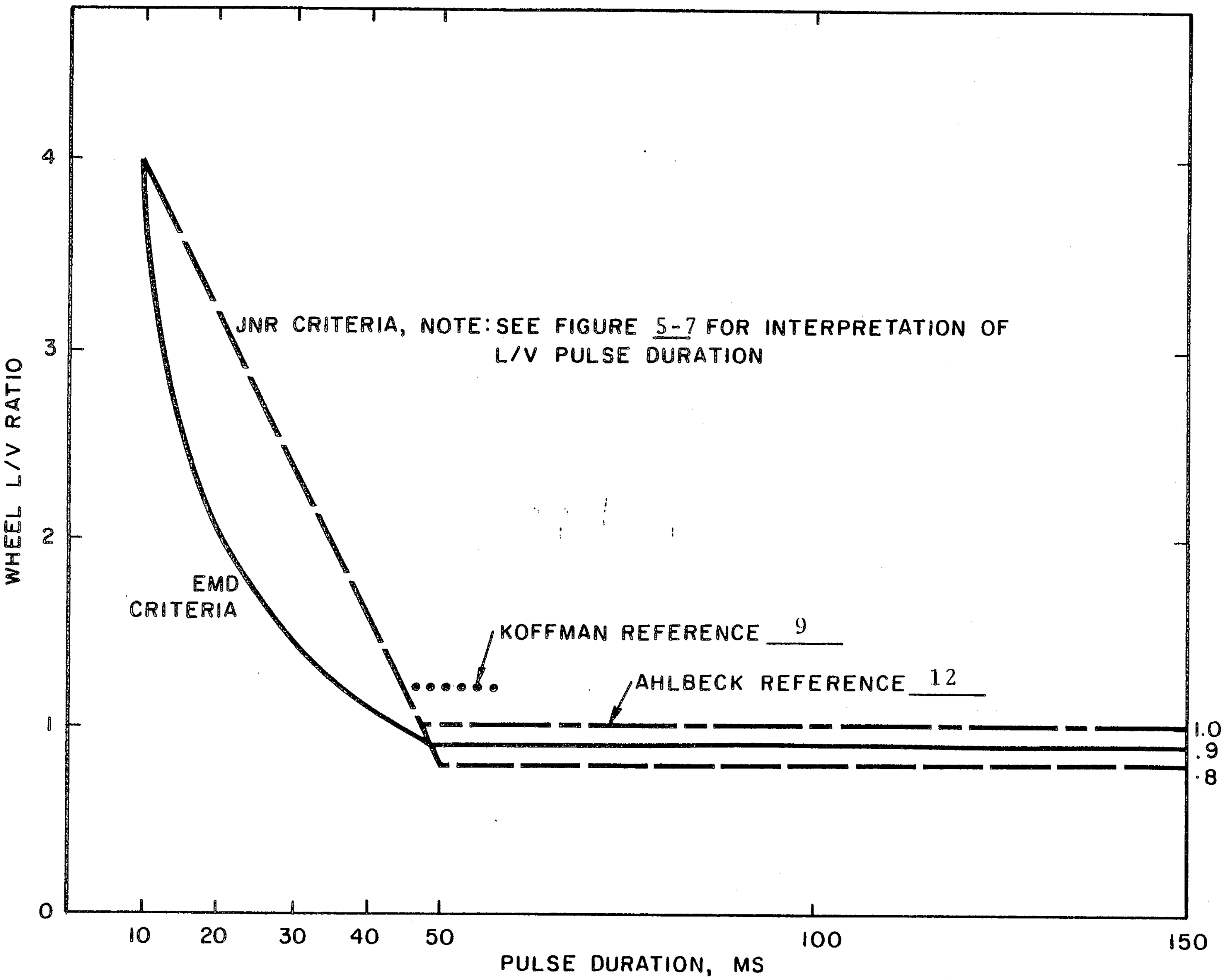


Figure 5-8. Comparison of Various Wheel Climb Criteria

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 large and the speed becomes very low but it should remain considerably less than the maximum reported values of 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 ($\alpha = 61^{\circ}$).

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.

5.3 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 intesects a point within the base of the rail. The limit of the purely geometrical resistance to rollover as shown in Figure 5-9 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 scrutenized 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 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

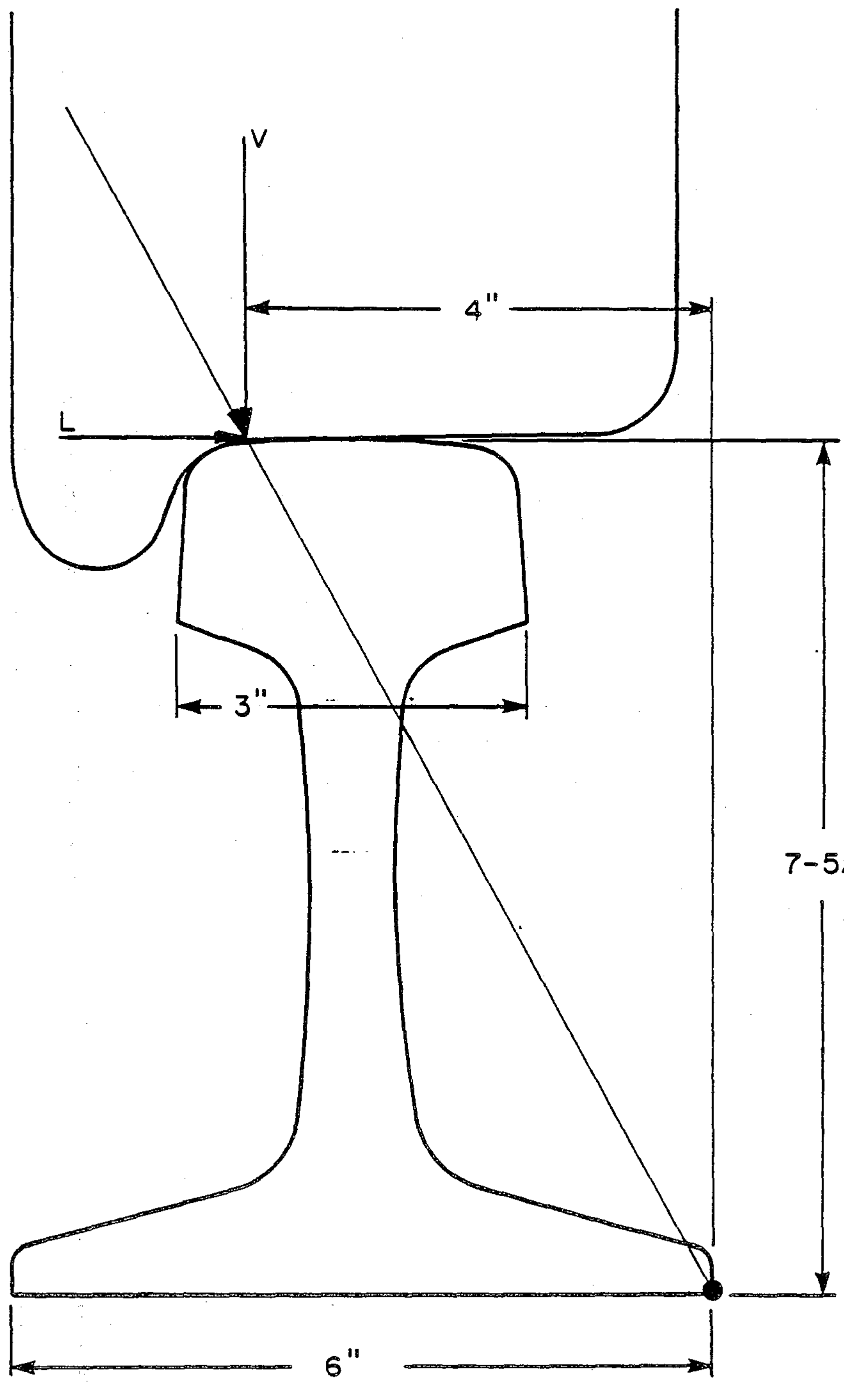


Figure 5-9. Cross Section Geometric Resistence to Rail Rollover

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 5-9 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 5-10 plots the criterion which may be stated:

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

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

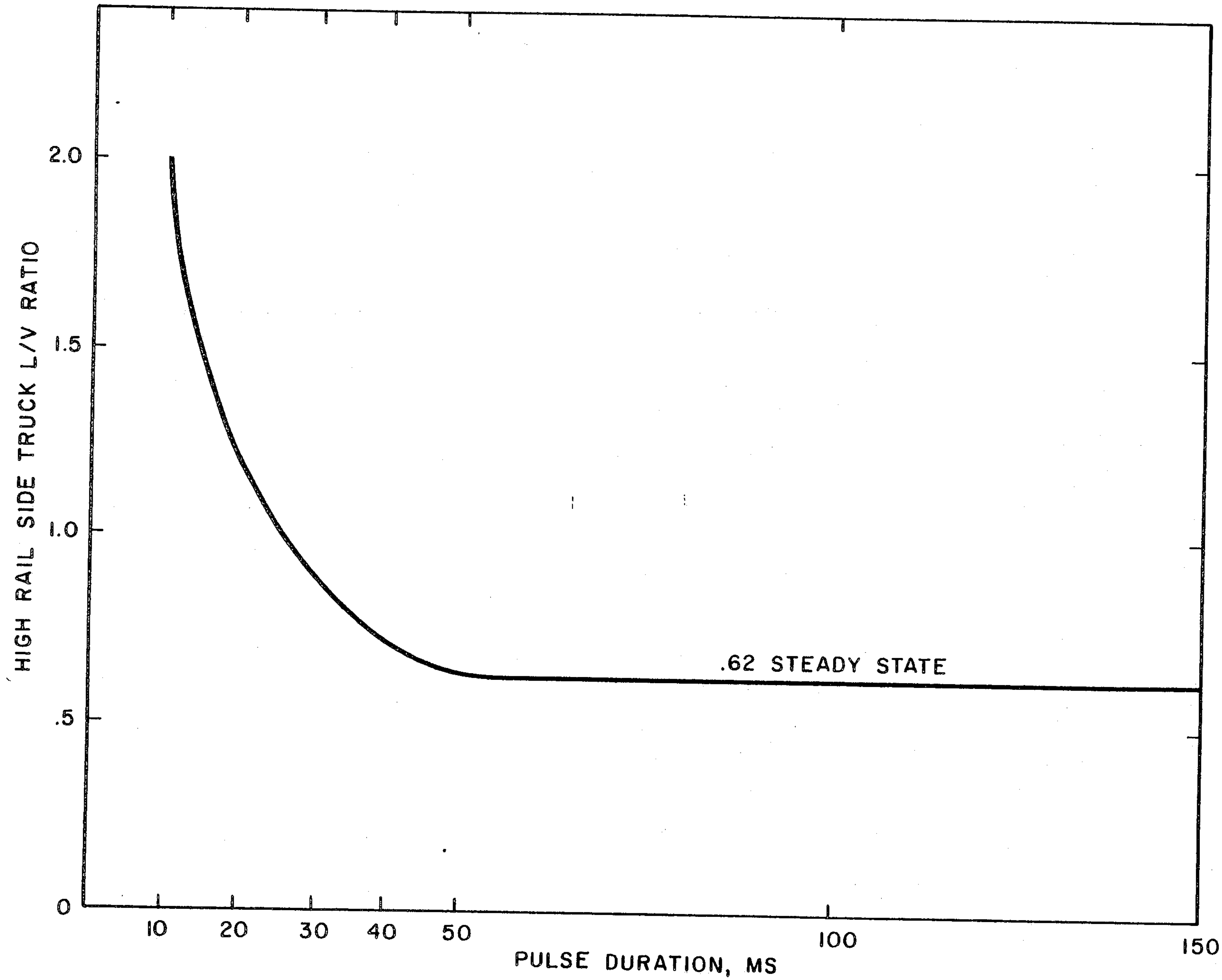


Figure 5-10. Rail Rollover Criteria, AEM-7 Specifications

5.4 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 zeo wind measurements and applied to traffic on CWR, 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:

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

where:

A = rail cross section area, in^2
$\Delta\theta$ = max temperature change after rail installation, $^{\circ}F$
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°F and D max of 4°.

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

Figure 5-11 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 Atmrak 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.

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 $T \geq 50ms$ 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 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: Fmax (truck) = .7nFmax (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.

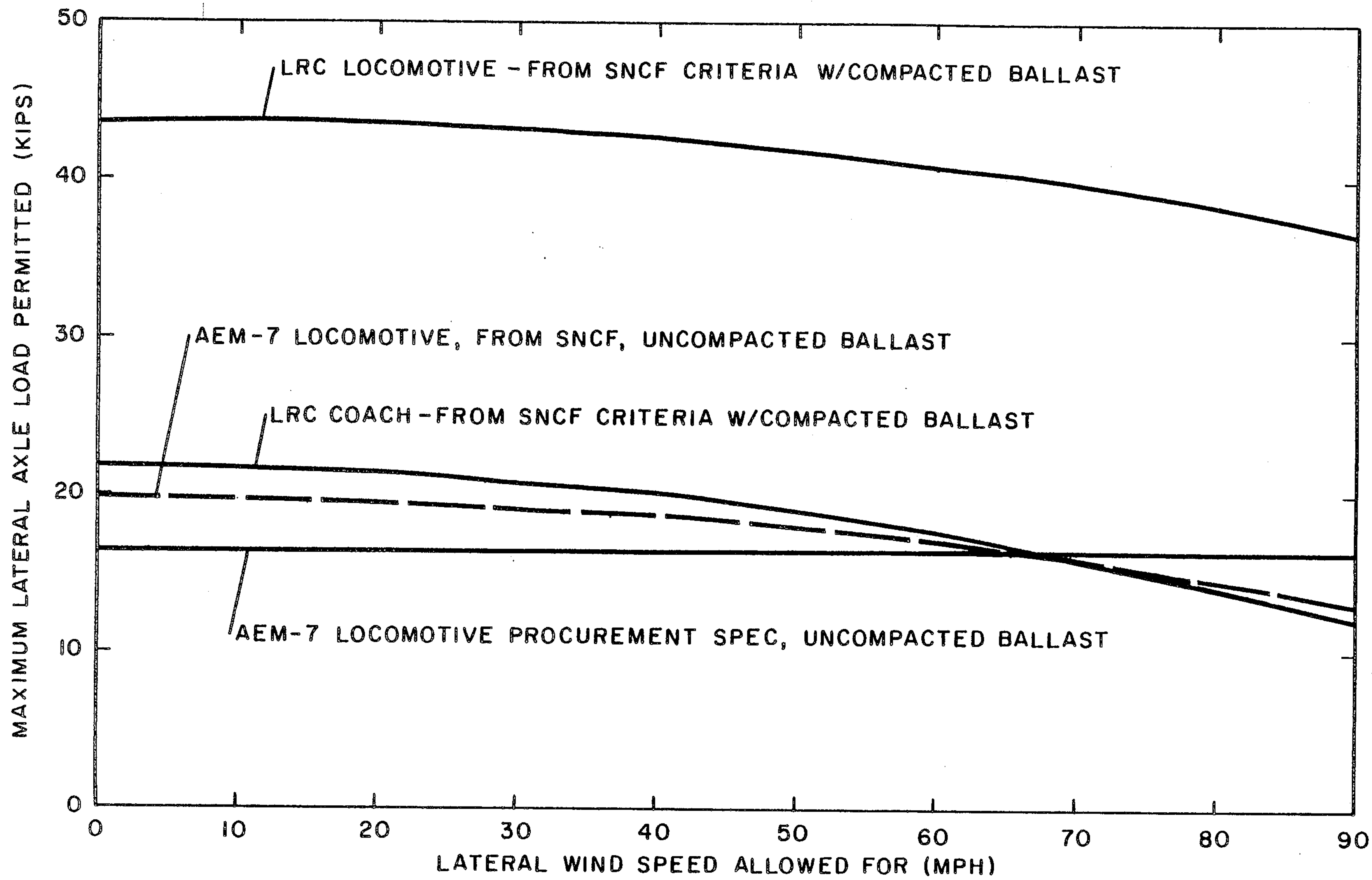


Figure 5-11. Comparison of Various Lateral Track Shift Criteria

The specific criteria proposed for U.S. service appears to be judgements by various investigators based mainly on the same 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.

5.5 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:

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

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 5-12 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 5-1 summarizes the lateral acceleration recommendations by several organizations (many of them quoted in Reference 21) for steady state lateral acceleration and other criteria.

Figure 5-13 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 it may be more valuable that 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

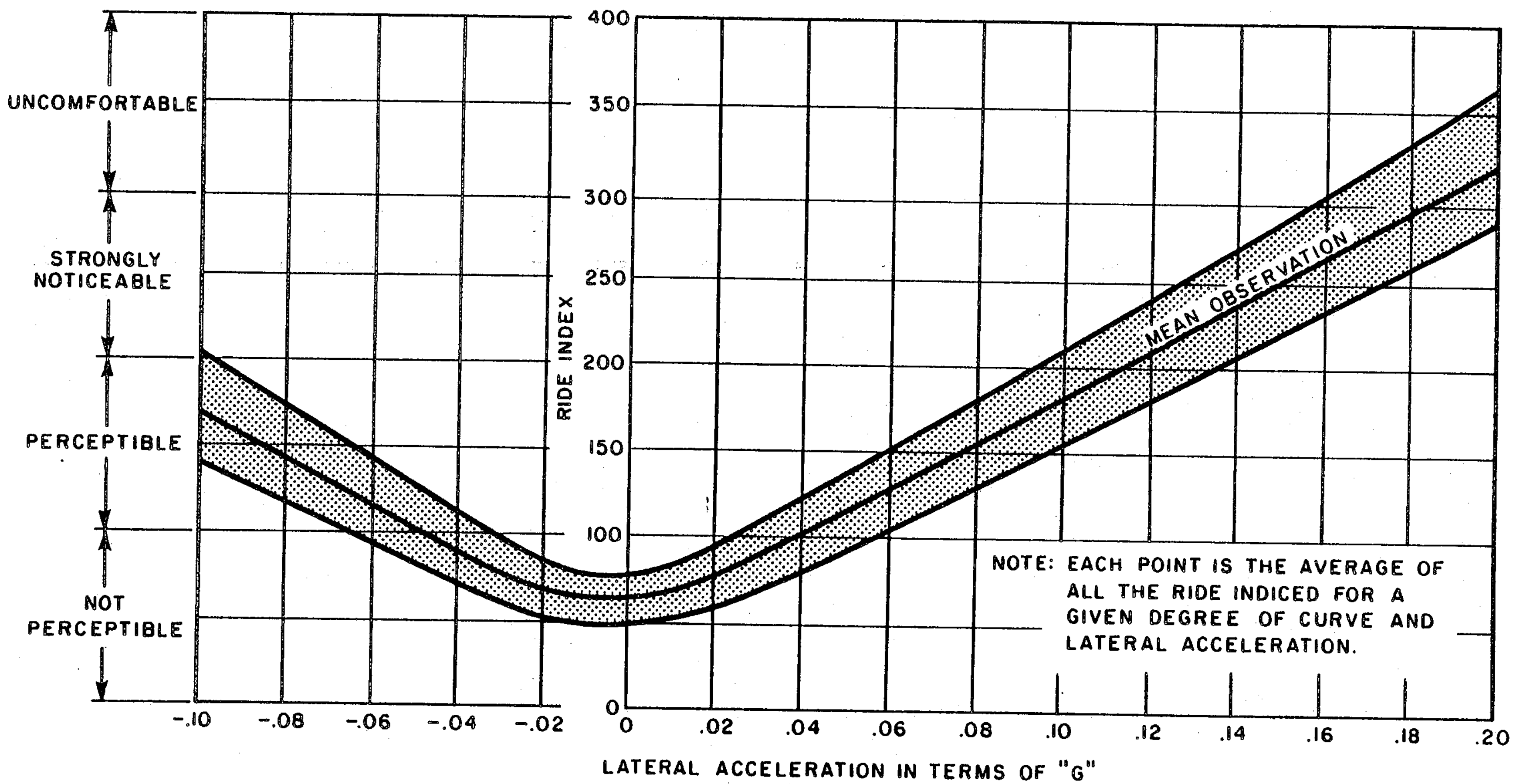


Figure 5-12. Relation of Subjective Comfort Rating to Carbody Lateral Acceleration for AAR Experiment

TABLE 5-1

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)	- -	- -

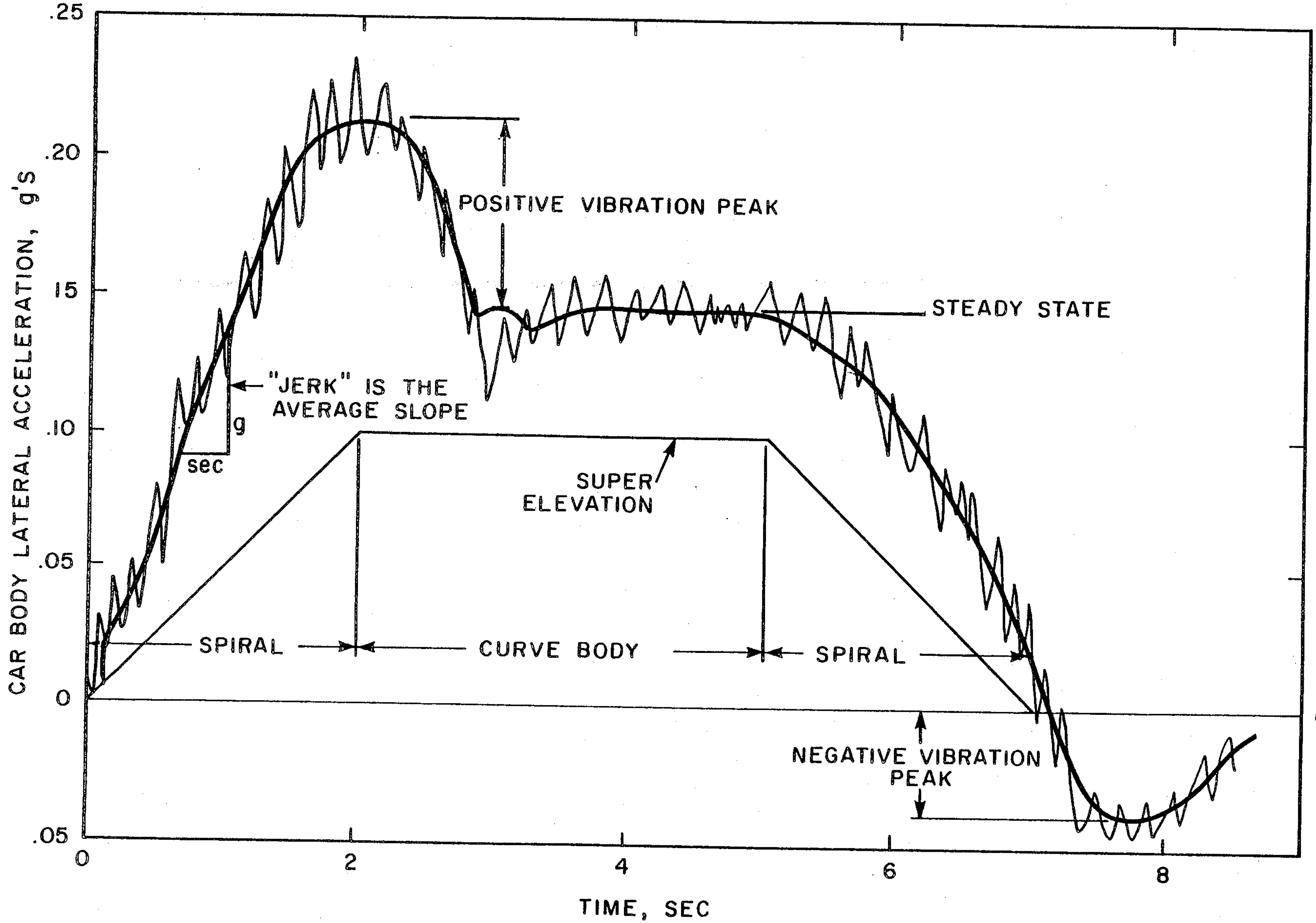


Figure 5-13. Features of Carbody Lateral Acceleration Referenced to Location on Curve

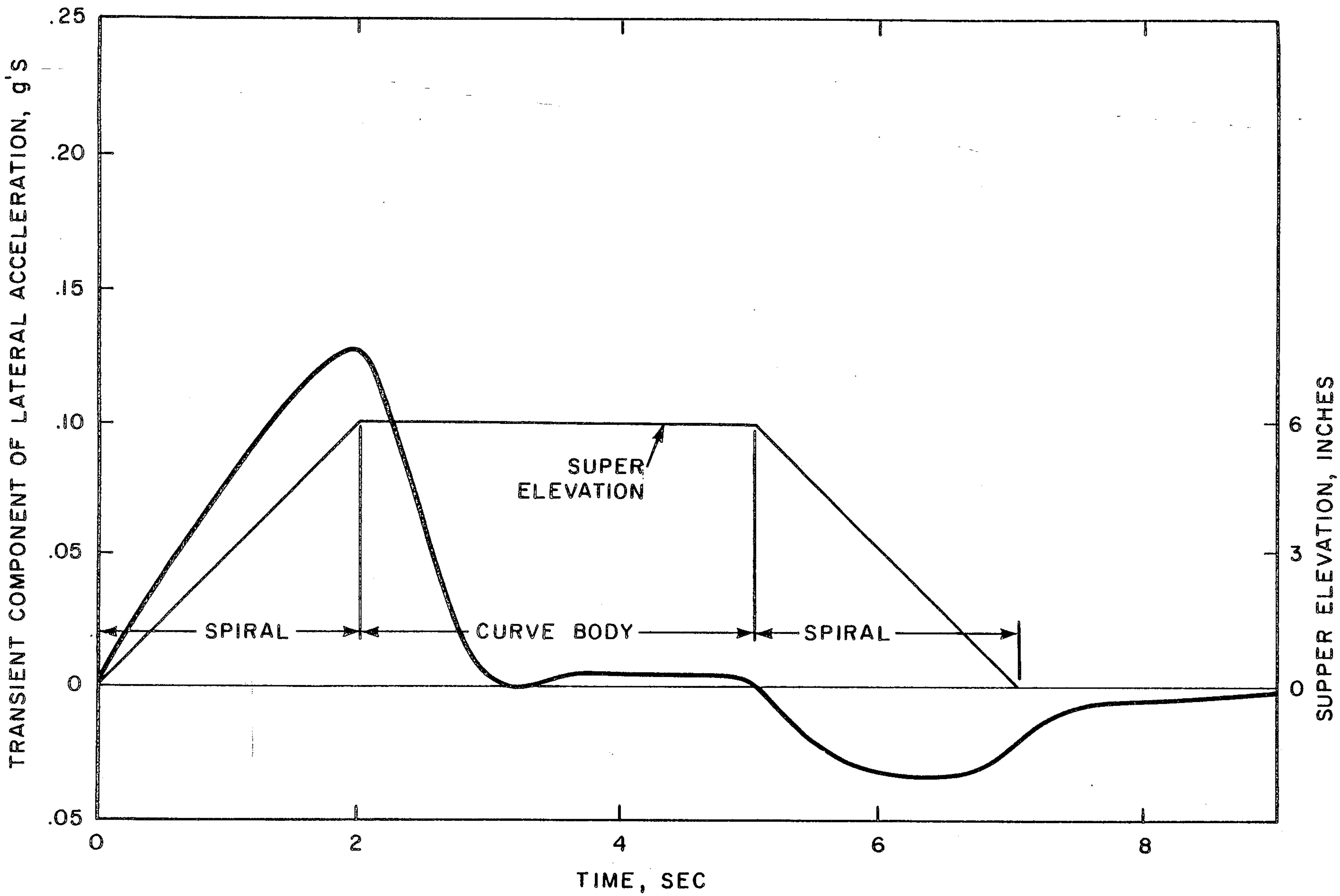


Figure 5-14. Typical "Transient Response Diagram" of "Carbody Vibration"

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 5-15 to judge the natural frequency response of proposed suspension systems to mathematically model steady sinusoidal track perturbations.

5.6 RECOMMENDED SAFETY AND RIDE COMFORT CRITERIA

VEHICLE OVERTURNING

A criterion based on the JNR load ratio standards is useful for comparison to steady state models and measurements and to peak measurements, and it offers a convenient method to handle the safety implications of high crosswinds. Peak measurements of about 50 ms duration should be used, and the lower of the two limits derived from steady state and peak comparisons should be taken when both measurements are available. The criterion may be stated by the following two conditions expressed in terms of weight vector intercept.

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

and

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

where:

V = the lateral wind speed in mph

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

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

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

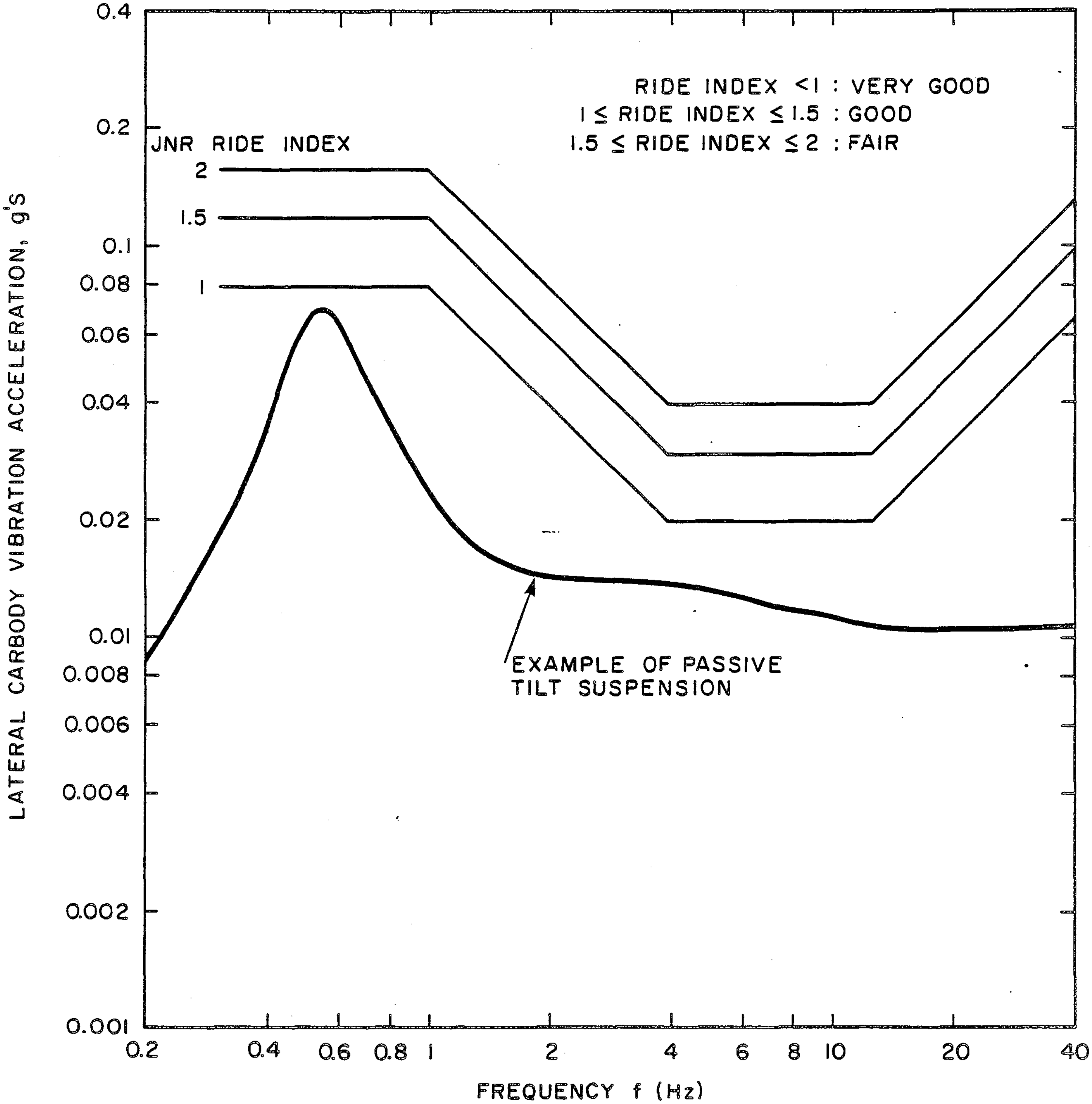


Figure 5-15. Suspension Frequency Response to Steady Sinusoidal Lateral Perturbation

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 it considers the AAR flange angle. It may be expressed as:

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

and

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

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 with conservative judgements of friction coefficient and angle of attack.

RAIL ROLLOVER

The rail rollover criterion based on rail section geometry and AAR measurements of the torsional support of the surrounding rail but which assumes zero pull out strength of the fasteners has been expressed:

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

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:

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

where

A = rail section area, in^2

$\Delta\theta$ = max temperature change after rail installation, ^oF

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^oF and D max of 4^o.

$$Fmax = .61P + 5800 - 1.28 \times 10^{-3}\ SV^2$$

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

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

where N is the number of axles per truck and one truck supports half the total wind load. Peak 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 experiment.

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 existing 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.